Implementation of SMED in the Landing Gear Maintenance Process

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Thesis to obtain the Master of Science Degree in

Aerospace Engineering

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Aos meus pais
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I could not have completed this thesis without the input and continuous support of many.

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Resumo

As organizações da indústria aeronáutica têm vindo a adoptar princípios *lean* para garantir que se mantêm competitivas, rentabilizando custos. As abordagens *lean* pretendem remover etapas específicas de processos que não criam valor para o produto final e optimizar etapas que o criam. Neste sentido, a ferramenta SMED serve para eliminar desperdícios gerados por actividades de troca de ferramentas, em ambientes de produção ou de manutenção. É uma ferramenta que tem como objectivo reduzir ao máximo o tempo de paragem de uma máquina, realizando o máximo de operações enquanto esta está em funcionamento. A empresa OGMA procurou aplicar este mesmo princípio e, deste modo, simplificar os processos de montagem, desmontagem e troca de ferramentas no seu banco de ensaios. Adicionalmente, foi requerida a criação de uma ferramenta de ligação à prensa, capaz de receber todo o *portfolio* de trens da OGMA. O trabalho descrito na presente tese seguiu os passos clássicos da ferramenta SMED, de forma a tentar atingir o primeiro objectivo. Princípios de projecto mecânico foram utilizados no desenvolvimento de soluções de optimização e da ferramenta de conexão universal pretendida. Com as soluções propostas, obtiveram-se ganhos de tempo de até 90%, reduzindo as variações nos processos da área de teste e os seus tempos totais de montagem, desmontagem e troca de ferramentas.

**Palavras-chave:** Trem de aterragem, Manutenção, Prensa Hidráulica, SMED, Projecto Mecânico, Ferramenta de Ligação.
Abstract

Companies in the aeronautical industry have been adopting lean principles to ensure that they remain competitive by keeping their processes cost effective. These approaches intend to remove non value added steps in processes and streamline activities which contribute positively to a final product. In this sense the SMED tool serves to eliminate wastes being generated by changeover activities in a production plant or maintenance facility. It is a tool which aims to keep idle times of machinery to a minimum, by conducting as much work as possible while it is running. OGMA intended to apply this very principle and streamline the set-up, removal and tool changeover processes at its test bench. Additionally and in line with this topic an universal connection fixture was requested which could be used for OGMA's entire landing gear portfolio. The work described in this thesis follows the classical steps of SMED in attempting to achieve the first request. Mechanical design was present in developing streamlining solutions and the universal connection fixture for the test bench. The devised solutions achieved time gains up to 90%, effectively reducing both variations in test area processes and their overall set-up, removal and tool changeover times.

Keywords: Landing Gear, Maintenance, Hydraulic Test Bench, SMED, Mechanical Design, Connection Fixture.
Contents

Acknowledgments .......................................................... v
Resumo ........................................................................... vii
Abstract ......................................................................... ix
List of Tables .................................................................... xiii
List of Figures ................................................................... xv
Nomenclature ................................................................... xix
List of Acronyms ............................................................. xxii

1 Introduction 1

1.1 Motivation ................................................................. 1
1.2 Topic Overview .......................................................... 3
1.3 Objectives .................................................................... 4
1.4 Thesis Outline .............................................................. 5

2 Background 6

2.1 Lean Approach ........................................................... 6
2.1.1 Muda .................................................................... 7
2.1.2 Mura .................................................................... 10
2.1.3 Muri .................................................................... 11
2.2 SMED .......................................................................... 12
2.2.1 The Stages of SMED ................................................. 14
2.3 Mechanical Design ...................................................... 17
2.3.1 FEA ..................................................................... 20
2.4 Current Situation at OGMA .......................................... 23
2.4.1 Existing Tools ........................................................ 23
2.4.2 Changeover and set-up duration ............................... 28

3 SMED Implementation .................................................. 31

3.1 Registered Durations and Actions ............................... 31
3.1.1 Set-up Times .......................................................... 32
3.1.2 Connection Tool Changes ........................................ 36
3.2 Converting Internal Activities to External Activities ....... 38
3.3 Proposed Improvements to Activities ........................... 41
3.3.1 Short to Medium Term Solutions ............................. 41
3.3.2 Long Term Solutions .................................................. 50

4 Universal Tool Development .......................................... 52
  4.1 FEA Meshes and Results ............................................. 54
  4.2 Upper Fixture .......................................................... 56
    4.2.1 Claw ............................................................. 57
    4.2.2 Beam ........................................................... 59
    4.2.3 Sliders .......................................................... 63
    4.2.4 Under Slider ................................................... 65
    4.2.5 Pins and Rod-end Bolt ........................................ 66
    4.2.6 Pin Connecting Part .......................................... 66
    4.2.7 Fitting Bushings .............................................. 67
  4.3 Lower Fixture .......................................................... 68
    4.3.1 Piston Connection ............................................. 69
    4.3.2 Inferior Shaft .................................................. 70
    4.3.3 Thrust Bearing ............................................... 71
    4.3.4 Support Body ................................................. 72
    4.3.5 Sliders .......................................................... 74

5 Concluding Remarks .................................................... 76
  5.1 Results ............................................................... 76
  5.2 Achievements ....................................................... 77
  5.3 Future Work .......................................................... 78

Bibliography ........................................................................ 79

A SkyCiv Software Validation ............................................ 82
  A.1 Section Builder ...................................................... 83
  A.2 Beam Calculator .................................................... 84

B Visual Operation Standard Example .................................. 86

C Relevant FBDs .................................................................. 88
  C.1 8mm and 10mm Pins ................................................ 88
  C.2 E190 MLG Beam Loading ......................................... 89
  C.3 E170 MLG Beam Loading .......................................... 90
  C.4 NLG Beam Loading .................................................. 92
  C.5 MLG Support Body Loading ....................................... 93
  C.6 NLG Support Body Loading ....................................... 94

D Finite Element Analyses and Results ................................ 96
List of Tables

2.1 Example of parallel operations (adapted from [15]). .......................... 16
2.2 Total set-up times for the different observed ship-sets. ...................... 29

3.1 ERJ145 SA Set-up Activities ................................................. 33
3.2 E190 MLG Set-up Activities .................................................. 34
3.3 E190 NLG Set-up Activities .................................................. 34
3.4 SA Removal Activities .......................................................... 35
3.5 E190 MLG Removal Activities .................................................. 35
3.6 E190 NLG Removal Activities .................................................. 36
3.7 SA to MLG Beam Tool Change ................................................... 37
3.8 SA to NLG Tool Change .......................................................... 37
3.9 NLG to SA Tool Change .......................................................... 37
3.10 Bolt mapping of the test area’s tools. ........................................... 40
3.11 Check-list for E190 MLG set-up, including tool changeover from SA tools. 41
3.12 Internal process times with 5% NVA time. .................................... 41

4.1 General recommendations for factor of safety selection [25]. ................. 53
4.2 Material Digital Logic example according to [26]. .................................. 54
4.3 Obtained relative property weights for the claw part. .......................... 58
4.4 Obtained relative property weights for the main beam part. .................. 63
4.5 Obtained relative property weights for the MLG/NLG slider part. .......... 64
4.6 Obtained relative property weights for the SA slider parts. .................... 65
4.7 Obtained relative property weights for the pin connecting part. ............. 67
4.8 Obtained relative property weights for the NLG fitting bushing. .......... 68
4.9 Obtained relative property weights for the piston connection part. .......... 70
4.10 Obtained relative property weights for the inferior shaft. ................... 71
4.11 Thrust bearing characteristics [29]. ........................................... 71
4.12 Obtained relative property weights for the MLG and NLG slider parts. .... 75

5.1 Results for the internal times. .................................................... 77

A.1 Section properties for 127×76×13 serial size section [33] and computed errors. 83

D.1 Refinement analysis results. ..................................................... 97
D.2 FEA results for each part. ......................................................... 97
List of Figures

1.1 Aerial view of OGMA and the Components’ building, circled in blue [1]........... 2
1.2 Isometric view of the ERJ145’s landing gear assembly, adapted from [2]........... 2

2.1 The influence of the different types of improvements to a changeover process [16]. ... 17
2.2 The different stages in design along with the different feedbacks and interactions [17]. ... 18
2.3 MLG set-up................................................................. 24
2.4 NLG set-up................................................................. 24
2.5 SA set-up................................................................. 24
2.6 NLG beam fixture..................................................... 25
2.7 MLG beam fixture..................................................... 25
2.8 Box assembly.......................................................... 25
2.9 Comparison between the old (top) and new (bottom) beams at the shop................. 26
2.10 NLG support fixture.................................................. 27
2.11 MLG support fixture................................................ 27
2.12 Final cross-section of the MLG support fixture........................................... 27
2.13 SA upper connection fixture....................................... 27
2.14 SA lower connection fixture....................................... 27

3.1 Proposed correction to part on E190 MLG car........................................... 42
3.2 Top view of proposed correction......................................................... 42
3.3 Detail to be corrected............................................................... 43
3.4 Schematic of the rod-end bolt [24].................................................... 45
3.5 Threaded knob [24]............................................................... 45
3.6 3D model of proposed changes....................................................... 45
3.7 Connection part detail.............................................................. 45
3.8 Resting car for the MLG support...................................................... 46
3.9 Resting car for the NLG support...................................................... 46
3.10 SLIC pin CAD model provided by Pivot Point™.................................... 47
3.11 Alternative connection 1............................................................ 50
3.12 Alternative connection 2............................................................ 50
3.13 Transport car used in ELEB......................................................... 51
3.14 CAD model of the ELEB car......................................................... 51
3.15 Schematic of the landing gear shop arrangement...................................... 51
4.1 Applied load and constraints of the MLG/NLG slider part. ........................................... 55
4.2 Schematic 3D model of the final upper fixture. ................................................................. 56
4.3 Location of the claw part. ................................................................................................. 58
4.4 Closer view of the claw part. ......................................................................................... 58
4.5 Dimensions of the claw’s section along with some relevant properties. ....................... 58
4.6 Location of the beam part. ............................................................................................. 59
4.7 Closer view of the beam part. ......................................................................................... 59
4.8 Dimensions of the beam’s cross-section along with some relevant properties. ............. 61
4.9 Location of the MLG/NLG slider part. ............................................................................ 64
4.10 Closer view of the MLG/NLG slider part. ....................................................................... 64
4.11 Location of the SA slider part. ....................................................................................... 65
4.12 Closer view of the SA slider part. .................................................................................. 65
4.13 Location of the under-slider part. .................................................................................. 65
4.14 Closer view of the under-slider part. ............................................................................. 65
4.15 Location of the pin part. ............................................................................................... 66
4.16 Closer view of the pin part. ............................................................................................ 66
4.17 Location of the pin connecting part. .............................................................................. 67
4.18 Closer view of the pin connecting part. ........................................................................ 67
4.19 Location of the MLG170 bushing part. .......................................................................... 67
4.20 Closer view of the MLG 170 bushing part. .................................................................... 67
4.21 Location of the NLG bushing part. ................................................................................ 68
4.22 Closer view of the NLG bushing part. ........................................................................... 68
4.23 Cross-section of the NLG fitting bushing part. ................................................................. 68
4.24 Schematic 3D model of the final lower fixture. ................................................................. 69
4.25 Location of the piston connection. .............................................................................. 69
4.26 Closer view of the part. ................................................................................................. 69
4.27 Location of the interior shaft part. .................................................................................. 70
4.28 Closer view of the interior shaft part. ............................................................................ 70
4.29 Location of the bearing part. ........................................................................................ 71
4.30 Closer view of the bearing part. .................................................................................... 71
4.31 Dimensions of the thrust bearing [29]. ........................................................................ 72
4.32 Location of the support body part. ................................................................................ 72
4.33 Closer view of the support body part. .......................................................................... 72
4.34 Dimensions of the support body’s cross-section along with some relevant properties. . 72
4.35 Location of the MLG slider part. .................................................................................... 74
4.36 Closer view of the MLG slider part. .............................................................................. 74
4.37 Location of the NLG slider part. .................................................................................... 74
4.38 Closer view of the NLG slider part. .............................................................................. 74
4.39 Location of the Shock Absorber slider part. ................................................................. 75
| D.1 | Claw | 98 |
| D.2 | Beam | 98 |
| D.3 | SA notch in the beam | 99 |
| D.4 | MLG/NLG slider | 99 |
| D.5 | SA slider | 99 |
| D.6 | Under-slider | 99 |
| D.7 | Connection part | 99 |
| D.8 | Singularities around the constraint in the under-slider | 99 |
| D.9 | NLG bushing | 100 |
| D.10 | Piston connection | 100 |
| D.11 | Inferior shaft | 100 |
| D.12 | MLG slider | 100 |
| D.13 | Support body | 100 |
| D.14 | Lower surface in the support body | 100 |
| D.15 | NLG slider | 101 |
| D.16 | SA slider | 101 |
| D.17 | Modified suspension ring | 101 |
| D.18 | Connection part of the ring | 101 |
| D.19 | Singularities in the connection part of the suspension ring | 101 |
| D.20 | Maximum combined stress about Z axis for the MLG car | 102 |
| D.21 | Maximum combined stress about Z axis for the NLG car | 102 |
Nomenclature

Greek symbols

\( \epsilon \)  
Strain

\( \phi D \)  
Bearing outer diameter

\( \phi d \)  
Bearing inner diameter

\( \phi \)  
Diameter

\( \rho \)  
Density

\( \sigma \)  
Stress

\( \sigma_b \)  
Bearing stress

\( \sigma_c \)  
Compressive strength

\( \sigma_f \)  
Fatigue strength

\( \sigma_m \)  
Bending moment stress

\( \sigma_X \)  
Maximum stress at the edge of the plate

\( \sigma_y \)  
Yield strength

\( \sigma_{bolt} \)  
Bolt stress

\( \sigma_{hole} \)  
Stress at the edge of the hole

\( \sigma_{proof} \)  
Proof load of a bolt

\( \tau \)  
Shear stress

Latin Symbols

\([K]\)  
Stiffness matrix

\( \{ R \} \)  
Force vector

\( \{ u \} \)  
Node values vector

\( A_t \)  
Tensile-stress area

\( B \)  
Bulk modulus

\( C_0 \)  
Static load rating
$C_r$ Dynamic load rating

$D$ Plate height

$d_n$ Distance to point $n$

$E$ Young's modulus

$F$ Flexural modulus

$F_i$ Bolt preload

$F_{in}$ Tensile load acting on bolt $n$ due to bending moments

$G$ Shear modulus

$h$ Notch depth

$HV$ Vickers hardness

$I_{ii}$ Second moment of area about axis $i - i$

$J_{IC}$ Fracture Toughness

$k$ Stiffness constant of a joint

$K_t$ Static stress concentration factor

$L_{10}$ Basic bearing life

$M$ Bending moment

$N_b$ Number of bolts

$P$ Tensile load acting on a bolt

$Q_i$ First moment of a section with respect to the neutral axis $i$

$R$ Load acting on a component

$r$ Notch radius

$R_1$ Left load transmitted by the landing gear

$R_2$ Right load transmitted by the landing gear

$R_{allow}$ Allowable load on a component

$R_{fail}$ Failure load for a component

$S_i$ elastic section modulus about axis $i$

$SF$ Safety factor

$T$ Bearing height
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$t$</td>
<td>Thickness</td>
</tr>
<tr>
<td>$T_o$</td>
<td>Torque</td>
</tr>
<tr>
<td>$V$</td>
<td>Shear load</td>
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</tbody>
</table>
## List of Acronyms

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>3D</td>
<td>Three-dimensional.</td>
</tr>
<tr>
<td>6S</td>
<td>Sort, Set in order, Shine, Standardize, Sustain, Safety.</td>
</tr>
<tr>
<td>AOG</td>
<td>Aircraft On Ground.</td>
</tr>
<tr>
<td>AQAP</td>
<td>Allied Quality Assurance Publications.</td>
</tr>
<tr>
<td>BMD</td>
<td>Bending Moment Diagram.</td>
</tr>
<tr>
<td>CAD</td>
<td>Computer-Aided Design.</td>
</tr>
<tr>
<td>CAE</td>
<td>Computer-Aided Engineering.</td>
</tr>
<tr>
<td>CAM</td>
<td>Computer-Aided Manufacturing.</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics.</td>
</tr>
<tr>
<td>d.o.f.</td>
<td>Degrees of freedom.</td>
</tr>
<tr>
<td>E170</td>
<td>EMBRAER E170 jet.</td>
</tr>
<tr>
<td>E190</td>
<td>EMBRAER E190 jet.</td>
</tr>
<tr>
<td>EASA</td>
<td>European Aviation Safety Agency.</td>
</tr>
<tr>
<td>ELEB</td>
<td>EMBRAER Liebherr Equipamentos do Brasil S.A..</td>
</tr>
<tr>
<td>EMBRAER</td>
<td>Empresa Brasileira de Aeronáutica S.A.</td>
</tr>
<tr>
<td>ERJ145</td>
<td>EMBRAER ERJ145 jet.</td>
</tr>
<tr>
<td>FAR</td>
<td>Federal Aviation Regulations.</td>
</tr>
<tr>
<td>FBD</td>
<td>Free Body Diagram.</td>
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<tr>
<td>FE</td>
<td>Finite Element.</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite Element Analysis.</td>
</tr>
<tr>
<td>ISO</td>
<td>International Organization for Standardization.</td>
</tr>
<tr>
<td>IST</td>
<td>Instituto Superior Técnico.</td>
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<tr>
<td>JIT</td>
<td>Just in Time.</td>
</tr>
<tr>
<td>MLG</td>
<td>Main Landing Gear.</td>
</tr>
<tr>
<td>Acronym</td>
<td>Description</td>
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<tr>
<td>---------</td>
<td>-------------</td>
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<tr>
<td>MRO</td>
<td>Maintenance, Repair and Overhaul.</td>
</tr>
<tr>
<td>NLG</td>
<td>Nose Landing Gear.</td>
</tr>
<tr>
<td>NVA</td>
<td>Non-Value Added time.</td>
</tr>
<tr>
<td>OEE</td>
<td>Overall Equipment Effectiveness.</td>
</tr>
<tr>
<td>OGMA</td>
<td>Oficinas Gerais de Material Aeronáutico - Indústria Aeronáutica de Portugal, S.A..</td>
</tr>
<tr>
<td>SA</td>
<td>Shock Absorber.</td>
</tr>
<tr>
<td>SFD</td>
<td>Shear Force Diagram.</td>
</tr>
<tr>
<td>SLIC Pin</td>
<td>Self Locking Implanted Cotter pin.</td>
</tr>
<tr>
<td>SMED</td>
<td>Single Minute Exchange of Die.</td>
</tr>
<tr>
<td>UTS</td>
<td>Ultimate tensile strength.</td>
</tr>
<tr>
<td>V &amp; V</td>
<td>Verification and Validation.</td>
</tr>
<tr>
<td>WIP</td>
<td>Work in Progress.</td>
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Chapter 1

Introduction

This thesis comes as a product of the ongoing partnership between Instituto Superior Técnico (IST) and OGMA - Indústria Aeronáutica de Portugal, S.A. (OGMA). This serves both parties as OGMA makes its facilities and site available to students from IST who wish to develop their master's theses, namely in the field of Aerospace Engineering. While the former benefits from young people who aim to produce significant gains for the company, working on their thesis topic, the latter provides its prospective engineers with the possibility to come into contact with a company operating in the aeronautical world. The present work is the result of one such arrangement, which included a four month internship and continuous personal supervision.

Originally a state-owned company whose services focused mainly on military aircraft, OGMA has since become a leading Maintenance Repair and Overhaul (MRO) centre, owned by the Brazilian aeronautical company "Empresa Brasileira de Aeronáutica S.A." (EMBRAER) - 65% - and the Portuguese State - 35%. OGMA's main business areas are MRO services, and aero-structures and components manufacturing. These two areas represent respectively 70% and 30% of the company's financial yield. Moreover, to this day, the company remains a competitor in both civil and military aircraft markets, having been awarded the Federal Aviation Regulation (FAR) 145 and European Aviation Safety Agency (EASA) 145 Repair Station, Allied Quality Assurance Publications (AQAP) 2110 and International Organization for Standardization (ISO) 9001-2008 Quality Management certificates. In addition it is also an authorized maintenance centre for such manufacturers as Lockheed Martin, EMBRAER and Rolls-Royce, among others. Its services are available in North and South America, the Pacific regions of Asia, Europe and Africa.

OGMA is the largest Portuguese company in its sector and also the main employer in the municipality of Vila Franca de Xira, in Lisbon's outskirts. The company enjoyed a 2016 revenue of 195,4 million Euros. Employing as many as 1734 workers from 12 different countries, its area extends over 440.000 m², with 11 maintenance hangars and a 3 kilometre long landing strip. An aerial view of its grounds can be seen in Figure 1.1.

1.1 Motivation

The motivation for the work developed at OGMA is related to its growing MRO business, more specifically with landing gear repair and testing. The landing gear is the undercarriage of an aircraft and plays an essential role in both take-off and landing operations. Being the point of connection between the aircraft's frame (fuselage, wings, engines, vertical and horizontal empennage) and the ground, it must
be conceived in such a way that it is able to withstand the impact force of landing without any damage.

Landing gears are subjected to enormous loads during their lifetime. They are also components prone to corrosion or impact damage, depending on how harsh the environmental conditions are. This means that periodical maintenance and inspection are necessary to ensure structural and functional integrity.

An aircraft landing gear ship-set consists of a nose gear assembly and two to four main gear assemblies, depending on the aircraft type. The main components of each assembly consist of the main fitting, the sliding tube, drag braces, struts which define the gear’s different positions, and various hydraulic actuation mechanisms that serve to lower and retract the different gears. Figure 1.2 shows a schematic view.

"Landing gear overhaul intervals are determined by the need to inspect, and if required, treat for corrosion" [3]. Overhaul intervals for landing gears are generally a function of either calendar or the gear’s number of flight cycles. For most models these values are between 10 to 12 years and 18,000 to 20,000 flight cycles, respectively [3]. When determining the timing for an overhaul, the most limiting
performance intervals determines the gear’s next intervention. For example, a landing gear with overhaul intervals of 10 years and 20,000 flight cycles, which completes 2,500 flight cycles per year, will be submitted for overhaul eight years after its last overhaul or entry into service. In this same example a landing gear operating below 2,000 flight cycles per year will have its overhaul calendar limited to 10 years.

OGMA’s landing gear overhaul and repair services are located in the Components’ Building, circled in blue in Figure 1.1. It is divided into shops, each of them dedicated to a specific type of component, namely avionics, electrical components, accessories, propellers and landing gears. The latter is where disassembly and subsequent assembly occur, as well as hydraulic test bench procedures. This last activity is mandatory and described in each aircraft’s maintenance manuals - [2], [4], [5], [6] and [7]. It is also the main focus of the present thesis.

While a few years back OGMA only performed landing gear MRO services for EMBRAER ERJ145 jets, it has since widen its portfolio to include EMBRAER E170 and E190 jets. This diversification has obviously meant difficulties and challenges related with the integration of larger products and more demanding testing, assembly and transportation requirements. The work requested by OGMA, resulting in this master’s thesis is part of an organized effort by the company to consolidate the existing guidelines and tackle these challenges, in particular with respect to the connection between the hydraulic test bench and the landing gears.

1.2 Topic Overview

The three different types of aircraft which have their landing gears subjected to testing on the hydraulic test bench are the ERJ145, the E170 and the E190. Their ship-sets differ in size, dimensions and procedures. The ERJ145 Main Landing Gear (MLG) assembly is comprised of three different main elements: Shock Absorber (SA), Main, and Secondary Side Struts. However, only the first component requires testing on the bench, hence will be the only one considered here.

For the E170 and E190 the ship-sets include both the MLG and the Nose Landing Gear (NLG) assemblies. Their dimensions are also considerably larger than those of the SA. These size and weight disparities make it difficult difficult to ensure that the set-up of these assemblies is done quickly and efficiently while maintaining the same text fixture. While the set-up of the SA of the ERJ145’s MLG is registered to be regularly accomplished in under ten minutes, when it comes to setting up the hydraulic tests for the remaining assemblies the recorded durations are far greater.

When conducting MRO to a landing gear ship-set there is a well defined sequence through which the components are put through before the landing gear can be dispatched and returned to the client. Once a gear enters the components building at OGMA its maintenance process begins.

Initially the landing gear is disassembled in all its constituent parts and the paint is removed. Afterwards there is the process of cadmium removal. This metal coats all the gear’s exposed surfaces and acts as a barrier against damages caused by environmental factors, namely corrosion, one of the greatest enemies of parts and assemblies whose working conditions are harsh and unfavourable.
Following suit, the existing dimensions are checked and the parts which do not exhibit the established clearances and fits must be replaced or repaired, when such action is possible. After this identification and renewal of the relevant components of the landing gear, these undergo an array of non-destructive testing procedures, including liquid and particles testing. Afterwards any corrosion present must be removed. The previous two steps are then performed again so as to assess the integrity and compliance of the intervened components.

Once all of this is done the relevant superficial treatments are reapplied to the individual elements of the assembly. The landing gear is then put back together and afterwards it is tested. If during the set of tests the landing gear is found to be in abnormal conditions then the root cause must be identified and dealt with. Once the assembly complies with all the pre-determined parameters it is packaged and shipped.

