

Development of the JS-2 landing gear system

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ABSTRACT

The aim of this study was to review literature for the most suitable landing gear components used in modern gliders in order to develop a landing gear system that will adhere to the geometrical constraints of the JS-2 fuselage as well as the safety and structural requirements of the EASA CS-22.

The JS-2 glider is a high performance glider designed to be more efficient and smaller than the previous models. Compared to the JS-1, the JS-2 fuselage is 50 [mm] narrower at the widest point. This has made it necessary to develop a new landing gear system that is able to fit within the narrower fuselage and still be able to reduce the landing acceleration to a level lower than 4.5 G's [-], when landing with a maximum all up mass of 600 [Kg] at a descent velocity of 1.77 [m/s].

A literature review was done on the different landing gear components that were able to be used within modern gliders. It was found based on the research and the JS-1 landing gear design that the most applicable landing gear components to be used in a modern glider with a single main wheel landing gear arrangement that is mechanically retracted were the compact hydraulic brake and a rubber shock absorber. During the investigation it was found that the key component within the JS-2 landing gear system was the rubber shock absorber that directed the study towards rubber behaviour during compression.

Due the complex nonlinear behaviour of rubber materials, it was decided to investigate and derive an constitutive hyperelastic material model that was able to describe the nonlinear behaviour of the polyurethane rubber, in order to be able to develop an efficient and compact shock absorber for the JS-2 landing gear. The material model that was selected from a list of several different material models was the Yeoh model. This decision was based on the model's ability to predict accurately the behaviour of large strain applications in different deformation modes and its availability in commercial FEA codes.

In order to use the selected model within a FEA code, its coefficients had to be derived and validated. The coefficients were derived from a single uniaxial compression test that was validated by modelling the JS-1 rubber shock absorber in the FEA code PATRAN, and by analysing it in FEA codes MARC and NASTRAN. The numerical deflection results obtained from the FEA codes were then compared with the actual compression deflection results of the JS-1 shock absorber. A satisfactory correlation between the experimental and numerical results was obtained. This concluded that it was possible to use the Yeoh material model for complex

geometries that are subjected to compression load, where the material coefficients were determined by a single uniaxial compression test.

Having validated that the polyurethane rubber was modelled correctly with the Yeoh material model, it was decided to use this model to partially develop the JS-2 shock absorbing system. This included the development of the shock absorber rubber elements and the selection of the wheel and the tyre that had an influence on the shock absorber properties. The developed shock absorbing system of the JS-2 consisted of a single Michelin 5.00-5" tyre and two rubber shock absorbers situated on both sides of the shock strut arm. Each shock absorber consisted of three rubber elements with a hardness of 50 [Shore A] and a Young's modulus of 7.5 [MPa]. It was then analytically and numerically proven that the developed shock absorption system adhered to the CS-22 certification specifications, with a limit landing gear load factor ($n_{LG\ LIM}$) of 2.794 [-] and an ultimate landing gear load factor ($n_{LG\ ULT}$) of 3.484 [-]. These load factors with the developed JS-2 rubber shock absorber were then used to develop the JS-2 landing gear structure with FEA code NX NASTRAN.

The JS-2 landing gear structure was developed to be able to withstand loads calculated with a limit landing gear load factor of $n_{LG\ LIM} = 3$ [-] and a maximum design weight of $m_m = 600$ [Kg]. The loads derived from these two variables were used within the FEA code NX NASTRAN to analyse the landing gear structure for two loading conditions required by the CS-22 certification specifications. The two conditions included a level and side landing condition. The JS-2 landing gear structure was then analysed with the FEA code to investigate the stresses that the structure experienced for both load cases. The numerical stress results were then used to develop the landing gear structure to be able to accommodate the selected landing gear components and to be able to obtain a minimum safety factor equal to, or higher than 1.5 [-]. It was found by analysing each component of the final landing gear structure that the structure adheres to the CS-22 requirements with a minimum load safety factor of 1.57 [-].

The stress results that were obtained from the FEA code NX NASTRAN were then compared to a stress result that was analytically calculated in order to establish if the results of the numerical calculations were accurate. The results of the two different calculation methods correlated satisfactorily with each other. This verified that the JS-2 landing gear structure was modelled accurately and that the numerical results are reliable and accurate enough to be used to prove that the JS-2 landing gear system adheres to the CS-22 safety requirements.

KEYWORDS:

Landing Gear; Sailplane; Glider; Shock Absorber; Energy Absorption; Non-linear; Linear; Rubber; Structure; Finite Element Analysis; Implicit; Static; Hyperelastic; Constitutive Material Model; Fuselage; Load Factor; Development; Compression; Impact.

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NOMENCLOSURE

ABBREVIATIONS

<i>2D</i>	Two-dimensional
<i>3D</i>	Three-dimensional
<i>AR</i>	Tyre aspect ratio
<i>CS-22</i>	Certification Specifications
<i>EASA</i>	European Aviation Safety Agency
<i>FA</i>	Front strut arm
<i>FEA</i>	Finite Element Analysis
<i>FEM</i>	Finite Element Method
<i>JS</i>	Jonker Sailplanes cc
<i>TT</i>	Tubeless Tyre
<i>RA</i>	Rear strut arm
<i>SA</i>	Shock absorber strut arm
<i>SF</i>	Safety Factor

ROMAN SYMBOLS

A	Cross-sectional area	$[m^2]$
C_{ij}	Material coefficients that describe the shear behaviour of the material	
D_R	Specific rim diameter	$[mm]$
d_i	Deflection	$[m]$
E	Young's modulus	$[Pa]$
E_c	Compression modulus	$[Pa]$
E_i	Energy	$[J]$
F	Force	$[N]$
G	Gravitational Force acting on the object due to gravity	
g	Standard gravitational acceleration	$[m/s^2]$
H_i	Horizontal reaction loads	$[N]$
I_i	Strain invariants	
I_x	Area moment of inertia	$[m^4]$
ID	Inside diameter	$[mm]$
KE	Kinetic Energy	$[J]$
K	Spring rate	$[N/mm]$
K_c	Compression stiffness	$[N/mm]$
K_z	Tangential vertical stiffness	$[Kg/mm]$
L	Lift ratio	
L_i	Length	$[mm]$

M	Bending moment	$[N/m]$
m_i	Mass	$[Kg]$
N	Order of terms in a strain energy density function	
n_i	Load factor	
OD	Outside Diamater	$[mm]$
P	Inflation pressure	$[KPa]$
P_i	Landing loads	$[N]$
S_N	Nominal section width of tyre	$[mm]$
S	Shape factor	
T	Thickness	$[mm]$
V_Z	Vertical descent velocity	$[m/s]$
V_{Stall}	Aircraft stall speed	$[m/s]$
V_i	Vertical reaction loads	$[N]$
W	Stored energy function per unit volume	$[J]$

GREEK SYMBOLS

ε	Strain	$[mm/mm]$
δ_{VC}	Vertical ground clearance	$[mm]$
δ_{MVC}	Maximum vertical ground clearance	$[mm]$
θ	Front strut arm angle according to ground	$[^{\circ}]$
α	Rear strut arm angle according to ground	$[^{\circ}]$
σ_i	Stress	$[Pa]$
σ_{Yield}	Yield Stress	$[Pa]$
σ_{max}	Maximum Stress	$[Pa]$
μ	Friction coefficient	
ρ	Density	$[kg/m^3]$
λ_i	Strain principles extension ratio	
ν	Poisson's ratio	
η_i	Efficiency	

CHAPTER 1 : INTRODUCTION

1.1. BACKGROUND

The JS-1 is a high performance 18m class glider developed and manufactured by Jonker Sailplanes in South Africa. The glider is currently considered to have the highest performance in the 18m class. In an attempt to stay ahead of the competition the design team at Jonker Sailplanes is currently busy with the development of the JS-2 sailplane.



Figure 1 - JS-1 High performance glider just before landing (Jonker Sailplanes C.C., 2013)

The JS-2 high performance glider is designed to be more efficient and smaller than the previous models. The main design consideration of the JS-2 and any other high performance glider is to create the least amount of drag for any given amount of lift. This is mainly achieved with its wide wingspan and its aerodynamically designed narrow cockpit that will give it the ability to efficiently climb in rising air and glide long distances at high speeds with a minimum loss in height.

Compared to JS-1 the JS-2 fuselage is 50mm narrower at the widest point. This influences all of the internal structure and internal control systems that must be housed in the fuselage. The main structural element that is housed in the central fuselage is the retractable landing gear system. The reduction in fuselage width translates to a requirement for a smaller and narrower landing gear system. The main function of the landing gear system is to reduce the landing load to an acceptable level and to transfer the acceptable load from the wheel to the airframe structure.

Introduction

A key element of the landing gear is the shock absorption elements. The purposes of the shock elements are twofold. The first is to reduce the landing loads to the structure by reducing impact acceleration. The second function is to absorb all the landing energy without bottoming out which will otherwise result in an increase in acceleration.

The acceleration and energy absorption requirements are specified in the EASA CS-22 document which contains the airworthiness requirements to which modern gliders are designed. The CS-22 specifies that the shock element must reduce the landing acceleration to a level lower than 4.5 G's [-] and that the energy that must be absorbed is to be calculated at the maximum all up mass when the aircraft lands at a vertical acceleration of 1.77 [m/s]. The shock element must be able to absorb 1.44 times the energy of the above landing specifications.

1.2. PROBLEM DEFINITION

The geometrical constraints of the JS-2 fuselage width with the CS-22 requirements make the current landing gear used in the JS-1 unsuitable for the JS-2. A new landing gear and shock absorption system must therefore be developed.

1.3. OBJECTIVE

The objective of the study is to redesign the JS-1 landing gear to fit the JS-2 fuselage with specific focus on the shock absorption system so that the CS-22 requirements are fully met. The following sub goals will enable this objective to be met:

- To investigate the different landing gear components found in modern gliders.
- To investigate the different constitutive hyperelastic material models used to describe the behaviour of elastomers in order to determine the most applicable model for the rubber that will be used in the JS-2 shock absorber.
- To develop and verify a FEA model that will accurately predict rubber behaviour when subjected to compressional loads, in order to develop the rubber shock absorber for the JS-2 landing gear system.
- To develop and verify a FEA model of a landing gear system that consists of mechanical braking and retracting systems with a single main wheel and rubber shock absorbing element, in order to develop a landing gear system for the JS-2 glider.
- To mimic real-life landing conditions of the landing gear system by means of the developed FEA models and modern FEA codes, in order to prove that the tyre and

rubber shock absorber together absorb the total amount of energy when landing at a weight of 600 [kg] and a descent velocity of 1.77 [m/s].

- To numerically and analytically prove that the developed JS-2 landing gear system adheres to the safety specifications of the second amendment of the CS-22.

1.4. CHAPTER OUTLINE

- **Chapter 2: Literature study**– This chapter will present a brief overview of the different components that are found in the landing gear systems of modern day gliders. This chapter will also investigate rubber as a shock absorbing material that will include the investigation of the different constitutive hyperelastic models that are available to describe the behaviour of rubber materials.
- **Chapter 3: Deriving the Hyperelastic Material model** –The focus of this chapter is to correctly determine and validate the coefficients of the hyperelastic material model used to numerically model the polyurethane rubber material that will be used in the shock absorber.
- **Chapter 4: Development of the Landing Gear Shock-absorbing system** – In this chapter the analysis and development of the shock-absorbing system of the JS-2 landing gear will be discussed.
- **Chapter 5: Development of the Landing Gear Structure** – The focus of this chapter is to develop the JS-2 landing gear structure that will adhere to Jonker Sailplanes and EASA CS-22 requirements by using FEA code. This chapter will also include the verification of the numerical FEA model result with analytical results.
- **Chapter 6: Conclusion and Recommendations** – This chapter consolidates the work by providing a conclusion and recommendations.

CHAPTER 2 : LITERATURE STUDY

2.1. LANDING GEAR SYSTEM FOR SAILPLANES AND POWERED SAILPLANES

The landing gear has been described as “the essential intermediary between the aeroplane and catastrophe” (Currey, 1988). The engineers designing the landing gear have constantly been faced with the challenge of achieving a satisfactory design that is based on the conflicting demands of both the structural engineers and aerodynamicists. Landing gears are required to meet the requirements of the airframe designers and aerodynamicists to somehow be stowed in areas that have a minimum effect on the basic airframe structure and aircraft drag without affecting the maximum weight of the glider and still have the capability to land on bare soil if necessary.

One of the most important responsibilities of the landing gear system is to have the capability to absorb the kinetic energy upon impact during landing to ensure that the loads due to the vertical decent are transferred to the aircraft’s structure at a tolerable level. This shock absorption is accomplished by its main spring being the shock absorber and, to a smaller extent, by the tyre pneumatics. The landing gear system is also responsible to maintain stability when absorbing side loads due to severe wind conditions, braking or a sideways landing.

Requirements such as minimum weight, components maximum strength, maximum reliability, low cost, airfield compatibility and others can negatively influence each other (Currey, 1988). To prevent that a single requirement, such as minimum cost, gets too much emphasis, aviation authorities prescribe requirements. Requirements for sailplanes and powered sailplanes have to follow the certification specifications from the European Aviation Safety Agency (EASA CS-22).

2.2. LANDING GEAR SAFETY CONSIDERATIONS

In order to ensure the structural integrity of the aircraft, as well as the safety of the passengers within it, the requirements of EASA CS-22 must be adhered to when developing the landing gear system (Federal Aviation Administration, 2013). The safety requirements of the EASA CS-22 that are applicable to the landing gear system, stipulate the following:

- Landing gear must be able to land on unprepared soft ground without endangering the aircraft’s occupants (CS 22.721 (a)).

- Landing gear must be equipped with a mechanical braking system (CS 22.721 (d)).
- Landing gear must be equipped with a shock-absorbing system that consists of the shock absorber and tyre, which are able to absorb the total amount of kinetic energy developed in a level landing to ensure that acceptable loads are transferred to the rest of the landing gear and airframe structure without being fully depressed (CS 22.725 (a), (b), (c)).

2.3. LANDING GEAR COMPONENTS

The modern glider's landing gear systems basically consist of a single main wheel with a pneumatic tyre, a shock absorber, a brake and a kinematic system that allows for the retraction and deployment of the landing gear during flight. Each of these components will be discussed in this section to ensure a full understanding of the function and their impact on the aircraft's safety in order to select the most appropriate type of components for the development of the JS-2 landing gear system.

2.3.1. TYRE

The tyre is one of the landing gear components that enable the aircraft to manoeuvre safely during taxiing and ensure the safety of the passenger and the aircraft when absorbing some of the dynamic and static loads during landing and take-off. The tyres provide a significant amount of shock absorption for the landing gear system during impact when landing. Tyre manufacturers design tyres according to a maximum allowable static load that the tyre can be loaded to. This maximum allowable static load is determined by its pneumatic pressure within the tyre. Thus, the tyre carries the load almost entirely by its internal pneumatic pressure.

When selecting a tyre for a landing gear system, one can choose between a conventional bias-ply tyre and radial tyre. The most popular type of tyre is the Radial tyre that has only recently been developed. There are several advantages to the radial type tyre when compared to the bias-ply tyre. Two of the most important advantages of the radial tyre are the extended lifespan and light weight compared to the bias-ply tyre, making the radial tyre the preferred tyre for newly developed aircraft.

Radial tyres are constructed with additional steel belts in the radial direction, which give the tyre an additional advantage of having a larger footprint area of about 10 [%], reducing the wearing of the tread with about 40 to 60 [%] (Currey, 1988). A larger footprint area improves flotation characteristics and reduces hydroplaning. Radial tyres can withstand higher overload

bearing stresses and can withstand under-inflation better. When the radial tyres do fail due to impact the reaction is less marked compared to the reaction of bias-ply tyres, and indications of damage on the radial tyres can be spotted much easier than on bias-ply tyres.

For conventional bias-ply tyres the belts run in varying angles, usually between 30 to 40 degrees. Bias-ply tyres are, however, still widely used on current commercial aircrafts (Goodyear, 2002). A schematic representation of the structure of bias-ply and radial tyre is shown in Figure 2.

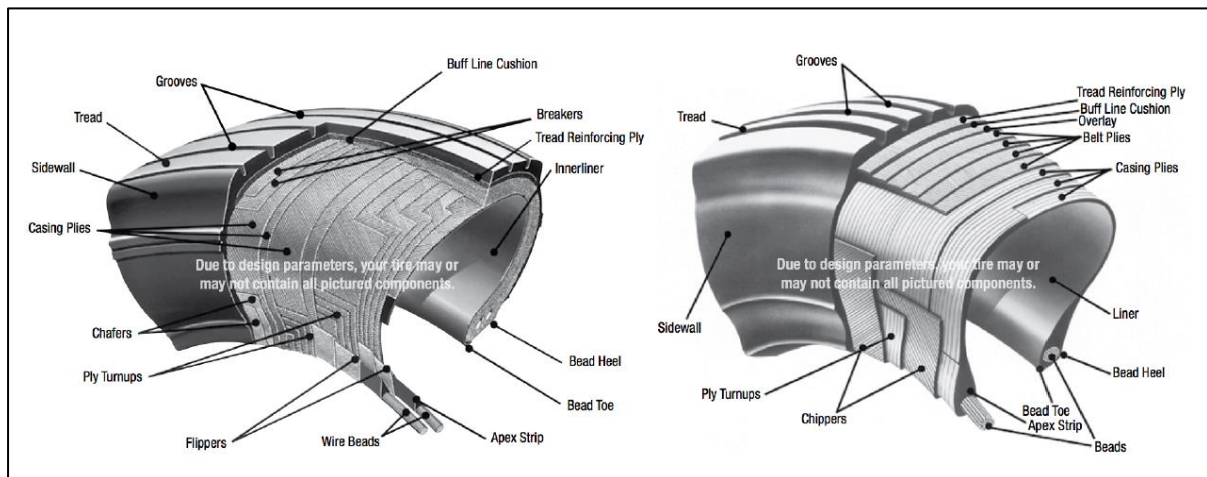


Figure 2: A schematic representation of the structure of a bias-ply tyre on the left and the radial tyre on the right (Goodyear, 2002)

Both bias-ply and radial tyres can be ordered to be manufactured with different specifications and options. These options include their maximum weight bearing loads or patterns or bulged shapes on the tyres surface to deflect water in a certain direction.

Aircraft tyre manufactures include Bridgestone, Goodyear, Dunlop and Michelin. All these tyre manufactures provide tyre-rated loads, pressures and dimensions of all currently available and certified aircraft tyres. From these available data the radius and shock absorption of the tyre can be determined for the main landing gear design. This determines the vertical position of the aircraft with respect to the ground that is needed to determine the vertical deflection of the shock absorber. The JS-1 landing-gear system is currently using a bias-ply type tyre from Michelin Air. A basic preferred system specification from Jonker Sailplanes is that the JS-2 landing gear is developed with the same wheel, tyre and braking system that are currently being used in the JS-1 landing gear, provided that it adheres to the JS-2 load requirements.

2.3.2. WHEELS

Technically, the term “wheel” refers to a circular metal/plastic object around which the rubber “tyre” is mounted. The brake system is mounted inside the wheel to slow the aircraft during landing. The wheel has to be dimensioned in such a manner that there is enough room to house the brakes and that the selected tyre will fit on it. This should be done while keeping in mind that the weight should be minimal and the life span of the wheel structure should be maximal.