All these different steps are explicit in each assembly’s descriptive or repair manual. As an example, the repair manual for the MLG of the E190 has the following sections (excluding Introduction and the Illustrated Parts List [4]):

1. Description and Operation;
2. Testing and Fault Isolation;
3. Disassembly;
4. Cleaning;
5. Inspection/Check;
6. Repair;
7. Assembly;
8. Fits and Clearances;
9. Special Tools, Fixtures, Equipments;
10. Special Procedures;
11. Storage.

It becomes now clear that the process of maintaining and repairing a landing gear assembly is a lengthy one for which companies like OGMA are held accountable at every step of the process.

1.3 Objectives

The objectives for this master’s thesis are twofold:

1. To apply and implement Single Minute Exchange of Die (SMED) to the KRATOS hydraulic test bench;

2. To develop an universal connection fixture which may incorporate all landing gears in OGMA’s MRO portfolio.

It must be noted that, as seen further in the document, the second main objective can be considered a subcase of the first. One of the stages of SMED includes the streamlining of all internal and external operations of a given process; of these, the set-up change presented itself as an activity which the host organisation wanted to be dealt with. Therefore, this topic is one main subject of the thesis as well as one main goal to be achieved by the end of internship.

The objectives here laid out are related to the testing part of the MRO procedure for a landing gear assembly. Therefore the existing repair manuals and registry of past testing procedures will be analysed.
for relevant information regarding maximum and service loads, their variation during testing and the movements of the landing gears, such as rotation of the wheel axle about the sliding tube’s centreline in an NLG.

A critical evaluation and assessment of the current solutions found at OGMA was done in order to determine the optimal course of action and subsequent implementation. All this considering the overall needs of the company and the main desired outputs expected of the work developed. Ultimately, this aims at reducing, as much as possible, the time it takes to remove a gear under testing and setting up a new one, ultimately reducing the overall time needed for a landing gear MRO.

1.4 Thesis Outline

This work is divided in five chapters, each describing a core aspect of the work developed at OGMA. The first chapter contains the introduction and current situation of the topic at hand, at OGMA. Its main objectives and intended results are also described and outlined.

Chapter two delves into the lean approach systems, to which the SMED tool belongs to, as well as the main principles behind it. The origin of SMED is also explored along with its different stages and implementation suggestions and techniques. An overview of mechanical engineering design is included as well. Related to this insight into finite element analysis (FEA) is also provided since this was a core tool used during the design of the universal set-up solution.

Chapter three presents and analyses the current situation at OGMA with regard to landing gear testing and the changeover times at the hydraulic test bench. The observed changeover times are shown as well as the discernible activities during the process. The different stages of SMED implementation are addressed, from the separation of activities all the way to the streamlining of operations.

Chapter four is entirely dedicated to the development of the universal set-up. It is divided between the top and bottom fixtures which compose the entire connection solution. Here the design process is approached at every step. This chapter however will not deal with the iterative component of this work except in cases where it is paramount to grasp the evolution of the developed concept, such as the evolution of the slider parts, to be addressed further ahead.

Chapter five presents the simulated resulting changeover times and the overall expected time gains. The final conclusions are also presented in this chapter, as well as possibilities for future work and improvements.
Chapter 2

Background

This chapter addresses the theoretical background behind lean approach systems, the SMED tool, fundamentals of mechanical design and an insight into Finite Element Analysis (FEA). In the end of the chapter the current situation at OGMA will be presented. This includes both the used tools and an initial insight into the duration of the set-up processes at the hydraulic test bench.

2.1 Lean Approach

The concept of management which OGMA has implemented in its different service areas is based upon the lean approach. This is an organizational strategy which aims to reduce and eliminate all kinds of waste produced during the company’s activity, through a continuous improvement mentality [8]. When this mindset is present at all levels of the company’s hierarchy, waste can be reduced at every stage of the company’s organisational structure, something that is difficult to accomplish when only macro viewpoints are considered. Lean systems are an ongoing effort to better products, services or processes which require “incremental” improvement over time, in order to increase efficiency and quality.

The lean approach originates from the Toyota Production System which is often referred to as Just In Time (JIT) Production. The Toyota Company became successful after World War 2 when it adopted a number of American production and quality techniques. However, unlike the American automotive industry, Toyota encouraged employees to be a part of the production process, having discovered that workers could contribute with much more than just muscle power [9]. The company introduced quality circles, groups of workers who meet to discuss workplace improvement and then make presentations to management with respect to the quality of production. Also, unlike the American manufacturers, Toyota developed manufacturing in smaller batches requiring a set of processes that reduced set-up and changeover times [9]. All these factors gave Toyota a competitive edge and, naturally, Toyota’s developments were subsequently adopted by other Japanese manufacturers albeit less successfully. This was because superficial measures were being adopted, rather than a systematic approach which included the underlying principles behind Toyota’s success. In the 1980s, American companies began to adopt some of the processes developed by Toyota. The term “lean manufacturing” was coined by James Womack [8], who helped diffuse the lean principles in United States companies and afterwards around the world.

Lean approaches employ different methodologies aiming to eliminate all factors creating wastes in time, effort or money, which add no value to the processes at hand. This is achievable by continuously analysing different existing processes and then redefining or scratching any steps that do not create any
value from a customer standpoint. This is one of lean’s core principles. According to [10], the sequence of core principles of the lean production system is:

1. Defining what is value from the customer’s standpoint;
2. Identifying each step in a selected business process and eliminating any step which does not create value;
3. Making the value-creating steps occur in tight sequence so as to maximize efficiency;
4. Repeating the first three steps on a continuous basis and across all different process areas of the organisation until all waste has been eliminated.

These lean principles ensure that the processes involved with bringing a product to market remain cost effective from beginning to end.

Lean production is the application of lean principles to production processes so as to eliminate waste being generated in a manufacturing process. This may include wastes created through unevenness in work loads, overburden and any work that does not add value. From the point of view of the final customer, “value” is any process or action that he would be willing to pay for. In essence, lean methodologies focus on making obvious whatever appendes value by decreasing everything else.

When Toyota developed its production system, three main enemies were identified which made their way into the foundations of what was to become the lean manufacturing system. These are, according to Womack et al. ([8]):

1. Muda - non-value adding work;
2. Mura - unevenness;

### 2.1.1 Muda

Muda is any activity or process that does not add value; a physical waste of time, resources and money which occurs when a given operation is being conducted. These wastes can be separated in seven different categories, each of them with its own definition. All of these different types of Muda lead to waiting times and longer lead times in processes. These categories can be organized in a mnemonic acronym - DOWNTIME. Ohno ([11]) lists the main wastes of Muda and Panneman ([12]) defines how to best combat them:

- **Defects** - products or services which are moved to the next process but do not meet customer specifications. These usually lead to rework so as to ensure that the customer receives the correct order. This has a double negative impact on the lead time of the overall process due to the existence of an internal feedback loop with the extra time needed to correct the problem. The two most effective tools to prevent the occurrence of defects are Poka Yoke and Standard Work.
  
  The first is a method which aims to eliminate the possibilities of outputting products with defects,
through either warning systems or control systems. Standard work is the description of the correct way to perform a certain task. It also helps prevent injuries and wastes in adjustments and the dependency on experience.

- **Overproduction** - producing product more than what has been ordered. This waste is related to the consumption of resources to generate products or services which will not be passed onto the next process stage. More often than not the lack of communication between consecutive process steps is at the origin of over-productive activity upstream. This will generate supplies and resources which are in excess and that the downstream process has no use for at the given moment. It can be combated using SMED and Kanban. The first is the main tool used in the present text and is explored in more detail in section 2.2. Kanban is a self-managing production system in which the material moves downstream and the information about what to produce moves upstream. Every workstation receives its own signal from its downstream workstation, which informs the upstream station on what and how much to produce.

- **Waiting** - time wasted while a machine doesn’t finish a task, a product doesn’t arrive, or any other cause of time waste. When a product or batch of material is waiting to be processed or shipped it is wasting time, which does not add any value to the customer’s final order. Time studies, takt time and line balancing are the tools used to combat waiting wastes. Time studies define the process in different types of time - process times, waiting times and lead times. The first is the sum of cycle times where value is added to the product, the second is its contrary and the third is the time between two consecutive starts to a process. Takt time is the cadence under which the customer actually demands the product and is the pace of production which the company should strive to meet. Faster production means waiting and inventory build-up and slower production keeps the customer waiting on his order. Finally line balancing ensures that all sequential workstations have equal cycle times.

- **Non-used Talent** - this waste is present whenever someone’s knowledge is not fully taken advantage of. Having employees with unused skills often keeps an organisation from maximizing its efficiency. Having a high skilled worker doing easy work or directing them towards short term correction activities rather than investing into long term improvements are just two different types of non-used talent inside an organisation. Ignoring the input of specialists also generates waste, especially if this leads to an investment being poorly aimed and ultimately useless. Training is one of the main tools used to combat this waste, mainly how to make use of tools like 6S, standard work, time boards and kaizen and prevent investments and changes to be ineffective. The practical problem solving technique of the “five whys” is also a powerful method to teach employees, helping them get to the root cause of a problem, rather than only a symptomatic treatment of an existing issue.

- **Transport** - the movement of an item between operations and different locations. During this movement the item remains unaltered between workstations. This only increases the lead time and man hours of the process without adding any value to the final customer. In order to reduce transport,
one must observe and trace the movement of a product along a process. This can be done by building a spaghetti diagram with the purpose of changing the layout of the plant so that subsequent workstations are placed close to one another. Another possibility is to implement work cells which are like small production lines in a U-shape where the input and output of a workstation are on the same side of the cell. Tool transport can be reduced by using 6S, equipping every workstation with standard locations to keep all tools - shadow boards are a common tool to achieve this.

- **Inventory** - work in progress (WIP) and stocks of finished goods and raw materials that are held by the company. This includes any products which may be waiting at a workstation without being worked on. This waste is related to the already mentioned overproduction. The more items are left in inventory, the longer an item has to wait before being worked on. Not only does this waste cause an increase in lead times but it also carries with it augmented costs related to material, depreciation, physical space needed, insurance and possible redundancies or damage to products in inventory. Work cells and Kanban reduce inventory. Additionally, One-Piece-Flow is another useful tool, reducing batches to only one item and cutting down the amount of WIP.

- **Motion** - the physical movement of a person or machine without actually working on the product. It differs from transport, which relates to the item or service being produced. If a worker wastes 5 minutes walking between different stations to collect all necessary equipment or tools then motion waste is in place. If however the worker does this while carrying the product then transport is also in place. While motion must be reduced it is advisable not to fully eliminate it since it may have unwanted consequences on a worker’s health in the long run. Motion can be reduced using standard work, 6S and spaghetti diagram methods, similarly to transport and other wastes.

- **Excess (over-)processing** - conducting operations on the product which are beyond what the customer requires. The inclusion of features which will be irrelevant to the final customer or the addition of packaging materials for internal transport which are subsequently removed and not usable in any other way are examples of excess processing. In this type of waste the operations of defect correction and repair may also be included since it is work which does not add anything of value, it merely assures the basic requirements for the product. Over-processing can be cut down to a minimum (ideally zero) by building a process map. This is a flow chart which details and describes all different actions during production of a product. They are then separated into six categories: process step, delay, inventory, decision, measurement and transport. This helps find and eliminate activities in the process map which do not add value to the final output of the overall process.

Two additional wastes are also frequently considered to be relevant in addition to the above list of wastes. These are:

- **Resources** - keeping unused machinery and lighting running when they are idle and not working on the product;

- **By-Products** - not taking advantage of whatever by-products a process produces.
Although waste must be eliminated, according to a lean production mindset, it is often not enough to simply eliminate a given wasteful activity. Muda comprises the waste part of a process which lean methods aim to eliminate. However, it is common for Muda to have a deeper and more complex cause behind it. It is insufficient to tackle only the manifestation of the problem and one must go deeper and combine both Muda, Muri and Mura in order to fully eliminate waste.

2.1.2 Mura

Mura, the second type of waste, is often at the origin of many of the different types of Muda - in short, Mura drives Muda. It can be found in customer demand, process times per product or the variation of cycle time for different operators. According to Hopp ([13]) there are two important laws concerning variance of production:

1. Variability will always degrade performance of a production system;
2. Variability in a system will be buffered by some combination of inventories, capacity or time.

In order to reduce the effect of these two buffering factors one must either smooth out customer demand or reduce variation in existing processes, respectively [12]. Failing to smooth demand over-stresses processes and the workforce, generating inventory and other wastes.

To reduce the variation in customer demand there must be a cooperative effort inside a supply chain. When information is not provided about current customer demand, inventory levels or orders placed for raw material, a fluctuation downstream creates massive variance upstream. This is due to the emergence of the bullwhip effect. This phenomenon leads to the appearance of ever increasing swings in inventory at a given point in the chain, when moving further upstream. There is a tendency for each link in a chain to order additional products/parts when an order can’t be met, due to an unexpected shift in customer demand, which is augmented when there is an accumulation of work to be completed or orders to be fulfilled. The longer the total lead time and delivery times between subsequent links, the higher the bullwhip effect. A large number of links in a chain also leads to a higher bullwhip effect.

In order to reduce the effect of shifting end consumer demands it is advisable to:

1. Reduce the number of links in a supply chain;
2. Reduce delivery times between links;
3. Create transparency between links in the supply chain when it comes to order portfolios.

For the case of process variance there are different methods to reduce it. There are four main methods to eliminate this form of variance:

1. Modular product design at design level;
2. Heijunka - production levelling in production planning;
3. Building Flow at production-level;
4. Standard work and 6S on workstation level.

Considering production design, the variation between products can be minimized by using modular designs. Using standard modules will reduce the number of possible material routings in the factory and a number of inventory items.

Heijunka serves to reduce the impact of customer variance. With Heijunka a fixed interval is chosen in which all product types can be produced. Shorter intervals mean products produced more often and shorter lead time for each product. Lead time reduction also causes the uncertainty in customer demand to be reduced. It must be noted that the variation in product mix does not have a high impact on a production process if the processing times are balanced for the different products.

The movement of products through the plant should also be optimized. Ideally, products flow through the plant, which means there is no waiting when they move between the necessary workstations. When the processing time of workstation 2 is larger than workstation 1, either every product coming from workstation 1 has to wait before station 2 can work on it, or station 1 has to wait for free capacity at workstation 2. This means that all operator handling should be optimized to minimize production variation at workstation level. Standard procedures and lay-out prevent differing work cycles for different operators conducting the same task, as well as time wasted searching for tools or materials.

2.1.3 Muri

Finally Muri is what happens when unnecessary stress is given to employees and processes. The quality of the final outputs of a process suffers whenever a machine or a worker is overburden with work. Muri is brought about by Mura alongside other failures in the overall system such as lack of training, non-existent work standard for a given process, incorrect tools being provided, and ill thought out performance indicators. When tackling Muri two different environments must be considered: the machine environment and the human environment.

In the first one Muri depends on the performance of the machine, as measured by its Overall Equipment Effectiveness (OEE), best described by King ([14]). This metric aims to prevent a machine breakdown and includes measures for availability, performance and quality. To improve OEE of a machine and prevent overburden, maintenance must be carried out. It can be of two different types - preventative or autonomous. In preventative maintenance a machine is kept in good operating conditions through inspection, detection and prevention of failures. The reasoning behind this type of maintenance is that it is cheaper to prevent malfunctions than to wait for the machine to breakdown and consequently need repair. Autonomous maintenance requires operators to carry out key maintenance tasks on a machine. This type of maintenance is based on the concept that operators who work with the machines daily are the first ones to detect irregularities. In order to assess the financial advantage of implementing preventative and autonomous maintenance, the costs of breakdowns (including cost of repair and lost production) can be compared with costs of inspections for the same period.

In the human environment a consequence of muri is absenteeism. This is the result of overworked employees who become unable to work for long periods of time. Tools that help reduce people related
muri are the team board, 6S, Standard work and using Jidoka principles. The first one includes metrics like the amount of over hours made, stress levels of individuals, or even the mood of team members in order to make these topics part of the team's daily communication. 6S describes the safest and most efficient lay-out for a workstation, which leads to an organized workstation and therefore reduces stress. Standard work reduces the possibility of injuries. Finally Jidoka contemplates the possibility of stopping a production line whenever a problem is encountered. All employees are notified about the location of the problem and focus is then placed on solving it. The jidoka principles are described in four steps:

1. Find a deviation;
2. Stop production;
3. Fix the problem;
4. Analyse the root cause of the problem and prevent it from happening again.

Ultimately the link between Muda and Muri is reciprocal: Muri causes Muda but Muda also brings about Muri when too much waste is removed from the process without treating the underlying cause. Mura, however, is more often than not the main cause for the existence of Muri in a process. This relation is addressed by Hopp ([13]).

### 2.2 SMED

As mentioned in subsection 2.1.1, SMED is a tool which aims at reducing waste in processes, mainly related to overproduction waste. Additionally it helps reduce lead times, cut down on non value added times (NVA) and overall cycle times. In line with what was described in subsection 2.4.2 SMED presented itself as a tool with great potential gains in changeover times at the hydraulic test bench. This section describes this tool's emergence, its fundamentals and different stages.

SMED means Single Minute Exchange of Die and its goal is to perform equipment changeovers in as little time as possible. While its name indicates changeover times of just seconds, SMED’s goal is to reduce this changeover period to single digit minutes, i.e. under ten minutes, [15]; not that seconds-long changeover periods are ever undesirable. The essence of this tool and its core principle is to conduct as much work as possible while the machine in question is running and as little as possible while the machine has been stopped, reducing non-productive time to a minimum. This first kind of work is referred to as external activities and the second one as internal activities. Additional simplification and streamlining of activities is a subsequently needed to fully implement this tool.

According to Shingo ([15]), the general benefits of applying SMED to a given process are:

- Lower manufacturing costs due to the reduced machine downtime;
- The possibility of producing smaller lot sizes;
- An improved responsiveness to customer demand driven by more flexible scheduling;
• Prevention of inventory build-up;
• Smoother start-ups by either cutting completely or reducing the number of trial runs, improving consistency and quality.

In order to determine which process should be the target of SMED, five different characteristics are sought out. These are duration, variation, opportunities, familiarity and constraint. The first one concerns how long does the current changeover operation takes; these values should leave room for improvement but not be too long so as to become overwhelming in scope. Secondly the process must present large variation in changeover times and be dependent on a multitude of external factors. There must be many different opportunities to perform the changeover so as to analyse both the current situation and the effect proposed changes may have on the process. Employees must also be familiar with the equipment in question for SMED to be efficient. Finally the equipment and the changeover in question must obviously be a constraint/bottleneck whose improvement will bring about immediate gains in lead time reduction and productivity.

SMED was originally developed in the 1950s by Shingeo Shingo [15]. His first step was at the Mazda plant in Hiroshima where he came up with the concept of distinguishing between internal and external operations. In his work he attempted (and ultimately succeeded) to reduce bottlenecks generated by large body-moulding presses. Shingo was able to ultimately remove the bottlenecks these presses presented merely by listing a sequence of check-up and preparation activities to be run while the presses were working. By simply applying this separation he was able to raise efficiency by about 50%.

A second moment relevant to the creation of SMED occurred at the Mitsubishi Heavy Industries shipyard in Hiroshima. During his work Shingo noticed that support procedures like markings for centring and dimensioning of engine beds were occurring at the planer table. His proposal was to simply conduct these marking operations on an additional planer table to perform set-up operations. This meant the possibility to switch between tables whenever a lot change was in order. Ultimately this conversion of an internal operation into an external one translated itself in a 40% increase in productivity at the shipyard.

Later on, Shingo was asked by the Toyota Motor’s main plant to reduce changeover times of a 1000 ton press, from around four hours to one and a half hours, so as to surpass the two hours changeover times of European companies such as Volkswagen. This was accomplished by employing his previously developed concepts on explicitly mapping and differentiating between internal and external operations, and six months worth of analysing and classifying work. However, once this was concluded it was not long before a request was put in to further reduce the changeover time to an apparently impossible 3 minutes. Here the concept applied at the Mitsubishi shipyard came into play - to convert internal set-up operations into external ones. With this concept he was able to further reduce the idle time of the press and ultimately reach the 3 minute goal for die change. This last concept is, according to Shingo himself, a benchmark method to effectively applying SMED to whichever changeover operations are considered.

Ultimately the development of SMED took Shingo about nineteen years to be completed. It was the succession of encounters with different projects which gave its author the insight and knowledge to develop “a scientific approach to set-up time reduction” [15]. This specific characterization of his
method is what distinguished Shingo from other engineers. While it is reported that Toyota had already implemented single digit minute changeovers, the fact is that Shingo’s work is applicable to any kind of process, rather than just the changing of press dies done previously at Toyota. This is a by-product of Shingo’s reported need to establish theoretical reasoning behind practical solutions [15].

The usefulness of this tool did not go unnoticed and Taiichi Ohno, one of the pioneers of the Toyota Production System, published an article a few years after Shingo’s die change reduction, highlighting the value and possible gains made by applying SMED [15]. Eventually SMED’s contribution to a company’s efficiency made its way out of Japan and this tool became a standard in many production companies processes with reported time gains which sometimes reach around 95% and rarely under 50%. Nowadays SMED is applied to many different processes which range from production environments to maintenance tasks, of which the current work is an example.

2.2.1 The Stages of SMED

The application of SMED to a process has different separate stages which are organized according to a logical sequence of actions to be taken. These will gradually produce changes on the paradigm of whichever process is being analysed at a given moment. While the SMED process is divided in stages, processes are divided in steps and types of activities. Shingo ([15]) distinguishes between four types of steps in a set-up process:

- Preparation, after-process adjustment, checking of materials/tools - ensuring that all parts and tools are in their supposed locations and in proper conditions. Removing and returning these items to storage and cleaning of machinery is also included;
- Mounting and removing blades, tools, parts - removal of tools after completion of a processing lot and attachment of tools for the following set;
- Measurements, settings, calibrations - centring, dimensioning and other measurements of temperature and pressure;
- Trial runs and adjustments - small adjustments which are made after a test part is produced; the more thorough the calibration step, the less time wasted on adjustments.

These steps and their sequence are applicable mainly to productive processes. However, it is possible to still match almost every different operation in a set-up process to one of the listed types. A process may even have some kinds of steps and dispense others. Regardless of what types of activities exist during a specific changeover, all set-up improvement studies follow the same four stages.

Firstly a process has all its different set-up operations bundled together without any kind of separation or distinction between them. This is the preliminary stage and any separation between internal and external set-up activities is non-existent. In order to begin the process of implementing SMED on a given process, Shingo recommends different methods and approaches, depending on the frequency of said process, the conditions of the shop floor being analysed and also the expertise of the team responsible [15]:
Continuous production analysis using a stopwatch – the best approach but not trivial and requires a great deal of skill to correctly use it, mainly to quickly distinguish when the previous operation has ended and a new one has begun;

Work Sampling Study – this method has the disadvantage of only being applicable when a great deal of repetition is present. It intends to quantify the idle and active time of an employee or a machine;

Interviewing Workers – useful in every instance since these are the people in direct contact with the operations at hand and their insight is often very practical and valuable towards the end goal;

Videotaping – this tool is extremely powerful and takes the most out of the previous method when the recording of the operation is shown immediately to the workforce. It also serves as a record against which future developments and advancements will be compared to.

Once these tools are used the first stage of SMED begins: separating internal and external set-up activities. This deals with the classification and segregation of every activity which must be conducted, or not, when the machinery used for the process is stopped. At this stage there must be an effort to assess whether an activity is actually an internal activity or an external activity being conducted at an inappropriate timing. Simply by doing this it was observed that internal set-up times could be usually reduced to 50% – 70% of their original time.

The classification step evolves into the next stage of SMED: converting external operations into internal operations. The present section already described how this was the game changer which allowed for the massive changeover time reduction at the Toyota Plant. Shingo himself reflected on the possibility of developing SMED a dozen years earlier, if he had grasped right away the importance of converting internal activities to external ones [15]. Hence the relevance of this stage must not be understated. To achieve this there are two main steps:

1. Internal operations must be re-examined so as to find and reclassify incorrectly defined activities;
2. When this is done, solutions must be found to aid and facilitate this conversion.

This must be done by constantly re-examining the function of a given operation. One must also strive not to be constrained or bound by old habits and “this is how it is done” philosophies. Here creativity is an asset, being able to look at the sequence of different operations and finding ways in which to turn the maximum number of them into external activities, without the need of having them precede one another. It is also important to be able to breakdown an activity into its steps and move the maximum number of them into an external environment. An example is to move a preheating activity to before the machinery is stopped; while the element in question is only mounted internally, its preparatory operations may occur externally.

Shingo ([15]) presents the following main types of techniques which can be used to convert types of activities:

• Advance preparation;
Table 2.1: Example of parallel operations (adapted from [15]).