Two types of designs for wheels are available at the moment, namely the A-frame type and the bowl-type wheels, shown in Figure 3 (Division, et al., 1987). The A-frame type can be made lighter than the bowl-type wheel with the disadvantage that it does not have much space to house the braking system. Therefore, when the braking requirements of the wheel system are very high the bowl-type wheel is the only option.

Both the A-frame and bowl-type wheels are usually constructed with forged aluminium. Other materials, such as magnesium, are not used in aircraft wheel design due to the serious problems with corrosion. Additionally, steel has the problem of increased weight and titanium is too costly for aircraft wheel design. The JS-1 landing gear is currently using the 5.00 – 5” [Inch] Beringer (JA-001) wheel which is of the bowl-type.

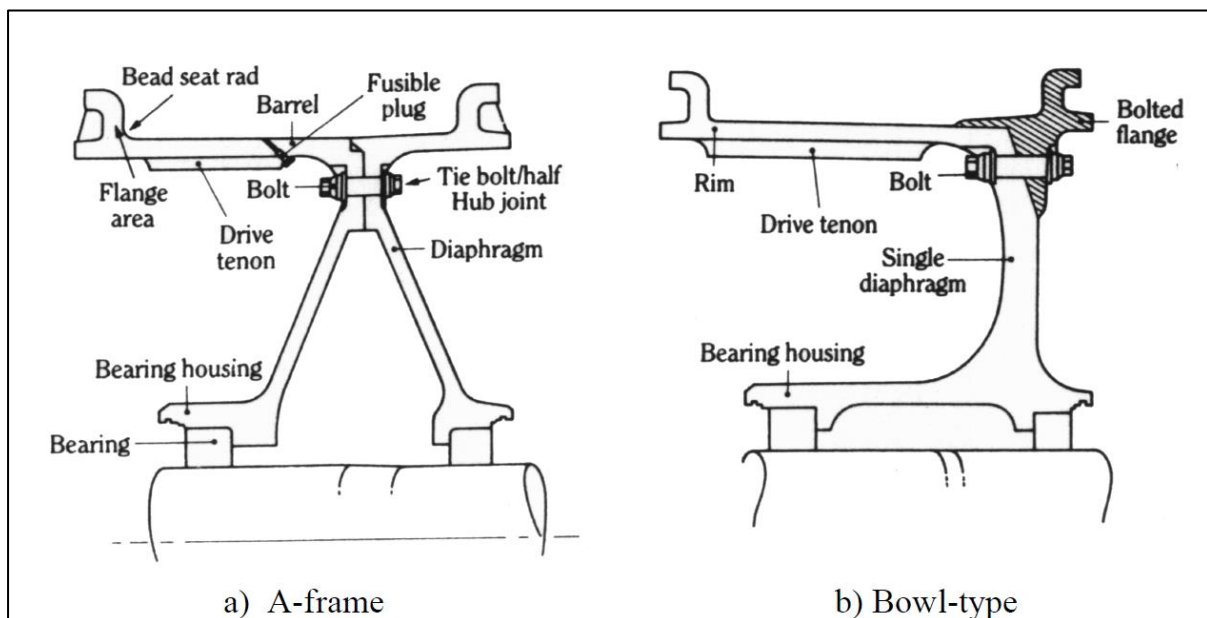


Figure 3: Basic configuration of wheel designs (Division, et al., 1987)

2.3.3. BRAKE SYSTEM

Brakes are used for stopping, speed control and keeping the aircraft in a parked position. Wheel brakes produce friction at the wheel assembly to slow or to stop rotation of the wheel. Light aircraft use a simple single disc type of brake but large transport aircraft require multiple discs to deal with the forces generated.

Simple braking systems comprise of a steel disk fixed to the wheel. A brake unit or calliper is equipped with friction pads operated by a hydraulic piston. When the brake pedals are pressed, the hydraulic pressure from the pedal transmits hydraulic pressure to the calliper position which squeezes the friction pads onto the discs. A schematic illustration of a single disc brake is shown in Figure 4.

The brake discs which the brake pads apply force onto are usually made from a form of cast iron called grey iron that is mostly found in motorcycles. Brake discs found in small aircraft are usually made from coated high strength steel or stainless steel to reduce weight and to ensure thermal stability and a long lifespan. The design of the disc varies somewhat. Some are simply solid and others are hollowed out with fins and vanes joining the disc contact surfaces together. This type of disc is usually called ventilated discs, and the design of these discs helps to dissipate generated heat. The weight and power of the vehicle determine the need for ventilated discs.

There are usually two brake pads per brake disc to ensure evenly wear on both sides of the disc. Depending on the properties of the material of both the pad and the disc and the configuration and the usage, pad and disc wear rates will vary considerably. The properties that determine material wear involve trade-offs between performance and longevity. Early brake pads were made from a material that contained asbestos that produces a dust that is harmful when inhaled. Newer brake pads are made of ceramics, Kevlar and other plastic compositions that have a greater lifespan and that are not so harmful when inhaled.

Recently new materials are introduced that have a lower weight and better material properties. One of these relatively new brake materials is carbon. Carbon has a high thermal conductivity and high specific heat that provide a more efficient and uniform heat distribution. When compared to steel disc brakes, carbon keeps most of its specific strength at high temperatures. Additional advantages are low maintenance; a longer service life and much lighter when compared to steel brakes.

Disadvantages include the larger volume that is required to achieve the same amount of energy absorption, the sudden loss of strength due to oxidation of the carbon, the loss of braking capabilities due to moisture and high initial cost to manufacture (Chai & Mason, 1996).

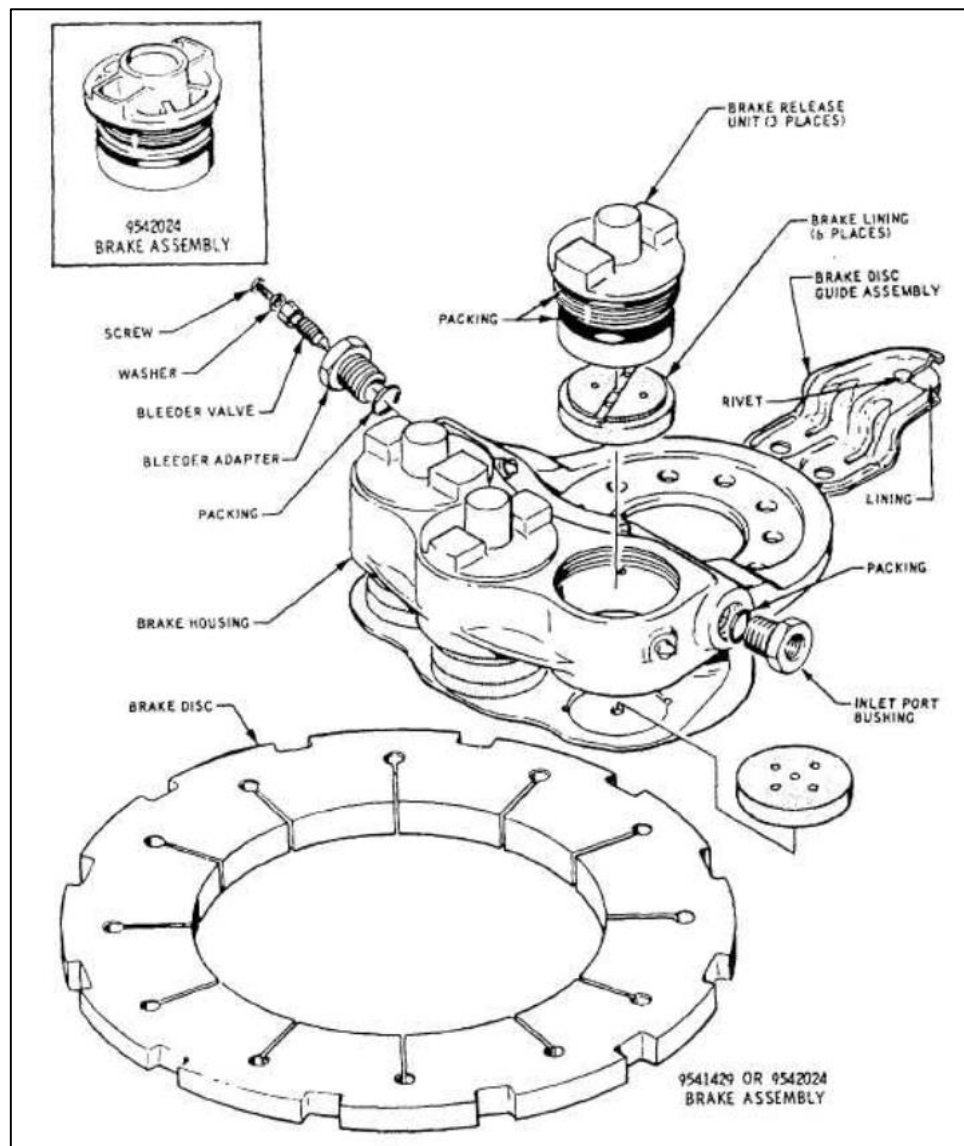


Figure 4: Schematic illustration of a single disc brake for small aircrafts (Navy Aviation, 2014)

The brake calliper is the assembly that houses the brake pads and pistons. These pistons are usually made of plastic, aluminium or chrome plated steel. There are basically two types of callipers, floating or fixed. A fixed calliper does not move relative to the disc and is thus less tolerant of disc imperfections. It uses one or more single or pairs of opposing pistons to clamp from each side of the disc, and is more expensive when compared to the floating calliper.

A floating calliper moves with respect to the disc, along a line parallel to the axis of rotation of the disc. The floating calliper also uses a piston on one side of the disc to push the inner brake

pad until it makes contact with the disc surface and simultaneously pulls the calliper body with the outer brake pad so pressure is applied to both sides of the disc.

The most common calliper designs use a single hydraulically actuated piston within the cylinder that increases the braking force severely. A recent development on the brake actuation system is the development of electric brake (Goodrich, 2012). Advantages of electric brake system include:

- reduced maintenance cost and the brake system can easily be replaced;
- higher reliability due to redundancy with multiple independent actuators installed on a single wheel;
- system health and brake wear are reported automatically.

The disadvantages of electrical brake system are that it is quite expensive to implement in small aircraft, and that the system requires larger space than the basic hydraulic system. The JS-1 uses a hydraulic brake system by Beringer that is incorporated into the Beringer wheel.

2.3.4. KINEMATICS

The design and analysis of landing gear parts relating to the retraction and extension of the gears is called kinematics. Stowage of the landing gear has to be possible within the available space while the increased weight due to structural reinforcements is minimal.

The goal is to make the retraction scheme as simple as possible, based on the economic considerations (Currey, 1988). A requirement that may increase complexity and cost is the requirement to limit the interference between the gear and the surrounding structure as much as possible. Also, the gear must be properly supported against side forces.

A retraction mechanism generally consists of a retraction actuator, a folding brace and a locking mechanism. The retraction of the gears positioned on the fuselage is most preferably done in the forward direction. This is to make sure that the gear can lock manually by gravity and air drag in the event of a hydraulic failure.

There are many different retraction schemes possible, shown in Figure 5, Figure 6, Figure 7 and Figure 8. To keep the retraction scheme as simple as possible each gear rotates about a single axis. On most forward retracting gears the shock strut is shortened during retraction to minimise the stowed space. Also drag struts and side struts fold away during retraction with a more complicated scheme than for the main gear. Figure 6 shows a typical aft retracting scheme

while Figure 7 illustrates an upward scheme with floating link and Figure 8 shows a forward scheme also with a floating link.

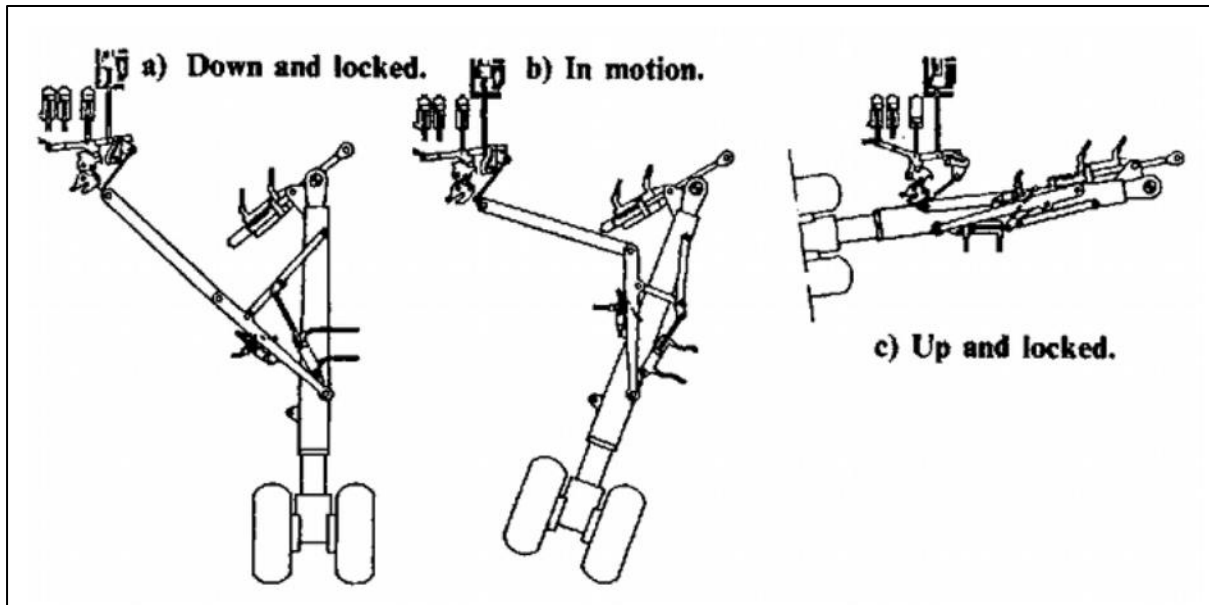


Figure 5 - Typical sideways retraction scheme of a wing mounted landing gear (Currey, 1988)

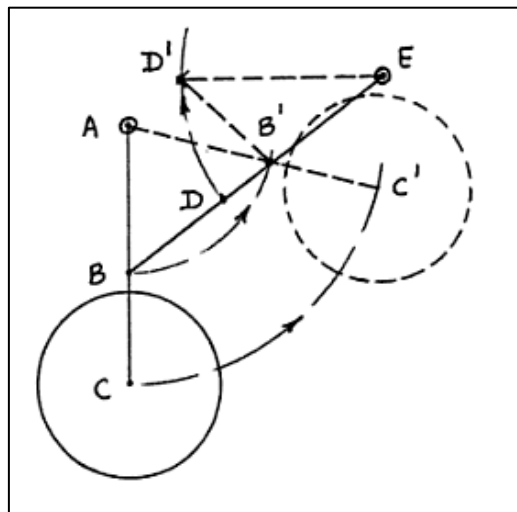


Figure 6 - Aft retracting gear with floating link (A and E are fixed points) (Roskam, 1986)

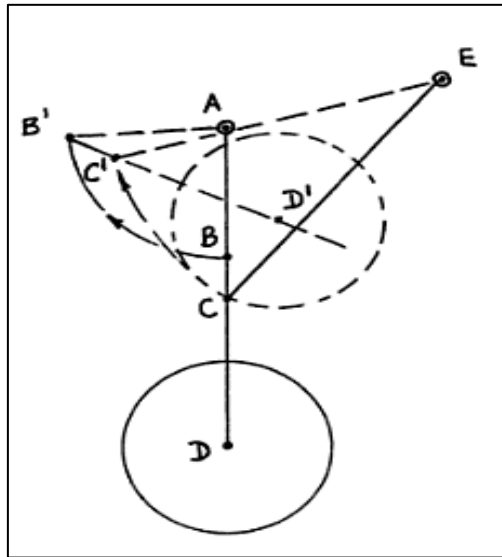


Figure 7 - Upward retracting gear with floating link (A and E are Fixed Points) (Roskam, 1986)

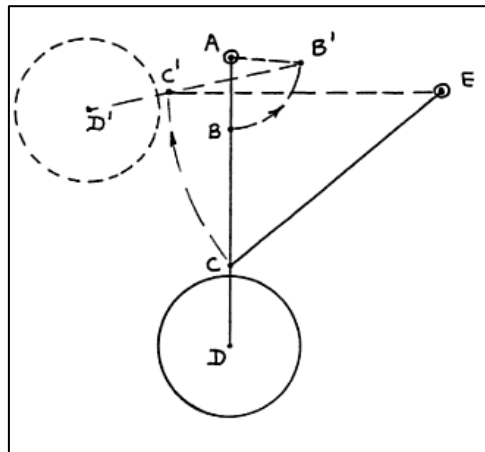


Figure 8 - Forward retracting gear with floating Link (A and E are fixed points) (Roskam, 1986)

The JS-1 uses a three point retracting system that is similar to the aft retracting type mechanism shown in Figure 6. There is no specific requirement on retraction mechanism to be used for the JS-2 except for the space requirements.

2.3.5. SHOCK ABSORPTION

Impact loads during landing and taxiing need to be absorbed by the landing gear and these loads need to be reduced to an acceptable level. There are several different types of shock absorbing elements. These can be divided into two basic types, based on the type of spring being used. The first type uses a solid spring made out of steel or rubber while the second type uses a fluid spring. The fluid spring uses air and gas or oil as the absorbing substance, or in some cases a mixture of oil and gas. This is generally referred to as an oleo pneumatic shock absorber (Currey, 1988).

Five different types of shock absorbers can be defined from the two classes. These are steel spring and rubber absorbers from the first group and air, liquid and oleo pneumatic from the second group. Figure 9 shows a comparison between these five absorbers based on efficiency and load capability.

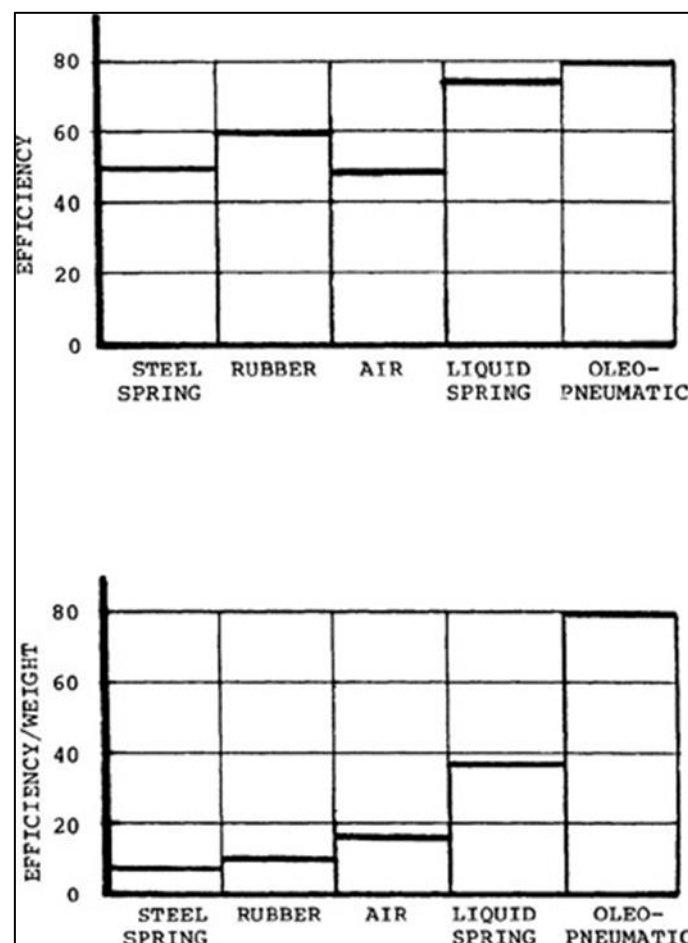


Figure 9 - Shock absorber efficiency (Currey, 1988)

In selecting the type of shock absorber it is also very important to give recognition to the simplicity, reliability, maintainability, and cost of the shock absorber as well as the efficiency while being compressed (Currey, 1988). Especially when selecting a shock absorber for smaller aircraft, like sailplanes or powered sailplanes, where the weight penalty is usually negligible and a low cost and reliability are necessary to ensure a profit. The basic characteristics of each of the five shock absorbers will now be discussed.

STEEL SPRING

The steel spring shock absorber can be used in two different arrangements. The first is a simple steel coil spring used to absorb the shock. This type of arrangement is rarely used in modern-day aircraft due to their extreme weight and their lack in efficiency. The second type uses a solid flexible steel strut or layers of solid flexible steel struts which connects the wheel arrangement to the aircraft's fuselage (Currey, 1988). The strut is usually mounted at a lateral angle to enable some vertical displacement of the aircraft through bending within the strut that acts as the shock absorber. As the strut deflects, the wheel undergoes an angle of travel that is non-vertical, illustrated in Figure 10, which causes wear on the sides of the wheel's surface.

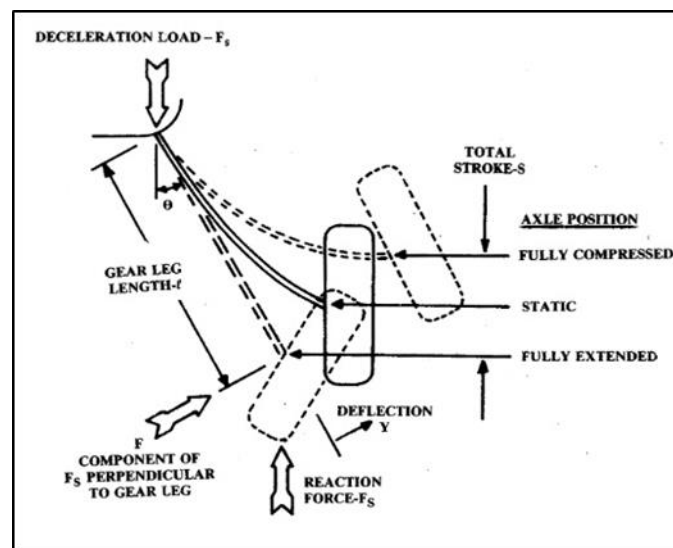


Figure 10 - Displacement of steel leaf spring shock absorber (Raymer, 1999)

Another negative aspect of the leaf spring shock absorber is that there is no damping of the shock-induced vibration, causing the aircraft to bounce during landing (Raymer, 1999). These types of shock absorbers are usually used for light aircraft equipped with non-retractable landing gear and are ideal for designs where simplicity, reliability and maintainability are to be

obtained. The Midwest “Mercury” Main gear is a good example of this type of shock absorbing arrangement, shown in Figure 11 below.

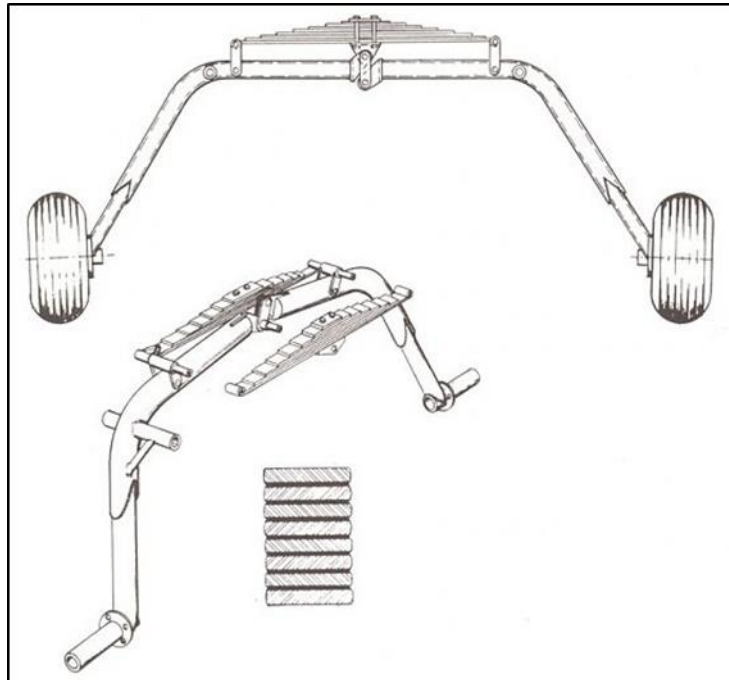


Figure 11 - Main Gear - Midwest Mercury (Currey, 1988)

RUBBER SPRING

Rubber has been applied to aeronautical design to absorb shock since the beginning of aviation in the early 1900s (Currey, 1988). Some of the earliest designs consisted of a levered bungee system that was basically a rope made by binding a multitude of thin rubber strands in a woven arrangement, combined with a metal strut to absorb energy. This configuration is similar to the steel leaf spring arrangement discussed in the previous section, as the vertical displacement of the aircraft also induces an outward movement of the landing gear wheel that will lead to wear on the outside surfaces of the wheel (Raymer, 1999).

In contrast to its steel spring counterpart there is improved energy absorption and damping ability due to the frictional forces between the rubber and rope strands. However, this arrangement is purely historical and was only used at the beginning of World War One. It can only be implemented on very light aircraft due to its low weight to shock absorbing efficiency. A few examples of this arrangement are shown in Figure 12 and Figure 13 (Currey, 1988).

Rubber can also be used in the form of blocks and disks. This allows an efficiency of around 60[%], which is largely due to the fact that the rubber medium can be more uniformly stressed in a solid form. An example of a typical shock absorber strut using stacked plates of rubber can

be shown in Figure 12. In order to achieve satisfactory shock absorption, each disk is generally no more than 5 [cm] thick (Currey, 1988). Rubber disks have been widely used for example in the previously referred to Twin Otter design shown in Figure 13 and in a typical rubber shock strut used in the Havilland design of the Mosquito aircraft shown in Figure 12. The rubber disk shock absorber is perfect to use for aircraft designs that want to eliminate the necessity to use strategic materials, to minimize cost and precision machining.

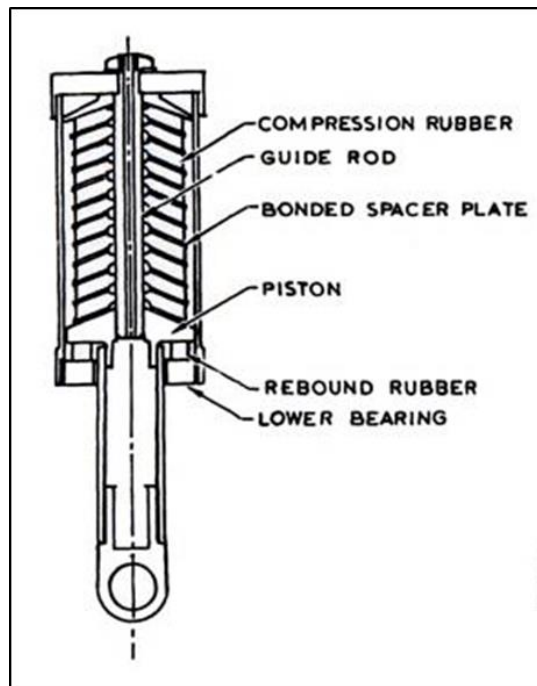


Figure 12 - Typical rubber shock strut (Currey, 1988)

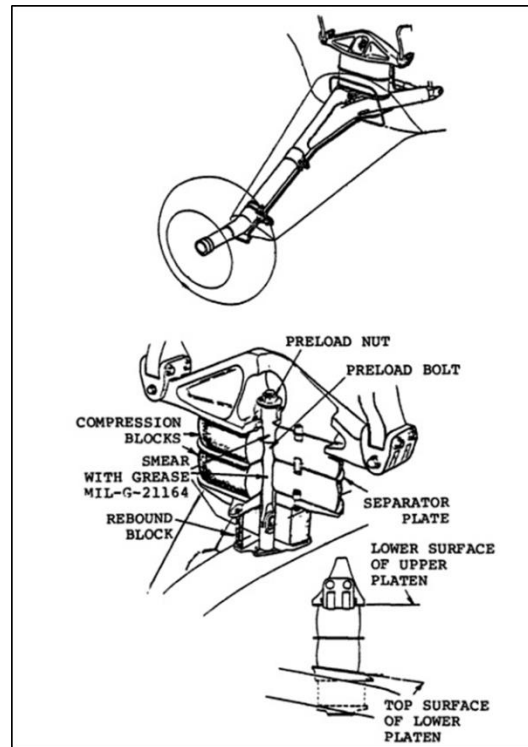


Figure 13 - DHC Twin Otter landing gear (Currey, 1988)

LIQUID SHOCK ABSORBERS

Liquid spring is an example of an oil-type shock absorber. This type of shock absorber was developed by Dowty and was first used in World War II (Currey, 1988), and their design is based on the fact that every liquid is compressible to some degree. This type of shock absorber is still being used today in levered suspension designs with an efficiency of about 75 to 90 [%]. Although their design and build is relatively the same in size and weight compared to the oleo-pneumatic shock absorbers, the efficiency is lower in comparison to the oleo-pneumatic shock absorbers. The shock absorbers' essential components are a cylinder filled with liquid, a piston and a special valve head that are shown in Figure 14. The liquid spring compresses fluid with the piston, occupying more of the cylinder's volume as a load is being applied to the piston. The piston's head houses a valve that opens when compressing and closes when retracting, which give it the ability to dampen the shock movement.

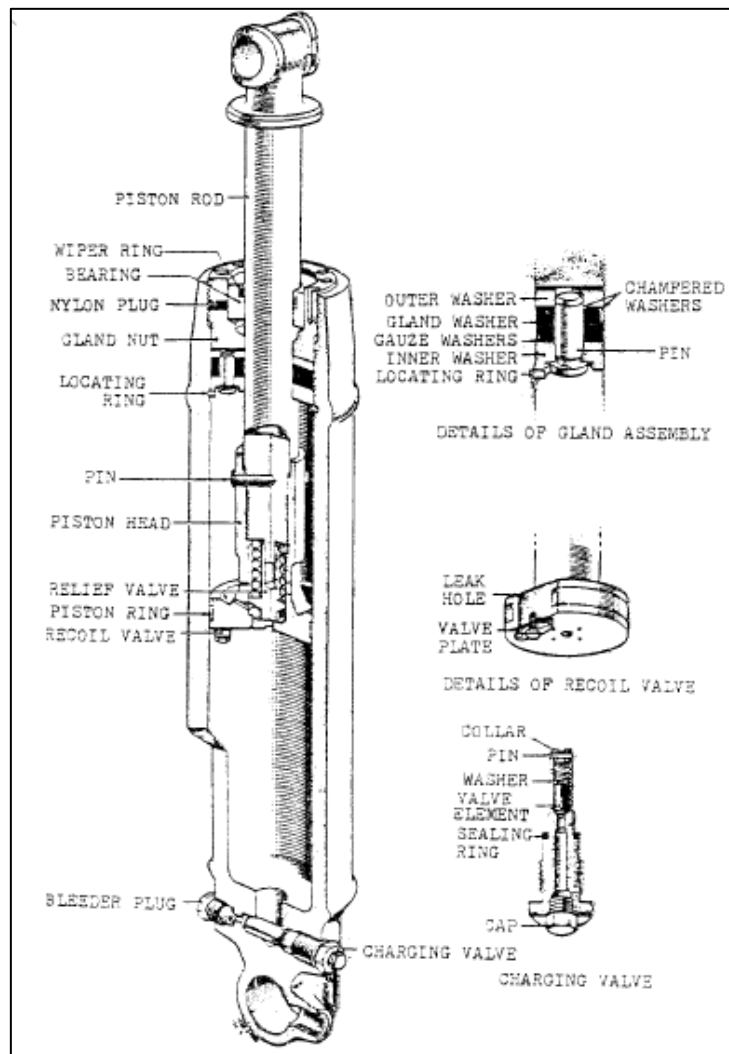


Figure 14 - Liquid spring (Currey, 1988)

The advantages of the liquid spring shock absorbers are their low fatigue due to their robust design and their relatively small size compared to steel spring absorbers. The disadvantages of a liquid spring according to Currey (1988) is that the liquid fluid volume changes at low temperature that affects the shock absorber's performance and that the shock absorber is only able to damp impact loads when the it is not fully depressed. The other disadvantage that makes this type of shock absorber quite expensive when compared to the steel and rubber shock absorbers, is that the fluid within the chamber must be sealed in such a way that it will not leak under high pressures; this is very important to ensure that the shock absorber is able to damp the impact load effectively. Although there are several disadvantages to the liquid shock absorber, it is still reliable, compact and rugged and can easily be used for light to heavy aircrafts.

AIR SHOCK

According to Currey (1988), air shocks, also known as pneumatic shock absorbers, have been used but not in recent times. Their design, build and operation are similar to oleo-pneumatic shock absorbers. The air shock tends to be heavier and much less efficient and reliable when compared to oleo-pneumatic shock absorbers. One of the biggest disadvantages of air springs is that there is no inherent means of lubricating the bearings within the shock.

OLEO-PNEUMATIC SHOCK

Most of the shock absorbers used in aircraft today are oleo-pneumatic shocks that are also known as air, oil shock absorbers. The oleo-pneumatic shock design, shown in Figure 15, absorbs energy by a compressed cylinder of air and oil. This is done by forcing oil from a lower chamber into an upper chamber of air through a compression orifice, compressing them together. The compression orifice could merely be a hole in the orifice plate. The diameter of the hole is thus varied by a metering pin with a varying diameter. The metering pin passes through the orifice hole, varying the area of the orifice hole to maximise the efficiency and giving a constant strut load during dynamic loads. The two substances can then either mix or stay separate while being compressed, depending on the design of the chamber. After the shock has dissipated the energy from the impact the air pressure forces the oil back into its chamber through the recoil valve.

According to Currey (1988), oleo-pneumatic shock absorbers have the highest efficiency and energy dissipation of all the shock absorber types. A typical efficiency of 80 to 90 [%] is realized in practise. One of the most popular advantages of this type of absorber is that it has the ability to return to its original state at a controlled rate, unlike a coil steel spring that rapidly releases the energy stored after being compressed.

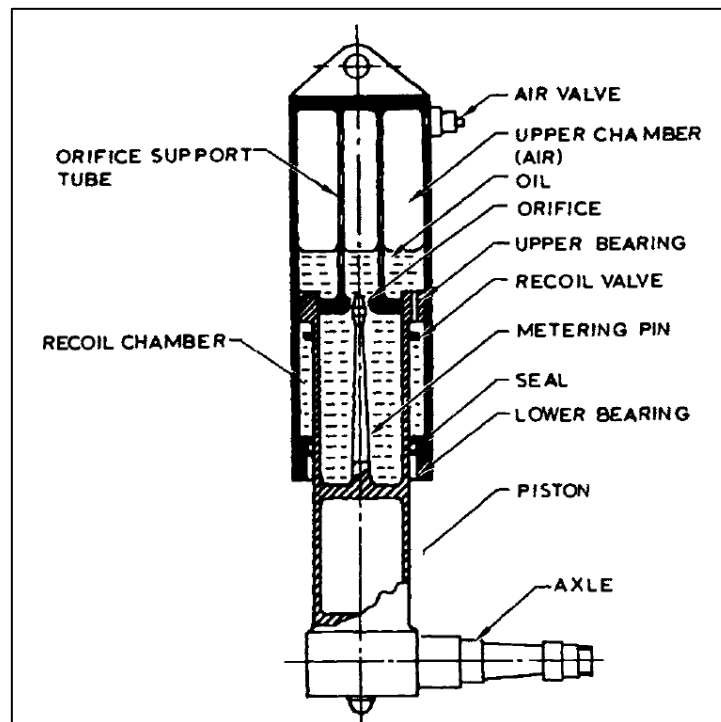


Figure 15 - Oleo-pneumatic shock absorber (Currey, 1988)

2.3.6. CONCLUSION

In the modern aviation field the landing gear system is a component of the aircraft that directly affects the performance of the aircraft's overall performance. The landing gear for modern gliders normally consists of a single main wheel with a pneumatic tyre, some shock element that is retractable in an effort to reduce drag. This system is normally manually operated by the pilot and must therefore be as simple and as light as possible (Federal Aviation Administration, 2013).

Due to the single wheel arrangement of the glider's main landing gear, the pneumatic tyre only has the ability to absorb some of the shock during landing, placing the rest of the shock absorbing responsibilities onto the shock element. When comparing the different shock absorbing elements, the mass requirement for modern gliders precludes pneumatic or oleo shock absorber elements and allows only rubber and spring steel to be possible candidates for the shock absorbing element. The analysis of a steel spring element is quite simple using normal linear stress assumptions. This is, however, not the case for rubber. Rubber requires special material modelling due to its unique material properties and nonlinear hyperelastic behaviour, making it quite difficult to model numerically (Yang & Charlton, 1993). As the JS-1

Literature Study

uses a rubber shock absorber, it was decided to follow this design philosophy and also to use a rubber shock element for the JS-2. The following section will discuss rubber complex behaviour and the modelling techniques pertinent to rubber shock absorbers.

2.4. RUBBER AS A SHOCK ABSORBING ELEMENT

Rubber has been used as an engineering material for the past 200 years and can mainly be divided into two main groups, known as natural- and synthetic rubber. Natural rubber is manufactured mainly from tree and plant sap and synthetic rubbers are produced from petroleum products (Yang & Charlton, 1993). Today the majority of rubber used in engineering applications is synthetic rubbers, due to the different additives that can be added to the chemical composition to manipulate the mechanical properties thereof (Håkansson, 2000).

Rubber's properties of elasticity and resilience have resulted in applications in load carrying structural bearings, springs, seals, shock absorber bushes, coupling and tyres. However, unlike metals, which require relatively few properties to characterize their behaviour, the behaviour of rubber is complex (Yang & Charlton, 1993).

2.4.1. MECHANICAL PROPERTIES OF RUBBER

According to Freakly and Payne (1978), there are several different characteristic features of rubber that make this material more popular in engineering applications. The most prominent characteristic of rubber is its elastic behaviour that gives it the ability to sustain large straining without deforming permanently. This is mainly possible due to the long tangled chains within the molecular structure of the compound that enable the material to undergo large deformations and to recover almost completely to its original shape when unloaded.

The second prominent characteristic feature of rubber is its almost zero volumetric change under hydrostatic pressure; this means that the rubber material is nearly incompressible. This is due to the large difference between its shear and bulk modulus, where the bulk modulus is an indication of how much resistance the compound has to uniform compression. For example, a typical carbon-black-filled rubber vulcanized used in typical engineering applications has a shear modulus of about 1 [MPa] and a bulk modulus of about 2000 [MPa] (Freakly & Payne, 1978). This large value of resistance to uniform compression compared to the shear stiffness indicates that the material is nearly incompressible.