<table>
<thead>
<tr>
<th>Task</th>
<th>Time [sec]</th>
<th>Worker 1</th>
<th>Worker 2</th>
<th>Buzzer</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>15</td>
<td>Lower ram</td>
<td>Prepare removal</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>20</td>
<td>Remove front bolts</td>
<td>Remove rear bolts</td>
<td>Yes</td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>Raise ram</td>
<td>Turn press switch off</td>
<td>Yes</td>
</tr>
<tr>
<td>4</td>
<td>20</td>
<td>Remove bolster setting pins</td>
<td>Prepare removal</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>60</td>
<td>Move bolsters</td>
<td>Remove bolts lower die</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>Attach cable</td>
<td>Attach cable to die</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>20</td>
<td>Hoist</td>
<td>Move die for mounting</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>30</td>
<td>Position die</td>
<td>Position die</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>20</td>
<td>Tighten front bolts</td>
<td>Tighten rear bolts</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>50</td>
<td>Move bolster</td>
<td></td>
<td>Yes</td>
</tr>
<tr>
<td>11</td>
<td>30</td>
<td>Set pins</td>
<td>Move crane</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>30</td>
<td>Set ram at bottom</td>
<td>Adjust ram stroke</td>
<td>Yes</td>
</tr>
<tr>
<td>13</td>
<td>50</td>
<td>Tighten front bolts</td>
<td>Prepare tightening</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>20</td>
<td>Raise ram</td>
<td>Tighten rear bolts</td>
<td>Yes</td>
</tr>
<tr>
<td>15</td>
<td>15</td>
<td>Test die action</td>
<td>Check switches and meters</td>
<td>Yes</td>
</tr>
<tr>
<td>16</td>
<td>40</td>
<td>Insert material</td>
<td>Check for safety and quality</td>
<td></td>
</tr>
<tr>
<td>Total Time:</td>
<td>470</td>
<td>Problems to watch for: (...)</td>
<td>Actions to be confirmed: (...)</td>
<td></td>
</tr>
</tbody>
</table>

- Jigs to perform alignment and adjustments externally;
- Modularization of equipment to eliminate internal reconfiguring activities;
- Modification of existing tools and machinery.

Once this stage is correctly completed, a further decrease in total internal set-up time must be already present. Sometimes single digit times can be reached immediately after conducting this stage. Even if this is the case, it is still advisable to implement the last step so as to further reduce set-up times and increase efficiency and productivity of the intended process.

The third and final stage is to streamline all aspects of the set-up. After separating and converting all different activities there must be an effort to streamline and further reduce the time needed for each one of the operations. In this way a detailed and in depth analysis of each operation is required. This and the previous stage are presented separately to clarify the distinct notions involved; however, they can be conducted simultaneously. Improvements should be applied to both external and internal operations. For the former type of activities improvements in transport, storage and workspace alterations may prove useful. Eliminating the need for multiple trips or presenting the equipment on time and in an organized manner are some examples which help streamline external operations.

Internal operation streamlining is the main focus of the last stage of SMED implementation, since it is the most direct way to reduce idle times of the machinery in question. One powerful way to achieve this is to implement parallel operations and assigning extra workers to assist in these operations. An example of a procedural chart with parallel activities conceived by Shingo can be seen in Table 2.1.

In addition the use of functional and quick change fastening methods also serves to improve internal set-up times greatly. In a SMED environment a solution which only requires one turn of the wrench is a much more desirable option since every second counts. Spring mechanisms and other one-motion methods are the preferred options for centring and securing parts interchangeable between different
set-ups. The elimination of adjustments through standardization, the use of centring jigs or stopping wedges, is also one way internal operations may be streamlined.

Improvements to the set-up process must be considered at every stage of this implementation, rather than just in localized moments. These can be categorized in two different ways:

- Human - achieved through preparation and organization;
- Technical - achieved through engineering.

Through experience it has been noted that human elements are faster and cheaper to improve than the technical elements of a set-up operation. Human improvements are therefore commonly called quick wins. This is why it is advisable not to focus solely on technical and mechanical improvements of a process. Figure 2.1 illustrates the example areas of opportunity when implementing SMED and how to take the most of all improvement investments.

2.3 Mechanical Design

To design is either to formulate a plan for the satisfaction of a specified need or to solve a specific problem. If the plan results in the creation of something having a physical reality, then the product must be functional, safe, reliable, competitive, usable, manufacturable, and marketable. It is a process which relies on innovation and is frequently highly iterative and obviously requires the ability to make decisions. The amount of information available when making decisions is often not ideal. One must remember to keep the possibility to adjust a previous decision as new information becomes available. Therefore, a design engineer must be at ease with as much of a decision making role as a problem solving role.

Mechanical phenomena are ideally compartmentalized in different subject areas but real world problems and their solutions do not fit so easily in separate categories. While mechanics of solids, momen-
tum transfer, heat transfer, material selection and statistical analysis can all be considered separate, a
development of a component may require a knowledge of all of them from the designer [17].

![Diagram of design process]

There is an ideal sequence of activities for a design process. However, things are seldom ideal and it is common for a designer to have to revisit previous stages before heading onto the following step in the process. This is shown in Figure 2.2. Often a designer has to reassess what was defined in previous phases and make decisions as to whether some features are maintained or scrapped, in light of new information. The large number of feedbacks and interactions between different stages highlights the importance of the decision making skill approached above. Budynas & Nisbett ([17]) state that a lack of confidence in deciding that an old route must be redefined could lead to a sense of being overwhelmed. Additionally if a designer lacks the courage to change the course of a project, either the process will be carried on until a dead end is met or the conceived solution will not fulfil the original need.

The design process begins with the identification of a need. Often enough the need is vague and little more than an uneasiness or the feeling that something ought to be different. This differs from the definition of the problem, which is the following stage. While the need can be a more general description, the problem definition must include the existing specifications for the object to be designed. These are related with minimum strength requirements, space and weight constraints and even the available materials. While requirements are related to the conditions that the product must withstand and set the goal for the designer, anything which limits the freedom of the design process is instead a constraint. These may result from the organisation which contracted the work, the available machinery and materials, or the nature of the problem itself. Whether they are requirements or constraints, characteristics which influence the overall design of an element are usually referred to as design considerations. There are many different ones and they must be given priority according to their importance to the final function of the element. Examples are [18]:

---

Figure 2.2: The different stages in design along with the different feedbacks and interactions [17].
1. Functionality; 6. Safety;
2. Strength; 7. Reliability;
3. Deflection; 8. Manufacturability;

The next stage is the synthesis step, also referred to sometimes as concept design. The quality of a proposed concept must be measured against previously defined metrics. Several schemes may be developed and assessed; those which do not "make it" can then be revised, improved or discarded. Those with final desirable metrics are then integrated into the analysis and optimization phase. This phase is interconnected with the previous one and is one of the most frequent feedback loops. It is common to apply this loop to individual components in sequence, rather than bundled together. This allows for more specific approaches to each component and better understanding of each part's function.

While initially a mathematical approach is in place, there is still the need to assess the real physical system being devised. For this there is the evaluation stage. This is the final proof of a good design and ideally includes prototype testing. This stage must address both engineering and non-engineering questions such as: reliability, maintenance, attractiveness, cost of production, possible failures, among others. This step may require the reassessment of a concept. If this happens the designer must have sufficient insight into the overall project to be able to implement local rather than general changes to the product, as advised by Budynas & Nisbett ([17]). If this is not the case then the entire process may have to begin anew; this must be prevented at all costs by being focused and critical throughout the entire project. Once again, decision making and problem solving skills are front and centre in the skill-set a mechanical designer must possess.

After evaluation there often is a final presentation. "This is a step whose importance should not be underestimated" [17]. Presenting a solution is all about selling a product. If a better product is poorly presented, it can lose to inferior products simply based on the lack of ability to convey its importance and advantages. If this stage is not correctly faced then the whole project becomes a waste of time and resources.

Nowadays, there is a wide variety of tools available to help in mechanical design problems. These range from technical information available in manuals, textbooks, standards and catalogues, to electronic and computational tools. Electronic tools are mostly related to the increased reliability of data gathered from experiments or other phenomena. Computational tools have also created a breakthrough in the design and analysis capabilities of mechanical designers when creating a solution. Computer-aided Engineering (CAE) applies to any kind of computerized application which may serve an engineer some specific purpose. These may be Computer-aided Design (CAD) software, engineering-based software or non-engineering-specific software [17].

CAD allows for the development of three-dimensional (3D) models which can later be used to create technical drawings. This type of tool also allows for exact calculations of the properties of a component, and helps in defining production methods for a developed part. Examples of engineering-based
software include finite element analysis (FEA), computational fluid dynamics (CFD) and programs for
dynamic force and motion simulations in mechanisms. Software not engineering-specific which may be
considered part of CAE, are tools for word processing, spreadsheets and mathematical solvers. Some
of these tools were used during the work preceding this thesis; Excel (spreadsheets), MATLAB (mathem-
atical solver), Siemens NX12.0 (CAD and FEA tool) and SkyCiv (section designer, beam calculator &
FEA). Some insight into the fundamentals of FEA is presented in subsection 2.3.1.

2.3.1 FEA

When faced with ideal components, a mechanical designer is usually able to make use of simplified
methods which have definitive solutions [17]. However, when those components leave the design paper
and become real parts, their behaviours and responses are seldom as simple as those portrayed in
simplified mathematical models. When faced with this situation, Budynas & Nisbett ([17]) state two
possible courses of action: experimentation or simulation.

The first method is the one which has the potential to turn out more realistic results. Human and
experimental errors play a part in the output of a given experience, be it due to poorly calibrated equip-
ment, incorrect procedures or inexperience in conducting experiments. Another big hurdle in conducting
experiments relates to the design of the experiment itself. Considering the designed part and the differ-
et load requirements it has to withstand, it is paramount that an experiment be conceived which mimics
and allows for the study of the response of the part in question. In view of the above and the costs of
experimentation (test subjects, equipment, lab access, etc...), it comes as no surprise that experimen-
tation is mainly used in situations where the most realistic results possible are desired and experiments
can be easily created and carried out. Even then, many experiments serve as a means of validating the
outputs of a computational tool.

The process of creation of a computational tool or method involves two distinct "confirmation" phases:
Verification and Validation (V & V). The first is associated with "identifying and removing errors in the
model by comparing numerical solutions to analytical or highly accurate benchmark solutions" [19].
This means that verification is mostly related with getting the underlying mathematical models right.
Validation on the other hand concerns "the process of determining the degree to which a model is an
accurate representation of the real world from the perspective of the intended uses of the model" [19].
This step is usually achieved by comparing experimental results of the intended model and assuring that
the developed model describes the occurring phenomena correctly. The present thesis will not delve into
V & V. Verification was performed for the SkyCiv tool in Appendix A because it is a relatively recent tool,
Validation was beyond the scope of the developed work. While mathematical predictions and simulated
results are presented and constitute a main component of the design process, the number of needed
experiments for the developed number of parts would be prohibitive.

Instead of using the previous considered method, engineers have been making use of FEA as a tool
to aid in the conception of parts and the prediction of their mechanical behaviours. This is a numerical
method which is commonly used in tandem with such tools as CAD and Computer-aided Manufacturing
(CAM). In recent years FEA has proven itself as one of the most accessible ways to aide in mechanical
design, with applications which may focus on static and dynamic load conditions, vibration analysis, buckling analysis, heat transfer, acoustics and even electrostatics and fluid dynamics. The present work focuses mainly on mechanical analyses since such factors as heat transfers between the environment and the conceived parts are non-determinant in the intended work conditions.

The overall objective of a FE analyst is to compute a field quantity, be it stress distributions, temperature fields or others. More specifically, the FE method is a piecewise polynomial interpolation, in which the desired field quantity is interpolated over each element as a function of its values at their respective nodes, according to Cook ([20]). Elements are the finite number of sub-domains in which a larger domain is divided into. These are connected between at the nodes, which fasten them together. Additionally, the nodes of an element ensure the compatibility between adjacent elements.

There is more complexity to the discretization of a structure than just separating it into different elements and then joining these at the nodes. This would be a weak representation of the actual domain due to the possibility of the appearance of gaps, concentrations at the nodes (such as strain in stress analyses) and the sliding of elements on one another. To prevent these problems each element is restricted in its mode of deformation; more specifically each side of an element has its deformed shape dependant on the number of degrees of freedom (d.o.f.) of the nodes attached to it [20]. This also ensures the already mentioned compatibility of neighbouring elements since these share sides and their respective nodes, and will therefore exhibit the same d.o.f. and ultimately allowable deformations.

The analysis of any problem includes 4 different stages, according to Reddy ([21]):

1. FE discretization;
2. Element equations;
3. Assembly of equations and solutions;
4. Convergence and error estimate.

While the first step may seem simple enough, it is necessary to have a clear understanding of what’s at play in the problem at hand. It is not enough to simply divide the domain in a finite number of elements without considering location, applied boundary conditions and expected results around a given region in the domain. A mesh of elements too coarse will likely misrepresent or neglect some variations in the desired field quantity which may be relevant to know. On the contrary, refining a mesh too much is also undesirable, since having more elements than those needed to accurately represent the field results in a waste of computational time and resources. This can be prevented by having a good grasp of the problem which will be analysed.

Over each element algebraic equations regarding the quantities of interest must then be developed. This is done by making use of the general governing equations for the given problem and applying them to every individual element in the mesh. These governing equations must take into account the restrictions and conditions imposed on the nodes belonging to each element. These equations also include physical properties which are derived from the material and shape of the larger structure. Afterwards these elemental equations are assembled into a matricial problem of the form
\[ [K] \{u\} = \{R\} \quad , \tag{2.1} \]

where \([K]\) is the Stiffness Matrix obtained by summing the individual stiffness of each element, \([R]\) is the Force Vector obtained in the same way as the previous one but this time summing the contributions of applied boundary conditions or gradients on the nodes, and \([u]\) the primary variable to be computed which is the value of the field quantity at the nodes [21]. These are general terms which can represent different types of quantities, depending on the nature of the problem. Structural problems, heat conduction, fluid and electrostatics are some of the areas that use this formulation in their solutions. For each area, \([K]\) (stiffness, conductivity, viscosity, etc...), \([R]\) (mechanical force, heat flux, body force, etc...) and \([u]\) (displacement, temperature, velocity, etc..) change accordingly.

The last step of the four is related to mathematical approaches used in solving the equations presented. The common ground is that there is a function dependant on the values of the field quantity. When this function is minimized it means that the values at the nodes exhibit the “correct” values for the problem formulated. This is based on the principle that a system in equilibrium will coincide with a minimum of this function. More often than not these used functions are related to the energy of the system in question. Obviously the final obtained values depend on all previous steps and the error generated during this solution process must then be estimated.

Cook ([20]) states there are three main types of error which occur when using a finite element analysis. The first is the modelling error. This error appears due to the difference between the mathematical model being solved and the actual physical problem. The more closely the mathematical model is to the physical one then the smaller this error will be. Again this demands a good grasp in what analyses may certain factors and effects be neglected. For example in elemental beam theory, which represents a beam by a straight line, it is incorrect to neglect the transverse shear deformations which occur in short beams but such an approximation is usually valid for the case of longer, slender beams.

The second type of error which emerges is the discretization error. This type of error is associated with the rounding off or truncating of numbers during computations. Since computers have a limited accuracy when solving equations the final solutions will not be 100% accurate. Usually this error is small.

Linear Static Analysis

The main type of analysis used in the present work is a Linear Static Analysis. It is used when the stresses in the structure for a given applied pressure/load are desired. Its primary variable is the
displacement at each node, caused by applied loads. For this analysis to be valid the loads must not induce significant inertia and damping effects. Cook ([20]) states that since this analysis is "static" it deals with approximate conditions in which the applied loads do not vary based on time, therefore ignoring the inertial and damping forces which would otherwise be at play.

As the rest of its name suggests, the main assumptions of a linear static analysis are that the relationship between the load applied and the response of the analysed object is linear. For this to be applicable some additional assumptions are needed. Small deformations is one of them, i.e. the characteristic length of the structure is much greater than the displacements of the structure. Additionally the material is considered to be elastic \( \sigma = E\varepsilon \), where \( \sigma \) is the stress in the structure, \( E \) is the Young’s modulus of the material and \( \varepsilon \) the computed strains) and the loads and constraints keep their original magnitude and direction.

Solving a linear static analysis involves Equation 2.1. Once the loading data, the constraints, the geometry and the material data are provided the solver computes the stiffness matrix and generates the boundary condition scheme that will allow it to solve the system. In the described case \( [K] \) is the material stiffness matrix, \( \{ R \} \) the loads applied to the model and \( \{ u \} \) the displacements which the solver attempts to find. After the solution process is concluded the information is post-processed and strains and stresses are computed from the final values.

These linear static analyses may be conducted using one-, two- or three-dimensional cases. For each case the number of dimensions at play for loads or constraints differs from the number of d.o.f. of the elements. During the course of the present work some analyses involved both 3D load scenarios and 3D elements, as well as situations with two-dimensional load cases and two-dimensional elements. The choice of dimensionality of an analysis falls back on the knowledge about the problem, on what is expected to be the model’s response, as well as the relevant parameters to be analysed.

Linear static analyses were conducted with both Siemens NX12.0 (SOL101) - the default case - and SkyCiv’s 3D structural software. When the latter is used this will be explicitly stated. For Siemens NX12.0 the solver used is Nastran. This thesis does not deal with the solution process within an FE solver since it is not in line with the present purposes.

### 2.4 Current Situation at OGMA

This section describes the tools which are used to connect landing gears to the hydraulic test bench. Additionally some insight into the durations of set-up processes is provided. In Figure 2.3, Figure 2.4 and Figure 2.5 the E190 MLG and NLG and an SA are shown set-up in the test bench, respectively.

#### 2.4.1 Existing Tools

Regarding the equipment used during testing in the landing gear shop the relevant parts to the present work are:

- The hydraulic test bench;
• The current beams used for the MLG and NLG;

• The support fixtures on which the assemblies will be placed on when in testing;

• The connection fixtures used for the SA.

The hydraulic test bench was acquired from “KRATOS Equipamentos” and is a fixture whose maximum applied load is of 50tf (tone-force) which amounts to approximately 490.333kN. It is an universal test bench with servo-hydraulic activation, composed of the top board and the bottom piston. The former can move upwards or downwards and is controlled by a side panel. The bottom piston is where force is applied during testing, in the upward direction, and is fully controlled by software developed by the company. Its movement has variable speed and may be locked in a given position required by the procedure, such as in angular amplitude tests when a NLG is going through the fault isolation process - [6] [7]. The test bench is also able to conduct cyclical tests which compress and decompress the SA or gear assembly in question, and record the variation in applied load according to the displacement of the bottom piston. It can additionally compare the applied force results to given performance curves, previously provided to the controlling computer.

There are two existing beam assemblies: one for the NLG and one for the MLG. The former is a component which was acquired by OGMA from the manufacturers of the landing gears - EMBRAER Liebherr Equipamentos do Brasil S.A. (ELEB). This beam is only fit to receive the NLG assemblies for the E170 and E190. Its shape can be seen in Figure 2.6. Obviously it is certified to withstand the major loads during testing of these components. The other existing beam is a solution created within OGMA for the sole purpose of connecting the MLGs to the hydraulic test bench. The complete beam assembly can be seen in Figure 2.7. The components in this image are:

1. Main beam;

2. Right suspension ring;

3. E170 MLG fitting bushings;
The beam fixture for the NLG is connected to the test bench through two steel plates. The lower one connects to the beam and the upper plate and the latter connects to the test bench’s connection point, in the upper board. The upper plate uses six bolts to connect to the upper board and the lower plate uses four bolts to connect both to the beam and the upper plate.

The MLG upper fixture is connected to the hydraulic test bench using a box assembly composed of four steel plates connected between them by employing high strength bolts: six bolts connect to the top board of the press, two bolts connect the side plates to the top plate and six bolts are used to fasten the bottom plate to the side plates.

Contrarily to the connection used in the NLG’s beam, this particular solution allows for the position of the beam to be varied longitudinally, depending on which gear is being tested. This is necessary since in both the E170 and E190 MLGs the sliding tube assembly is not centred relative to their upper connection points - pintle pins, as can be seen in Figure 2.3. This gives rise to issues during both set-up and testing procedures, since this deviation has consequences in the landing gear’s centre of gravity position and ultimately in how the applied loads by the test bench are transmitted to the connection.
fixture. To account for this factor the connection box is placed as close as possible to the edge of the beam connected to the pintle pin the nearest to the gear’s sliding tube position, which happens to be the left side of the beam. The position of the wheel axle relative to the sliding tube is also a factor to take into account since the connection point of the top board of the test bench is directly above its lower piston.

Problems arose at OGMA with the beam used for the MLGs. During testing, the assembly installed in the press, an E190 MLG, was subjected to an applied load which surpassed the maximum allowable loads for this beam. As a result the left side of the beam entered the plastic region of strain and acquired a permanent vertical deformation due to local yielding. Although this value was small and imperceptible if not looking at the beam in the right angle, the fact remains that yielding of the part was determined to be a “safe” type of failure which beckoned the replacement of the beam. This is common practice in mechanical design so as to avoid large permanent deformations and possible losses in mechanical resistance [22]. This kind of defect can be thought of as a result of the Muri being introduced in the process. Over-stressing the connection tool used led to the appearance of the referred defect.

As a response OGMA tasked one of its engineers to design and produce an alternative beam which maintained the dimensions of the deformed beam but was required to resist the maximum loads applicable by the hydraulic test bench. An increased safety factor applied during conception ultimately led to the new beam being considerably heavier and harder to connect to the test bench. The difference between both old and new beams can be seen Figure 2.9, the older beam is the lighter one on top, which can be seen as having had material remove so as to attain an I-shape, with a top and bottom flange and a web. The new beam is below, is darker and has no material removed besides what was needed to install the suspension rings.

![Figure 2.9: Comparison between the old (top) and new (bottom) beams at the shop.](image)

Once the used beam is pushed into the desired position the bolts of the side and bottom plates are further tightened so as to secure the beam’s position. The suspension rings of the beam are fastened to the main beam via a bolt and nut system. To secure the suspension rings another pair of nut and bolt sets is used, this fastens the under-rings to the upper half. It is easily inferred that the under-rings only “work” when the landing gear is suspended, waiting to be tested or removed; the upper part must resist the larger loads occurring during testing. The last component of the beam assembly are the adaptive bushings which are used to reduce the inner radius of the suspension ring and compensate for the difference in upper widths between the E170 and E190 MLGs.

The support fixtures also differ between those of the NLGs and of the MLGs. These are shown in Figure 2.10 and Figure 2.11. In these figures two parts must be addressed, one corresponding to each
assembly. Part 1 in Figure 2.10 indicates the wheel axle protections. These are installed in the wheel axle of the NLG to avoid damage and prevent the wheel axle from sliding off the support, during testing. Parts 2 are the sacrificial bushings used for the MLG wheel axles. These are made from bronze which has a lower hardness that that of the material of of the wheel axle of the MLGs - “high tensile steel” [4] [5]. This means that damage to the wheel axle is avoided and hence the need to rework or correct any defects which may originate from testing.

The main difference between the two presented assemblies is the rotational d.o.f.. Since the NLG testing procedure includes angular amplitude tests, its support fixture must allow a rotational movement about the central axis of its sliding tube, the reason for the cylinder and sprocket in its lower face. The same does not happen for the MLGs, since the wheel axle has no freedom to rotate about this axis. Another factor which must be addressed is that Figure 2.11 represents the original concept of the MLG support. However, at OGMA a modification was made and three steel plates were welded to this part to increase its resistance to the applied loads. The final cross-section of the support fixture can be seen in Figure 2.12 with the welded plates highlighted in blue.
Finally one must consider the connection tools of the SA. Like the previous sets, these are also two but considerably smaller since the dimensions and weight of the SA are also inferior to those of the MLGs and NLGs. The upper and lower parts can be seen Figure 2.13 in and Figure 2.14, respectively. The upper connection is made up of one plate and a connection part. The plate is connected to the test bench’s upper board through six bolts and the connection part is fastened to this plate by the use of two bolts. The SA is connected to this part via a connecting rod which has a threaded end to fasten this component to the SA’s upper hole.

The lower connection fixture is made up of only one major part, connected to the piston of the test bench via a bolted connection. The lower hole of the SA is secured to this part using the rod that can be seen in Figure 2.14. The upper bolts can be tightened so as to secure this connection rod.

The existence of three different connection fixtures for the hydraulic test bench is a factor which forces a change every time a different landing gear is tested. This means that time and personnel must be assigned to the task of changing both the upper and lower fixtures ahead of the landing gear set-up process. This takes up a considerable amount of time - e.g. the time it takes to change between the SA connection tools and the NLG tools is around 1 hour. Although this operation may be done in parallel with the final assembly of the landing gear, it is still a lengthy process which should be optimized and shortened. In light of this it was requested that one of the outputs of the present work be a design for an universal connection tool for testing, with every type of landing gear. To this effect the mechanical design insights gained in section 2.3 are applied.

### 2.4.2 Changeover and set-up duration

In order to assess the previous time situation at OGMA a series of time registries were conducted. It must be understood that since OGMA provides a service, the rate at which landing gears begin the MRO process and the variety of gears are independent factors. While smooth activity rates are desirable, the monthly real amount of work involved in the MRO of landing gears is not constant. In fact a ship-set may have its normal maintenance period overtaken by a more urgent ship-set which must be concluded on time. This happens with the Aircraft on Ground (AOG) situation. When this classification is issued to a given ship-set it means that the landing gears are needed immediately for the aircraft to be able to safely return to service. This is an emergency situation which can cost as much as 925.000 Euros per day to the operating company [23]. For this reason some ship-sets are concluded in shorter periods than others and there is the possibility that work on the first ship-set has to be slowed down in order to conclude the one in AOG conditions.

The above issues and the erratic frequency at which ship-sets may enter for maintenance operations creates problems in the number of opportunities to observe, register and detail the changeover process. This particular difficulty in applying the first stage of SMED implementation to the changeover process, arises from the difference between the kind of business conducted at OGMA and the case studies at the origin of the SMED tool. Indeed, while a production plant may enjoy some periods of uninterrupted production rate with several die changes according to the main plan of the organisation, a maintenance company depends on which customers it attracts. Moreover, once an MRO is conducted for a ship-set,
Table 2.2: Total set-up times for the different observed ship-sets.