Although rubber is a highly elastic material it is not perfectly elastic. Its behaviour is mainly influenced by the type of load being applied to it. The stiffness of the rubber is also affected by other factors like the frequency and amplitude when cyclic loading is applied. A higher frequency will increase the stiffness of the rubber, where cyclic loading with higher amplitudes will decrease the stiffness of the material. Another unique characteristic of rubber is when cyclic

loading is applied to it a slight difference is observed between the loading and unloading curves in their stress strain diagrams. This phenomenon is called hysteresis, and has a greater effect on filled rubber, illustrated in Figure 16 (Håkansson, 2000).

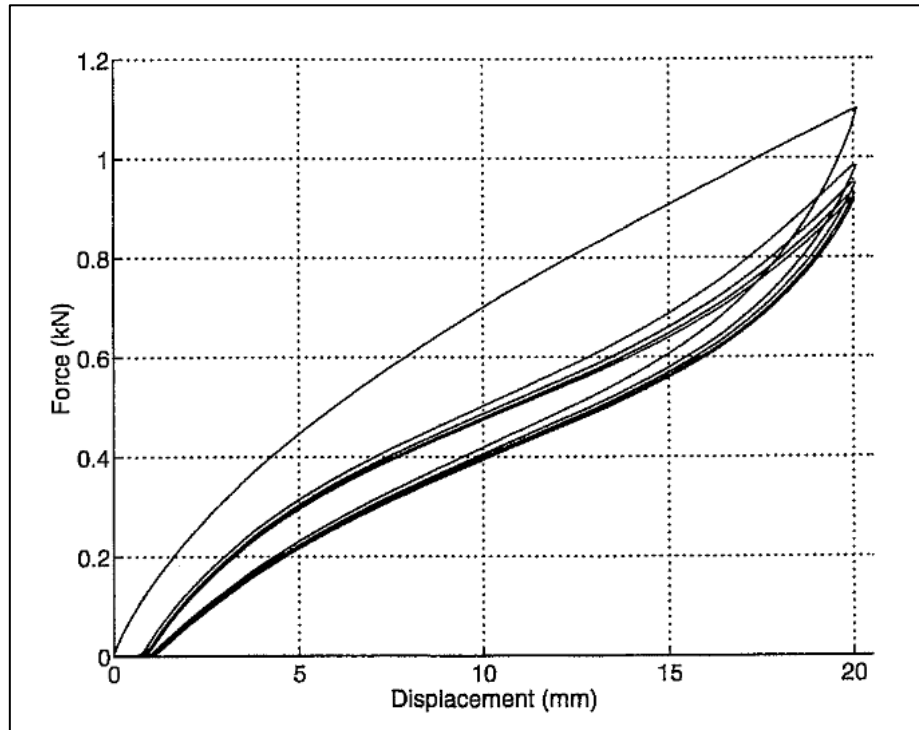


Figure 16 - Force-displacement relations for a carbon-black-filled rubber in planar-tension (Per-Erik, 1997)

In cycling loading there is always a part of energy that is not recoverable. This energy is mainly dissipated as heat, and is represented by the enclosed area between the loading and unloading curves in Figure 16. In free vibration cyclic loading this causes the amplitude of the vibrations to decrease, this effect is also known as damping that can be increased when fillers are added to the rubber compound (Freakly & Payne, 1978).

Rubber's mechanical behaviour is thus further complicated by the effect of temperature, strain history, loading rate and the amount of strain. This has meant that for the history of rubber as engineering material, its applications have been developed and optimized by means of trial and error, rather than by fundamental understanding.

2.4.2. METHODS TO DEVELOP RUBBER SHOCK ABSORBERS

As previously stated the rubber shock absorber is one of the most difficult and complex systems to model due to its non-linear behaviour. There are two methods to develop and model a rubber shock absorbing element: physical modelling of the element based on practical data, and nonparametric modelling based on experimental data (Cui, et al., 2010).

The physical model of a rubber shock absorber is a detailed description of the shock absorber element that is used to describe the behaviour of the absorber in several different operating conditions. The physical method is thus a very reliable and accurate method to use from a theoretical point of view. However, it usually is very complex and time consuming when used to develop a design. This method can also become very expensive especially when a small change was made to the design of the shock absorber that requires an adjustment to the model that can only be changed by the manufacturer (Weigel, et al., 2002), (Cui, et al., 2010).

In contrast to the physical model method, the nonparametric model method is a very accurate and much cheaper method to use to describe the behaviour of the rubber shock absorber. The nonparametric model is based on mathematical means that can only describe the behaviour of the system for limited operating conditions that have been tested through practical experiments. However, the nonparametric method generates an efficient model that can easily be adapted when the design is changed (Cui, et al., 2010). The mathematical theory behind the nonparametric modelling method used to describe the behaviour of the rubber shock absorber will now be discussed in the following section.

2.4.3. NONPARAMETRIC MODELLING METHOD FOR RUBBER

Many theoretical nonparametric models have been developed in an attempt to characterize the mechanical behaviour of rubber-like materials. These attempts fall primarily into two categories. The first category is based on statistical thermodynamics, and the other is based on Rivlin's phenomenological theory that treats the material as a continuum (Yang & Charlton, 1993).

According to Treloar (1958) and Rosen (1971), the thermodynamic approach is based on the observation that the rubber elastic force arises almost entirely from the decrease in entropy with increase in extension. This method, according to Shaw and Young (1987), is accurate to describe low strain applications of around 50 [%] strain. The majority of the research work

done to characterize the nonlinear behaviour of rubber and elastomers has been focused on the development of the phenomenological method based on continuum mechanics that was concluded from the observation of rubber under various conditions of homogeneous strain.

The phenomenological theory assumes that rubber is an isentropic material. This assumption can be said to remain valid, only if the long molecular chains of the rubber polymer are assumed to be randomly orientated when in an unstrained state and orientated into the direction that it is being stretched when in strain. These assumptions are fundamental to describe the elastic characteristic properties of rubber when explained in terms of a strain energy function based on the strain invariants I_1 , I_2 and I_3 . This theory offers a mathematical framework to describe rubbery behaviour based on continuum mechanics and is the starting point of any kind of modelling. The basic strain energy density function (W) in terms of strain invariants is as follow:

$$W = f(I_1, I_2, I_3) \quad (2.1)$$

Where;

W – Stored energy function per unit volume,

I_1, I_2 and I_3 – The three invariants of the green deformation tensor

given in terms of the principle stretch ratios λ_1, λ_2 and λ_3 by:

$$I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \quad (2.2)$$

$$I_2 = \lambda_1^2 \lambda_2^2 + \lambda_2^2 \lambda_3^2 + \lambda_3^2 \lambda_1^2 \quad (2.3)$$

$$I_3 = \lambda_1^2 \lambda_2^2 \lambda_3^2 \quad (2.4)$$

Equation (2.1) can also be represented as:

$$W = \sum_{i+j+k=1}^N C_{ijk} (I_1 - 3)^i (I_2 - 3)^j (I_3 - 1)^k \quad (2.5)$$

By considering that rubber is incompressible, the third invariant $I_3 = 1$, thus decreasing Equation (2.5) to:

$$W = \sum_{i+j=1}^N C_{ij} (I_1 - 3)^i (I_2 - 3)^j \quad (2.6)$$

2.4.4. STRAIN BASED NONLINEAR MATERIAL MODELS FOR RUBBER

A great amount of works has been done and literature published on the different models that can be used for the modelling of rubber material. The choice of the model to use for modelling depends mainly on the application of the rubber and the data available to determine the material's coefficients (Lemaitre, 2001).

The modelling and design of a hyperelastic rubber material, basically come down to the selection of the correct strain energy function and an accurate determination of the strain energy material coefficients (Garcia Ruiz & Suarez Gonzalez, 2006). There are several different forms of strain energy functions available to model incompressible and isotropic elastomers. However, there are only a few models that will be able to describe the material behaviour correctly and accurately for the desired loading condition (Markmann & Verron, 2006).

Boyce and Arruda (2000), and Seibert and Schoche (2000) compared five different models respectively that describe the deformation behaviour of rubber by means of experimental data. Markmann and Verron (2006) did another study on twenty different hyperelastic models for rubber-like materials and classified them according to their abilities to fit to experimental test data.

The hyperelastic models that are being reviewed are the physical-based models that treat the rubber as a nearly incompressible material based on continuum mechanics without reference to molecular concepts. The models to be discussed are the models that are most commonly used in commercial FEA programs that are based and presented either by strain invariants or by stretch ratios, and in some cases by both, for example the Mooney-Rivlin and Neo-Hookean models (Ali, et al., 2010).

RIVLIN MODEL

The Rivlin model also known as the polynomial model is based only on the first and second invariant I_1 and I_2 shown in the incompressible form as follow:

$$W = \sum_{i,j=1}^N C_{ij} (I_1 - 3)^i (I_2 - 3)^j \quad (2.7)$$

Where;

W – The strain energy density,

N – The number of terms within the strain energy functions,

C_{ij} – Material coefficients that describes the shear behaviour of the material,

I_1 and I_2 – The first and second invariants of strain, which may be expressed as principle stretch ratios.

The Rivlin model is usually used to model the stress-strain behaviour of filled elastomers, with at least four or greater order terms (N).

OGDEN MODEL

This model proposes the strain energy function based on the principle stretches ($\lambda_1, \lambda_2, \lambda_3$) for incompressible materials that assume that $\lambda_1 \lambda_2 \lambda_3 = 1$. One of the main advantages of using these principle stretches is that they are directly measurable. The relation of the Ogden strain energy model is as follow:

$$W = \sum_{i=1}^N \frac{2\mu_i}{\alpha_i^2} (\bar{\lambda}_1^{\alpha_i} + \bar{\lambda}_2^{\alpha_i} + \bar{\lambda}_3^{\alpha_i} - 3) \quad (2.8)$$

Where;

λ_i – The principle stretches,

α_i and μ_i – Material constants describing the shear behaviour of the material.

It is found that the Ogden's model is used more often than the Rivlin model. This model is much more accurate when fitting experimental data, especially when several experimental tests are available. This model can be used for small, moderate and large strain deformations (Ali, et al., 2010).

MOONEY-RIVLEN MODEL

The Mooney-Rivlin model strain energy function can be presented by principle strain invariants based on the Rivlin model as well as by the three principles stretches based on the Ogden model. Although there is a difference in terms of their formula, the results obtained from them are the same (Ali, et al., 2010). With $N = 1$ the Rivlin model is changed into the Mooney-Rivlin model, proposed as follow:

$$W = \sum_{i,j=0}^N C_{ij} (I_1 - 3)^i (I_2 - 3)^j \quad (2.9)$$

Where;

C_{ij} – Material constant that describes the shear behaviour of the material and $C_{00} = 0$ (Markmann & Verron, 2006),

I_1 and I_2 – The first and second invariants of strain, which may be expressed as principle stretch ratios.

The Mooney-Rivlin model is usually used with one order term (N) for incompressible materials and can be simplified as follow:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) \quad (2.10)$$

The first order Mooney-Rivlin model according to the Ogden model based on principles stretches can be presented by setting $N = 0$, $\alpha_1 = 0$ and $\alpha_2 = -2$. The Mooney-Rivlin model is obtained as follow:

$$W = \frac{\mu_1}{2}(\bar{\lambda}_2^2 + \bar{\lambda}_2^2 + \bar{\lambda}_3^2 - 3) - \frac{\mu_2}{2}(\bar{\lambda}_2^{-2} + \bar{\lambda}_2^{-2} + \bar{\lambda}_3^{-2} - 3) \quad (2.11)$$

Where;

$$\begin{aligned} C_{10} &= \frac{\mu_1}{2} \text{ and,} \\ C_{01} &= -\frac{\mu_2}{2} . \end{aligned}$$

The first order Mooney-Rivlin and Ogden model are two of the most favourite constitutive models to use for moderate to large strain deformations (Ali, et al., 2010). Their disadvantages are that their material parameters can only be obtained by experiments that are not physically based and that the fitting of their parameters can become very complex when the number of parameters becomes too large (Bol & Reese, 2003).

NEO-HOOKEAN MODEL

The Neo-Hookean model is the simplest hyperelastic model to use for elastomeric materials when material data is insufficient. As with the Mooney-Rivlin model the Neo-Hookean model can be presented by principle strain invariants based on the Rivlin model as well as by the three principles stretches based on the Ogden model. The Rivlin model can be reduced by omitting the second invariant. The reduced Rivlin model is changed into the Neo-Hookean model, as follow:

$$W = \sum_{i=1, j=0}^N C_{ij} (I_1 - 3)^i \quad (2.12)$$

Where;

C_{ij} – Material constant that describes the shear behaviour of the material and ,

I_1 – The first invariants of strain, which may be expressed as principle stretch ratios.

The Neo-Hookean model is usually used with one order term (N) for incompressible materials and can be simplified as follow:

$$W = C_{10}(I_1 - 3) \quad (2.13)$$

This model is only applicable and offered in the first strain invariant (I_1) (Timbrell, et al., 2003). It is noteworthy that Rivlin arrived at his model from mathematical arguments involving symmetry and material incompressibility. Thus illuminating the doubt about the need for second strain invariant term (I_2). The Neo-Hookean model can also be obtained by the Ogden model by setting $N = 1$, $\alpha_1 = 2$, and is as follow (Bol & Reese, 2003):

$$W = \frac{\mu_1}{2} (\bar{\lambda}_2^2 + \bar{\lambda}_2^2 + \bar{\lambda}_3^2 - 3) = C_{10}(I_1 - 3) \quad (2.14)$$

The Neo-Hookean model is found to generate accurate results for small strain deformations (Ali, et al., 2010). The model can derive its material coefficient from stress-strain data from a single mode of deformation. This advantage comes with the disadvantage that the results from the material model lose its accuracy (Gent, 2012).

YEOH MODEL

The Yeoh model strain energy function is obtained from the reduced Rivlin model by neglecting the second strain invariant and by setting $N = 3$. The Yeoh model is then as follow:

$$W = \sum_{i=1}^3 C_{i0} (I_1 - 3)^i \quad (2.15)$$

The Yeoh model is usually used with three order terms (N) for incompressible materials and can be simplified as follow:

$$W = C_{10}(I_1 - 3) + C_{20}(I_2 - 3)^2 + C_{30}(I_3 - 3)^3 \quad (2.16)$$

The Yeoh model is also one of the favourite models used to model and describe the hyperelastic properties of rubber elastomers. One of the advantages of using this model is that this model is

able to predict the behaviour of rubber for a much wider deformations range than the other compared models. The second advantage, as with the Neo-Hookean model, is that it is able to predict the behaviour in different deformation modes by using data from a single deformation mode like uniaxial compression or tension (Gent, 2012) (Ghosh, et al., 2003).

ARRUDA AND BOYCE MODEL

The Arruda and Boyce model is a very physical model, based on molecular chain networks. The theory behind this model is that the strain energy is assumed to be equal to the sum of the strain energies of the individual chains that are orientated randomly. The model is as follow:

$$W = \mu \sum_{i=1}^5 \frac{C_i}{\lambda_m^{2i-2}} (I_1 - 3)^i \quad (2.17)$$

With:

$$C_1 = \frac{1}{2}, C_2 = \frac{1}{20}, C_3 = \frac{11}{1050}, C_4 = \frac{19}{7000}, C_5 = \frac{519}{673750} \quad (2.18)$$

Where;

- μ – The initial shear modulus,
- λ_m – The locking stretch, the point where the stress-strain curve will rise significantly,

This model has shown sufficient accuracy in small and large strain problems despite neglecting the second invariant(J) (Seibert & Schoche, 2000).

CONCLUSION

Each of these material models for rubber has a mathematical form with a different set of parameters which have been established by fitting a curve through experimental data. According to Sharma (2003) some of these strain energy functions, such as the Neo-Hookean, Yeoh, and Arruda-Boyce hyperelastic models, are independent of the second stretch invariant. If it is supposed that they are well independent, a single test such as a uniaxial tension test will only be needed to determine the material's response. The model obtained from the single test mode will only be able to present an approximate representation of the material's response in multiple different modes as with a multi-parameter model where it will be able to present the material's behaviour in full detail. Although this method is only an approximation, it is still much easier and less time consuming (Marlow, 2003). It is now important to discuss the availability

and application of these hyperelastic models in commercially available FEA codes before a decision can be made on which model to use to develop the rubber shock absorbing element.

2.4.5. HYPERELSTIC MATERIAL MODELS USED IN FEA PROGRAMS

FEA of elastomers became a reality for rubber and elastomer component design early in the 1970s with commercial FEA codes such as MARC (Finney & Gupta, 1977). Since that time additional FEA codes, such as ANSYS and ABAQUS, have also incorporated the ability to be able to analyse hyperelastic rubber materials. Hyperelastic material models are widely used in FEA codes. These hyperelastic models are used to give a relation between the stress and strain of rubber materials in a FEA. The rubber materials strain energy parameters can be determined by fitting the strain energy function to stress-strain data.

Most commercially available FEA codes such as MARC, PATRAN, NASTRAN, ABAQUS and ANSYS offer full nonlinear rubber analysis with some of the more basic hyperelastic material models, for instance the Mooney-Rivlin, Ogden, Neo-Hookean, Yeoh and Arruda and Boyce models. These codes have the ability to input user-defined material models that can be included as a simple FORTRAN based sub-routine if the user wishes to include a strain energy function that is not supported (Ali, et al., 2010) (Yang & Charlton, 1993).

These FEA codes usually require an input of test data of several different deformation modes depending on the analyses that are done. These tests are more commonly known as uniaxial tension and compression, pure shear (also known as planar tension) and equi-biaxial tension tests. Most of the FEA codes automatically execute the curve fitting routine based on the least square type regression, to determine material model coefficient values from the test data. The material model coefficients are then used in the material model of the FEA codes to interpolate and extrapolate stress-strain values for other deformation modes throughout the solution. There, however, are some FEA codes that will allow the user to insert the relevant material model coefficients directly, if the coefficients have been determined by independent means (Yang & Charlton, 1993).

2.4.6. FEA MODELLING TECHNIQUES FOR HYPERELASTIC MATERIALS

There are several numerical methods used to solve complex rubber designs. FEA is one of the most accurate and versatile methods to use. In order to ensure that the FEA is applied to the rubber material in a confident and accurate manner that will derive accurate results it is necessary to model the rubber hyperelastic material with the correct element type that will not influence the material's behaviour.

All of the commercial FEA codes typically provide a rich element library in order to suit different types of analyses. The two basic element types are 2D and 3D elements. 2D Elements can be triangular or quadrilateral in shape and the 3D elements may be tetrahedral or hexahedral (known as brick elements) in shape. These elements may use linear or quadratic interpolation techniques. All of these types of elements may be used for rubber materials simulated in linear and nonlinear analyses, provided that they are formulated with displacement as the only variable. However, when incompressible or nearly incompressible rubber materials are involved it is recommended to use hybrid or Hermann elements, which are formulated with displacement and pressure variables (Gent, 2012).

According to Gent (2012), when modelling a rubber part as a 2D surface or 3D solid, quadrilateral or hexahedral elements are generally preferred over triangular or tetrahedral elements. This is due to the reason that the triangular or tetrahedral elements tend to be too stiff.

It is preferable to model rubber material that is subjected to contact with first order elements. The reason for this is that first order elements require a higher mesh density that will improve the contact area for contact analysis. The use of first order elements will decrease the computational resources that will improve the run time when modelling large complex 3D models.

The other method used to reduce the run time when large complex rubber parts are being analysed is to model the part with reduced integration elements. However, this will have to be done with caution due to the reason that first order elements and reduced integration elements may cause uncontrolled distortion of the mesh known as 'hour-glassing'. This problem does not always occur, thus it is advised to check your results carefully (Gent, 2012).

2.5. SUMMARY

In his chapter the responsibilities of a landing gear system for a modern glider were discussed. Additionally, its sub components and elements were investigated in order to ensure that the most appropriate components will be considered for the development of the JS-2 landing gear system. It was found that a simple single main wheel landing gear that is rear wards retractable offers the most promise for a modern glider. Due to the space and simplicity requirements it was decided to focus on polyurethane rubber as the shock absorbing material for the main landing gear shock absorber.

It was found that properties of rubber materials are quite complex and their behaviour difficult to predict due to their nonlinear and nearly incompressible nature. It was found that several hyperelastic material models based on strain energy density functions were developed for rubber materials. All these material models consist of several coefficients that are used to describe the material's behaviour through strain energy function. The only method to predict the values of these material coefficients is with experimental stress-strain data. In order to determine the material coefficients for Rivlin and Ogden models, which consist of the first and second strain invariants, it is necessary to perform experimental stress-strain measurements from several different modes of deformations.

However, this is not the case for the Neo-Hookean, Yeoh and Arruda-Boyce models. These models neglect the second strain invariant making it possible to obtain their material coefficients from a single experimental stress-strain deformation measurement.

Out of these few material models that only need the experimental stress-strain data from a single mode of deformation, the Yeoh model was selected. The Yeoh model is selected because of its ability to accurately predict the behaviour for large strain applications in different deformation modes by using data from a single experimental deformation mode like uniaxial compression or tension test as well as its availability in commercial FEA codes (Gent, 2012).

The following chapter will focus on the development of the Yeoh material model, to ensure that the coefficients of the Yeoh material model are correctly determined. The derived Yeoh material model will then be validated to ensure that the Yeoh material model can accurately predict the behaviour of the rubber that will be used in the JS-2 rubber shock absorber, in order to be used to develop the JS-2 rubber shock absorber.

CHAPTER 3 : DERIVING THE HYPERELASTIC MATERIAL MODEL

3.1. INTRODUCTION

The modelling of a complete landing gear system will invariably require the mechanical properties of the shock absorbing elements. In this chapter, the material model coefficients will be determined for polyurethane rubber which will act as the shock absorbing element for the JS-2 landing gear.

The focus of this chapter is specifically to determine correctly and validate the coefficients of the hyperelastic material model that is used to numerically model the polyurethane rubber material. As discussed in the literature survey, the Yeoh material model will be used to numerically model the behaviour of the polyurethane rubber. It was found that the Yeoh material model will be the most applicable strain energy function to numerically determine and predict the behaviour of the polyurethane rubber in order to develop the JS-2 shock absorber.

This chapter, therefore, describes the Yeoh model as a strain energy density equation that will include the description of the Yeoh material coefficients (C_{10} , C_{20} and C_{30}). These will be used within the FEA code, PATRAN, NASTRAN and MARC to predict the rubber nonlinear behaviour.

This chapter will also include the methodology used to determine the Yeoh material model coefficients (C_{10} , C_{20} and C_{30}) from a single uniaxial compression test. This chapter will also validate this methodology that is used to determine the coefficients and whether it is accurate enough to numerically predict the behaviour of polyurethane rubber when assigned to more complex geometries that are subjected to compression loads.

3.2. YEOH MATERIAL MODEL

The Yeoh model is a phenomenological model also known as the reduced third-order polynomial form that is used to model and describe the deformation of isotropic incompressible rubber-like materials. The model is based on Ronald Rivlin's observation that the elastic properties of rubber may be described by using a strain energy density function (W) in terms of the strain invariants (I_1, I_2, I_3). The strain energy density function (W), explained in *Chapter 2*, implemented in most commercially available FEA codes capable of handling and simulating nonlinear material behaviour is given by the following equation:

$$W(I_1, I_2, I_3) = \sum_{i,j,k=0}^{\infty} C_{ijk}(I_1 - 3)^i(I_2 - 3)^j(I_3 - 1)^k \quad (3.1)$$

Where;

W – is the stored energy function per unit volume,

I_1, I_2 and I_3 – is the three invariants of the Cauchy-Green deformation tensor

given in terms of the principle stretch ratios λ_1, λ_2 and λ_3 by:

$$I_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2$$

$$I_2 = \lambda_1^2\lambda_2^2 + \lambda_2^2\lambda_3^2 + \lambda_3^2\lambda_1^2$$

$$I_3 = \lambda_1^2\lambda_2^2\lambda_3^2$$

If it is assumed that the rubber-like material is completely incompressible, as explained in *Section 2.4.3. Nonparametric Modelling Method for Rubber* then $I_3 = 1$, in order to illuminate the third term invariant (I_3) completely changing *Equation 3.1* to:

$$W(I_1, I_2) = \sum_{i,j=0}^N C_{ij}(I_1 - 3)^i(I_2 - 3)^j \quad (3.2)$$

; where C_{ij} is the model's unknown material coefficients. The explicit version of this equation, with terms having an index sum less or equal to three, is written as follow:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) + C_{20}(I_1 - 3)^2 + C_{11}(I_1 - 3)(I_2 - 3) + C_{02}(I_2 - 3)^2 + C_{30}(I_1 - 3)^3 + C_{21}(I_1 - 3)^2(I_2 - 3) + C_{12}(I_2 - 3)^2(I_1 - 3) + C_{30}(I_2 - 3)^3 + \dots \quad (3.3)$$

The third order deformation or James-Green-Simpson model for incompressible rubber-like material is found by using the terms that include: I_1, I_2, I_1^2, I_1^3 and $I_1 I_2$, reducing the equation to:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) + C_{20}(I_1 - 3)^2 + C_{11}(I_1 - 3)(I_2 - 3) + C_{30}(I_1 - 3)^3 \quad (3.4)$$

By leaving out the terms that include the second invariants (I_2) in *Equation (3.4)*, it is possible to get the strain energy function for the Yeoh model. This model is more versatile than the others since it has been demonstrated to fit various modes of deformation using data obtained from a single uniaxial test that in the end leads to reduced requirements for material testing. The strain energy function based only on the first invariant (I_1), known as the Yeoh model, is represented by the following equation:

$$W(I_1) = C_{10}(I_1 - 3) + C_{20}(I_1 - 3)^2 + C_{30}(I_1 - 3)^3 \quad (3.5)$$

Where C_{10}, C_{20} and C_{30} is the rubber material coefficients that will be used to predict the behaviour of the rubber material being used to develop the shock absorbing elements. It is important to know that the coefficients C_{10} and C_{30} influence the behaviour of the rubber at low and high strain rates respectively. These three material constants will be determined by an experimental test that is discussed in the following section.

3.3. METHODOLOGY USED TO DETERMINE YEOH MATERIAL COEFFICIENTS

This section will discuss the methodology used to determine the Yeoh material model coefficients (C_{10}, C_{20} and C_{30}) from a single uniaxial compression test. This section will include the description of the uniaxial compression test, the derivation of the relationship between theoretical and experimental stress in terms of principle stretches (λ), and a description of the least square fitting method used to determine the closest fit for the Yeoh material model to the experimental uniaxial compression data.

3.3.1. UNIAXIAL COMPRESSION TEST

The test described here is to define the material coefficients of the Yeoh hyperelastic material model that is used within the nonlinear FEA, PATRAN and NASTRAN codes to simulate the behaviour of rubber in a compression strain state. The compression test is designed to give a

uniaxial deformation. In the uniaxial deformation two principal stretches are equal to each other and the other stretch is determined from the assumed condition that the material is incompressible.

The compression test is based on ASTM D575-91 that is the standard test method for rubber properties in compression. The standard covers the test procedure for determining the compression-deflection characteristics of rubber compounds. The rubber-compression specimen, according to the standard, consists of a cylinder with a diameter of 28.6 ± 0.1 [mm] and a thickness of 12.5 ± 0.5 [mm].

The rubber specimen is compressed between two rigid flat parallel plates that are shown in Figure 17. Sheets of sandpaper are placed on the two parallel plates where the rubber specimen comes in contact with them, in order to ensure that there is not any lateral slippage of the rubber at the contact surfaces. The force (P), shown in Figure 17 (b), is then applied to the rubber specimen and the deflection of the specimen is then recorded. The recorded data of the uniaxial compression test is presented as a nominal stress- strain data set.

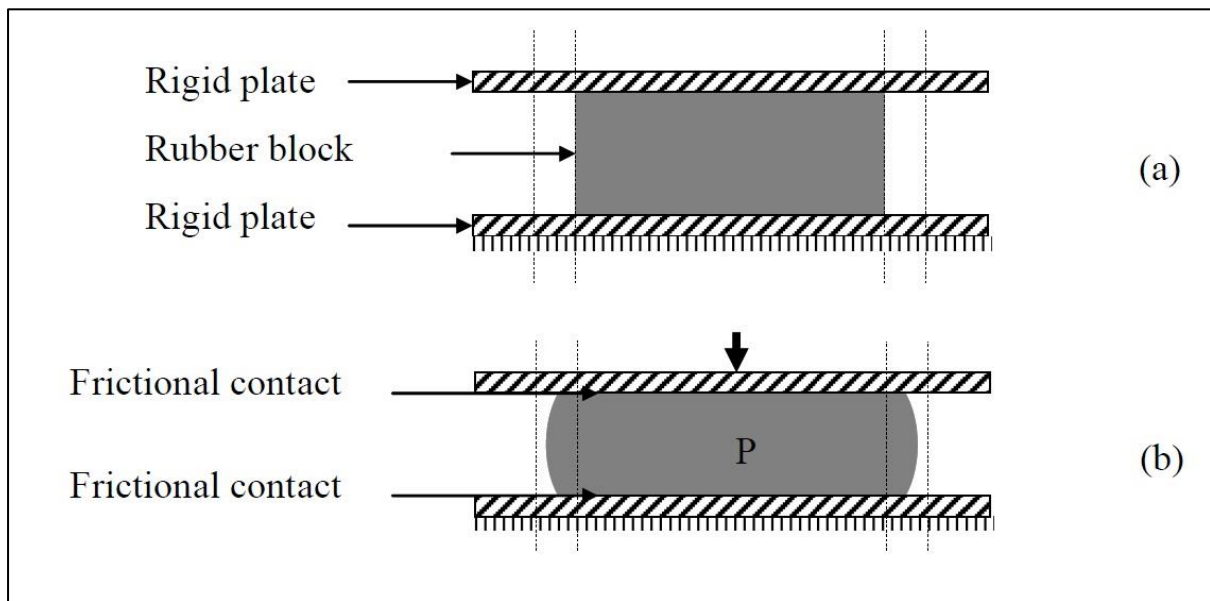


Figure 17 - Schematic diagram of deformation of rubber sample under vertical load (a) Before loading, (b) During loading

In order to determine the three coefficients of the Yeoh material model from the experimental stress-strain ($\sigma - \epsilon$) data an equation has to be generated where the nominal stress (σ) of the uniaxial compression test data is described in terms of strain invariants (I_1, I_2 and I_3) and stretches. Further details on the uniaxial compression test are found in *APPENDIX A*.

3.3.2. NOMINAL STRESS RELATION FOR UNIAXIAL DEFORMATION DATA

The nominal stress (σ_U) for the uniaxial deformation test, defined by the force (P) per original cross section area (A_{cross}), can be obtained by using the general strain energy density function (W) of the Yeoh model in terms of strain invariants (I) or stretches (λ) (Per-Erik, 1997). This nominal stress is given by the following equation:

$$\sigma_U = \frac{P}{A_{cross}} = \frac{\partial W}{\partial \lambda_U} \quad (3.6)$$

Where the general strain energy density function (W) is the Yeoh material model given by Equation (3.5) as a function of the first strain invariant (I_1), defined by the stretch in the uniaxial direction (λ_U). The stretch (λ_U) in the uniaxial direction is determined by the prescribed displacement which gives us a measure of deformation. The stretch (λ_U) in the uniaxial direction is calculated as:

$$\lambda_U = \frac{L + \delta}{L} = \varepsilon + 1 \quad (3.7)$$

Where;

- ε – is the nominal strain for the uniaxial compression test data pair [mm/mm],
- L – is the original length of the compression specimen [mm],
- δ – is the prescribed displacement [mm].

With incompressibility assumed the deformation modes for the uniaxial compression test described in terms of principle stretches (λ_1, λ_2 and λ_3) are:

$$\begin{cases} \lambda_3 = \lambda_U \\ \lambda_1 = \lambda_2 = \frac{1}{\sqrt{\lambda_U}} \end{cases} \quad (3.8)$$

By inserting the values into the three invariants the condition of incompressibility is fulfilled because ($\lambda_1 \lambda_2 \lambda_3 = 1$):

$$\begin{cases} I_1 = \frac{2}{\lambda_U} + \lambda_U^2 \\ I_2 = \frac{1}{\lambda_U^2} + 2\lambda_U \\ I_3 = 1 \end{cases} \quad (3.9)$$

Using the first invariant in *Equation 3.9* and the equation of the strain energy density function of the Yeoh model as a function of the first strain invariant *Equation 3.5*, it is possible to get the strain energy equation as a function of the compression stretch (λ_U) deformation:

$$W(\lambda) = C_{10} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right) + C_{20} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right)^2 + C_{30} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right)^3 \quad (3.10)$$

By using this expression of the Yeoh model and *Equation 3.6*, the nominal stress (σ_U) for uniaxial compression case can be presented as a function of stretch in the uniaxial direction:

$$\sigma_U = \frac{\partial \left(C_{10} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right) + C_{20} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right)^2 + C_{30} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right)^3 \right)}{\partial \lambda}$$

$$\sigma_U = 2 \left(\lambda_U - \frac{1}{\lambda_U^2} \right) \left(C_{10} + 2C_{20} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right) + 3C_{30} \left(\frac{2}{\lambda_U} + \lambda_U^2 - 3 \right)^2 \right) \quad (3.11)$$

Since the number of stress equations will be greater than the number of unknown coefficients, a fitting procedure must be performed to determine the Yeoh material coefficients. The Yeoh strain energy function (W) is linear in terms of its material coefficients. Therefore, the fitting procedure that will be used to determine the material coefficients will be the least-square fitting procedure.

3.3.3. FITTING PROCEDURE

The fitting procedure is based on the relative error between theoretical data and experimental data, where the theoretical stress data is fitted against experimental stress data obtained from the uniaxial compression test. The material coefficients are then selected where the relative error (R_{Error}) between the two data sets is at its minimum.

The procedure is illustrated by fitting the Yeoh material model with the experimental compression test data of a rubber with the hardness of 60 [Shore A]. The procedure is illustrated by using data of uniaxial compression test data shown in Figure 18. The small red circles shown in Figure 18 correspond to the stress strain values $i = 1, \dots, n$, where σ_i^{Test} is the nominal stress value of the test data and ε_i is the corresponding strain value for n number of data points. The Yeoh material model fitted to the experimental data is illustrated by a blue curve shown in Figure 18.

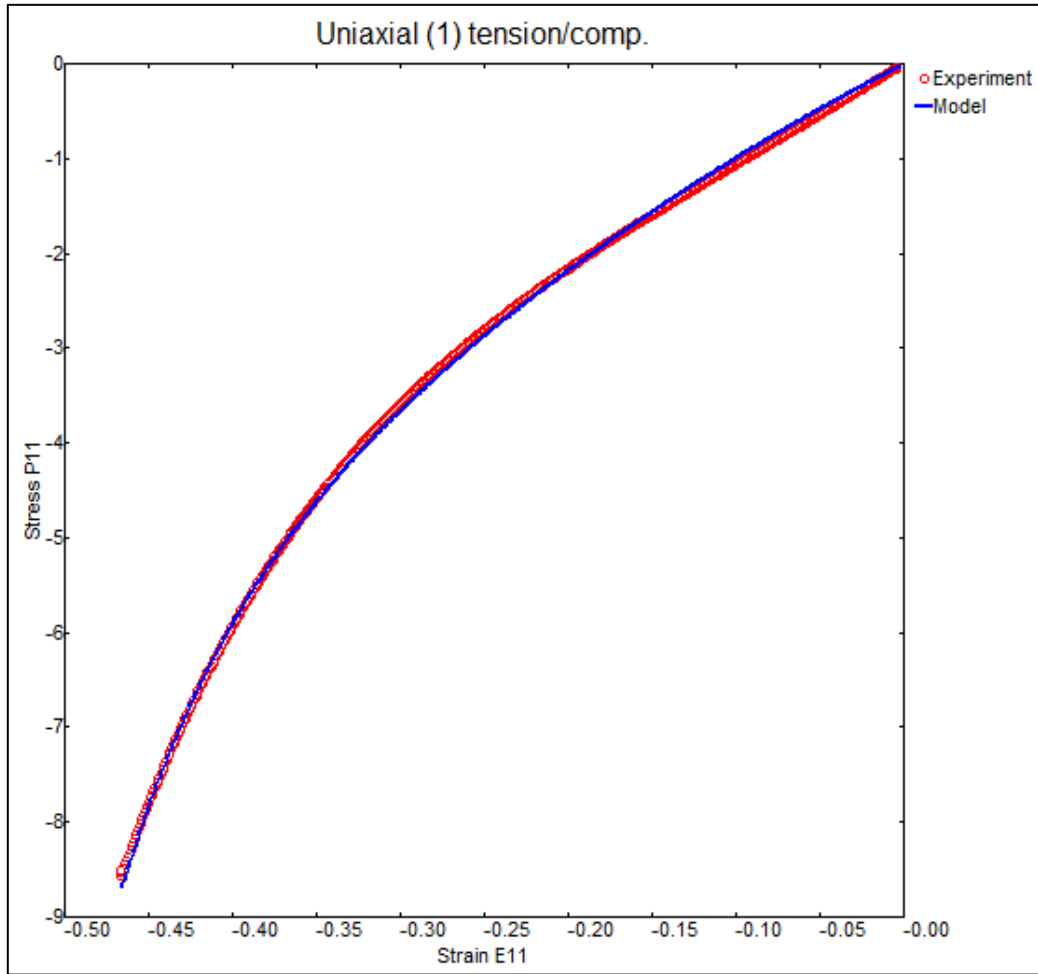


Figure 18 - Fitting of the three parameter model to experimental data

By using the nominal stress equation (σ_U) shown in *Equation 3.11*, the three material coefficients of the Yeoh material model can be generated from the fitted curve. The three material coefficients are calculated as follow:

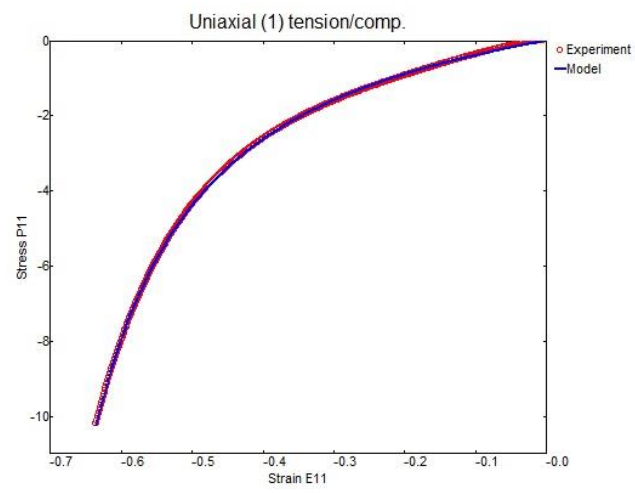
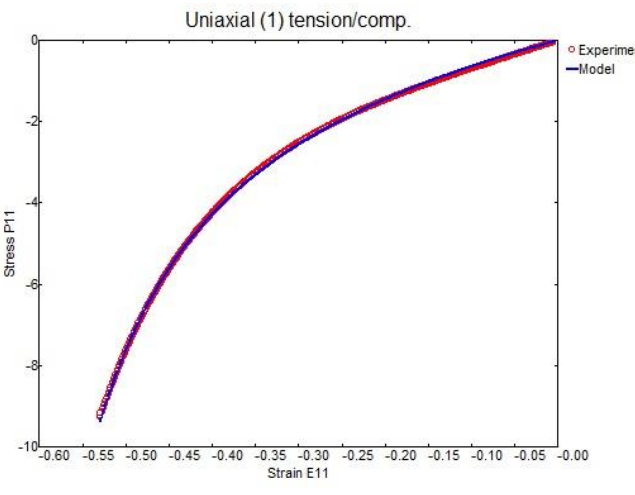
$$C_{10} = 1.498 \text{ [MPa]} \quad C_{20} = -0.281 \text{ [MPa]} \quad C_{30} = 0.172 \text{ [MPa]}$$

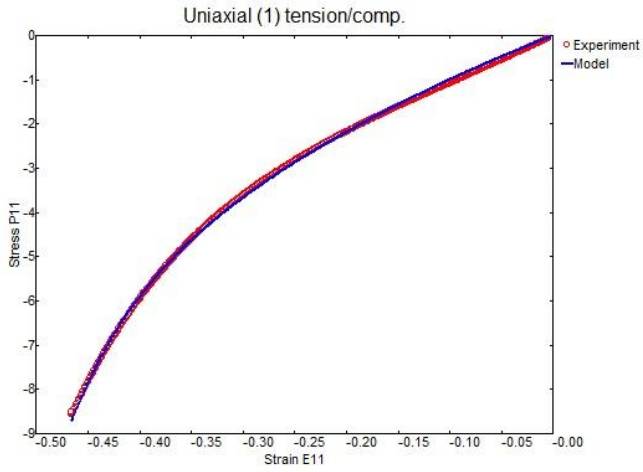
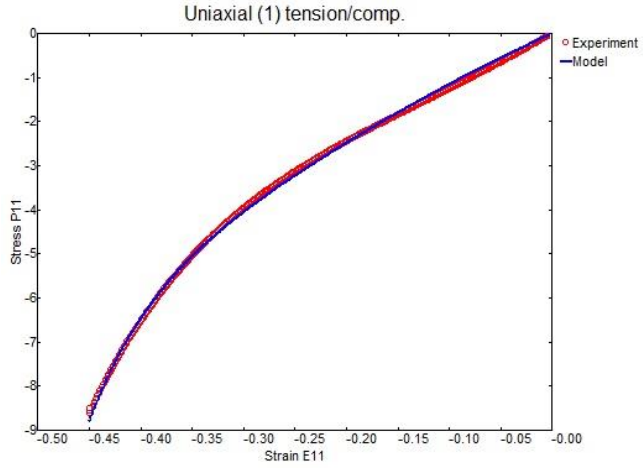
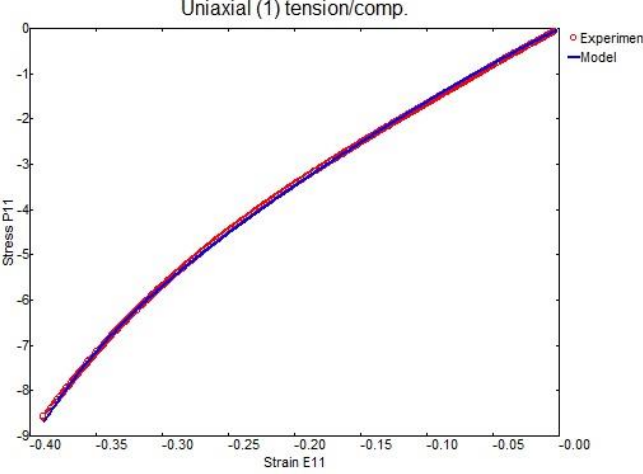
These material coefficients can then be used within a FEA code to model a hyperelastic rubber-like material if proven stable. The fitted results for five different samples will now be discussed.

3.3.4. FITTED RESULTS AND DISCUSSION

The Yeoh model material constants are derived for five rubber polyurethane samples, each with a different hardness value. These samples were tested by a uniaxial compression test and the results thereof are fitted by the least square procedure described in the previous section in order to obtain the material coefficients of the Yeoh material model. The results of the five polyurethane rubber samples with different hardness are shown and compared in Table 1.

Table 1 - Fitted data results

Hardness [Shore A]	Fitted Yeoh Material Model	R_{Error} [%]	Yeoh Material Constants [MPa]
40		0.93	$C_{10} = 0.585$ $C_{20} = 0.009$ $C_{30} = 0.004$
50		1.2	$C_{10} = 0.958$ $C_{20} = -0.027$ $C_{30} = 0.043$

Hardness [Shore A]	Fitted Yeoh Material Model	R_{Error} [%]	Yeoh Material Constants [MPa]
60		1.5	$C_{10} = 1.498$ $C_{20} = -0.281$ $C_{30} = 0.172$
65		2.1	$C_{10} = 1.749$ $C_{20} = -0.500$ $C_{30} = 0.297$
80		0.81	$C_{10} = 2.454$ $C_{20} = -0.717$ $C_{30} = 0.454$

3.3.5. CONCLUSION

For all the samples with each a different hardness the Yeoh material model showed a good fit with good progression outside the interval of experimental data where the maximum relative error value of all the different tests is below 2.2 [%].

The next step before a complete shock absorber can be modelled is to validate the stability and ability of the Yeoh material model when used to numerically predict the behaviour of the polyurethane rubber. The reason is that the test data were compiled from simple a uniaxial-compression test while the geometry and load case for a real-life shock absorber differ significantly from that.

3.4. VALIDATION OF THE YEOH MATERIAL MODEL

In this section the stability and ability of the Yeoh material model will be determined. This is necessary to establish if the Yeoh material model derived from a single uniaxial compression test can give a good numerical prediction of the behaviour of the polyurethane rubber when assigned to more complex geometries subjected to compression loads.

3.4.1. VALIDATION PROCEDURE

The Yeoh material model ability and stability will be validated by modelling the JS-1 rubber shock absorber and numerically simulating the behaviour of the polyurethane rubber when being compressed. The reason for using the JS-1 shock absorber geometry is that this absorber and shock elements are available and that the absorber element for the JS-2 will probably be of similar design.

The polyurethane rubber will be modelled by the Yeoh material model that will be implemented as a user-defined hyperelastic material in the nonlinear FEA PATRAN code and simulated in the MARC and NASTRAN codes. The load and deflection results of the experimental compression test and numerical simulation will then be compared that will determine if the Yeoh material model has the ability to give a stable prediction of the behaviour of polyurethane rubber when applied to more complex geometries.

3.4.2. JS-1 RUBBER SHOCK ABSORBER

The JS-1 shock absorber is a mechanical shock element that primarily uses polyurethane rubber as the shock absorbing medium to absorb the impact during landing. This JS-1 shock absorber consists of two 65 [Shore A] polyurethane rubber discs, each with a height of 50 [mm] and inner and outer diameter of 19.1 [mm] and 50.8 [mm]. The two polyurethane rubber discs are separated by a stainless steel divider washer that is slightly bigger than the discs. The two discs and washer are kept in line by a low alloy steel centre tube and are compressed by two medium carbon steel end caps, one on the bottom and one at the top that are illustrated in Figure 19.

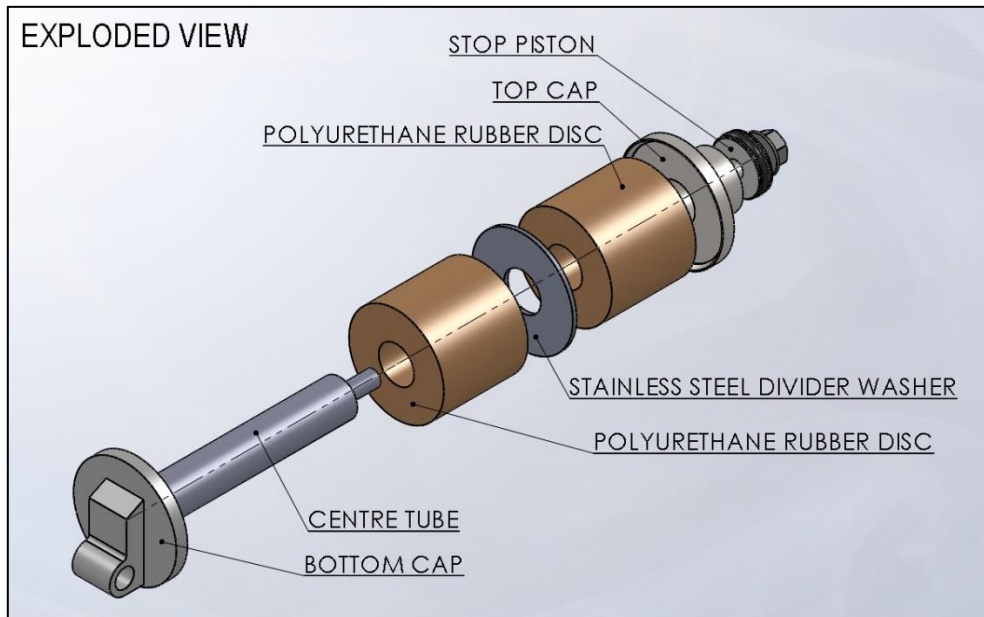


Figure 19 - JS-1 Shock exploded view (Jonker Sailplanes C.C., 2013)

3.4.3. MODELLING OF JS-1 RUBBER SHOCK ABSORBER

SOLID MESH AND MODEL CONSTRAINTS

The complete JS-1 shock absorber, as discussed in the previous chapter, involves basically four materials, polyurethane rubber, low alloy steel, medium carbon steel and stainless steel. Due to the great difference in stiffness between the metal and the rubber parts and because the polyurethane rubber's behaviour is the only material under investigation, it was decided to model the polyurethane rubber and stainless steel divider washer as deformable bodies and the rest of the parts as rigid surfaces.

The polyurethane rubber and stainless steel divider washer deformable bodies were modelled using 4-node quadrilateral hybrid solid elements with a global element edge length of 4 [mm], shown in Figure 20.

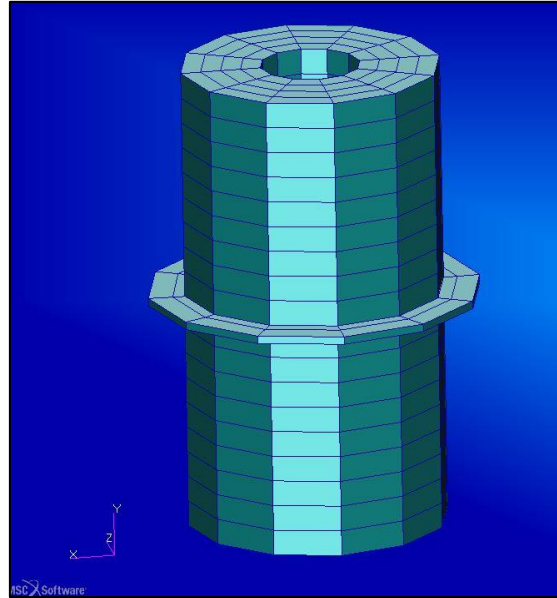


Figure 20 - FE mesh of JS-1 rubber shock absorber elements

These two rubber elements and the centre washer were then constrained between two rigid surfaces at the top and bottom as shown in Figure 21. It was further constrained axially by a cylindrical rigid surface situated in the middle of the discs that represents the centre tube of the JS-1 shock absorber. In order to represent the actual behaviour of the experimental test of the JS-1 shock absorber being compressed, the bottom and cylindrical centre tube rigid surfaces were constrained in all directions, shown in Figure 21. The upper rigid surface applying the load was constrained in all directions except for the vertical direction, allowing the load to be applied to the polyurethane rubber discs.

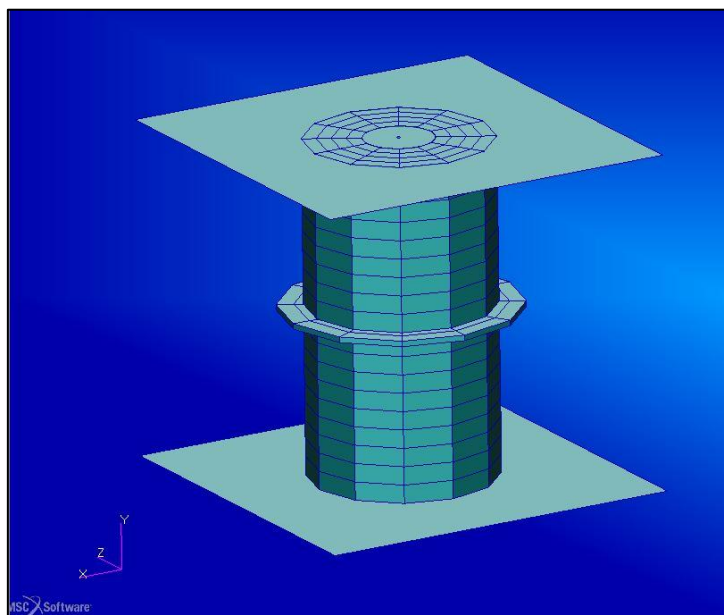


Figure 21 – JS-1 FE model constraints

MATERIAL PROPERTIES ASSIGNMENT

The properties of the stainless steel were assigned to the numerical model of the divider washer by using PATRAN isotropic linear elastic material model where the Young's modulus (E) is equal to $E = 200 \text{ [GPa]}$ and Poisson ratio (ν) is equal to $\nu = 0.29 \text{ [-]}$. The material properties of the polyurethane rubber were assigned to the rubber discs by using PATRAN nonlinear nearly incompressible hyperelastic material model known as MATHE. The hyperelastic material model MATHE with only one polynomial order had to be adapted in order to accommodate the Yeoh material model. This was done by inserting a zero value for constants C_{01} and C_{11} . In order to accurately simulate the nearly incompressible behaviour of the material, the poisson's ratio was selected equal to $\nu = 0.499 \text{ [-]}$ and the mass density of the material was selected according to the manufacturer's data that is equal to $\rho = 1.017 \times 10^{-6} \text{ [}\frac{\text{Kg}}{\text{mm}^3}\text{]}$. The Yeoh material coefficients of a hardness of 65 [Shore A] were used to predict the behaviour of the polyurethane rubber discs. The values of the Yeoh material coefficients of the selected polyurethane rubber derived are shown in Table 2:

Table 2 - Yeoh Material Coefficient for JS-1 shock absorber discs

Hardness [Shore A]	Yeoh Material Coefficients		
	C_{10} [MPa]	C_{20} [MPa]	C_{30} [MPa]
65	1.749	-0.500	0.297

LOADS APPLIED TO THE MODEL

In order to improve the accuracy of the simulation it was decided to model the friction between the rubber and its contacting surfaces. It was decided to use a friction coefficient of $\mu=0.5 \text{ [-]}$ wherever the rubber disc elements came in contact with other rigid surfaces.

The compression load was applied to the model by assigning the loads to the top rigid surface. The load was then systematically applied in the vertical direction that compressed the rubber discs. The load being applied was equal to the load that was used in the practical test equal to $F = 8500 \text{ [N]}$. The simulation was then run as an implicit nonlinear simulation (SOL 600) that was able to solve the nonlinear behaviour of hyperelastic materials. The maximum deflection obtained from the SOL 600 simulation is shown in Figure 22.

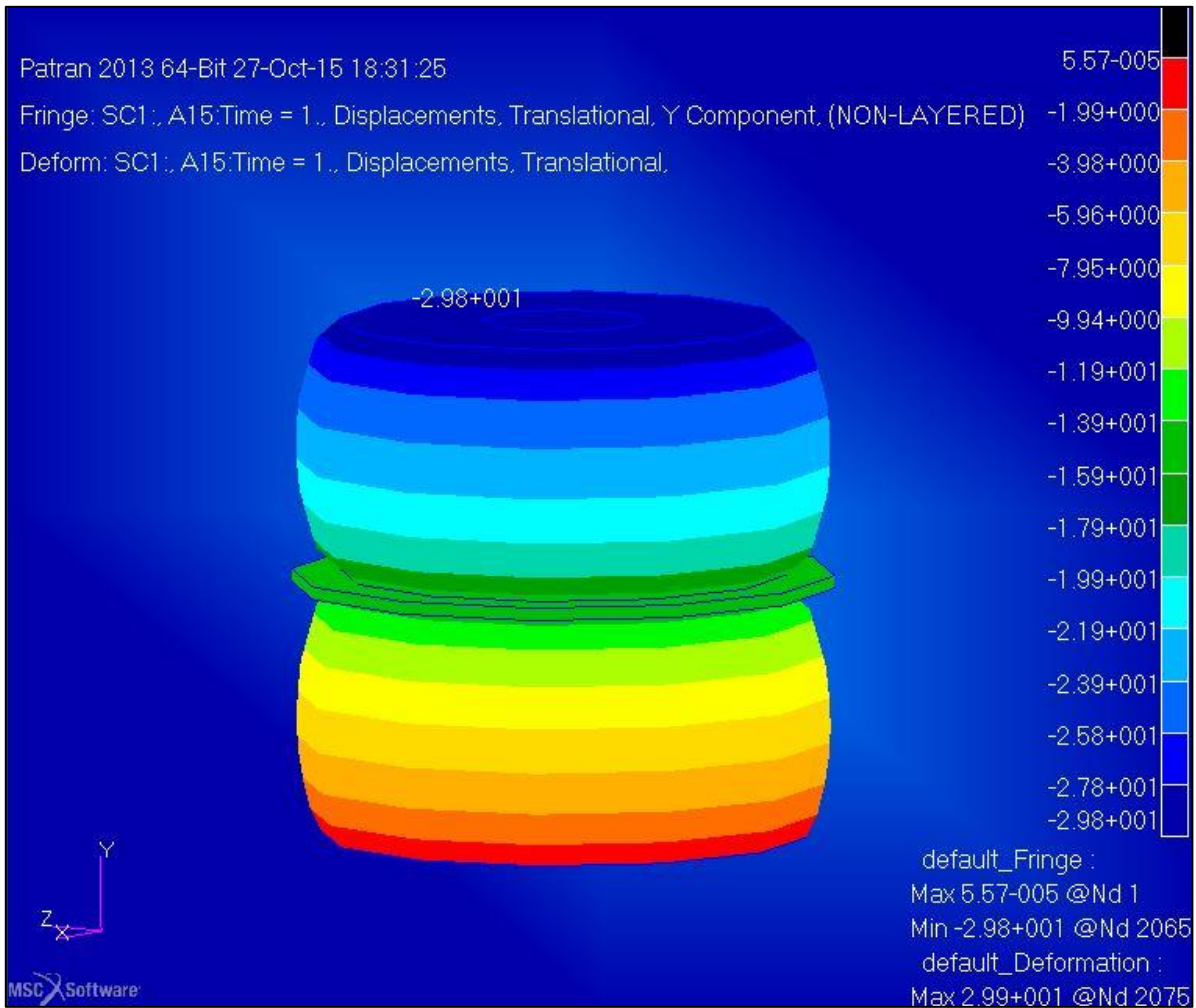


Figure 22 - JS-1 Rubber element FEA result

3.4.4. LOAD-DEFLECTION RESULTS

The load-deflection data of the experimental compression test and numerical FEA solution was compared by plotting the load-deflection curves against each other; this was done to distinguish the difference between the two data sets. The recorded experimental load-deflection data of the JS-1 shock absorber can be found in *APPENDIX B*. The comparison of the results between the FEA and the experimental compression test is shown in Figure 23 and in Table 3.