<table>
<thead>
<tr>
<th>Landing Gear/Ship-set</th>
<th>Set-up Time (external + internal) [hrs.:min:sec]</th>
</tr>
</thead>
<tbody>
<tr>
<td>MLG Left/01</td>
<td>10:35:00</td>
</tr>
<tr>
<td>MLG Right/01</td>
<td>1:00:00</td>
</tr>
<tr>
<td>NLG/01</td>
<td>1:30:00</td>
</tr>
<tr>
<td>MLG Left/02</td>
<td>1:19:09</td>
</tr>
<tr>
<td>MLG Right/02</td>
<td>1:00:29</td>
</tr>
<tr>
<td>NLG/02</td>
<td>2:24:26</td>
</tr>
<tr>
<td>SA</td>
<td>11:47 (averaged over 50 data points)</td>
</tr>
</tbody>
</table>

that specific ship-set becomes covered for the next 8 to 10 years.

With this in my mind it comes as no shock the limited number of observations which were possible to register at OGMA, during the present internship. Indeed, while at OGMA, only 2 E190 ship-sets were observed (and no E170 ship-sets). Nevertheless, the number of ERJ145 landing gears sent to the components’ building was considerable and the set-up and removal procedure was observed many times. The extracted overall set-up times of the observed E190 ship-sets and the average set-up time for the tested SA can be seen in Table 2.2.

The first value which stands out is the enormous amount of time the first MLG of the first ship-set took to be set-up in the hydraulic test bench. This is due to problems with connecting the newly developed upper connection fixture to the test bench. However, the remaining set-up times, are around the same overall time, except for the SA set-up time which takes on average 12 minutes to be completed. Indeed most of the time wasted during this set-up consists on what could be considered external operations. More specifically, if the previous test were of a larger landing gear (an MLG or NLG) then the connection fixture would have to be switched to the one corresponding to the SA. Since originally OGMA did not include these latter types of gears in its portfolio, the older times balance out the occasional increase in the new set-up times.

One other aspect to have in mind is that, in most cases, the variance of the duration of a set-up is unfortunately dependant on the experience of the workers involved. One good example of this is the time it takes for the SA changeover. While the first internal recorded times, which date as far back as 2011, are almost all around the 15 minute mark, the registered internal times during the internship at OGMA were held frequently under 10 minutes. This shows the effect of growing experience and knowledge on the changeover times. Another way to confirm this is the reduction in set-up times for the MLGs which start at about 10 hours long and afterwards are as "short" as 60 minutes. It must be stated that this dependence is undesirable and SMED aims precisely at cutting down on the influence that experience may have on a process’s duration.

Recalling the gathered data, it also seems natural that the biggest potential for improvement lies within the set-up of the larger gear assemblies. Because of this and the fact that the average set-up time for a SA takes routinely around ten minutes, the present work will focus mainly on the set-up process of larger assemblies. Initially a reduction of the registered times to a minimum will be attempted using the existing equipment, whether by replacing fastening methods, reducing the number of operations or standardizing the ideal piece of machinery for a specific job. Once this is achieved, the introduction of some medium- and long-term solutions will be laid out, which involve changing existing tools or conceiving
entirely new ones, as is the case of the connection beams and bottom fixtures.
Chapter 3

SMED Implementation

This chapter addresses the implementation of SMED. It is divided into three parts. The first describes the current situation in terms of changeover times and their separation into external and internal activities; section 3.2 applies the second main stage of SMED implementation described in subsection 2.2.1. The internal operations which were converted into external ones are presented. In the final part the different concepts explored to streamline activities and ultimately reduce their time are explored.

Before presenting these results, one must distinguish external and internal operations. In a production plant it is easier to divide these two kinds of activity according to the classical definition presented in subsection 2.2.1. In the specific case of the hydraulic test bench, it is normal for this one to be idle for most of the time of an MRO procedure. The factor which determines when a landing gear may be set-up on the press, is often not the availability of the latter but whether the assembly of the landing gear has been concluded.

Due to the above reasons it was decided that the separating criterion between internal and external activities should be the landing gear assembly. In this way, any activity that could be performed while the landing gear was still being assembled was considered an external activity, and operations which could only take place once the assembly was finished and ready for installation into the hydraulic test bench would be characterized as internal. Thus an operation such as changing the upper connection fixture on the test bench is considered an external activity but the lower one may then be internal since in some cases the support part cannot be assembled prior to the set-up of the landing gear.

3.1 Registered Durations and Actions

In this section tables with the sequence of operations undertaken during set-up are presented. Additionally, making use of the distinction between internal and external operations, in the present context, the tool changeover times are also shown.

These last times are not presented alongside the set-up activities for reasons which are threefold. First and foremost, the duration of these operations is not necessarily at play when a landing gear is being set-up for testing. Whether or not the connection fixture needs to be changed ahead of a planned testing procedure, depends on the previous tests conducted at the press. If the previous landing gear matches the current one being prepared for testing, there is no need to employ resources into a changeover operation.

The second reason is due to the fact that these operations are already regarded and executed as external ones. It must be noted that these actions were undertaken while the landing gears were being
worked on, when observations were made at the landing gear shop. One exception must be mentioned however, as one of the observed set-up operations included the changing of connection tools. This was due to causes related with the quality and readiness of the tools available, already referred to in subsection 2.4.1. An issue with the replacement beam caused the tool changeover times to “leak” into the set-up and testing times, increasing the overall set-up times (seen in Table 2.2). This factor was so important that the majority of the time was dedicated to solving the problem created by the defect. This problem was not present in following set-up operations.

Finally, for clarity reasons, the tool changeover times were not presented alongside the set-up times. This would cause too much information within the same table possibly leading to confusion. Additionally a separate analysis of tool changeover times and operations allows for a better understanding of solutions developed in order to reduce the duration of these activities. When analysing these last processes the distinction between internal and external activities is not addressed since the entire process occurs as an external set of operations. Therefore, these are addressed in section 3.3.

With this in mind the presented sets of activities are separated according to this classification. Firstly the sequence of activities undertaken once the landing gears had been finished is shown; afterwards the registered tool changes and their times are addressed. In line with this, the times presented for the internal set-up operations includes only the times corresponding to these internal activities, not considering the external total time.

The presented times for each case are related to one assembly/set-up process. This may be done due to the consistency of the activities at play, and the time variations being attributable to the non-value added (NVA) time. NVA time concerns the time wasted by non-value added activities, such as searching for tools and wrenches or wrong set-up procedures which demand that the process be restarted. Although in some presented situations the NVA time is small, it can grow to be quite large, as can be seen by the difference between MLG set-up times in Table 2.2.

### 3.1.1 Set-up Times

As was written in subsection 2.4.2, only E190 MLGs and NLGs could be observed and analysed, alongside with many ERJ 145 SAs. For the SAs the set-up procedure is rather straightforward and well-defined. The sequence of activities and their respective durations is shown in Table 3.1. In the described situation, representative of the set-up process as a whole, NVA time is small and related mainly to wrench and tool search. Possible causes for NVA increase are poorly machined holes, the wrong fits and clearances on bushings or other defects which may cause the need to rework can vastly increase the duration of NVA time and thus the set-up time. The negative effects of the presence of defective parts and how to deal with them have been addressed in subsection 2.1.1.

In this and subsequent tables, the filling procedure is the pumping of hydraulic fluid into the landing gear assembly. This happens in the test bench for both the SA and the NLGs. The fluid is drawn from a deposit into a pump which pushes the fluid into the interior of the landing gear. This hydraulic fluid is an integral part of a shock absorbing assembly in reducing the transmitted loads to the airframe and the gear’s velocity of compression during touchdown.
Table 3.1: ERJ145 SA Set-up Activities

<table>
<thead>
<tr>
<th>Activity</th>
<th>Time [min:seg]</th>
<th>Activity</th>
<th>Time [min:seg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Fasten upper hole</td>
<td>0:53</td>
</tr>
<tr>
<td>Adjusting upper board height</td>
<td>0:10</td>
<td>Remove transport car</td>
<td>0:34</td>
</tr>
<tr>
<td>Remove upper &amp; Lower Connection Rods</td>
<td>0:05</td>
<td>Connect fluid cables</td>
<td>0:43</td>
</tr>
<tr>
<td>Tighten lower securing nuts</td>
<td>0:05</td>
<td>Turn fluid pump on</td>
<td>0:05</td>
</tr>
<tr>
<td>Move fluid deposit &amp; Connect to Pump</td>
<td>0:30</td>
<td>Filling procedure</td>
<td>4:20</td>
</tr>
<tr>
<td>Install valve part</td>
<td>0:20</td>
<td>Place lower connection rod</td>
<td>0:30</td>
</tr>
<tr>
<td>Move transport car to test area</td>
<td>0:45</td>
<td>NVA Time</td>
<td>2:30</td>
</tr>
<tr>
<td><strong>Total Time</strong></td>
<td><strong>12:30</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

For the MLG of the E190 the sequence of activities is quite different and more complex. These can be seen in Table 3.2. It can be observed in this table that the number of activities is quite larger than in the case of the SA. The total set-up time is also larger, due to the increased size and weight of the equipments in question. From the transport car to the connection tools, everything is heavier when it comes to the MLG. This means that extra personnel is needed for set-up activities and that the operation itself is more complex as well.

The only landing gear set-up left to analyse is the NLG. This set-up process has some advantages and disadvantages when compared to the previous type of gear. The first advantage is the reduced size which leaves more room to conduct operations when the test bench's top board is in the most elevated position. Another advantage is the usage of standard beams whose connection is simpler and quicker than those for the MLG. This means less time wasted in adjustments and repetitive operations.

The greatest disadvantage facing the NLG set-up process is the lack of an adequate transport car. Because the filling procedure for these landing gears must be carried out inside the press, any connection to a transport car which doesn’t prevent the sliding tube from collapsing is inadequate. Because of this a makeshift transport car has had to be employed. It only has the ability to carry the assembly on an horizontal position, which ultimately brings about problems when connecting the gear's pintle pins to the upper beam.

The registered actions and their times can be seen in Table 3.3. Just by analysing the described actions it can be deduced the crudeness of the current set-up process for this gear. The reliability on manpower and worker strength is a disadvantage which ultimately introduces variance in the set-up times according to the available number of personnel. The presented times were achieved by a team made up of two technicians and two engineers.

In the presented set of actions, it can clearly be seen that the lengthiest task is related to the transport car. As explained above this solution is not ideal and it consumes both time and effort. In fact, during the observed set-ups the common factor of both time wastage and personnel complaints was the lack of a dedicated car for the NLG. This issue is addressed in section 3.3.

The second greatest challenge faced was again the NVA time. In the particular case this was related with searching for wooden wedges, adjusting the placement of the wheel axle protections and the pintle pin protective bushings. This kind of wasted time must also be reduced to a minimum, ideally zero.
While the set-up time for this landing gear does not take as long as the MLG, room for improvement also exists.

Table 3.2: E190 MLG Set-up Activities

<table>
<thead>
<tr>
<th>Activity</th>
<th>Time [min:seg]</th>
<th>Activity</th>
<th>Time [min:seg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Move assembly car</td>
<td>0:45</td>
<td>Raise upper board</td>
<td>0:05</td>
</tr>
<tr>
<td>Rotate landing gear in car</td>
<td>3:35</td>
<td>Place support on pallet-carrier w/ wooden wedges</td>
<td>2:30</td>
</tr>
<tr>
<td>Remove left under-ring</td>
<td>0:15</td>
<td>Attach Girdles to Support</td>
<td>4:34</td>
</tr>
<tr>
<td>Remove piston protections</td>
<td>1:25</td>
<td>Raise support w/ crane</td>
<td>0:05</td>
</tr>
<tr>
<td>Centre pintle pins</td>
<td>1:12</td>
<td>Remove wedges</td>
<td>0:02</td>
</tr>
<tr>
<td>Place ladders</td>
<td>0:08</td>
<td>Lower support onto pallet-carrier</td>
<td>0:05</td>
</tr>
<tr>
<td>Remove nitrogen tanks</td>
<td>0:10</td>
<td>Remove support resting car</td>
<td>0:03</td>
</tr>
<tr>
<td>Change transport beam bolts</td>
<td>0:50</td>
<td>Move pallet-carrier to test area</td>
<td>0:30</td>
</tr>
<tr>
<td>Soft fastening left under-ring</td>
<td>0:29</td>
<td>Centre &amp; place metal wedges</td>
<td>2:18</td>
</tr>
<tr>
<td>Install right suspension ring</td>
<td>4:01</td>
<td>Remove pallet-carrier</td>
<td>0:10</td>
</tr>
<tr>
<td>Fasten right under-ring</td>
<td>4:33</td>
<td>Raise piston</td>
<td>5:59</td>
</tr>
<tr>
<td>Under ring fastening adjustments</td>
<td>1:32</td>
<td>Place inferior circular plates</td>
<td>0:45</td>
</tr>
<tr>
<td>Remove ladders &amp; other tools</td>
<td>0:29</td>
<td>Fasten central bolt</td>
<td>1:00</td>
</tr>
<tr>
<td>Unfasten oblique component</td>
<td>1:22</td>
<td>Remove Wedges and Lower Piston</td>
<td>0:28</td>
</tr>
<tr>
<td>Remove oblique part</td>
<td>0:30</td>
<td>Lower gear and place on support</td>
<td>0:05</td>
</tr>
<tr>
<td>Remove transport car</td>
<td>0:33</td>
<td>NVA Time</td>
<td>30:17</td>
</tr>
<tr>
<td>Rotate transport beam</td>
<td>8:24</td>
<td><strong>Total Time</strong></td>
<td><strong>79:09</strong></td>
</tr>
</tbody>
</table>

Table 3.3: E190 NLG Set-up Activities

<table>
<thead>
<tr>
<th>Activity</th>
<th>Time [min:seg]</th>
<th>Activity</th>
<th>Time [min:seg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Place gear on car</td>
<td>8:00</td>
<td>Raise upper board</td>
<td>0:36</td>
</tr>
<tr>
<td>Move car to test area</td>
<td>0:46</td>
<td>Remove wedges</td>
<td>0:05</td>
</tr>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Remove car</td>
<td>0:05</td>
</tr>
<tr>
<td>Place wooden wedges on car</td>
<td>1:20</td>
<td>Place wheel axle protections</td>
<td>1:30</td>
</tr>
<tr>
<td>Lower upper board</td>
<td>1:58</td>
<td>Lower upper board</td>
<td>0:40</td>
</tr>
<tr>
<td>Lift gear (w/ hands)</td>
<td>0:30</td>
<td>NVA Time</td>
<td>7:00</td>
</tr>
<tr>
<td>Connect pintle pins</td>
<td>1:30</td>
<td><strong>Total Time</strong></td>
<td><strong>25:00</strong></td>
</tr>
</tbody>
</table>

Together with the set-up operations, the removal sequences must also be mentioned because they are also a part of the MRO process. Reducing these times is also desirable and useful to further increase efficiency and reduce the overall time needed for testing.
Table 3.4: SA Removal Activities

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Remove fluid tubing</td>
<td>0:45</td>
<td>Lower upper board</td>
<td>1:30</td>
</tr>
<tr>
<td>Remove valve part</td>
<td>0:15</td>
<td>Place SA on car</td>
<td>1:00</td>
</tr>
<tr>
<td>Move transport car to test area</td>
<td>0:10</td>
<td>Remove upper connection rod</td>
<td>0:30</td>
</tr>
<tr>
<td>Remove lower rod</td>
<td>0:05</td>
<td>Remove car</td>
<td>0:45</td>
</tr>
<tr>
<td>Raise upper board</td>
<td>0:15</td>
<td>NVA Time</td>
<td>4:40</td>
</tr>
<tr>
<td>Move car underneath SA</td>
<td>0:05</td>
<td>Total Time</td>
<td>10:00</td>
</tr>
</tbody>
</table>

Table 3.5: E190 MLG Removal Activities

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Place support on pallet-carrier</td>
<td>0:32</td>
<td>Raise upper board</td>
<td>0:05</td>
</tr>
<tr>
<td>Raise upper board</td>
<td>0:05</td>
<td>Remove pallet-carrier &amp; support</td>
<td>0:10</td>
<td>Place ladders</td>
<td>1:20</td>
</tr>
<tr>
<td>Unfasten lower support</td>
<td>0:26</td>
<td>Move transport car to test area</td>
<td>1:01</td>
<td>Remove right suspension ring</td>
<td>1:25</td>
</tr>
<tr>
<td>Move pallet-carrier</td>
<td>0:20</td>
<td>Lower upper board</td>
<td>0:20</td>
<td>Connect gear to transport beam</td>
<td>0:33</td>
</tr>
<tr>
<td>Raise piston</td>
<td>1:30</td>
<td>Place connection pin</td>
<td>0:05</td>
<td>Remove ladders</td>
<td>0:20</td>
</tr>
<tr>
<td>Place metal wedges</td>
<td>0:05</td>
<td>Centre oblique part &amp; lower gear</td>
<td>1:35</td>
<td>Move car out of test area</td>
<td>1:19</td>
</tr>
<tr>
<td>Lower piston</td>
<td>1:10</td>
<td>Fasten oblique part</td>
<td>3:53</td>
<td>NVA Time</td>
<td>5:15</td>
</tr>
<tr>
<td>Remove circular plates</td>
<td>0:05</td>
<td>Remove under-rings</td>
<td>1:40</td>
<td>Total Time</td>
<td>24:14</td>
</tr>
</tbody>
</table>
Table 3.6: E190 NLG Removal Activities

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Move transport car to test area</td>
<td>1:00</td>
</tr>
<tr>
<td>Raise upper board</td>
<td>0:25</td>
<td>Centre car and gear</td>
<td>8:00</td>
</tr>
<tr>
<td>Raise piston</td>
<td>1:00</td>
<td>Place protections on wheel axle</td>
<td>0:05</td>
</tr>
<tr>
<td>Move support resting car</td>
<td>0:30</td>
<td>Support wheel axle on car</td>
<td>0:20</td>
</tr>
<tr>
<td>Unfasten support from circular plate</td>
<td>5:00</td>
<td>Connect rod to gear</td>
<td>3:30</td>
</tr>
<tr>
<td>Lower piston &amp; place support on car</td>
<td>1:00</td>
<td>Remove car</td>
<td>5:00</td>
</tr>
<tr>
<td>NVA Time</td>
<td>28:50</td>
<td>Total Time</td>
<td>55:00</td>
</tr>
</tbody>
</table>

In the presented tables one see that removal procedures for both MLG and NLG include the removal of a portion of the connection tools. This is a constraint which is hard to avoid due to the dimensions of the tested assemblies and the available transport tools. Besides the height of the landing gears, the minimum height of the transport cars is also a constricting factor. Additionally, even if "taller" tools are acquired, allowing for the supports to remain in the press during removal processes, the danger of injuries to personnel would increase, together with the difficulty of the set-up process itself.

3.1.2 Connection Tool Changes

In this subsection the observed changeover processes are presented. In particular the changeover from the SA tool to the MLG tools (Table 3.7), from the SA tool to the NLG fixture (Table 3.8), and from the NLG fixture to the SA tool (Table 3.9).

Some remarks are in order about the presented data. Regarding Table 3.7 only the set-up of the upper beam is addressed. This is because the transport car for the MLG of the E190 does not allow the support part to be previously installed, as was referred in subsection 3.1.1. This is the reason why Table 3.2 also includes the installation of the support part and Table 3.5 its removal operation. However, this is not the case for the set-up of the NLG. In fact, the absence of a dedicated transport car for the NLG allows for the set-up process of this gear to occur with the support part already connected to the piston. It can be seen in Table 3.8 that the installation of the support part is included.

The reverse does not hold true, unfortunately. If the support part is installed, the car used to transport the NLG after testing is completed, is unable to connect to the nose gear. Because of this, and as seen in Table 3.6, the lower part must be disconnected from the piston before the gear can be removed from the test bench. Additionally, and for all collected data, the time indicated for the activity "Turn the Test Bench on" may or may not be present, according to whether it was already active. In view of its reduced dimensions, the used tools for the SA are not a constraint to the used transport car.

The variability in times of all nine processes presented is an example of Mura. This affects negatively the landing gear maintenance system of OGMA, therefore it is important to reduce it and standardize it below ten minutes, as was requested.
### Table 3.7: SA to MLG Beam Tool Change

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Move resting car</td>
<td>1:58</td>
</tr>
<tr>
<td>Lower upper board</td>
<td>0:05</td>
<td>Fasten upper &amp; side plates to upper board</td>
<td>3:05</td>
</tr>
<tr>
<td>Unscrew lower part (upper connection)</td>
<td>1:54</td>
<td>Centre beam</td>
<td>2:25</td>
</tr>
<tr>
<td>Remove lower part</td>
<td>0:10</td>
<td>Fasten lower plate</td>
<td>11:30</td>
</tr>
<tr>
<td>Unscrew upper part (upper connection)</td>
<td>2:50</td>
<td>Adjust longitudinal beam position</td>
<td>0:30</td>
</tr>
<tr>
<td>Remove upper part</td>
<td>0:10</td>
<td>Tighten plates</td>
<td>0:55</td>
</tr>
<tr>
<td>Unscrew lower connection part</td>
<td>1:34</td>
<td>Remove support car</td>
<td>0:10</td>
</tr>
<tr>
<td>Remove lower connection part</td>
<td>0:10</td>
<td>Raise upper board</td>
<td>0:10</td>
</tr>
<tr>
<td>Remove right suspension ring (MLG Beam)</td>
<td>0:30</td>
<td>NVA Time</td>
<td>5:24</td>
</tr>
<tr>
<td><strong>Total Time</strong></td>
<td><strong>34:34</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 3.8: SA to NLG Tool Change

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Raise upper board</td>
<td>0:24</td>
</tr>
<tr>
<td>Unscrew lower part (upper connection)</td>
<td>3:04</td>
<td>Remove resting car</td>
<td>0:10</td>
</tr>
<tr>
<td>Remove lower part</td>
<td>0:10</td>
<td>Tighten lower plate</td>
<td>0:27</td>
</tr>
<tr>
<td>Unscrew upper part (upper connection)</td>
<td>1:16</td>
<td>Place bottom circular elevator parts</td>
<td>0:05</td>
</tr>
<tr>
<td>Remove upper part</td>
<td>0:10</td>
<td>Fasten lower circular plate</td>
<td>0:47</td>
</tr>
<tr>
<td>Unscrew lower connection part</td>
<td>0:15</td>
<td>Move resting car</td>
<td>0:47</td>
</tr>
<tr>
<td>Remove lower connection part</td>
<td>0:15</td>
<td>Raise piston</td>
<td>3:13</td>
</tr>
<tr>
<td>Separate upper plates (NLG Beam)</td>
<td>1:01</td>
<td>Centre &amp; fasten support to lower plate</td>
<td>13:12</td>
</tr>
<tr>
<td>Fasten upper plate to upper board</td>
<td>2:18</td>
<td>Lower piston</td>
<td>0:23</td>
</tr>
<tr>
<td>Centre resting car</td>
<td>4:15</td>
<td>NVA Time</td>
<td>13:25</td>
</tr>
<tr>
<td>Fasten lower plate</td>
<td>13:16</td>
<td><strong>Total Time</strong></td>
<td><strong>59:53</strong></td>
</tr>
</tbody>
</table>

### Table 3.9: NLG to SA Tool Change

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn test bench on</td>
<td>1:00</td>
<td>Unscrew upper plate</td>
<td>1:20</td>
</tr>
<tr>
<td>Centre car</td>
<td>0:30</td>
<td>Remove upper plate</td>
<td>0:10</td>
</tr>
<tr>
<td>Lower upper board</td>
<td>1:00</td>
<td>Fasten upper plate (SA upper connection)</td>
<td>2:20</td>
</tr>
<tr>
<td>Unscrew lower plate</td>
<td>2:30</td>
<td>Fasten lower plate (SA upper connection)</td>
<td>3:00</td>
</tr>
<tr>
<td>Raise upper board</td>
<td>0:10</td>
<td>Fasten lower connection part</td>
<td>0:45</td>
</tr>
<tr>
<td>Remove car</td>
<td>0:10</td>
<td>NVA Time</td>
<td>8:20</td>
</tr>
<tr>
<td><strong>Total Time</strong></td>
<td><strong>20:15</strong></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3.2 Converting Internal Activities to External Activities

Having already distinguished which activities were internal and which were external, the following step concerns their conversion. One of the objectives of SMED implementation is to try and reduce, as much as possible, the number of activities which only occur once the landing gear is finished.

Starting with the set-up process of the SA shown in Table 3.1, the activities to be converted were found to be:

- Turning test bench on;
- Lowering upper board;
- Removing upper and lower connection rods;
- Tightening lower securing nuts;
- Move fluid deposit and connect it to the pump;
- Install the valve part;
- Turn fluid pump on.

These simple conversions allow for the total internal time to decrease from 12 minutes and 30 seconds to 10 minutes and 20 seconds - a 17% gain. This value is quite close to the intended ten minute mark.

Looking at the MLG of the E190, the convertible internal activities are:

- Removal of the left under-ring;
- Removal of the piston protections;
- Removal of the nitrogen tanks;
- Changing of the transport beam bolts;
- Pre-rotating the transport beam into the desired position;
- Placing support on the pallet-carrier;
- Attaching girdles to support;
- Raising support with the crane;
- Removing wooden wedges;
- Lower support onto pallet-carrier;
- Remove support resting car;
- Place lower circular plates.
Subtracting the combined durations from the total time result presented in Table 3.2, one obtains 60 minutes and 1 second. Just with this conversion 20 minutes of set-up time are gained, a gain of 23% right out of the gate.

For the set-up process of the NLG (Table 3.3), the convertible actions are:

- Turning the test bench on;
- Placing wooden wedges on car;
- Lower the upper board;
- Place the wheel axle protections.

Removing the sum of these times from the overall internal set-up time a value of 20 minutes and 40 seconds is obtained - a gain of 17%.

Obviously, the removal processes must also be scrutinized and put through this conversion step. Starting with Table 3.4, which refers to the removal of the ERJ145 SA, the actions to be converted are:

- Remove valve part;
- Move transport car to test area.

This translates into a gain of 4% resulting from a time reduction to around 9 minutes and 30 seconds. In this case the single digit mark has already been achieved. However, these times can be further reduced by cutting down on the non-value added time.

For the removal of the MLG of the E190, the following operations may be turned from internal into external:

- Turn test bench on;
- Place ladders.

For the NLG, the activities are:

- Turn test bench on;
- Move transport car to test area;
- Place protections of wheel axle support.

These time reductions however, are not as effective in cutting down the internal time, as what happened with the set-up operations. A common factor to all different processes must be pointed out as its influence is considerable: the NVA time.

An analysis of the information in Table 3.1 through Table 3.6 shows these times can become as large as 30 minutes. Regardless of the absolute value of these NVA times, the fraction of the overall time they take up is a better measurement of their influence on the considered processes. The NVA time percentage ranges from 20% to 52%. Cutting this time down would represent a great improvement of
Table 3.10: Bolt mapping of the test area's tools.

<table>
<thead>
<tr>
<th>Bolt Size</th>
<th>Nr. of Bolts</th>
<th>Set-ups</th>
</tr>
</thead>
<tbody>
<tr>
<td>M20</td>
<td>5+2</td>
<td>Test bench connection point + SA lower plate (upper connection)</td>
</tr>
<tr>
<td>M16</td>
<td>1+1</td>
<td>MLG &amp; NLG Support + SA support</td>
</tr>
<tr>
<td>M12</td>
<td>2+4+4</td>
<td>MLG suspension ring + under-ring connection + NLG lower plate (upper beam)</td>
</tr>
<tr>
<td>M10</td>
<td>4+6+4</td>
<td>Side plates + Bottom plates (box assembly) + NLG support lower circular plate</td>
</tr>
<tr>
<td>M6</td>
<td>4+4</td>
<td>SA Securing bolts (lower connection) + Piston protections</td>
</tr>
</tbody>
</table>

the set-up and removal process times. In some instances, such as the SA set-up and removal, process times would be within the single digit mark just by eliminating these times.

NVA times are addressed with the standardization of processes. This involves the creation of visual standards which depict the correct sequence of actions in order to complete an activity in the proper manner. These standards should be filled with imagery of both the incorrect and the correct ways to perform a task. Large posters, rather than small pieces of paper, are preferred. In the list of described processes the activities which are recommended to be helped through visual standards are:

- SA filling procedure;
- Removal of piston protections;
- No right suspension ring during set-up and removal;
- Longitudinal position of the beam;
- Transport beam bolts;
- Angular position of the transport beam;
- Support location for installation with metal wedges;
- NLG supported on wooden wedges;
- Wheel axle protections position;
- NLG installed on transport car.

An example of a visual standard for some activities can be seen in Appendix A.

A check-list of all the needed tools is another useful type of document. Before set-up or removal activities this helps to inspect whether they are available and ready to be used. A way to achieve this is by mapping the tools used during set-up operations. Starting with a bolt mapping exercise of the test area and its components, it is possible to determine which tools are needed for the different processes. The bolt mapping for the test area is presented in Table 3.10. With the information in this table it is easy to come up with a check-list for a given set-up. Of course it must also include other tools that are used during set-up such as wedges, hammers, pallet-carrier, among others. An example of a check-list for the set-up of the E190 MLG is presented in Table 3.11.

Another factor which may help reduce NVA times is the analysis of the processes themselves. By describing and breaking down a process into a set of activities and operations, and projecting them to the personnel, failed attempts due to incorrect procedures or wrong sequences can be avoided. Combining this with a visual standard and a practical check-list for each process, it is expected that the NVA time
Table 3.11: Check-list for E190 MLG set-up, including tool changeover from SA tools.

<table>
<thead>
<tr>
<th>Tool</th>
<th>Use</th>
<th>Check</th>
</tr>
</thead>
<tbody>
<tr>
<td>M20 Hex Key</td>
<td>Remove SA upper plates &amp; Connect box assembly</td>
<td>✓</td>
</tr>
<tr>
<td>M16 Hex Key</td>
<td>Remove SA lower support &amp; Connect MLG Support</td>
<td>✓</td>
</tr>
<tr>
<td>M12 Hex Key + M12 Wrench</td>
<td>Connect right suspension ring &amp; Fasten under-rings</td>
<td>✓</td>
</tr>
<tr>
<td>M10 Hex Key</td>
<td>Fasten side and bottom plates of box assembly</td>
<td>✓</td>
</tr>
<tr>
<td>M6 Hex Key</td>
<td>Remove piston protections</td>
<td>✓</td>
</tr>
<tr>
<td>Pallet-Carrier</td>
<td>Transport and install MLG support</td>
<td>✓</td>
</tr>
<tr>
<td>Wooden Wedges</td>
<td>Place MLG support on pallet-carrier</td>
<td>✓</td>
</tr>
<tr>
<td>Girdles</td>
<td>Raise support and remove wooden wedges</td>
<td>✓</td>
</tr>
<tr>
<td>Metal Wedges</td>
<td>Install MLG support on piston</td>
<td>✓</td>
</tr>
</tbody>
</table>

Table 3.12: Internal process times with 5% NVA time.

<table>
<thead>
<tr>
<th>Process</th>
<th>Time [min:sec]</th>
</tr>
</thead>
<tbody>
<tr>
<td>SA set-up</td>
<td>8:14</td>
</tr>
<tr>
<td>MLG set-up</td>
<td>31:13</td>
</tr>
<tr>
<td>NLG set-up</td>
<td>14:21</td>
</tr>
<tr>
<td>SA removal</td>
<td>5:10</td>
</tr>
<tr>
<td>MLG removal</td>
<td>19:56</td>
</tr>
<tr>
<td>NLG removal</td>
<td>27:29</td>
</tr>
</tbody>
</table>

can be reduced to nearly zero.

Considering this, but keeping NVA time at a conservative value 5% of the total process time, the internal times of the presented processes then become much lower. This is shown in Table 3.12. In this table, one can see that the reduction of non-value added activities is paramount to achieve reduced process times. In fact, the activities for the SA meet the SMED goal just by removing these waste times. Additionally the set-up time for the NLG is rather close to the ten minute mark. The remaining durations must be reduced by employing solutions to streamline their activities.

3.3 Proposed Improvements to Activities

This section proposes some solutions intended to reduce the overall set-up, removal and tool change-over times described in the previous subsection. These solutions follow a time sequence, with short to medium term solutions are presented before long term concepts. Each solution is backed by its underlying reasoning.

3.3.1 Short to Medium Term Solutions

Corrections to existing tools

Firstly, there is the issues of problematic details in the existing tools, namely the transport car of the E190 MLG and the resting car of the NLG connection tools. These details are related to the levelling of the transport car, replacing existing wheels, the rotation of the landing gear on the car and the removal
of the oblique part. Levelling the car is essential to ease its motion around the shop and the test area. This can be done by removing the wheel which is not in contact with the ground and reinstalling it with an additional plate between it and the car. This levelling can save time in the adjustments and movements of the car when set-up and removal is under way.

Wheels must be replaced in the resting car for the NLG connection tools. With the current configuration only two of the wheels exhibit rotational freedom about their vertical axis. Replacing the two remaining wheels to also exhibit this freedom reduces the car’s restriction of movement, which becomes possible in every direction. Levelling and replacing wheels can be considered short term solutions.

The rotation of the landing gear in the car should also be eased. The gear is rotated with aid from the crane. The problem arises when, in the final stages of the rotation, the lower pin of the landing gear collides with a component of the car. Then the pin has to be placed by hand, requiring a great deal of manpower and time. This fitting part is intended to prevent the gear from rotating back into the horizontal position. This part has a threaded connection to the car and thus may be raised or lowered. To avoid this collision but still maintain the purpose of this part it is suggested that about a quarter of its surface be removed. This allows the gear to rotate more freely between vertical and horizontal positions. After rotation the part is given a half turn and the gear is then prevented from rotating back. In Figure 3.1 and Figure 3.2 the part is shown, additionally the location of the material to be removed is presented painted in white.

![Figure 3.1: Proposed correction to part on E190 MLG car.](image1)

![Figure 3.2: Top view of proposed correction.](image2)

It is also advisable to remove a pin connected to the oblique part of the car and the smoothing out of the part around which the gear rotates. This first pin prevents the easy removal of the oblique part, necessary to install the MLG in the upper connection fixture. Without this removal a component of the landing gear hits the transport car and it cannot be set-up on the test bench. The second component should be machined and smoothed so as to reduce the possible area of collision with the landing gear. The component is shown in Figure 3.3 where the material recommended to be removed is also displayed in white. If this is done, the effort required to remove the oblique part can be reduced an thus the time taken for the separation between landing gear and transport car.
Changes to the MLG upper fixture suspension ring

One concept which may offer great gains in time is the redesign of the suspension ring assembly. Analysing the fastening times for both MLGs and NLGs one can conclude that there is a great disparity between them, the MLG securing operation takes about 6 minutes and the NLG about 1 minute and a half. This is due to the different designs of each connection fixture. The proposed design is therefore intended to mimic the solution existing for the NLG.

The design was conceived with the intention to utilise the existing suspension rings and under-rings. The suggestion is to machine the vertical holes into an horizontal rip with two horizontal holes. In these holes a steel pin is to be fitted. One of the pins connected to the upper suspension ring is linked to a connection part which is also linked to the pin in the under-ring. On the other side the pin is connected to a rod-end bolt. To secure the under-ring this bolt is fastened to a threaded steel knob, as with the NLG.

The pins were chosen to be of $8mm$ and $10mm$ in diameter, the former used for connection with the connecting part and the latter to be connected to the rod-end bolt. A smaller pin was desired so as to reduce the risk of failure in the suspension rings “bottom feet. However, a rod-end bolt with a hole of that diameter did not present the necessary resistance to withstand the loads at play, namely in what concerned fatigue resistance. In spite of this the originated bearing stresses are not critical and do not present risk of failure for the loading in question. The maximum weight was found for the case where the entire weight of the E190 MLG - $320kg$ [4] - and its transport beam - $40kg$ - are suspended on only one of the suspension rings. To this weight a safety factor ($SF$ - see chapter 4) of 2 was also applied, giving the maximum loading of $7.063kN$.

Using a free body diagram (FBD) analysis and Equation 3.1 [18] the maximum stresses in the pins may be computed. This is because the pins are resisting exclusively shear loads. The outputs of an FBD are a shear force diagram (SFD) and a bending moment diagram (BMD). The initial conditions and the outputs of the conducted FBDs can be seen in Appendix C. $\tau_{\text{max}}$ is the maximum shear stress.

$$\tau_{\text{max}} = \frac{4 \cdot V}{3 \cdot π(\phi/2)^2} \text{ [MPa]},$$

(3.1)
In this equation $V$ is the shear load in $N$ and $\phi$ is the pin’s diameter in $mm$. $V$ is the maximum value found in the obtained SFD of the FBD analysis. Applying the maximum weight to these pins it was predicted that the maximum shear stresses would be about $92.718 MPa$ for the thinner pins and $59.339 MPa$ for the thicker pins.

These pins must be the first elements to fail. Hence the obtained maximum value should be multiplied by a factor of 0.90 giving a maximum allowable shear stress of $83.446 MPa$ and $53.405 MPa$, respectively. Knowing that the maximum allowable shear stress is usually taken as half of the material’s yield stress ($\sigma_y$), this means that the material chosen for the pin in question should have a $\sigma_y$ of about $166.892 MPa$ and $106.810 MPa$, respectively.

Applying a pressure on the bottom ring corresponding to the maximum weight distributed over the under-ring’s inner surface, the maximum stress in the modified suspension ring was found to be on the shoulder of the upper suspension ring - about $229.398 MPa$. Additionally, the maximum displacement was determined to be about $0.867 mm$, lower than $1 mm$. These values were obtained through a linear static analysis of the entire assembly in Siemens NX12.0.

To compute the stress in the rod-end bolt, the expression

$$\sigma_{bolt} = \frac{P \cdot k + F_i}{A_t} \quad [MPa],$$

(3.2)
can be used [17]. Here $\sigma_{bolt}$ is the tensile stress of the bolt, $P$ is the applied load in $N$, $F_i$ is the bolt preload in $N$, $k$ is the connections stiffness constant which may be taken as 0.2 and $A_t$ the bolt’s tensile stress area in $mm^2$. Budynas & Nisbett [17] recommend that the preload be taken as $0.75 \cdot A_t \cdot \sigma_{proof}$, $\sigma_{proof}$ is the bolt’s proof strength and according to [17], it can be taken as 0.85 of the bolt’s tensile strength, because only this value is provided in [24], hence $\sigma_{proof} = 0.85 \cdot 517.107 MPa$. For this given bolt $A_t = 58 mm^2$ (M10 bolt [17]),

$$\sigma_b = \frac{P \cdot k + 0.75 \cdot A_t \cdot \sigma_{proof}}{A_t} = 354.011 MPa.$$  

(3.3)

This means that the endurance strength of the used bolt must be at least this value, given the cyclic nature of the loading on the under-ring. This can be achieved with a strength grade of ISO 8.8 [17].

For the connection part, the maximum stress obtained with a linear static analysis was about $275.102 MPa$ and the maximum displacement $0.0186 mm$, a very small value. This analysis focused solely on this part and simulated its contact with other parts making use of boundary conditions.

The final design is made up of the suspension ring, three steel pins 45$mm$ long - two with 8$mm$ diameter and one with 10$mm$ diameter - one connection part, one M10 rod-end bolt with a strength grade of at least ISO 8.8 and an M10 threaded steel knob to secure the connection. The scheme of dimensions for the rod-end bolt selected is shown in Figure 3.4. The proposed type of knob is shown in Figure 3.5. A 3D model of the intended end result is shown in Figure 3.6 and the devised connection part is presented in Figure 3.7.
Redesign resting cars for supports

One other suggestion for gaining time during set-ups and removals is to redesign the resting cars for the supports. This would mean that while the supports weren’t installed, they would be placed on these new cars. The cars allow for an easy installation while the landing gears are suspended in the upper board. Removing the need for pallet-carriers, for example, would mean a greater cut down on set-up times.

The concepts were intended to transport the supports up to the piston and, once the support was fastened to the piston, be removable with the piston raised. These cars are use materials available at OGMA, namely steel tubing and wheels. The steel tubing has a quadrangular shape and is available in two different sections: 60x60x3.2 and 40x40x3.2 (height x width x thickness).

In order to assess if the allowable height and other dimensions of these cars is enough a 3D linear static analysis using SkyCiv’s 3D structural software was conducted. The 3D part of this analysis concerns both the directions in which the loads act and the transmission of deflections throughout the elements. The latter are 3D beam elements. The section characteristics of the available steel tubing at OGMA were also used to compute the stresses in the structures when the supports were resting on them.

For the NLG car the height limitations were close to none as the NLG is short enough to be raised while in the test bench, and still leave plenty of room below it. For the MLG car, accounting for the wheels available at OGMA, with a height of about 120mm, it was determined that the maximum height of the structure of the car should not exceed 110mm. This value accounts for the maximum height of the test bench, the height of the E190 MLG [4] and also the height of the wheels which are to be installed in
the support car. Assuming 60x60x3.2 tubing, the maximum height taken for the car was 50mm.

The developed structures for the cars can be seen in Figure 3.8 and Figure 3.9. The car developed for the NLG support is not as simple as the one for the MLG because of the lower sprocket of this support (see Figure 2.10). This diameter is larger than the width of the support, so a solution had to be devised allowing for the removal of the car once the support was installed.

A 3D model was constructed for each car, with beam elements which connected nodes between them. The cross-section of each beam element was pre-defined. The loading applied to the structures was the weight of each car, distributed over the support length. These loads were 2.404kN/mm for the MLG support car and 3.063kN/mm for the NLG support car.

After analysis it was determined that the bending stresses in these cars would not surpass 7.714 MPa and 25.517 MPa and that shear stresses would remain below 2.847 MPa and 6.433 MPa, respectively. These values are well within the mechanical resistance of the steel tubing used. The maximum deflections obtained were also small - 0.034mm and 0.275mm respectively - in line with the assumptions of a linear static analysis. These values were extracted from an analysis hence their precision.

**Reduction of the diversity of bolts in the test area**

As was shown in Table 3.10 there is a wide diversity of bolts used in the test area. This means that a large number of keys and wrenches are needed when setting up either the landing gears or the connection fixtures. Additionally some of the connections employ more bolts than needed and others employ solutions which take too long, like the example given of the under-ring fastening operation.

Firstly quick change fasteners were looked into to replace some of the existing tools. One solution found was the SLIC pin. These are removable pins which take up considerably less time to remove and also provide a good level of resistance to the loads involved during testing and assembling. It is a commercially available pin developed by Pivot Point™ and can be seen in Figure 3.10. The candidates chosen to be replaced by these pins were the suspension ring connection bolts and the bottom plate connection bolts of the NLG support.

A linear static analysis was conducted for the upper part of the suspension ring using the maximum loading of 480kN. The reaction load obtained was 17.333kN, making use of Equation 3.1 the maximum shear stress in the pins during testing reaches the maximum value of 204.343 MPa. Using the same logic as for the pins of the suspension ring, the material chosen for the pin in question should have a \( \sigma_y \) of about 367.818 MPa.

For the pins of the NLG support disc it can be assumed that the torque at play during testing \( T_o \) is
equal to $300N \cdot m$, with a factor of safety of 1.5 [6] [7]. Taking this value the total force during rotation is found to be $30kN$, given that

$$To = N/R \quad [N \cdot mm],$$

(3.4)

and $R \approx 100mm$. This value can be divided by the number of bolts to get the shear load on each pin: $7.500kN$. These pins have a diameter of $10mm$ which means that a shear stress of $127.324 MPa$, obtained with Equation 3.1, must be withstood by the pins used for the connection between the NLG support part and the lower part which connects to the piston. Following the previous logic the material of the pins chosen for this connection should present a yield stress of about $229.183 MPa$.

![SLIC pin CAD model provided by Pivot Point™.](image)

Looking into the reduction of the number of keys and wrenches, a conversion of certain threaded holes is in place, the bolt mapping of the test area is described in Table 3.10. The selected conversions are:

- Side and lower plates (box assembly) $M10 \rightarrow M12$;
- Piston connection (MLG, NLG and SA support) $M16 \rightarrow M20$.

These conversions reduce to three the existing sizes of bolts in the test area. A reduction in the number of certain bolts is also useful. In this sense it would be advantageous to analyse whether the number of bolts needed for the press connection can be only four, rather than six, similarly the bottom plate could also need only four bolts and not six.

To determine whether or not the number of bolts can be reduced, the stress in each bolt was computed for the extreme loading cases. To compute the number of bolts needed when the landing gear is suspended on the upper beam Equation 3.5 [17] is used. In this equation $N_b$ is the number of bolts needed and $P$ will be taken as $9.771kN$, the maximum weight that can be suspended on the box assembly which accounts for landing gear and the remaining connection fixture.

$$N_b = \frac{P \cdot k}{\sigma_{proof} A_i - F_i}$$

(3.5)
For the case of the bolts connected to the press, \( k \) can be taken as 0.2 because all components are made of steel, \( \sigma_{\text{proof}} = 650 \text{MPa} \) for these bolts and \( A_t = 245\text{mm}^2 \), \( F_i \) was computed following the same logic as for the suspension ring rod end bolt. With these values \( N_b = 0.049 \). For the bottom plate \( k = 0.2 \) as well, \( \sigma_{\text{proof}} = 970 \text{MPa} \) and \( A_t = 58\text{mm}^2 \) which computes \( N_b = 0.139 \). The differences in \( \sigma_{\text{proof}} \) are due to the different strength grades the bolts belong to.

These values mean that six bolts is far more than what is needed for both these bolted connections. However, it is not enough to compute this value because when the landing gear is being tested, the applied loads cause moments which originate additional loads that the connections must withstand. To account for this the moments generated by the loads transmitted to the beam on the centre of the plates must be computed. Then Equation 3.6 [17] is applied and the total loads acting on each bolt are summed. In this formula \( n \) is the current bolt being analysed, \( M \) is the moment acting on the centre of the plate in \( N \cdot \text{mm} \) and \( d_i \) is the distance to the centre of the plate of bolt \( i \) in \( \text{mm} \). In the present cases \( d_i \) is constant and equal to \( d_n \) because the bolts are distributed symmetrically on the plates in question.

\[
F_n = \frac{M \cdot d_n}{\sum_i d_i^2} \quad [N]
\]

(3.6)

After computing the load for each bolt, Equation 3.2 is used and the stress in each bolt is obtained. With both these equations it is therefore possible to determine the loads acting on the bolts for both MLG testing scenarios. The moments are computed using the distance between each edge of the beam where the loads are applied and the position of the box assembly and its centre point.

Next, the question was to determine the magnitude of the loads at play during testing. To reach these values a simplification of the MLG assemblies was in order. This was done by considering the assemblies as a set of connected beams in equilibrium, with the load being applied on the point where the wheel axle should be. This analysis is not needed for the NLG because of the symmetry of its structure, as well as the fact that the connection to the upper board is directly above the wheel axle support point. This gear is not relevant for the problem at hand but its transmitted loads were computed nonetheless since they will be made use of further ahead.

Considering rigid connections and simple supports on the nodes representing the pintle pins, the following transmitted loads were found, for an applied force of 480\( kN \):

- **E190 MLG**: \( R_{\text{left}} = R_1 = 395.539[kN] \) & \( R_{\text{right}} = R_2 = 75.829[kN] \);

- **E170 MLG**: \( R_{\text{left}} = R_1 = 367.155[kN] \) & \( R_{\text{right}} = R_2 = 105.180[kN] \);

- **NLG**: \( R_{\text{left}} = R_1 = R_{\text{right}} = R_2 = 236.961[kN] \).

Using these computed values, it is possible to obtain, for the E190 MLG:
\[ M = R_2 \cdot 691.4 - R_1 \cdot 113.6 = 7494940 \quad [N \cdot mm]; \quad (3.7) \]
\[ F_n = \frac{M \cdot 44.780}{4 \cdot 44.780^2} = 41843.122 \quad [N]; \quad (3.8) \]
\[ \sigma_{bolt,n} = \frac{F_n \cdot 0.2 + 0.75 \cdot 245 \cdot 650}{245} = 521.658 \quad [MPa]. \quad (3.9) \]

For the E170 MLG:

\[ M = R_2 \cdot 616 - R_1 \cdot 166.5 = 3326573 \quad [N \cdot mm]; \quad (3.10) \]
\[ F_n = \frac{M \cdot 88.256}{4 \cdot 88.256^2} = 18571.756 \quad [N]; \quad (3.11) \]
\[ \sigma_{bolt,n} = \frac{F_n \cdot 0.2 + 0.75 \cdot 245 \cdot 650}{245} = 502.661 \quad [MPa]. \quad (3.12) \]

These operations show that four bolts are enough to secure the top plate of the box assembly. In the above equations 44.780\text{mm} is the distance \( d_n \) between each bolt and the centre of the plate.

For the bolts of the bottom plate \( d_n = 88.256\text{mm} \), and considering the E190 MLG case,

\[ M = 7494940 \quad [N \cdot mm]; \quad (3.13) \]
\[ F_n = \frac{M \cdot 88.256}{4 \cdot 88.256^2} = 21230.681 \quad [N]; \quad (3.14) \]
\[ \sigma_n = \frac{F_n \cdot 0.2 + 0.75 \cdot 58 \cdot 970}{58} = 800.710 \quad [MPa]. \quad (3.15) \]

For the E170 MLG,

\[ M = 3326573 \quad [N \cdot mm]; \quad (3.16) \]
\[ F_n = \frac{M \cdot 88.256}{4 \cdot 88.256^2} = 9423.079 \quad [N]; \quad (3.17) \]
\[ \sigma_n = \frac{F_n \cdot 0.2 + 0.75 \cdot 58 \cdot 970}{58} = 759.993 \quad [MPa]. \quad (3.18) \]

By analysing the obtained results it is possible to conclude that the reduction from six bolts to four bolts on both plates in question will not reduce the beam fixture’s ability to withstand the loads at play. This conclusion results from the fact that the analysed case is the limit one. With all these changes in place to the bolts used, time reductions are expected to occur both during connection tool changeover and during set-up and removal operations.
3.3.2 Long Term Solutions

Redesign of the oblique part

One of the main issues regarding the set-up operation of the E190 MLG is the collision which occurs between the landing gear and the central part around which the landing gear rotates, shown in Figure 3.3. So much so that it requires the entire oblique part of the transport car to be removed in order to take the car out of the test area and raise the landing gear. To address this issue the oblique part and its connection to the landing gear must be reviewed in order to ease the separation between the two components. This implies redesigning this part and then reinstalling it in the transport cars. A new shape allowing the car to be removed while the landing gear is being raised and alternative ways on how to fasten this component to the landing gear are shown in Figure 3.11 and Figure 3.12.