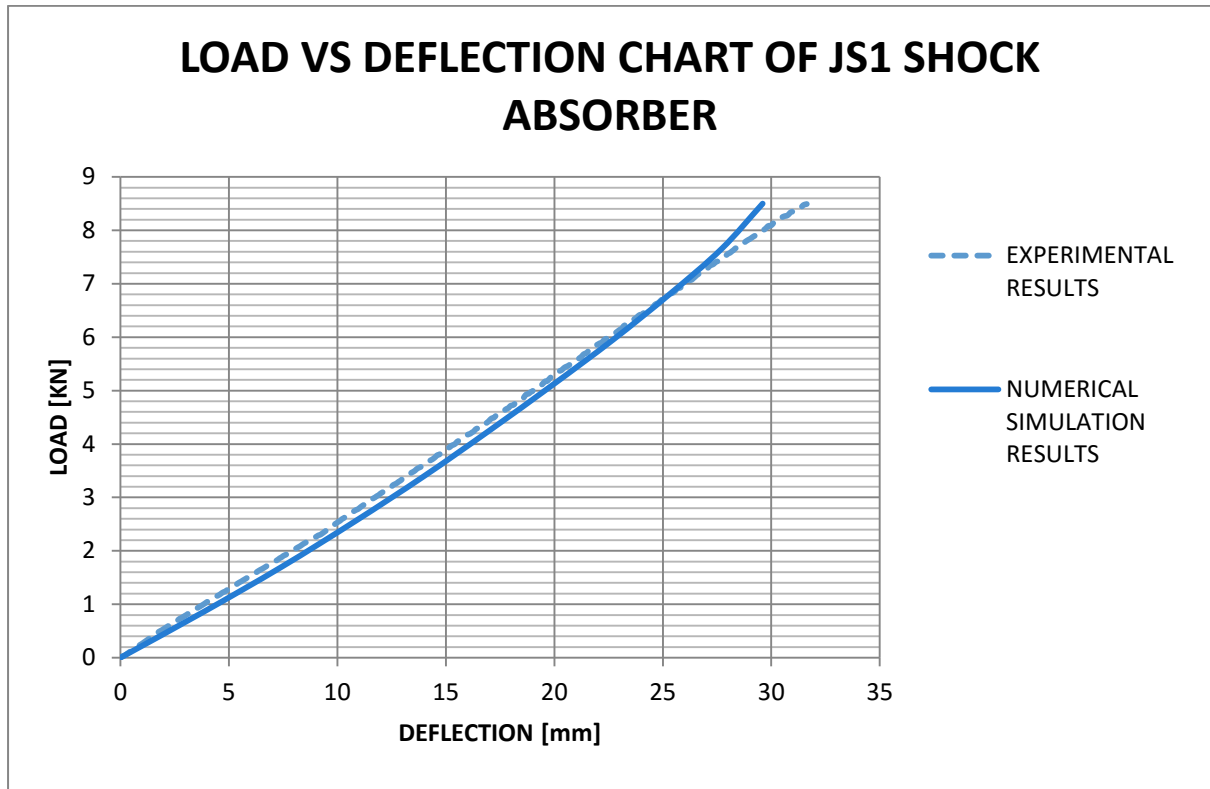


Figure 23 - Load deflection curve of JS-1 shock absorber

Table 3 - Compression load deflection comparison

Compression Load	Experimental Deflection results	Numerical Simulation Deflection Results	Percentage difference to experimental results
[N]	[mm]	[mm]	[%]
850	3.212	3.6	11.39%
1700	6.67	7.23	8.06%
2550	10.04	10.6	5.43%
3400	13.25	13.7	3.34%
4250	16.33	16.8	2.84%
5100	19.4	19.8	2.04%
5950	22.3	22.65	1.56%
6800	25.3	25.3	0.00%
7650	28.29	27.9	1.39%
8500	31.6	29.8	5.86%
Average difference			4.19%

Figure 23 shows that the calculated and measured data correlates well up to the 25mm deflection position. Table 3 shows that the average difference is 4.19 [%] with a maximum difference of 11.4 [%] at the low deflection range. According to Freakly and Payne (1978), the allowable difference in results is equal or below 15 [%], concluding that the FEA model, with the three term Yeoh model, is suitable to be used in the landing gear analysis.

3.5. SUMMARY

This chapter has validated that the Yeoh material model derived from a single uniaxial compression test can numerically model the polyurethane rubber behaviour when subjected to compression loads. This was shown by determining the Yeoh material model coefficients with a single uniaxial compression test and then numerically modelling the JS-1 shock absorber with the derived Yeoh material model and subjecting it to compression load in the commercially available FEA PATRAN code and comparing it to the experimental compression results.

Having validated that the polyurethane rubber was modelled correctly with the Yeoh material model in the commercially available FEA PATRAN code, it is now possible to use the validated Yeoh material model to accurately and numerically develop the polyurethane rubber element in the JS-2 shock absorber by using the FEA code.

CHAPTER 4 : DEVELOPMENT OF THE LANDING GEAR SHOCK ABSORBING SYSTEM

4.1. INTRODUCTION

In this chapter the analysis and development of the shock absorbing system of the JS-2 landing gear will be discussed. This includes the development of the polyurethane rubber shock absorber element and the selection of the wheel and the tyre according to the CS-22 certification specifications.

There are several different methods that can be followed to develop these mechanical subsystems. The method used in this study is based on the design requirements of Jonker Sailplanes and the CS-22 certification specifications.

It will be shown analytically that the requirements of the EASA CS-22 certification specifications are met. In order to do so, basic design parameters that have an influence on the development of the shock absorbing system must be discussed.

4.2. DESIGN PARAMETERS

Several different design parameters must be considered when a shock absorbing system is developed. The initial layout influences all of the other parameters and will, therefore, be considered first. Other parameters are the required load factors from the CS-22 specification, materials used and the shock absorption qualities thereof.

4.2.1. INITIAL LAYOUT

In order to be able to develop the shock absorbing system it is important to get an overall picture of the landing gear layout. This step is necessary to ensure that all the shock absorbing limitations and restrictions are kept in mind when developing the shock absorbing system.

The landing gear is developed for a single-seat all composite glider certified according to EASA CS-22 certification specifications for sailplanes and powered sailplanes. These certification specifications define the shock absorbing requirements that are legally imposed on the landing

gear. The initial layout of the retractable landing gear is a single main wheel configuration that is positioned forward of the aircraft centre of gravity position, shown in Figure 24.

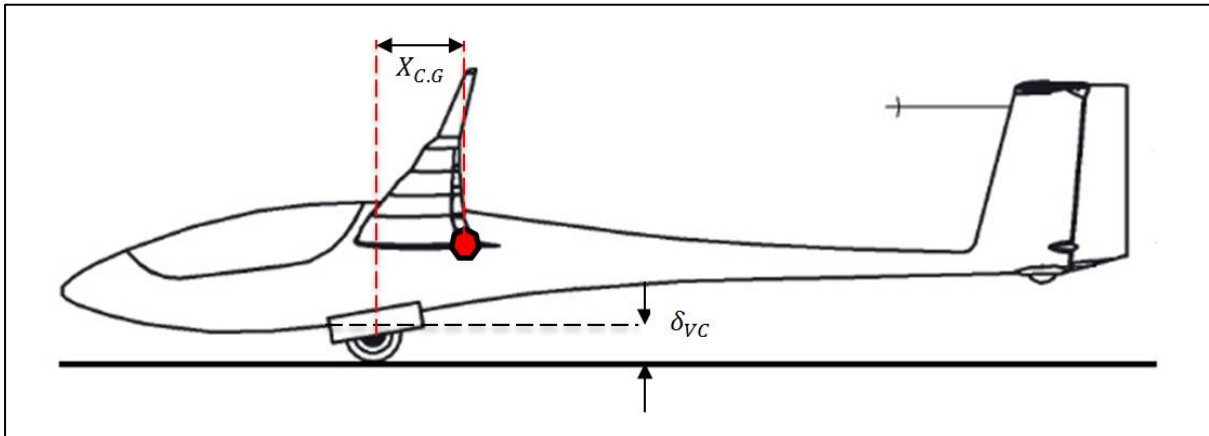


Figure 24 - Initial position of the landing gear

In order to assure that the sailplane's fuselage and additional components are not damaged during landing a minimal clearance between the fuselage and ground surface has to be adhered to when designing the landing gear system. As illustrated in Figure 24, the minimum allowable vertical ground clearance for the JS-2 is defined by Jonker Sailplanes as 100 [mm].

The specific kinematic geometry that is going to be used for the retracting system is a three arm setup with folding rear arm as shown in Figure 25. This layout is similar to the layout used on the JS-1 and several other modern gliders.

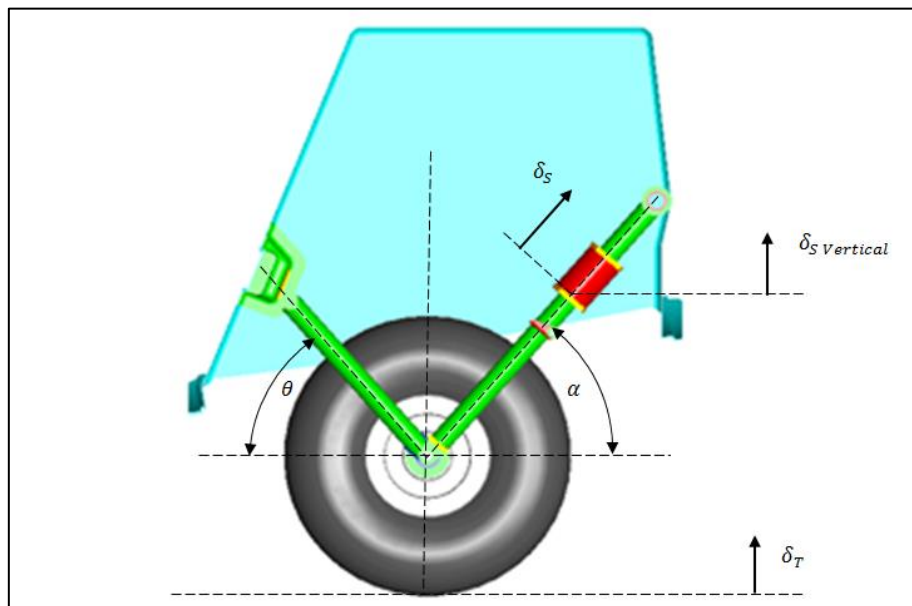


Figure 25 - Initial layout of shock absorbing system

The shock absorption is provided by two rubber shock absorbing elements in the rear strut arm of the landing gear and the single main landing gear tyre. These elements must absorb all the energy without being fully depressed as required by CS22.725 (b).

4.2.2. LOAD FACTORS

In order to ensure that the loads being transferred to the airframe structure are safe, it is necessary to use the correct load factors to develop the landing gears shock absorbing system. Load factors applied to the landing gear should not be confused with the aircraft load factors. According to ground load conditions and assumptions of paragraph CS 22.473 (a) to (c) of Subpart C of the CS-22 certification specifications there are three different types of load factors used to design the landing gear system.

These include a vertical inertia load factor (n_{VI}) acting through the centre of gravity position of the aircraft, a landing gear load factor (n_{LG}) acting onto the landing gear structure and a ground reaction load factor (n_{GR}) acting onto the landing gear main wheel at the point where it comes into contact with the ground. Each of these load factors can be further divided into a limit and ultimate (Currey, 1988). According to paragraph CS 22.301(a) of Subpart C the limit load factors will represent the maximum loads that will be expected in service and the ultimate load factors will represent limit loads multiplied by a prescribed factor of safety. To get a better understanding of each of these load factors see Figure 26.

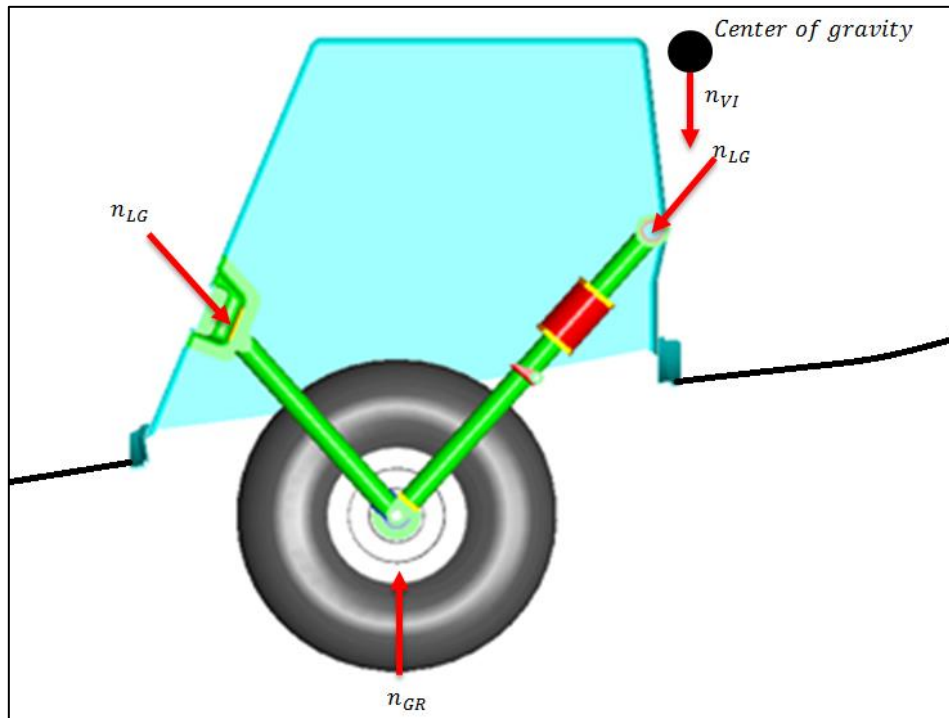


Figure 26 - Load factor layout

By assuming that the aircraft is a rigid body with no relative acceleration between the centre of gravity position and the landing gear attachment point to the aircraft, it can be deduced that the load factor at each of these positions is equal to each other. This relation can be illustrated by the following equation:

$$n_{VI} = n_{LG} \quad (4.1)$$

Where;

n_{VI} – is the vertical inertia load factor [-],

n_{LG} – is the landing gear load factor [-].

According to paragraph CS 22.473(c) of Subpart C in CS-22, another relationship between the ground reaction load factor and the vertical inertia load factor can be achieved. The relation includes that the ground reaction load factor may be equal to the vertical inertia load factor if it is assumed that the weight of the aircraft will be balanced by the lift of the sailplane's wings during the landing impact. This relation can be illustrated by the following equation:

$$n_{GR} = n_{VI} - 1 \quad (4.2)$$

Where;

n_{VI} – is the vertical inertia load factor [-],

n_{GR} – is the ground reaction load factor [-].

The selected limit vertical inertia load factor ($n_{VI\ LIM}$), according to paragraph CS 22.473(b) of Subpart C in CS-22, which is used to design the structure of the landing gear may not be less than 3.0 [-], nor may the limit ground reaction load factor ($n_{GR\ LIM}$) be less than 2.0 [-] when selecting the main landing gear wheel. The corresponding ultimate vertical inertia load factor ($n_{VI\ ULT}$) may not exceed a value of 4.5 [-], according to paragraph CS 22.473(b) and CS 22.725(c) of CS-22 certification specifications, nor may the ultimate ground reaction load factor ($n_{GR\ ULT}$) exceed 3.5 [-] when implementing *Equation 4.2*.

4.2.3. SHOCK ABSORPTION

According to the CS-22 certifications specifications, Subpart D, paragraph CS 22.725(a), the shock absorbing system, including the tyre, must be capable of absorbing a maximum amount of energy developed in a landing without being fully depressed. The total amount of energy is a sum of the kinetic and potential energy, where the kinetic energy is dependent on the design maximum weight and the descent velocity of the aircraft and the potential energy is dependent on the design maximum weight and the deflection of the landing gear.

By using the energy conservation law, the amount of energy absorbed by the landing gear can be determined by the total deflection of the landing gear. The approximate deflection required by the main landing gear can be determined by an equation based on the law of energy conservation as follow:

$$\frac{1}{2}m_m V_Z^2 + m_m g d = m_m g n_{LG \ LIM} \eta_t d_t + m_m g n_{LG \ LIM} \eta_s d_s + L m_m g d \quad (4.3)$$

with;

$$\text{Kinetic Energy} = 1/2 m_m V_Z^2$$

$$\text{Gravitational Potential Energy} = m_m g d$$

$$\text{Energy absorbed by the main landing gear tyre} = m_m g n_{LG \ LIM} \eta_t d_t$$

$$\text{Energy absorbed by the main landing gear shock absorber} = m_m g n_{LG \ LIM} \eta_s d_s$$

$$\text{Energy absorbed by wings lift} = L m_m g d$$

where;

- m_m – is the design maximum weight [kg],
- V_Z – is the descent velocity [m.s⁻¹],
- g – is the standard gravitational acceleration [m.s⁻²],
- d – is the total landing gear deflection ($d_s + d_t$) [m],
- d_s – is the shock absorber system deflection [m],
- d_t – is the main landing gear tyre deflection [m],
- $n_{LG \ LIM}$ – is the limit landing gear load factor [-],
- η_s – is the shock absorber absorption efficiency [-],
- η_t – is the main landing gear wheel absorption efficiency [-],
- L – is the lift ratio [-].

Equation 4.3 differentiates between the energy absorbed by the shock absorber and the energy absorbed by the main wheel tyre. This is necessary to adequately incorporate the different shock absorbing efficiencies of the shocks and tyre in the balanced energy equation.

This equation can be simplified by applying paragraph CS 22.473 (c) as prescribed in Subpart C of the CS-22 certification specifications. The assumption states that the lift of the sailplane wing will balance the weight of the sailplane throughout the landing impact. This assumption implies that the Lift ratio (L) in Equation 4.3 can be assumed to be equal to 1 [-]. By applying this assumption the balanced energy equation can be rearranged and written as follow:

$$\frac{1}{2}m_m V_Z^2 = g n_{LG\ LIM} m_m (\eta_s d_s + \eta_t d_t) \quad (4.4)$$

where;

- m_m – is the design maximum weight [kg],
- V_Z – is the descent velocity [m.s⁻¹],
- g – is the standard gravitational acceleration [m.s⁻²],
- d_s – is the shock absorber system deflection [m],
- d_t – is the main landing gear tyre deflection [m],
- $n_{LG\ LIM}$ – is the limit landing gear load factor [-],
- η_s – is the shock absorber absorption efficiency [-],
- η_t – is the main landing gear wheel absorption efficiency [-].