![Figure 3.11: Alternative connection 1.](image1)

![Figure 3.12: Alternative connection 2.](image2)

New NLG transport car

As stated before in section 3.1, one major issue regarding the processes concerned with NLGs is the available transport cars. During set-up a makeshift car has to be used to transport the gear to the test bench. To remove the gear a car is used which is originally intended for the E170 MLG. One desired solution to this issue is the conception of a new transport car. The latter should mimic the solution used at ELEB, which uses a strapped connection to prevent the gear to collapse on its wheel axle. In addition this car exhibits smaller dimensions than those of the existing cars which facilitates movement and reduces the effort needed to move the landing gear in the shop. This car can be seen in Figure 3.13 and in Figure 3.14 a developed CAD model where its vertical d.o.f. can be seen. The bottom configuration of the car could possibly allow for the landing gear to be set-up and removed with the support fixture installed in the test bench.

![Figure 3.13: New NLG transport car.](image3)

![Figure 3.14: Developed CAD model.](image4)

Changing the entrance of the test area

This proposal is one which would take the longest to apply. The test area is a squared area, with metal walls and two sliding doors - one leading to the control room and one leading to the shop. The hydraulic test bench is at the centre of this test area. As seen in Figure 3.15, the entry door for the shop faces directly a meeting room and not the shop itself. This arrangement forces the assembled gears to be unnecessarily moved around, generating transport wastes.

An entrance which leading directly to the shop would be preferable with no need to turn and adjust the transport cars - the gears would enter the test area directly. This solution would also force the test
bench to be reinstalled, so as to face the new entrance to the test area - hence the additional effort required in achieving this concept.

Overall this proposed change would reduce time and effort needed in moving the transport cars around the landing gear shop. This factor is non-determinant for a small gear like the SA but it is relevant when a landing gear weighing about $350\text{kg}$ has to be moved towards the entrance of the test area.

**Create an universal connection fixture**

This final solution is intended to reduce and nearly eliminate the duration of the tool changeover times. This tool, its function and how to use it are discussed in chapter 4.
Universal Tool Development

Considering what was referred to in section 2.3 about the Mechanical Design process, the same sequence of steps can be applied to the universal tool development. This development follows the main procedural stages presented in section 2.3. In this sense the present need relates to a faster and simpler changeover process in the connection fixtures. With some additional and more specific information, a problem definition can be created:

A connection fixture must be created which includes both upper and lower connections equipped for the Shock Absorber of the ERJ145 and the main and nose landing gears of both Embraer E170 and Embraer E190 aircraft. This tool must withstand the maximum loads applicable during testing, with an adequate safety factor, and also exhibit rotational freedom around the vertical axis, for the bottom connection. Additionally, its connection to the hydraulic test bench must be as fast as possible.

The main issues to be dealt with result from this definition. The first is adaptability. As seen from Figure 2.6 to Figure 2.14, there is a wide variety of shapes for the different connection fixtures which attach landing gears to the hydraulic test bench. The devised solution must take this into consideration. Additionally, as mentioned in subsection 2.4.1, the conceived fixture must account for the deviation of the centre of gravity of the MLGs of the E170 and E190 aircraft (see Figure 2.3). This means that the relative position between the test bench’s upper connection point and the midpoint of the upper assembly of the fixture must be variable.

In terms of heights of the landing gears, only the largest of these is determinant since the press can imprint vertical movement on both the upper board and the lower piston. The MLG of the Embraer 190 is the “tallest” landing gear with a fully extended height of 2183.3\(\text{mm}\) from the centre of the pintle pin to the centre of the wheel axle [4]. Meanwhile, the measured maximum height difference between the upper and lower points of the test bench is 3125\(\text{mm}\). This means that the total “height” of the connection tool cannot exceed \(3125 - 2183.3 = 941.7\text{mm} \approx 940\text{mm}\). This “height” is considered to be the distance between the connection point of the upper board and the central axis of the pintle pins of the landing gear, added to the distance between the lower piston of the test bench and the central axis of the wheel axle of the same landing gear.

One other important set of data is the maximum loads the structure must withstand. While analysing the procedural files at OGMA, filled according to what is observed during testing, it was found that the maximum expected load was of about 24269.5\(\text{kg} \cdot \text{f}\). This value was the maximum load applicable during “Proof Pressure Test Shock Absorber Assembly”[4] procedures for the MLG of the E190. This value
Table 4.1: General recommendations for factor of safety selection [25].

<table>
<thead>
<tr>
<th>Application</th>
<th>SF</th>
</tr>
</thead>
<tbody>
<tr>
<td>For use with highly reliable materials where loading and environmental</td>
<td>1.3 - 1.5</td>
</tr>
<tr>
<td>conditions are not severe and where weight is an important consideration</td>
<td></td>
</tr>
<tr>
<td>For use with reliable materials where loading and environmental conditions</td>
<td>1.5 - 2</td>
</tr>
<tr>
<td>are not severe</td>
<td></td>
</tr>
<tr>
<td>For use with ordinary materials where loading and environmental conditions</td>
<td>2 - 2.5</td>
</tr>
<tr>
<td>are not severe</td>
<td></td>
</tr>
<tr>
<td>For use with less tried and for brittle materials where loading and</td>
<td>2.5 - 3</td>
</tr>
<tr>
<td>environmental conditions are not severe</td>
<td></td>
</tr>
<tr>
<td>For use with materials where properties are not reliable and where</td>
<td>3 - 4</td>
</tr>
<tr>
<td>loading and environmental conditions are not severe, or where reliable</td>
<td></td>
</tr>
<tr>
<td>materials are used under difficult loading and environmental conditions</td>
<td></td>
</tr>
</tbody>
</table>

amounts to about 238kN which can be rounded off to 240kN. Applying to this value the safety factor of 2 the maximum withstood load for the connection fixture becomes 480kN.

The safety factor (or factor of safety) in engineering is usually expressed by the relation

\[ SF = \frac{R_{\text{fail}}}{R_{\text{allow}}} \]  

(4.1)

where \( R_{\text{fail}} \) is the failure load of the structure or component and \( R_{\text{allow}} \) is the allowable load, according to Beer et al. ([18]). When selecting the value for \( SF \), general recommendations were found, according to the type of loading, material and loads involved. These are seen in Table 4.1.

With this information a safety factor of 2 was selected. This value was chosen because the environment where loading occurs is controlled and non-damaging to the parts in question, due to corrosion, heat or impacts. Another reason for the value of \( SF \) has to do with the fact that a choice of reliable materials is favoured, rather than a choice of innovating and riskier materials. This work presents in some cases suggestions for materials that may be employed in certain components, as well as relative weights for material selection criteria for most of the developed parts.

The method used to determine these relative weights was material digital logic as described by Farag ([26]). To apply this method, the different relevant material properties are chosen initially, and then are compared one against the other for a total number of \( n \cdot (n - 1)/2 \) where \( n \) is the number of characteristics chosen. When evaluating between two different characteristics the one deemed the most important receives a score of 1 and the other a score of 0. In the end the total score of each characteristic is added up and divided by the total number of comparisons giving its final relative weight. The sum of all scores must equal one (or one hundred percent). These scores do not have any units ans can be presented as percentages. An example is shown in Table 4.2.

To obtain the score of a given material the relative weights are multiplied by the value of each property and then they are all added up, except for the density where the lowest value is the most desirable, which is therefore subtracted. The same is done for each property to be minimized. Looking at the properties in the example it is clear that the orders of magnitude of some properties are usually larger than others. While values for fracture toughness are usually around \( 10 - 100 \text{ MPa} \cdot \text{m}^{0.5} \), the values for \( \sigma_y \) are usually
Table 4.2: Material Digital Logic example according to [26].

<table>
<thead>
<tr>
<th>Property Comparison</th>
<th>Fracture Toughness ($J_{IC}$)</th>
<th>Yield Strength ($\sigma_y$)</th>
<th>Young's Modulus ($E$)</th>
<th>Density ($\rho$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>6</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Final Score</td>
<td>33%</td>
<td>16.7%</td>
<td>16.7%</td>
<td>33%</td>
</tr>
</tbody>
</table>

in the range $200 - 1000 \text{ MPa}$. Farag ([26]) states that in order to avoid the influence of larger values it is advisable to normalise the values obtained using the largest value for that given property. In this way the property values are weighted and the difference is attenuated between orders of magnitude.

During conception ordinary materials such as steels and aluminium alloys were considered and their characteristics employed. The final material selection depends also on availability, cost and other characteristics such as machinability and formability. These informations are more easily obtained by OGMA engineers and so, at the end of the internship, the final decisions regarding materials were transferred to OGMA. Nevertheless suggestions were handed over as well as factors to consider such as maximum stresses and deflections expected to occur during loading.

The sections below cover the synthesis, analysis and evaluation stages of the process described in section 2.3. However, for the sake of brevity, backtracks and restructuring of different concepts are not addressed. These different steps are analysed component by component rather than globally.

### 4.1 FEA Meshes and Results

As was stated in subsection 2.3.1, FEA was used as a tool to analyse the responses and behaviours of the different developed components. For each component a model was developed, a mesh created and then the loading and constraint conditions defined. All this in order to extract the maximum stresses and displacements occurring in each part.

The first step to run an FEA is the creation of the model, this was achieved using the CAD tools of both Siemens NX12.0 and SkyCiv’s 3D Structural Design. Afterwards the mesh must be created, according to Cook ([20]). For the Siemens NX12.0 a base parameter which measures the dimension of the individual elements in the mesh exists - the element size. This dimension may be increased or decreased in order to obtain a coarser or finer mesh, respectively. To determine the ideal element size mesh refinement studies must be carried out. The principle behind this study is that the finer mesh (with a greater number of d.o.f.) will exhibit the most reliable results, however an intermediate solution is desired which is not too computationally demanding, while accurately representing the reactions of the model at hand.

Because it would be impractical to run mesh refinement studies for each component, a mesh refinement study was carried out for the MLG/NLG slider part (subsection 4.2.3). Most of the parts developed have a similar characteristic length as the slider and the same element size can be applied when
analysing these parts. The selected element size can also be applied to both larger parts (beam - subsection 4.2.2, support body - subsection 4.3.4), at the cost of computational time. The mesh refinement studies were ran for an original shape, previous to the subsequent optimization process. This model was similar to the suspension ring shown in Figure 3.6, without the under slider. Figure 4.1 shows a later stage of the slider part, the original model is similar to this one but without the added ribs and with a smaller thickness of the cylindrical portion, 18\text{mm} to be exact.

Linear static analyses were ran, for a pressure of 5\text{MPa}. This small pressure was selected to ensure that the behaviour of the part remained in the elastic linear domain. A representation of both the load applied and the boundary conditions of each surface is presented in Figure 4.1. It must be noted that the shape presented is the final shape of the slider part, however the boundary constraints and the application of load is the same as the ones applied during the refinement study. In this figure the pressure is applied in the inner surface of the cylindrical portion of the part and the constraints serve to simulate the contact with the beam and the pin connection between these two parts. The light blue regions have the longitudinal displacement in the $Z$ direction and rotational displacements about the $X$ and $Y$ axes constrained. The pink regions have the longitudinal displacement in the $X$ direction and rotational displacements about the $Y$ and $Z$ axes constrained. The dark blue region has its longitudinal movement in the $Y$ axis constrained, as well as the rotational one about the $X$ and $Z$ axes. The pressure applied is presented in red.

The selected elements were 3D tetrahedral elements with 10 nodes each. A table with the results of the refinement study is presented in Appendix D. The selected element size was of 5\text{mm}, which presented the best compromise between computational time and absolute errors for both maximum stress and maximum displacement. All the subsequent analyses used this element type and size. The material chosen for the elements was one available in the NX12.0 library of materials - a stainless steel AISI SS 304-Annealed. This material was chosen so as to represent the application of steel to the developed parts. However, if after an initial analysis the yield stress of the material was surpassed (276\text{MPa}), a high strength steel was used, so as to ensure that the component remained in the linear elastic region. This material was the high strength steel - AISI Steel 4340. One additional case was for
the MLG/NLG slider where Aluminium 2014 was used in the analyses. This material was employed to observe the maximum displacements which would occur for a material with a lower Young’s modulus.

The stress state for each developed component is presented in Appendix D, as well as a table with the maximum stresses and displacements for each case. Additionally some insight is given into some issues which appeared during testing of different parts. These are related to the nature of the FEA method and must be addressed in order to correctly analyse the outputs of the analyses.

Mesh refinement was not necessary for the cases where SkyCiv’s 3D Structural Software was used because the analyses were done considering each element one of the constituent beams of the cars. This was possible because the relevant results were only relating to the translational components of the beams. Additionally, the extracted deflections and stresses were well below the length of the beam and its yield stress, respectively.

### 4.2 Upper Fixture

![Figure 4.2: Schematic 3D model of the final upper fixture.](image)

The chosen design for the top assembly consists of:

1. Connection part (claw);
2. Main beam;
3. 3 slider parts, 2 for the pintle pins, each with an under slider part and respective pins and connection methods, and a slider for the Shock Absorber;
4. 4 fitting bushings, 2 for each additional landing gear - Embraer 170 MLG and both nose landing gears.

A schematic view can be seen in Figure 4.2\(^1\). Each component of this assembly is described below.

\(^1\)Throughout the document, images are presented whose depicted parts contain lines over their surfaces. These lines do not exist in the final models and served only the purpose of allowing some areas to be split between different surfaces, so as to apply boundary conditions, loads or pressures. When attempting to hide these section splits the used software would hide the whole part so their presence is inevitable.
4.2.1 Claw

This part has a channel-shaped section so that it can be connected to the top board of the press and still allow the relative movement of the beam, the flange has the same thickness as the empty space of the claw section. The section dimensions have been chosen so that the reaction bending moments are supported without yielding of the materials. That is because this is this piece that withstands the forces and moments of reaction generated during testing.

According to FBDs (obtained in Appendix C), the maximum reaction moment the claw has to endure is of $5265.2\, N\cdot m = 5.2652 \times 10^3 kN\cdot mm$. This value occurs during testing of the Embraer 170 MLG, which can be seen in section C.3.

$$\sigma_{m,\text{max}} = \frac{M}{S_i} \, [N/mm^2] \leftrightarrow [MPa]. \quad (4.2)$$

$\sigma_{m,\text{max}}$ is the maximum bending stress, $M$ is the bending moment in $N\cdot mm$ and $S_i$ is the section’s modulus about the $i$ axis in $mm^3$ which in the present case is the $Z$ axis. Using the characteristics of the sections, observable in Figure 4.5, and Equation 4.2 [18] the maximum theoretical bending stress about the $Z$ axis is $35.164\, MPa$, $S_Z = 1.49731 \times 10^5 mm^3$, as can be seen in Figure 4.5.

The length of the flange (25 mm in Figure 4.5) was chosen to help increase the section’s modulus to better resist the factor described above, and also support the maximum weight that is expected to be suspended on the beam. This weight includes the maximum weight of all the landing gears, which corresponds to the MLG of the E190 (320 kg), the weight of the transport beam for this same landing gear (40 kg), as well as the weight of the remainder of the upper assembly, including beam, sliders, under-sliders and additional components such as pins which amounts to about 138 kg. This computed maximum weight (considering the worst case scenario where every part is made of steel) was determined to be approximately 4.885 kN. Applying the already referred $SF = 2$ this value becomes 9.770 kN.

By running a static linear analysis on the stress state of the claw part when the weight is suspended on its flanges, it was found that the maximum stress was 28.720 MPa. This value is well below the yield strength of any common manufacturing metallic material. Obviously, the compressive loading is the critical case. An analysis was also ran for this case, with a compressive load of 480 kN. For this last case the maximum stress found was 38.763 MPa. For the first analysis the maximum displacement was $8.723 \times 10^{-3}$ and in the second one the maximum value was $4.186 \times 10^{-3} mm$. Both these values were obtained by considering the part to be made of steel and are much lower than zero. Therefore, they can be neglected.

The holes on top of the part were selected according to the location of the holes in the upper board’s connection point. As mentioned in section 3.3 the number of required bolts remains at four. In line with this only four holes are projected in the part and these have the same dimensions as those of the existing top plate of the box connection assembly. The same bolts are also used, for the sake of economy and making the most of what is already in place.

In addition, two lateral holes on each side of the part were projected, in order to restrict the lateral
movement of the beam, when in the desired position. The reasoning behind their location and dimension is presented in subsection 4.2.2. Pins with a radius of $7.5\text{mm}$ should be used for this restraining purpose.

An additional fact that must be stated with relation to the claw part is the resistance of the used pins. These must be made from a highly resistant material like a stainless steel which is able to endure the shear stresses in the pins. It was found that these stresses can reach values around $620\text{MPa}$, for the case of maximum loading case. This was obtained by integrating the shear reactions along the notches of the beam, obtaining the total shear force $V = 82.172\text{kN}$ and then using Equation 3.1.

Using the same logic applied in subsection 3.3.1 the material chosen for the pin in question should have a $\sigma_y$ of about $1116\text{MPa}$. Again, this requires a highly resistant material be used for these pins.

For the claw part the important characteristics and their relative weights are shown in Table 4.3. On this table $E$ is the Young’s modulus, $B$ is the bulk modulus of the material, $\sigma_c$ is its compressive strength, $\sigma_f$ is the fatigue resistance of the material, $UTS$ is the ultimate tensile strength, $J_{IC}$ is the material’s fracture toughness and $\rho$ is the material’s density.

<table>
<thead>
<tr>
<th></th>
<th>$E$</th>
<th>$B$</th>
<th>$\sigma_y$</th>
<th>$\sigma_c$</th>
<th>$\sigma_f$</th>
<th>$UTS$</th>
<th>$J_{IC}$</th>
<th>$\rho$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>10.714%</td>
<td>17.857%</td>
<td>17.857%</td>
<td>25%</td>
<td>10.714%</td>
<td>3.571%</td>
<td>7.143%</td>
<td>7.143%</td>
</tr>
</tbody>
</table>

Table 4.3: Obtained relative property weights for the claw part.
4.2.2 Beam

The beam is the central piece of the upper fixture assembly as it connects the sliders to the claw and also resists most of the loads present during testing. Its location may be seen in Figure 4.6 and a closer view is shown in Figure 4.7. It is obviously the larger part of those that make up the upper connection assembly. Its section was conceived so as to mate with both the upper and lower parts that connect to it and its dimensions were chosen so as to accommodate all the different landing gears which are tested on it.

When considering and conceiving the upper assembly of the connection tool a fundamental piece of information was the upper widths of the landing gear assemblies. Unfortunately, more often than not, the available manuals do not present the distance between the edges of the pintle pins (but present the distance between the holes where these pins are fitted). Additionally the diameters of the pintle pins were not provided. To circumvent this setback some measurements were required on the standard material existing in the landing gear shop. These components were designed specifically for their respective landing gears by the same company which produces them. These tools were the transport beams and the connection fixtures used for the nose landing gears, among others. Another useful source of information was provided by the available technical drawings for the existing tool used for the MLGs, as described in subsection 2.4.1.

Using these sources of information it was found that the length of the existing MLG beam is $850\text{mm}$. This value however does not match the total distance between pintle pins for either MLG. Indeed these values were determined to be $865\text{mm}$ for the E190 gear and $798\text{mm}$ for the E170 MLG. This disparity in values was found not to be problematic by employing a solution which presented the same distance between its edges and allowing the usage of the already existing fitting bushings. In fact these bushings balanced out the difference in width of both MLGs and allowed for a smaller gear to be tested using the same tool.

By measuring the connection fixture, the width of the NLGs was determined to be $859\text{mm}$. Due to the small difference between both upper tools it was decided to employ a similar solution to the existing and conceive a separate fitting bushing for the NLGs. This means that the beam would only exhibit two possible connection points for larger gears, thus reducing the existence of holes and their role in reducing the resistance of the beam.

In subsection 3.3.1 the transmitted loads by the landing gears to the connection fixture have been
computed. With these values it was possible then to conduct an FBD analysis of the beam and its reaction to the loading in question. Three parameters had to be determined beforehand, these are:

1. Length of the beam: chosen to be 900 mm;
2. Length where the load is applied: chosen as 45 mm, the same value of the suspension rings’ thickness for the original MLG upper fixture;
3. Positions of the claw: found by measuring the distance from the edge of the left pintle pin to the central axis of the sliding tube and from this axis to the central axis of the wheel axle.

The distances from the left edge of the beam to the central point of the claw were found to be:

- MLG E190: 182.4 mm;
- MLG E170: 213.7 mm;
- NLG: 450 mm - symmetric landing gears.

These analyses were conducted with a single fixed support on the central point of the claw part. These FBDs and their outputted SFDs and the BMDs can be seen in Appendix C. With both of these it is possible to compute both $\tau_{\text{max}}$ and $\sigma_{m,\text{max}}$ at a given point [18]. This $\tau_{\text{max}}$ is found differently than the one presented in Equation 3.1. The $\tau_{\text{max}}$ for the beam is given by

$$\tau_{\text{max}} = \frac{V \cdot Q_{i,\text{max}}}{I_{ii} \cdot t} \text{ [MPa];} \tag{4.3}$$

where $Q_{i,\text{max}}$ is the maximum first moment in mm$^3$ with respect to the neutral axis of the cross-section. This value is found when the point being considered is in the neutral axis of the cross-section. $I_{ii}$ is the centroidal moment of inertia of the entire cross-section about the same axis and $t$ is the thickness of the cross-section at the point where the stress is being calculated. The maximum bending stress $\sigma_{m,\text{max}}$, can be calculated using Equation 4.2.

With both these stresses the combined stress at a given point can be computed with Equation 4.4 [18]. $\sigma_{\text{max}}$ is the maximum stress that is computed by combining both the maximum bending stress $\sigma_{m,\text{max}}$ and the maximum shear stress $\tau_{\text{max}}$.

$$\sigma_{\text{max}} = \frac{\sigma_{m,\text{max}}}{2} + \sqrt{\left(\frac{\sigma_{m,\text{max}}}{2}\right)^2 + \tau_{\text{max}}^2} \text{ [MPa]} \tag{4.4}$$

To reach all these values it is necessary to inspect the cross-section of the beam. The final cross-section is shown in Figure 4.8, alongside some of its computed characteristic values. From this figure $I_{ZZ}$, $S_Z$ and $Q_Z$ are extracted to obtain the stresses mentioned above.

With the available information in the diagrams and Figure 4.8, the maximum stresses and displacements (see Appendix C) occurring in the beam were computed and found to be:

- MLG 190: 209.610 MPa and 1.125 mm;

\[\text{In Figure 4.8 } I_{ZZ} \text{ is referred as } I_Z\]
Figure 4.8: Dimensions of the beam’s cross-section along with some relevant properties.

- MLG 170: 215.545 MPa and 1.382 mm;
- NLG: 257.761 MPa and 0.810 mm.

For the fastening of the different parts to the main beam, a variety of solutions was adopted. For the sliders a pin connection was chosen and for both the claw part and the SA slider, notches mating with pins were considered. All these modifications to the geometry of the beam can be considered stress concentrators. Large stress gradients in small, localized areas of a structure, originate high stresses which are called stress concentrations. Around a change in geometry of a loaded structure, the stress-flow is interfered with, which leads to the appearance of high stress gradients where the maximum stress and strain can become larger than the average or nominal values based on simple calculations [27].

To evaluate and account for the influence of certain geometry changes in loaded components the static stress concentration factor $K_t$ is used. This value represents the relation between the maximum stress originated by a geometry change and the nominal stress, usually computed with simple relations and formulas. $K_t$ is obtained by [27]

$$K_t = \frac{\sigma_{\text{max}}}{\sigma_{\text{nom}}}.$$  

Some simplifications can be made regarding the loading conditions of the beam and the resulting stress concentrations originated by geometry changes. For example the two extreme holes on the web can be analysed as a “central circular hole in a member of rectangular cross-section” [27] whose loading scenario is “elastic stress, in-plane bending” [27]. With this in mind, the same can be applied to the notches on the flanges of the structure. These can be simplified as “one U-notch in a member of rectangular section” [27] with an “elastic stress, in-plane bending” [27] loading.

For the first case two different stresses can be computed. The first at the edge of the hole and the
second at the edge of the plate which is the maximum stress expected (generic point X). These stresses are computed as [27]

\[
\sigma_{\text{hole}} = K_t \cdot \frac{12M\phi/2}{t \left[ D^3 - (\phi)^3 \right]} \quad [\text{MPa}], \quad K_t = 2;
\]

and

\[
\sigma_X = K_t \cdot \frac{6MD}{t \left[ D^3 - (2\phi/2)^3 \right]} \quad [\text{MPa}], \quad K_t = 4\phi/2D.
\]

In these expressions \( M \) is the moment applied to the plate in \( N \cdot mm \), \( \phi \) is the hole’s diameter in \( mm \), \( t \) is the plate’s thickness in \( mm \) and \( D \) is the plate’s height in \( mm \).

For the notches \( \sigma_{\text{max}} = K_t \sigma_{\text{nom}} \) [27], where \( \sigma_{\text{nom}} \) is computed for each loading case. \( K_t \) must be computed from Equation 4.8 and Equation 4.10 through Equation 4.12 [27].

\[
K_t = C_1 + C_2 \left( \frac{h}{D} \right) + C_3 \left( \frac{h}{D} \right)^2 + C_4 \left( \frac{h}{D} \right)^3;
\]

\[
C_1 = 0.721 + 2.394\sqrt{h/r} - 0.127h/r;
\]

\[
C_2 = -0.426 - 8.827\sqrt{h/r} + 1.518h/r;
\]

\[
C_3 = 2.161 + 10.968\sqrt{h/r} - 2.455h/r;
\]

\[
C_4 = -1.456 - 4.535\sqrt{h/r} + 1.064h/r.
\]

Here, \( h \) is the notch’s depth and \( r \) is the notch’s radius. Since only ratios are used the units are irrelevant as long as they are the same for all different parameters. For the MLG and NLG notches the values of \( h/r \) and \( h/D \) are respectively 2.563 and 0.683. For the SA notch the value of \( h/r \) is 2.5 and \( h/D \) is 0.357 because the lower flange is thicker than the upper flange. The notches for the claw part can be determined to have a stress concentration factor of \( K_t = 1.352 \) and the stress concentration factor for the SA notch is \( K_t = 1.813 \).

Due to the proximity between the positions for the MLGs, the connection points between the claw and the beam were determined so that the positions for both MLGs had a point in common. The influence of this proximity of multiple notches on the stresses originated in the beam must be considered. The notches for the MLGs and NLGs are only distanced about 30 mm from each other. This distance is too small to assume their influences as separate.

With this in mind, the concentration factor for the MLG and NLG notches was determined to be \( K_t \approx 1.7 \). This value was extracted from the information available in [28] using the distances between notches as well as their dimensions. The parameters used were \( d/h \) and \( (D - r)/r \), where \( d \) is the distance between the centre of the notches\(^3\). These ratios were determined to be 1.463 and 2.75, respectively. From these values, the increased stresses generated by these geometry changes can be

\(^3\)In [28] presented these ratios are \( b/t \) and \( a/\rho \) respectively.
Table 4.4: Obtained relative property weights for the main beam part.

<p>| | | | | | | |</p>
<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$F$</td>
<td>$\sigma_y$</td>
<td>$\sigma_F$</td>
<td>$\sigma_f$</td>
<td>UTS</td>
<td>HV</td>
<td>$\rho$</td>
</tr>
</tbody>
</table>

computed. It is expected that the maximum stresses occurring in these locations are:

- Left hole: $\sigma_X = 59.288\,MPa$;
- Right hole: $\sigma_X = 35.517\,MPa$;
- MLG notches: $\sigma_{\text{max}} = 247.740\,MPa$;
- NLG notches: $\sigma_{\text{max}} = 318.588\,MPa$;
- SA notches: $\sigma_{\text{max}} = 339.765\,MPa$.

The stresses for the notches are the maximum stresses found during the whole analysis of the beam. Running linear static analyses for the different loading conditions it was found that the maximum stress occurring in this part was in the SA notch and of about $530.883\,MPa$. The maximum displacement was $1.217\,mm$, a rather small value in comparison with the beam’s dimensions.

When it comes to possible materials for this part, it is advised that it be produced in a high-strength material - in order to endure the high stresses during maximum loading. In Table 4.4 $F$ is the material's flexural modulus, $\sigma_F$ is its flexural strength and $HV$ the Vickers hardness of the material.

### 4.2.3 Sliders

There are two different types of sliders: one for the main and nose landing gears and another for the Shock Absorber. They are analysed separately.

#### MLG and NLG Slider

This part connects the pintle pins of the landing gears and the upper beam. Its dimensions were thought out in accordance with those of the upper beam. Also, these dimensions were selected so as to avoid any conflict between components of the landing gears and the upper fixture. The main components which had to be considered were the transport beams. Because of this there was a minimum allowable distance between the central axis of the pintle pin and the bottom face of the upper beam.

When designing this part, several linear static analyses were conducted with the purpose of minimizing the maximum stresses during loading. This process started off with a part whose dimensions were similar to the suspension ring of the existing MLG connection tool. Afterwards the thickness of the ring was increased until an optimum value. Then, the possibility of including holes to reduce the part’s weight, the introduction of ribs and widening the top contact area were all investigated. The resulting optimized design is shown in Figure 4.10. This process was conducted in order to determine whether a lighter material such as aluminium alloy could be used for this part. Since this kind of material is also less resistant it was necessary to first reduce the strength requirements.
Table 4.5: Obtained relative property weights for the MLG/NLG slider part.

<p>| | | | | | | |</p>
<table>
<thead>
<tr>
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<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$E$</td>
<td>$B$</td>
<td>$\sigma_y$</td>
<td>$\sigma_c$</td>
<td>$\sigma_f$</td>
<td>UTS</td>
<td>HV</td>
</tr>
<tr>
<td>7.143%</td>
<td>10.714%</td>
<td>17.857%</td>
<td>25%</td>
<td>7.143%</td>
<td>3.571%</td>
<td>10.714%</td>
</tr>
</tbody>
</table>

For the maximum loading scenario of $480kN$, which is unlikely to happen but allows to predict for unusual and extreme loading conditions, the maximum stress determined with a linear static analysis was $176.234MPa$. This value means that a material such as an aluminium alloy may be used in producing this part, as long as it has high strength characteristics, such as the 7075 T651 aluminium alloy. This is important because a significantly reduced weight for this part means an increased easiness and speed when assembling the upper fixture for testing. Additionally workers are prevented from having to endure large efforts in non-natural positions and thus the risk of injury is reduced. The maximum displacement found for the case when aluminium was used was $0.713mm$ which was reduced when analysing the part with steel $0.274mm$. Both values are below $1mm$ and non-determinant.

The connection of this part to the beam uses a removable pin. It was designed with a $6mm$ radius, the same for the current suspension ring. Under maximum loading ($480kN$), the pin must endure a $400MPa$ shear stress, obtained with Equation 3.1 and the integrated shear loads. If maximum service loads are considered ($395.539kN$), the shear stresses in the pin decrease to about $335MPa$. The first value should still be used, if possible, in order to safeguard from possible malfunctions and unexpected testing conditions. With the same reasoning as in subsection 3.3.1 the material chosen for the pin in question should have a $\sigma_y$ of about $720MPa$. However, as long as the yield stress is above $603MPa$ the pin should be able to safely endure the loading conditions.

The determined relative weights for the relevant material properties are presented in Table 4.5.

This part is connected to an under slider by pins, a connection part, a rod end bolt and a screw cap. These components only support the suspended weight. The fitting bushings must also be included when it is necessary to account for the difference between diameters.

**Shock Absorber Slider**

This part was conceived to serve the same purpose as the previous one. However, neither the shape, the size or even the connection method for this part are the same as those used in the other type of slider.

The developed part has two holes in its inferior which match the bolts used in connecting the upper tool of the SA. In this way the available equipment can be used and waste of material and tools is prevented. This is possible due to the small dimensions of the SA’s connection tools. Additionally the
lateral movement restriction is done through vertical pins with a radius of $5\text{mm}$.

It was found that the maximum stresses for this part were $126.777\, MPa$ and the maximum displacement $3.520 \times 10^{-2}\, mm << 0$ (steel was considered). It must be stated however, that this loading is highly unlikely to occur when the SA is being tested. The used value of $480\, kN$ is quite larger than the maximum allowable value during testing of $220\, kN$. The pin used to secure the slider to the beam is expected to withstand a maximum shear stress of about $11\, MPa$, using Equation 3.1. The lateral movement of the part in question can be easily restrained. For this part the relative weights for each relevant material property may be seen in Table 4.6.

<table>
<thead>
<tr>
<th>$B$</th>
<th>$\sigma_y$</th>
<th>$\sigma_c$</th>
<th>$\sigma_f$</th>
<th>UTS</th>
<th>HV</th>
<th>$\rho$</th>
</tr>
</thead>
<tbody>
<tr>
<td>14.286%</td>
<td>9.524%</td>
<td>23.810%</td>
<td>9.524%</td>
<td>19.048%</td>
<td>19.048%</td>
<td>4.762%</td>
</tr>
</tbody>
</table>

4.2.4 Under Slider

This part was conceived from the original under-ring part of the MLG tool. Its thickness was chosen so as to match the thickness of the slider. It is a relatively simple part which should be produced in the same material as the MLG/NLG slider. The lighter this part, the easier it is to fasten the pintle pins of the landing gears to the connection tool.

The stresses in the under slider are only expected to be bearing and contact stresses. The first ones occurring in its inner surface and the holes where pins would be installed, and the second ones occurring in the surface below the fastening knob used to secure the connection between upper and under sliders.

When loading conditions were simulated in a linear static analysis, using the weight value of $7.063\, kN$,
computed in subsection 3.3.1 it was found that the maximum stresses for this part did not surpass 68 MPa in the support area, a low value in accordance with the low loading case in question. However, in the area in contact with the knob the maximum stresses observed reached 187.257 MPa. Still, these values are not as high as the ones predicted for other parts. Considering aluminium as the material that makes up this part, its maximum displacements were found to be 0.389 mm. Lower than 1 and therefore non-determinant.

4.2.5 Pins and Rod-end Bolt

These pins and their dimensions were selected similarly to how the pins in subsection 3.3.1 were chosen. However, concerns about the resistance of the parts were reduced because the "feet" of the developed sliders were made thicker than those of the original suspension rings. This means that larger holes can be drilled into these details of the developed parts and still retain a certain level of confidence that the part will not fail.

These pins were chosen to be 10 mm diameter steel pins, to safely endure the loads and the stresses in play during testing. The maximum stress in these pins is the same as those in subsection 3.3.1.

Also similarly to the solution devised in the previous chapter, the chosen bolt is also a partially threaded M10 rod end bolt. A threaded steel knob is also intended. The reasoning behind the choice of parts is the same as the one used for the renewed suspension ring in subsection 3.3.1.

4.2.6 Pin Connecting Part

This part, while similar to the one developed in the previous chapter, is slightly different. The main differences are the dimensions of the holes which mate with the pins and the overall length of the part. Because the sliders developed in the current chapter are thicker than the suspension rings which make up the MLG connection fixture, the connection part between pins must be longer to accommodate this change.

From a purely theoretical perspective it could be predicted that the bearing stress would be the determinant value in play during loading. Using the load acting on one slider as $7.063 kN$ and the dimensions of the hole of this connecting part, the bearing stress found was $\sigma_b = \frac{R}{\pi d^2} = 36 MPa$ [18], with $R$ the load in $N$. With a linear static analysis the maximum predicted stresses were 140.078 MPa, a slightly higher value but still small. The maximum displacement of this part, if it is made of steel, is
The results of the material digital logic for the connecting part are shown in Table 4.7.

\begin{table}[h]
\centering
\begin{tabular}{|c|c|c|c|c|c|c|}
\hline
$E$ & $\sigma_y$ & $\sigma_c$ & $\sigma_f$ & UTS & HV & $\rho$ \\
\hline
\hline
\end{tabular}
\caption{Obtained relative property weights for the pin connecting part.}
\end{table}

\section{4.2.7 Fitting Bushings}

These parts are intended to adapt the smaller pintle pins into the sliders. While the pintle pins of the E190 MLG have an outer radius of $112.5\text{mm}$, the MLG of the E170 aircraft has a $79.75\text{mm}$ pintle pin radius\footnote{For both E190 and E170 MLGs, the presented outer radii concern those of the transport beams. Since it is intended that the transport beam is installed during testing, their outer dimensions were the considered ones.} and the NLGs have a $40\text{mm}$ pintle pin radius. Additionally, there is a disparity in widths for the various landing gears, as was addressed in subsection 4.2.2.

These bushings are intended therefore to eliminate these differences and allow for the different gears to be assembled on the same fixture. While the bushing for the E170 was already available, the bushing for the NLGs had to be developed. Their cross-section is visible in Figure 4.23. By running a linear static analysis on this part, using the maximum expected load of $236.961\text{kN}$ which is the maximum for the NLG testing procedure, it was found that the maximum stress was around $120.813\text{MPa}$. In the case that this part is produced in steel, its maximum displacements are $1.240 \times 10^{-2}\text{mm}$, this value is quite small and can be neglected.
Table 4.8: Obtained relative property weights for the NLG fitting bushing.

<table>
<thead>
<tr>
<th>$B$</th>
<th>$\sigma_y$</th>
<th>$\sigma_c$</th>
<th>$\sigma_f$</th>
<th>UTS</th>
<th>HV</th>
<th>$\rho$</th>
</tr>
</thead>
</table>

Figure 4.21: Location of the NLG bushing part.

Figure 4.22: Closer view of the NLG bushing part.

Figure 4.23: Cross-section of the NLG fitting bushing part.

For this part, in terms of material, it is only recommended that a hard material be used or surface treatments be applied. Since the maximum stress is low and below most materials’ yield strength or fatigue strength, the mechanical characteristics of the material are non-determinant. The superficial hardness however, is a factor to take into account because of the possibility of impacts or collisions during set-up and removal operations. A harder outer surface protects the part and avoids damage should any of these harmful interactions take place. The relative weights for relevant material properties were obtained for the NLG bushing and can be seen in Table 4.8.

### 4.3 Lower Fixture

The design which was ultimately chosen for the bottom assembly includes (see Figure 4.24):

1. Piston connection;
2. Inferior shaft;
3. Thrust bearing;
4. Support body;

5. 3 different types of sliders, which give a total of 5 sliders.

One differentiating characteristic of the bottom fixture is related with material selection. Since this part is not suspended and is fixed to the bottom piston of the test bench, its overall weight is non-determinant. Therefore, density plays a smaller role in material choice than it played for the upper fixture’s parts. This factor is involved in material suggestions for this assembly.

![Figure 4.24: Schematic 3D model of the final lower fixture.](image)

This assembly has however, one additional constraint not referred to in the upper fixture - the assembly’s total height. The upper fixture was designed to accommodate the transport beams of the MLGs, during set-up and testing. Its "height" (distance from the central axis of the pintle pin to the connection point of the upper board) is a consequence of this factor. As a consequence, the support assembly becomes constrained in its maximum "height" (distance from the lower piston to the central axis of the wheel axle), and parts had to be adjusted to fit this need. Thankfully this constraint is only involved in the MLGs. Taking the height of this assembly, the maximum height of the connection assembly, considering the E190 MLG case, is determined to be 904 mm, equal to the maximum value allowed.

### 4.3.1 Piston Connection

![Figure 4.25: Location of the piston connection.](image)  ![Figure 4.26: Closer view of the part.](image)

This part is connected to the piston of the test bench. Its purpose is to transmit the forces to the remaining lower fixture, which in turn transmits them to the landing gear. The part has one central hole which is a through hole with the same dimensions as the hole (and bolt) used for the piston connection.
Table 4.9: Obtained relative property weights for the piston connection part.

<table>
<thead>
<tr>
<th>Property</th>
<th>Relative Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E$</td>
<td>14.286%</td>
</tr>
<tr>
<td>$G$</td>
<td>10.714%</td>
</tr>
<tr>
<td>$\sigma_y$</td>
<td>14.286%</td>
</tr>
<tr>
<td>$\sigma_c$</td>
<td>21.429%</td>
</tr>
<tr>
<td>$\sigma_t$</td>
<td>14.286%</td>
</tr>
<tr>
<td>UTS</td>
<td>7.143%</td>
</tr>
<tr>
<td>HV</td>
<td>14.286%</td>
</tr>
<tr>
<td>$\rho$</td>
<td>3.571%</td>
</tr>
</tbody>
</table>

With the introduction of this part the connection to the piston would need a bolt with a total length $(L + h)$ of at least 76\text{mm} with a threaded length $(L_T)$ of at least 26\text{mm}. Its height was chosen so as to not conflict with the piston protections and avoid the need for their removal. It must be stated that a reduction of this part’s height would mean the reduction of the total height of the connection assembly. This may happen if it is decided that the piston protections do not need to be installed. Additionally it has four holes in its upper face which connect it to the inferior shaft. These holes are intended to house quick change pins, such as the ones presented in subsection 3.3.1.

Considering a maximum torque of $300 N \cdot m$ the maximum shear stress these pins have to withstand is about 21 MPa. This value for the torque was chosen based on the maximum resistance torque allowed. The maximum resistance torque to the rotation of the gear’s sliding tube and wheel axis, is specified at $201 N \cdot m$ - [6] [7]. From this value a safety factor of 1.5 was applied and the value of $300 N \cdot m$ obtained. With this torque the same steps as in subsection 3.3.1 were applied and a shear force of 1250 N per pin found, since the distance between the pins and the centre of the part is $60\text{mm}$. This results in a shear stress of 21.22\text{MPa} per pin, using Equation 3.1. Accordingly the used pins should have at least $\sigma_y = 37.8\text{MPa}$.

When maximum loading is in play and steel as the considered material, this part exhibits a maximum stress of around 139.11\text{MPa} and a maximum displacement of $2.710 \times 10^{-2}\text{mm}$, a non-determinant value. This value was obtained through a linear static analysis of this part. The relative weights for the properties relevant for this part can be seen in Table 4.9, where $G$ is the material’s shear modulus.

### 4.3.2 Inferior Shaft

![Figure 4.27: Location of the interior shaft part.](image)

![Figure 4.28: Closer view of the interior shaft part.](image)

This part has the inverted shape of the previous part. Its upper shaft is to be installed inside the thrust bearing (subsection 4.3.3). This part also transmits the loads from the test bench to the base support part of the lower assembly. It is connected to the inferior part through the already referred pins. The maximum stresses present are about $278.31\text{MPa}$ and the maximum displacements $2.150 \times 10^{-2}\text{mm} << 0$. The results of the material digital logic conducted for this part are shown in Table 4.10.
Table 4.10: Obtained relative property weights for the inferior shaft.

<table>
<thead>
<tr>
<th></th>
<th>B</th>
<th>G</th>
<th>σ_y</th>
<th>σ_c</th>
<th>σ_f</th>
<th>UTS</th>
<th>HV</th>
<th>ρ</th>
</tr>
</thead>
</table>

4.3.3 Thrust Bearing

A thrust cylinder bearing was chosen based on force requirements required during the testing, the angular velocity and the number of rotations the support carries out throughout their life. It was selected from an on-line catalogue [29]. A cylinder bearing is recommended because it withstands better the axial loads at play over its lifetime. The clearances must be such that the interior shaft is axially constrained and the bearing does not have an axial translational d.o.f. inside the lower cylinder of the base of the support. The dimensions presented for these parts are already in accordance with the selected thrust bearing.

For an intermittently operating bearing, where reliability is important, a Basic Rating Life ($L_{10}$) of about 12000 is recommended [30]. This means that there is a 90% probability that the bearing will only fail after 12000 hours. Considering a maximum torque of 300$N\cdot m$, as well as a maximum speed of about 10$RPM$ during testing and a axial maximum load of 480$kN$, the minimum Dynamic Load Rating ($C_r$) was computed and found to be 683.436$kN$ [30].

In some cases the Static Load Rating ($C_0$) may be used. This happens when the bearing in question rotates at a low-speed, with small oscillations or even when they are stationary under load. A low-speed bearing is usually considered when its rotational velocity does not exceed 10$RPM$ [31]. It is then possible to use the $C_0$, rather than, the $C_r$ constant when choosing a bearing. This is advantageous because $C_0$ is always higher than $C_r$ for a given bearing. Hence a smaller bearing could be chosen. The characteristics and dimensions for the chosen bearing are found in Table 4.11 and Figure 4.31.

Table 4.11: Thrust bearing characteristics [29].

<table>
<thead>
<tr>
<th>φd</th>
<th>φD</th>
<th>T</th>
<th>$C_0$</th>
<th>$C_r$</th>
</tr>
</thead>
<tbody>
<tr>
<td>85mm</td>
<td>150mm</td>
<td>39mm</td>
<td>995$kN$</td>
<td>257$kN$</td>
</tr>
</tbody>
</table>
4.3.4 Support Body

This part has a similar function to the beam of the upper fixture. Accordingly, its dimensions were determined so as to withstand the bending moments and shear forces which occur during testing. The increase in stresses originated by the holes on the part’s sides was also taken into consideration. Besides being a lipped channel section beam with side holes, the part also has a lower cylinder which serves as an outer fitting for the bearing described above. The dimensions of the channel section which makes up this part are shown in Figure 4.34.
In this part the loading is more simple than with the beam section. Because of the symmetry of
the wheel axle the applied load is distributed equally to this component and the transmitted loads are
equal for both sliders, so an equilibrium analysis is not needed. Using FBD analyses and the proce-
dure in subsection 4.2.2 the maximum expected stresses occurred for the MLG loading case and were
110.355MPa, the maximum displacements were obtained for the same case and are 0.237mm, well
below 1mm (Appendix C). The length chosen for this part was 800mm.

The sliders which will serve as contact points for the wheel axles, are to be installed similarly to the
upper fixture's sliders. However, for the upper tool the fixation was made using a pin which went through
both the slider and the beam. For the support the sliders are retained using four different steel pins,
two on each side. Their diameter was designed to be 8mm and their length should be half of the lower
thickness of the slider parts. Having obtained a total shear load of 17.917kN the maximum shear stress
found for pins with half the total length is 237.592MPa, following the same logic as in previous sections.

If the pins are chosen so that only two pins for each slider are needed, then the shear stress they
would have to withstand is 475.184MPa. This requires a much stronger type of material for the pins. For
the former case it is then advised that the pins be made of a material which has a $\sigma_y \approx 427.666MPa$,
following the same logic presented in previous parts.

The holes introduced in this part act as stress concentrators, similar to those of the upper fixture's
beam. Following the same analysis undertaken in Figure 4.8 and the same equations, the maximum
stress created by these holes was found as 208.213MPa. Besides these lateral holes which fasten the
support sliders, the support body also has a central threaded hole. This hole is for the installation of the
SA slider. Its dimensions are the same as the central hole in the current bottom connection fixture of the
SA at OGMA.

Running a linear static analysis for the worst case scenario (MLG loading case) it was found that
the maximum stress in this part was 542.232MPa and the maximum displacement 0.276mm, well below
1mm. This value of maximum stress is addressed in Appendix D.

Because of the high expected stresses this part is recommended to be produced in steel. Besides
presenting greater mechanical resistances, steels are also desirable for the part in question because
their elevated flexural modulus translates into smaller deflections at the edges of the part, during testing.
This minimizes the possibility of appearance of non-vertical load components and hence the reactions
the fastening pins must withstand. The relative weights table coincides with the one presented in sub-
section 4.2.2, Table 4.4.
4.3.5 Sliders

MLG and NLG Sliders

These parts were designed with the same purpose and logic as the parts in subsection 4.2.4. The difference is the variety in parts and positions, which the beam assembly does not present. There are three different types of support assembly sliders - one for MLGs, one for NLGs and one for the SA. The two first types were designed so that the bottom part of their volume coincided with the empty space of the support body’s channel section. The SA slider was produced with a similar goal as the upper one, in subsection 4.2.4.

For the larger gears’ sliders their diameters and dimensions were extracted from the already existing tools, so as to adapt seamlessly to the set-up. The MLG slider has the same holes as the current support fixture, which allow the sacrificial bronze parts to be installed. The upper diameter of the NLG part was kept equal as the one of the standard support part (see Figure 2.10), to accommodate the wheel axle protections and their external dimensions.

The maximum stresses found for a critical stress situation were $129.934 \text{ MPa}$ for the MLG slider part. For the NLG the maximum stresses were $133.934 \text{ MPa}$. The maximum displacements obtained (using aluminium) were $3.980 \times 10^{-2} \text{ mm}$ and $8.700 \times 10^{-2} \text{ mm}$, respectively. With these values, and knowing that the stresses are mostly compressive, it would be possible to consider a lighter material, such as aluminium, to produce these parts, which would aid in set-up procedures. This possibility is particularly interesting for the MLG slider, as there is no need to protect it against impacts or other contact defects, thanks to the sacrificial bronze components. For the NLG slider a similar solution is possible since this
part is in contact with the wheel axle protections. Regardless of this, a relative weights table was also
devised for these parts - Table 4.12.

<p>| | | | | |</p>
<table>
<thead>
<tr>
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<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>B</td>
<td>$\sigma_y$</td>
<td>$\sigma_c$</td>
<td>$\sigma_{f}$</td>
</tr>
<tr>
<td>7.143%</td>
<td>14.286%</td>
<td>17.857%</td>
<td>25%</td>
<td>10.714%</td>
</tr>
</tbody>
</table>

Table 4.12: Obtained relative property weights for the MLG and NLG slider parts.

Shock Absorber Slider

The SA slider part was conceived with a threaded through hole which works to fasten both the lower
fixture of the SA to the slider and the slider to the support body. Its thickness was chosen to allow
the set-up of this lower fixture without creating interference or contact issues with the inner surface of
the channel section of the support body. For this fastening of the three parts a bolt of approximately
$(L + h) = 130\,\text{mm}$ is needed. It need not be fully threaded, as long as its thread is in contact with the
thread in the slider part and support part.

For this part the material digital logic table can be taken as the same for the upper SA slider (Ta-
ble 4.6). However, the Shock Absorber slider part should be manufactured in steel, so as to increase
the resistance of the threaded fastener. For this part the maximum stresses found were $82.306\,\text{MPa}$ and
the displacements $6.638 \times 10^{-3}\,\text{mm}$ - residual values that can be neglected.
Chapter 5

Concluding Remarks

In this final section three points are in order. Firstly the results for the processes exposed in section 3.1 are presented. These are simulated results which take into account the improvements proposed in section 3.3. To these times an NVA time of 5% is added in order to account for unexpected situations and possible unplanned delays. Afterwards the main achievements of the present work are detailed. Finally some suggestions to possible future work are introduced.

5.1 Results

Table 5.1 displays the computed results for the internal set-up times. These times include all the proposed improvements, except for the universal beam, the change in the oblique part of the MLG E190 transport car and the change of entrance location for the test area. The first is not considered because its influence is felt mainly in the streamlining of external operations. The second is excluded because this correction must be studied and analysed further before implementing any solution of its sort. If this is not studied and designed correctly the problems may appear during either assembly of set-up processes. For example if the connection point is not designed correctly it may break when the landing gear is rotating or it may let the landing gear slip and fall down. Even though this is a solution which, if implemented, represents a reduction of internal time of about 2 minutes, its analysis would be prohibitive considering the available time during which the present work was developed. Changing of the test area entrance is not considered because this is a lengthy and complex operation which requires the input of managerial structures and positions, since it involves the changing of a working shop. Its effects are also mainly related to transport operations and therefore its influence does not factor into the predicted times.

In Table 5.1 the values were computed using the simulated results and the measured times presented in the tables in section 3.1. The greatest time gain occurs for the E190 MLG set-up activity. This is also the process where the effect of the proposed changes is expected to be felt more strongly, namely in what relates with connecting the landing gear to the connection fixture. The time reduction involved in the transport car for the support part is also considerable.

The time reduction in external activities is less determinant than the previous ones due to the fact that the former can be processed while the landing gears are being worked on. Still, some reductions of up to 60% are expected, namely in the time spent in tool changeovers. Less time fastening bolts and performing adjustments will drive these external times down. Ultimately the implementation of the universal connection fixture will translate itself in a further reduction of these changeover times.
Table 5.1: Results for the internal times.

<table>
<thead>
<tr>
<th>Process</th>
<th>Time [min:sec]</th>
<th>Overall Time Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Set-up MLG E190/E170</td>
<td>08:17</td>
<td>90%</td>
</tr>
<tr>
<td>Set-up NLG E190/E170</td>
<td>11:28</td>
<td>54%</td>
</tr>
<tr>
<td>Set-up SA</td>
<td>7:45</td>
<td>38%</td>
</tr>
<tr>
<td>Removal MLG E190/E170</td>
<td>7:14</td>
<td>70%</td>
</tr>
<tr>
<td>Removal NLG E190/E170</td>
<td>21:58</td>
<td>60%</td>
</tr>
<tr>
<td>Removal SA</td>
<td>5:36</td>
<td>44%</td>
</tr>
</tbody>
</table>

NLG times are the hardest to predict since the effect of the new transport car on set-up and removal processes is difficult to quantify. It depends on whether the support part can remain on the test bench when removing the landing gear, how long the operation of strapping the sliding tube to the car will take and the mobility of this very tool. Because of this a more conservative approach to time reduction was preferred so as to avoid over-stating the effect this change may have on the process at hand.

It is also important to refer that the duration of the E170 processes are mere estimates which lack observation in order for these results to be more reliable. Unfortunately, as already mentioned, no E170 ship-sets gave entrance into OGMA’s landing gear shop and therefore no set-up or removal sequences were observed.

The NVA times is also a parameter which is mainly expected to decrease with the practical applications of visual standards and the reduction of the number of tools needed. The elimination of support parts and components, such as the pallet-carrier and the support wedges, helps reduce time which is mainly wasted in reallocating components or searching for the needed parts to finish the set-up or removal processes.

5.2 Achievements

The present work set out to implement changes and solutions to established set-up processes that would cut down on the time wasted in these processes. In fact, comparing the medium to long term expected times with the initial extracted times the reduction amounts to around 90%. The main goal is to reduce the bottleneck created by the variability in durations in the set-up processes.

The variations in how long it takes to connect a landing gear to the hydraulic test bench have been smoothed out with the introduction of changes such as the reduction of bolts and respective tools used, the standardization of the process, corrections applied to current tools, among others. The acquisition of new tools is also a step which can further improve the set-up times for the considered processes.

The work which led to the present master’s thesis wasn’t however, successful in applying SMED to all processes, namely the removal activities. Additionally it must be recognized that the emphasis put into the conversion of internal activities into external activities might have been stronger. This comes as a result of the multidisciplinary nature involved in the completion of the proposed objectives for the internship which originated the present thesis. Further insight into the possibilities of conversion of these activities is advised, specially by a trained eye which is able to find solutions where they may seem to be absent. This goes in line with the continuous improvement mentality associated both to lean production systems and SMED. Nonetheless, considerable time gains can be achieved from the implementation of
the proposed changes to activities.

The present work was also successful in developing a design for an universal connection fixture. This tool and its upper and lower assemblies may allow for a cut-down in both external and internal times. The external times are related to the tool changeover periods who are all but eliminated with the introduction of this tool. Instead of unfastening and removing assemblies which weigh around 60 kg, it is enough to remove two or three pins and readjust the position of the components according to the intended testing procedure. This frees up time and resources which can be utilized in assembly activities or other continuous improvement process.

The development of the connection tool allowed for an insight into mechanical design and the need to both dimension and devise all the different components involved in the chosen concept. The work developed includes, among others,

- Bolt dimensioning;
- Bearing dimensioning;
- Loading and stresses;
- Optimization;
- Material selection;
- Finite element analysis.

Ultimately the work developed provides OGMA considerable insight into what is expected to happen when the developed solution is implemented in the test area and the landing gear maintenance process. Most of the possible failure cases were analysed and the major factors at play during testing were considered, to create a tool as complete as possible.

5.3 Future Work

There are some possibilities for future improvement which can be addressed. The factors related to the continuing improvement of the processes at hand should be contemplated. Namely the possibility of further conversion of internal activities into external activities is a point of interest for OGMA, in order to reduce the set-up times in the landing gear shop. The introduction of new structures such as the modified transport cars for the E190 MLG and some spatial changes such as changing the entrance to the test area, seem clearly relevant, advising a complete analysis of the applicability of the presented possibilities. This should be done by professionals within the organisation.

The analysis of the manufacturability of the developed connection solution should also be performed. This includes both the final selections of materials, which includes cost factors not considered in the work described, and how to produce and manufacture the different components. This final step is of paramount importance to ensure that both superficial and mechanical characteristics of the used materials remain viable and relevant for the intended applications. Additionally it is recommended that
the connection solution between the beam and the claw parts for the upper assembly be revised. This with the intent to reduce the stresses at play during testing, namely in the securing pins which, as seen in subsection 4.2.1, require a yield stress of about 1100 MPa. This may not be easy to obtain. Once this is completed, the manufacturing of the devised tool can begin and the universal solution implemented.

One other area where OGMA may gain time in landing gear testing is the testing procedures themselves. This does not relate to the present work but should be mentioned nonetheless. It was observed at OGMA that the average length of the testing period for MLGs and NLGs was of about 16 hours. Taking the simulated set-up and removal times it can be seen that these amount to, in the worst case scenario, 2.2% of the total lead time of the testing process. The set-up processes represent therefore a fraction of the total time that a landing gear spends at the hydraulic test bench.

This means that tackling the testing process is of paramount importance in the reduction of the landing gear maintenance lead time. A dedicated analysis of the testing procedures and the durations of the different steps during the overall time spent by the landing gear assemblies in the hydraulic test bench, would optimally result in a lead time reduction. The consequences of this would be an increased capacity for OGMA at the test bench and ultimately increased productivity and profit.

Overall, the proposed achievements and objectives for this master’s thesis were met. In addition, it seems fair to conclude that new opportunities for further improvements, within landing gear maintenance processes at OGMA, have also been suggested in the course of the present work.
Bibliography


80


Appendix A

SkyCiv Software Validation

SkyCiv is an Australian based software company devoted to developing structural analysis software on the cloud. It provides subscribers and free users with 6 different kinds of application directed mainly to civil and mechanical engineering problems. These include beam, truss and shaft calculators, connection builder as well as section builder and design check software. All of these tools are available on a user’s dashboard, as seen in Figure A.1. SkyCiv additionally makes resources like section library or tutorial examples available to its subscribers. After a solution it is also possible to generate a custom report with the information obtained from one of its tools. SkyCiv is entirely cloud based, this includes both codes and applications as well as user databases.

In this appendix the validation of the tools available from SkyCiv will be shown in the perspective of the developed work. Namely the section builder, the beam calculator and the structural 3D application. This last tool was used for modelling and design of the support cars for both the MLG and NLG support parts. For the section builder a known beam section will be constructed and its computed properties will be compared to the section’s known values. The beam calculator and its results will be compared against two known solutions found in [18].

For the 3D structural tool, [32] includes 9 different verification models. Each one of them presents an image of the developed structure, the main outputs to validate and the error generated from comparing the tool’s analysis results and those of a third party software. This documentation is publicly available and contains detailed information regarding each one of the validation models. Hence, there is no further need to check whether the results extracted from this tool may be used.

![Figure A.1: User dashboard of SkyCiv.](image-url)
A.1 Section Builder

The beam section selected was the 127x76x13 serial size section from [33]. Its dimensions and some of its properties are shown in Table A.1. The section is presented in Figure A.2.

The developed section and its computed properties using the SkyCiv tool are presented in Figure A.3. It must be noted that the $X$ axis in Figure A.2 corresponds to the $X$ axis in Figure A.3, therefore the properties which refer to the $Z$ axis in Figure A.3 will be presented and evaluated as if it were the $X$ axis, similar to what happens in Figure A.2. The error between the tabulated values and the calculated ones may be seen in Table A.1.

As can be seen the presented errors are quite low but a possible cause for their appearance is the

Table A.1: Section properties for 127x76x13 serial size section [33] and computed errors.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value [units]</th>
<th>Property</th>
<th>Value [units]</th>
<th>Property</th>
<th>Error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>127 [mm]</td>
<td>A</td>
<td>16.5e2 [mm²]</td>
<td>A</td>
<td>0.0843</td>
</tr>
<tr>
<td>B</td>
<td>76 [mm]</td>
<td>$I_{XX}$</td>
<td>473e4 [mm⁴]</td>
<td>$I_{XY}$</td>
<td>3.165</td>
</tr>
<tr>
<td>t</td>
<td>4 [mm]</td>
<td>$I_{YY}$</td>
<td>56e4 [mm⁴]</td>
<td>$I_{YY}$</td>
<td>0.441</td>
</tr>
<tr>
<td>T</td>
<td>7.6 [mm]</td>
<td>$S_X$</td>
<td>75e3 [mm³]</td>
<td>$S_X$</td>
<td>2.287</td>
</tr>
<tr>
<td>r</td>
<td>7.6 [mm]</td>
<td>$S_Y$</td>
<td>15e3 [mm³]</td>
<td>$S_Z$</td>
<td>2.187</td>
</tr>
</tbody>
</table>
numerical error originated when computing the values of the variables of interest. Another possible error is the rounding off of values presented for the selected beam section, for simplicity reasons. Nevertheless, the section builder tool also presents the possibility to analyse all the calculations done to reach each value, which meets the formulations used for each property. Because of all this it is possible to state that the section builder tool and its value for different properties are valid and can be used in the present work.

A.2 Beam Calculator

![Figure A.4: The beam in question and obtained diagrams.](image)

![Figure A.5: Diagrams obtained with SkyCiv's beam calculator tool.](image)

To validate the outputs of this tool, two sample exercises were chosen from [18]: sample problem 5.4 and sample problem 9.1. These deal with shear force diagrams, bending moment diagrams, bending stresses and beam deflection. For the first problem the relevant parameters regarding the beam and the loading conditions are:

- Section: W360x79 standard of the Canadian Institute of Steel Construction (CISC), available in SkyCiv's section database;
- Beam length: 9 m;
- Supports: simply supported with pinned support at \( x = 0 \) m and roller support at \( x = 9 \) m;
- Loading: 20 kN/m distributed load from \( x = 0 \) m to \( x = 6 \) m;
- Reactions: left reaction - \( L_1 = 80 \) kN, right reaction - \( L_2 = 40 \) kN;
Maximums: shear force - $V_{max} = 80kN$; bending moment - $M_{max} = 160kN \cdot m$; bending stress - $\sigma_{max} = 126MPa$.

The defined free body diagram and the shear force and bending moment diagrams obtained may be seen in Figure A.4.

The free body diagram for the problem at hand, with the computed reactions and the diagrams obtained in the SkyCiv beam calculator application can be seen in Figure A.5. With this tool $\sigma_{max} = 125.31MPa$, which translates into an error of $0.548\%$. This error is very close to zero which helps to confirm the validity of the tool used, for the purpose at hand.

Afterwards the sample problem 9.1 was analysed, in order to verify whether the beam calculator tool computes displacements correctly. The parameters for this problem are:

- Section: W14x68 standard of the American Institute of Steel Construction (AISC), available in SkyCiv’s database;
- Material characteristics: $E = 29e6psi$;
- Beam length: $228in.$;
- Supports: simply supported with pinned support at $x = 0in.$ and roller support at $x = 180in.$;
- Loading: $50kips$ concentrated load at $x = 228in.$;
- Reactions: left reaction - $L_1 = -13.333kips$, right reaction - $L_2 = 63.333kips$;
- Maximum: positive deflection - $y_{max} = 0.238in.$, negative deflection - $y_{min} = 0.418in.$.

Generating the same free body diagram and solving the system, the maximum positive deflection obtained was $0.238in.$ and the maximum negative deflection was $-0.418in$. These values are exactly the ones intended, meaning that the tool used is also valid for deflection determination. The results of the analysis may be seen in Figure A.6.
Appendix B

Visual Operation Standard Example

Below an example of a possible operation standard is presented. It concerns the set-up process of the E190 MLG and some of its details. These are the removal of the right suspension ring, the rotation of the transport beam to the correct side and the removal of the piston protections. This operation standard was developed making use of the template provided by OGMA for the very same purpose.

In the presented example the importance of visual cues. Clear images accompanied by simple descriptions are more useful and intelligible than only written commands or poorly selected images. The standards developed should strive to keep this kind of presentation, without including excessive amounts of information for each action. It is preferable to have more documents with fewer actions described in each document than a smaller number of standards where the relevant actions are unclear or cluttered together in an unintelligible manner. Ideally the presented standard would exhibit only two actions, it was chosen to have three to have more examples to show in the present work.

The developed standard is in English to match the language in which this document was written.
<table>
<thead>
<tr>
<th>Nº</th>
<th>ACTION (WHAT TO DO AND HOW)</th>
<th>KNOW HOW (QUALITY POINTS)</th>
<th>MACHINE / TOOLS / TEMPLATE</th>
<th>VISUAL CUES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Centre landing gear</td>
<td>Remove right suspension ring (red) and keep left suspension ring (yellow) before moving landing gear to test area</td>
<td>KRATOS Test Bench</td>
<td>![Image]</td>
</tr>
<tr>
<td>2</td>
<td>Transport beam position</td>
<td>Opposite side to the connection point (blue)</td>
<td>KRATOS Test Bench</td>
<td>![Image]</td>
</tr>
<tr>
<td>3</td>
<td>Piston protections</td>
<td>Remove white piston protections before moving landing gear to test area</td>
<td>KRATOS Test Bench</td>
<td>![Image]</td>
</tr>
</tbody>
</table>
Appendix C

Relevant FBDs

In this appendix the relevant developed FBDs will be presented, along with the outputted SFDs and BMDs. Additionally the load and bending moment reactions will be shown for every case. The cases of loading with the SA set-up on the press was not considered because this is a situation where the loading point coincided with the support point. Hence the load reaction will be exactly the same as the applied load and the bending moment reaction will be zero.

Each section contains the loads and support cases, the reactions, the SFD, the BMD and the displacement diagram. For the pins the first 4 figures are common since these do not depend on the section of the pin, only the displacement diagram differs. The loads for the beam were determined in subsection 3.3.1 and the distributed loads are the loads for each case distributed over $45\text{mm}$, the thickness of the slider. For the support body the load applied is $480\text{kN}$ and distributed over the length of the slider for each case.

All the obtained displacements in the following diagrams were obtained by considering that the beams were made of steel. A material such as aluminium would present greater displacements.

C.1 8mm and 10mm Pins

Figure C.1: Loads and Support for the pins.

Figure C.2: Reactions for the pins.
C.2 E190 MLG Beam Loading

Figure C.7: Loads and Support for the E190 MLG.

Figure C.8: Reactions for the E190 MLG.
C.3 E170 MLG Beam Loading
Figure C.13: Reactions for the E170 MLG.

Figure C.14: SFD for the E170 MLG.

Figure C.15: BMD for the E170 MLG.

Figure C.16: Displacement for the E170 MLG.
C.4 NLG Beam Loading

Figure C.17: Loads and Support for the NLGs.

Figure C.18: Reactions for the NLGs.

Figure C.19: SFD for the NLGs.

Figure C.20: BMD for the NLGs.
C.5 MLG Support Body Loading

For the support body only one FBD was conducted for the MLGs because the connection points and the applied loads are the same.

![Diagram of MLG Support Body Loading](image)

Figure C.22: Loads and Support for the MLGs.

![Diagram of Free Body Diagram (FBD)](image)

Figure C.23: Reactions for the MLGs.
C.6 NLG Support Body Loading

Figure C.26: Displacement for the MLGs.

Figure C.27: Loads and Support for the NLGs.

Figure C.28: Reactions for the NLGs.
Figure C.29: SFD for the NLGs.

Figure C.30: BMD for the NLGs.

Figure C.31: Displacement for the NLGs.
Appendix D

Finite Element Analyses and Results

This appendix deals with the process to obtain solutions with FEA. Here is detailed the refinement process, the problems encountered in the developed models and the results obtained for each part. These parts include the components of the upper and lower assemblies of the universal connection tool as well as the modified suspension ring presented in subsection 3.3.1, the developed connection part for this modification and the transport cars presented in the same subsection.

The results for the mesh refinement are shown in Table D.1 and the extracted values from each analysis in Table D.2. For the mesh refinement presented the material considered was the stainless steel AISI SS 304-Annealed.

With this chosen mesh size analyses were ran for each separate component, excluding the thrust bearing. Figures are presented for each part, regarding the stress distribution and the results are found in Table D.2. In this table some stress values must be addressed, the ones marked with an asterisk. Because of the nature of both FEA and the problem’s definitions, when an abrupt change in d.o.f. occurs between adjacent elements, a singularity can be created. This means that the strain gradient between both adjacent elements is infinite and therefore the stress at that location will also tend to infinite as the mesh becomes finer. This may occur at 90 degree corners in parts or near boundary conditions - what happens in these cases.

The introduction of a boundary condition which is bordered by elements without d.o.f. leads to the appearance of singularities at the border between the restrained surface and the free surface. This is an approximation of the model which is not valid because a fixed constraint forces the strain of a given node to zero, equivalent to having a part with infinite resistance (and zero strain) in contact with the node in question. This is not true because the contact which the imposed constraints intend to simulate are with parts that will have finite resistances and, therefore, exhibit strains and stresses (compressive in the present case).

In spite of all this the values obtained away from the singularities may still be used, provided that they are outside the area of influence of the singularity. Because of all this the values presented were extracted from other locations of interest, for example the SA notch of the beam for the NLG case and in the area surrounding the face where the load is applied, in the support body, for the MLG case. These details are shown in Figure D.3 and Figure D.14. This same situation occurred in the MLG/NLG slider and the SA slider, in the constrained inner faces, in the under-slider part, around the circular constraint created to simulate the contact with the threaded knob, shown in Figure D.8, and in the connection part devised for the modified suspension ring. For these parts the relevant values were extracted from the surfaces where the load is applied, away from the influence of the created singularity.
### Table D.1: Refinement analysis results.

<table>
<thead>
<tr>
<th>Element Size (mm)</th>
<th>Number of Elements</th>
<th>Number of Nodes</th>
<th>Time (min:seg)</th>
<th>Maximum Stress (MPa)</th>
<th>Error (%)</th>
<th>Maximum Displacement (mm)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>35</td>
<td>1460</td>
<td>3164</td>
<td>0:18</td>
<td>99.44</td>
<td>11.4</td>
<td>0.143</td>
<td>2.7</td>
</tr>
<tr>
<td>25</td>
<td>2358</td>
<td>4951</td>
<td>0:05</td>
<td>105.7</td>
<td>5.8</td>
<td>0.145</td>
<td>1.4</td>
</tr>
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<td>3622</td>
<td>7315</td>
<td>0:07</td>
<td>111.15</td>
<td>0.9</td>
<td>0.145</td>
<td>1.4</td>
</tr>
<tr>
<td>15</td>
<td>6492</td>
<td>12325</td>
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<td>113.94</td>
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<td>0.146</td>
<td>0.7</td>
</tr>
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<td>10</td>
<td>13796</td>
<td>24761</td>
<td>0:18</td>
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<td>0.146</td>
<td>0.7</td>
</tr>
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<td>7.5</td>
<td>25606</td>
<td>43997</td>
<td>0:35</td>
<td>120.26</td>
<td>-7.2</td>
<td>0.147</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>66290</td>
<td>109217</td>
<td>2:16</td>
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<td>0.4</td>
<td>0.147</td>
<td>0</td>
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<td>4</td>
<td>108313</td>
<td>174975</td>
<td>10:26</td>
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<td>0.147</td>
<td>0</td>
</tr>
<tr>
<td>3.5</td>
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<td>114.15</td>
<td>-1.7</td>
<td>0.147</td>
<td>0</td>
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<tr>
<td>3</td>
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<td>336775</td>
<td>37:00</td>
<td>112.25</td>
<td>-0.04</td>
<td>0.147</td>
<td>0</td>
</tr>
<tr>
<td>2.5</td>
<td>311464</td>
<td>488523</td>
<td>44:43</td>
<td>114.82</td>
<td>-2.3</td>
<td>0.147</td>
<td>0</td>
</tr>
<tr>
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<td>774980</td>
<td>84:00</td>
<td>112.21</td>
<td>-/</td>
<td>0.147</td>
<td>-/</td>
</tr>
</tbody>
</table>

### Table D.2: FEA results for each part.

<table>
<thead>
<tr>
<th>Part</th>
<th>Claw</th>
<th>Beam</th>
<th>MLG/NLG Slider</th>
<th>SA Slider</th>
<th>Under-Slider</th>
<th>Connection Part</th>
<th>NLG Bushing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. Stress (MPa)</td>
<td>38.763</td>
<td>530.883</td>
<td>176.234</td>
<td>126.777</td>
<td>187.257</td>
<td>140.078</td>
<td>120.813</td>
</tr>
<tr>
<td>Max. Displacement (mm)</td>
<td>0.004</td>
<td>1.217</td>
<td>0.713</td>
<td>0.035</td>
<td>0.389</td>
<td>0.009</td>
<td>0.012</td>
</tr>
<tr>
<td>Part</td>
<td>Piston Connection</td>
<td>Inferior Shaft</td>
<td>Support Body</td>
<td>MLG Slider</td>
<td>NLG Slider</td>
<td>SA Slider</td>
<td>-/</td>
</tr>
<tr>
<td>Max. Stress (MPa)</td>
<td>139.117</td>
<td>278.315</td>
<td>542.232</td>
<td>129.934</td>
<td>133.934</td>
<td>82.306</td>
<td>-/</td>
</tr>
<tr>
<td>Max. Displacement (mm)</td>
<td>0.027</td>
<td>0.022</td>
<td>0.276</td>
<td>0.040</td>
<td>0.087</td>
<td>0.007</td>
<td>-/</td>
</tr>
<tr>
<td>Part</td>
<td>Suspension Ring</td>
<td>Ring Connection Part</td>
<td>MLG Car</td>
<td>MLG Car</td>
<td>-/</td>
<td>-/</td>
<td>-/</td>
</tr>
<tr>
<td>Max. Stress (MPa)</td>
<td>229.398</td>
<td>275.102</td>
<td>7.714</td>
<td>25.517</td>
<td>-/</td>
<td>-/</td>
<td>-/</td>
</tr>
<tr>
<td>Max. Displacement (mm)</td>
<td>0.867</td>
<td>0.019</td>
<td>0.034</td>
<td>0.275</td>
<td>-/</td>
<td>-/</td>
<td>-/</td>
</tr>
</tbody>
</table>
In Figure D.2, Figure D.12, Figure D.13 and Figure D.18 the location of the singularities may be observed.

For the modified suspension ring an assembly analysis was conducted and the bolt and knob set replaced by a simpler part with the goal of restricting the movement of the under slider. It can be seen in Figure D.17. The obtained results for the developed connection part are shown in Figure D.18, in this part it can also be seen the effect of a singularity originated in the border between two surfaces with different constraints. The fact that this analysis is of a single part and that the contacts with other parts are only simulated using boundary conditions explains the discrepancy between the values obtained for the assembly analysis in Figure D.17 and the part analysis in Figure D.18. In spite of all this the obtained values remain small and do not vary much from one to another.

The figures with the combined stresses for the resting cars can be seen in Figure D.20 and Figure D.21, for the MLG and NLG resting cars, respectively. In Figure D.20 the parabolas describe the evolution of the combined stress in the horizontal beam and the blue rectangles the same in the vertical beams of the car. The red rectangle represents the loading on the car. Figure D.21 presents a 3D model view for the NLG car because a representation equal to Figure D.20 would be too confusing.

![Figure D.1: Claw.](image1)

![Figure D.2: Beam.](image2)
Figure D.3: SA notch in the beam.

Figure D.4: MLG/NLG slider.

Figure D.5: SA slider.

Figure D.6: Under-slider.

Figure D.7: Connection part.

Figure D.8: Singularities around the constraint in the under-slider.
Figure D.9: NLG bushing.

Figure D.10: Piston connection.

Figure D.11: Inferior shaft.

Figure D.12: MLG slider.

Figure D.13: Support body.

Figure D.14: Lower surface in the support body.
Figure D.15: NLG slider.

Figure D.16: SA slider.

Figure D.17: Modified suspension ring.

Figure D.18: Connection part of the ring.

Figure D.19: Singularities in the connection part of the suspension ring.
Figure D.20: Maximum combined stress about $Z$ axis for the MLG car.

Figure D.21: Maximum combined stress about $Z$ axis for the NLG car.