With the wing lift balancing the weight of the sailplane throughout the landing impact, the only energy left that the tyre and shocks have to absorb is the kinetic energy that is only dependent on the maximum landing weight (m_m) of the sailplane and the vertical descent velocity (V_Z). These two parameters are specified in Subpart C and D, paragraph CS 22.725(b) and CS 22.473(b) of the CS-22 certification specifications.

These paragraphs conclude that while the weight of the sailplane is balanced by the lift of the wing the value of the kinetic energy that has to be absorbed must be determined under the assumption that the weight of the sailplane corresponds to the design maximum weight (m_m) at a constant decent velocity of 1.77 [m/s]. The design weight of the sailplane (m) is thus assumed to be equal to the design maximum weight (m_m) throughout the development of the landing gear.

4.2.4. SHOCK ABSORPTION TEST

According to the CS-22 the landing gear shock absorbing system has to ensure that the system will be able to supply sufficient shock absorption. In order to ensure this the CS-22 certification specifications prescribed in paragraph CS 22.723 indicate that the landing gear must be able to absorb 1.44 times the maximum amount of energy described in CS 22.473 without failing.

In order to prove that this design will adhere to the CS-22 specifications the shock absorbing system will be developed to be able to absorb an ultimate amount of energy of 1.44 times the kinetic energy that is developed by the aircraft without bottoming out or exceeding the load factor requirements of the CS-22.

By applying this assumption, the balanced energy equation can be rearranged and written as follow:

$$1.44 \times \left(\frac{1}{2} m_m V_Z^2 \right) = g n_{LG\ ULT} m_m (\eta_s d_s + \eta_t d_t) \quad (4.5)$$

where;

- m_m – is the design maximum weight [kg],
- V_Z – is the descent velocity [m.s⁻¹],
- g – is the standard gravitational acceleration [m.s⁻²],
- d_s – is the shock absorber system deflection [m],
- d_t – is the main landing gear tyre deflection [m],
- $n_{LG\ ULT}$ – is the landing gear load factor [-],
- η_s – is the shock absorber absorption efficiency [-],
- η_t – is the main landing gear wheel absorption efficiency [-].

As shown in *Equation 4.5*, the amount of energy being absorbed by the shock absorbing system is a function of its efficiency and its vertical displacement. Now that the maximum amount of energy that has to be absorbed by the landing gear system can be calculated, the shock absorbing system can be developed according to the total allowable vertical displacement of the sailplane and the reaction load factor limitations stated by the CS-22 certification specifications.

4.2.5. INITIAL PARAMETERS

In order to develop the polyurethane rubber shock absorber and to select the correct tyre, an appropriate set of initial parameters have to be selected. One of these parameters that have to be selected is the ultimate initial landing gear load factor. According to Currey (1988), a general guide for selecting a landing gear load factor used to calculate the stroke of the shock absorbing system is between 2 [-] and 4[-]. The load factor that is chosen is 3.5 [-], which is within the requirements of the CS-22 specifications. The parameters selected to develop the shock absorbing system, is shown in Table 4:

Table 4 - Initial design parameters

Parameter	Description	Value	Unit
$n_{LG\ ULT\ Initial}$	The initial ultimate landing gear load factor, selected within the CS-22 requirements.	3.5	[-]
m_m	The design maximum weight, JS requirement.	600	[Kg]
V_Z	The vertical descent velocity, CS 22 requirement.	1.77	[m/s]
g	The standard gravitational acceleration.	9.81	[m/s ²]
δ_{MVC}	The minimum vertical clearance between fuselage and the ground, JS requirement.	100	[mm]

The ultimate amount of energy(E_{ULT}) that has to be absorbed by the tyre and the rubber shock absorber can be calculated by using the first section of *Equation 4.5* and the initial parameters shown in Table 4:

Table 5 - Ultimate and maximum energy

Maximum amount of energy (E_{Max})	Ultimate amount of energy (E_{ULT})
$E_{Max} = \left(\frac{1}{2} m_m V_Z^2 \right)$ $E_{Max} = \frac{1}{2} \times 600 \times 1.77^2$ $E_{Max} = 939.86 [J]$	$E_{ULT} = 1.44 \times E_{Max}$ $E_{ULT} = 1.44 \times \left(\frac{1}{2} m_m V_Z^2 \right)$ $E_{ULT} = 1.44 \times \frac{1}{2} \times 600 \times 1.77^2$ $E_{ULT} = 1353.41 [J]$

This displacement together with the load determined from the ultimate ground reaction load factor specified in the CS-22 will be used to select an appropriate tyre for the design. The shock absorbing capabilities of the tyre selected will then be used to develop and determine the size of the rubber shock absorbing elements.

4.3. MAIN LANDING GEAR TYRE AND WHEEL

Tyres and wheels that are primarily used for sailplanes are selected based on their static and dynamic loads as well as their braking capabilities of the wheel's internal braking mechanism. Tyres typically have two load ratings: maximum load refers to the maximum static load (limit load) the tyre is capable of withstanding, and the ultimate load, also known as the bottoming load, refers to the load carried by the tyre at ultimate deflection. Similar to the tyre, the landing gear's wheel also has two load ratings; static load rating (limit load) and a ultimate load rating that has to be equal or exceed the loads of the tyre (Pauliny & Juracka, 2014).

Therefore, in this section the required ultimate load rating and braking energy according to the design requirements are calculated in order to select the most appropriate tyre, wheel and brake set for the landing gear. Where the ultimate load rating concludes if the tyre and wheel are strong enough in a structural view point and the maximum corresponding displacement of the tyre together with its calculated shock absorbing efficiency determines the ultimate amount of energy it will absorb.

4.3.1. REQUIRED ULTIMATE LOAD RATING

As discussed previously in *Section 4.2.2. Load Factors*, the load factor applicable to the tyre and wheel is the ground reaction load factor (n_{GR}). The ultimate ground reaction load factor ($n_{GR\ ULT}$) specified by the CS-22 will thus be used with the design maximum weight (m_m) in order to determine the minimum required ultimate load rating (L_{ULT}). The minimum required ultimate load rating is calculated as follow:

$$L_{ULT} = m_m n_{GR\ ULT} \quad (4.6)$$

$$L_{ULT} = 600 [kg] \times 3.5 [-]$$

$$\therefore L_{ULT} = 2100 [kg]$$

Where;

L_{ULT} –	is the required ultimate load rating [kg],
m_m –	is the design maximum weight, equal to 600 [kg],
$n_{GR\ ULT}$ –	is the ultimate ground reaction load factor, equal to 3.5[-].

The required ultimate load rating will now be used to select the appropriate tyre and wheel with an equal or higher limit load rating (L_{ULT}).

4.3.2. TYRE AND WHEEL SIZING

In the initial design phase we can determine the tyre size by using a statistical approach. Given below is an equation developed to estimate the tyre size according to the aircraft's maximum design weight. The main landing gear tyre size will be predicted by using Taylor's diameter and width equations that are developed from recorded data (Taylor, 1976-77). The equation used to determine the tyre diameter of the main landing gear wheel (Taylor, 1976-77) is as follow:

$$D = Am_m^B \quad (4.7)$$

Where;

D –	is the tyre diameter[Inch],
m_m –	is the design maximum weight [kg],
A –	is equal to 1.51[-] when calculation the tyre diameter,
B –	is equal to 0.349[-] when calculating the tyre diameter.

The equation used to determine the tyre width of the main landing gear wheel (Taylor, 1976-77) is as follow:

$$W = Am_m^B \quad (4.8)$$

Where;

W –	is the tyre width[Inch],
m_m –	is the design maximum weight [kg],
A –	is equal to 0.715[-] when calculation the tyre diameter,
B –	is equal to 0.312[-] when calculating the tyre diameter.

By using the maximum design weight (m_m) defined in Table 4, the predicted tyre size calculated by using Equation 4.7 and 4.8 is shown in Table 6.

Table 6 - Predicted tyre size

	Predicted Tyre Diameter [Inches]	Predicted Tyre Width [Inches]
Maximum design weight (m_m) of 600 [kg]	14.1	5.2

Based on the maximum design weight (m_m), ultimate load rating (L_{ULT}) and the estimated size calculations in Table 6 the following two tyres are taken into consideration for the main landing gear tyre:

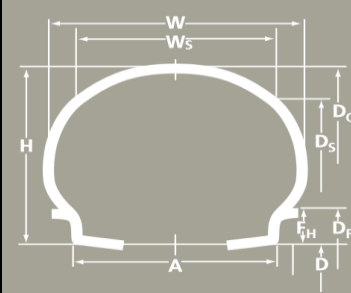
Table 7 - Main landing gear tyre specifications

Brand	Michelin Air	Goodyear
Type	BIAS Type III	BIAS TYPE III
Qualification standard	MIL-T-504	-
Size	5.00-5 [Inch]	5.00-5 [Inch]
Ply rating	10 [-]	10 [-]
Speed rating	250 [Km/h]	190 [Km/h]
TL/TT	TT	TT
Static load rating @ 6 Bar inflation pressure	975 [kg]	975 [kg]
Ultimate load rating @ 6 Bar inflation pressure	2600 [kg]	2630 [kg]
Tyre Dimension inflated to 6 Bar		
Tyre diameter	14.20 [Inch]	14.20 [Inch]
Tyre width	4.95 [Inch]	4.95 [Inch]

Although the specifications of the Michelin tyre are quite similar to those of the Goodyear tyre, the Michelin tyre is selected due to its higher speed rating and its availability. Now that the tyre data is known the landing gear wheel is selected to match the selected tyres and to meet or exceed their load requirements. According to CS-22, Subpart D, paragraph CS 222.731 (b), the maximum ultimate load rating of the wheel must be equal or exceed the maximum bottoming

load of the tyre. The wheel also has to meet the tyre's rim description. The rim description of the selected tyre is shown in Table 8:

Table 8 - Michelin Air 5.00-5 rim description

Michelin Air 5.00 – 5 BIAS TYPE III TYRE				
A Width Between Flanges [mm]	D Specified Rim Diameter [mm]	F_H Flange Height [mm]	D_F Outer Flange Diameter [mm]	
88.9	127	19.05	165.1	

Based on the tyre specifications and the CS-22 consideration, the following two wheels meet the requirements and are compared in Table 9:

Table 9 - Main landing gear wheel and brake set specifications

Wheel Specifications		
Brand	Beringer Wheels and Brakes	Matco MFG
Part	JA-01	WHLW50LXT
Type	Tubeless Wheel	Tube Wheel
Size	5.00-5 [Inch]	5.00-5 [Inch]
Speed rating	250 [Km/h]	-
Static load rating	650 [kg]	640 [kg]
Ultimate load rating	2600 [kg]	2095 [kg]
Brake Set Specifications		
Brake calliper	EA-006	WHLB5T
Brake disc	DSC-006	WHLD5T
Braking torque	2450 [Inch-lbs]	6143 [Inch-lbs]
Maximum kinetic energy rating at rejected take-off (RTO)	140000 [ft-lbs]	337932 [ft-lbs]
Weight of wheel and brake set	1.977 [Kg]	4.173 [Kg]

The decision of which wheel to select is primarily influenced by its ability to accommodate the selected tyre, and secondly by the braking capability of the internal braking system (Chai &

Mason, 1996). The braking capability of the braking system is based on its ability to absorb the kinetic energy generated by the aircraft's maximum landing weight (m_m) and its horizontal stalling speed (V_{Stall}). The total amount of kinetic energy that has to be absorbed is determined by using the following equation (Chai & Mason, 1996):

$$KE = 0.0443 m_m V_{Stall}^2 \quad (4.9)$$

Where;

KE – is the total kinetic energy [ft-lbs],

V_{Stall} – is the horizontal stalling speed [knots],

m_m – is the design maximum weight [lbs].

The total kinetic energy that has to be absorbed for this scenario is calculated, and the results are shown in Table 10.

Table 10 - Calculated kinetic braking energy

m_m		V_{Stall}		KE
[Kg]	[lbs]	[Km/h]	[Knots]	[ft-lbs]
600	1322.77	90	48.596	138 385.029

Both of the wheels' brake sets that are referred to in Table 9, comply with the braking requirements shown in Table 10, making them both eligible for selection. The final decision in selecting the appropriate set will be based on the overall weight of each system. The wheel and brake set selected is the Beringer set that is half of the weight of the Matco set, as shown in Table 9.

Now that the tyre, wheel and brake set are selected for the development of the shock absorbing system, the amount of energy that the tyre is capable of absorbing can be determined by using the data given by the tyre manufacturer.

4.3.3. TYRE ENERGY ABSORPTION

The maximum amount of energy that the tyre is capable of absorbing is defined by the tyre's allowable vertical displacement (d_t) before it bottoms out and the tyre's shock absorbing efficiency (η_T). The ultimate amount of energy that the tyre is capable of absorbing is prescribed according to the energy conservation theory previously discussed in Section 4.2.2. Load Factors, and is shown as follow:

$$E_T = m_m g n_{LG \text{ } ULT \text{ } Initial} \eta_t d_t \quad (4.10)$$

Where;

- E_T – is the ultimate amount energy absorbed by the tyre [J],
 m_m – is the design maximum weight [kg],
 g – is the standard gravitational acceleration [m.s^{-2}],
 d_t – is the maximum vertical deflection of the tyre before bottoming out [m],
 $n_{LG \text{ } ULT \text{ } Initial}$ – is the initial ultimate landing gear load factor [-],
 η_t – is the tyre shock absorber absorption efficiency [-].

The data used to calculate the maximum amount of energy that the tyre is capable of absorbing in Equation 4.10 are shown in Table 11 as follow:

Table 11 - Tyre energy absorption data

Variable	Description	Value	Unit
$n_{LG \text{ } ULT \text{ } Initial}$	The initial ultimate landing gear load factor, from Section 4.2.5.	3.5	[-]
m_m	Design maximum weight, JS-2 design specification.	600	[kg]
g	The standard gravitational acceleration.	9.81	[m/s^2]
Tyre data			
η_t	Main landing gear wheel absorption efficiency (Currey, 1988).	0.47	[-]
d_t	The maximum vertical deflection of the tyre, before bottoming out (Michelin, 2015).	77	[mm]

The maximum amount of energy that the tyre is capable of absorbing (E_T) according to Equation 4.10 and the values shown in Table 11 are calculated as follow:

$$E_T = m_m g n_{LG \text{ ULT Initial}} \eta_t d_t$$

$$E_T = 600 [kg] \times 9.81 \left[\frac{m}{s^2} \right] \times 3.5[-] \times 0.47[-] \times 0.077[m]$$

$$\therefore E_T = 745.55 [J]$$

This is the ultimate amount of energy that the tyre is capable of absorbing for the selected load, determined by the selected load factor. Before this data will be used to develop the rubber shock absorbing element, the stiffness of the tyre has to be determined in order to be able to deflect 77 [mm] for the selected load factor of 3.5 [-].

4.3.4. TYRE VERTICAL STIFFNESS

The stiffness of a tyre is mainly influenced by the tyre's internal pressure. As the tyre's pressure increases the stiffness also increases. To ensure that the tyre is able to deflect 77 [mm] vertically when the load, calculated by the selected load factor, is applied, the tyre has to be inflated to the correct internal pressure. By assuming that the tyre deflection is primarily linear, the tangent vertical stiffness (K_Z) can be determined in order to be used to calculate the internal pressure. The relation between static displacement and load is thus given as follow:

$$K_Z = \frac{F}{d} \quad (4.11)$$

Where;

F – is the load, determined by the initial selected load factor [Kg],

d – is the deflection [mm],

K_Z – is the tangential vertical stiffness [kg/mm].

According to Padula (2006), the stiffness of pneumatic tyres is controlled primarily by their inflation pressure and the tyre's dimensions. Rhyne (2004) developed a model that is able to predict the tangent vertical stiffness (K_Z) of a ring-shaped pneumatic tyre according to its inflation pressure.

Padula (2006) adopted Rhyne's (2004) model and developed a new model that is able to calculate the tangent vertical stiffness (K_Z) for any given tyre size and inflation pressure. The model used to calculate the tangent vertical stiffness (K_Z) expressed in ways that are useful to the engineering bodies according to Padula (2006) is shown below:

$$K_Z = 0.00028P \sqrt{(-0.004 AR + 1.03)S_N \times \left(\frac{S_N AR}{50} + D_R\right)} + 3.45 \quad (4.12)$$

Where;

- K_Z – is the tangential vertical stiffness [kg/mm],
 P – is the tyre inflation pressure [KPa],
 AR – is the tyre aspect ratio[-],
 S_N – is the nominal section width of the tyre [mm],
 D_R – is the specific rim diameter [mm].

By using the data obtained from the tyre manufacturer (Michelin, 2015), the inflation pressure of the tyre can be determined with a maximum deflection at an initial landing gear load factor of 3.5 [-]. The data used in *Equation 4.11 and 4.12* to calculate the inflation pressure of the tyre are shown in Table 12 below:

Table 12 - Tyre geometry

Variable	Description	Value	Unit
$n_{LG\ Initial}$	The initial landing gear load factor, discussed in Section 4.2.5.	3.5	[-]
m_m	Design maximum weight, a JS design specification.	600	[kg]
Tyre Geometry			
S_N	The nominal section width of the tyre (Michelin, 2015).	127	[mm]
D_R	The specific rim diameter (Michelin, 2015).	127	[mm]
AR	Tyre aspect ratio (Michelin, 2015).	93	[-]

The tangent vertical stiffness (K_Z) according to *Equation 4.11* and the values shown in Table 12, is calculated as follow:

$$K_Z = \frac{L}{d}$$

$$K_Z = \frac{600 [kg] \times 3.5 [-]}{77 [mm]}$$

$$\therefore K_Z = 27.27 [kg/mm]$$

By using the obtained tangent vertical stiffness (K_Z) and the tyre geometry the required inflation pressure can be calculated by rearranging *Equation 4.12* as follow:

$$K_Z = 0.00028P \sqrt{(-0.004 AR + 1.03)S_N \times (\frac{S_N AR}{50} + D_R)} + 3.45$$

$$P = \frac{K_Z - 3.45}{0.00028 \times \sqrt{(-0.004 AR + 1.03)S_N \times (\frac{S_N AR}{50} + D_R)}}$$

$$P = \frac{27.27 [\frac{kg}{mm}] - 3.45}{0.00028 \times \sqrt{(-0.004 \times 93[-] + 1.03)127[mm] \times (\frac{127[mm] \times 93[-]}{50} + 127[mm])}}$$

$$\therefore P = 488.30[KPa]$$

Now that the main landing gear tyre and wheel have been selected and their capabilities are known the rubber shock absorbing element can be developed in order to adhere to the CS-22 certification specifications.

4.4. RUBBER SHOCK ABSORBING ELEMENT

In this section the rubber elements of the two shock absorbers situated in the shock strut arm will be developed according to its shock absorbing capabilities that are mainly determined by the rubber material mechanical properties and geometry.

This section will include the calculation of the required shock absorber stroke, the methods used to determine the initial size of the rubber shock absorbing element and the numerical FEA that will determine how much energy the polyurethane rubber elements will absorb.

4.4.1. REQUIRED SHOCK ABSORBER STROKE

In order to develop the shock absorber it is important to establish the required vertical displacement to ensure that a safe reaction load factor is obtained and enough energy is absorbed during landing. The required vertical displacement also known as the stroke is dependent on the shock absorbing efficiency of the rubber being used and the amount of energy that has to be absorbed. The energy that has to be absorbed by the rubber shock absorber is prescribed in *Equation 4.3* according to the energy conservation theory previously discussed in *Section 4.2.2. Load Factors* :

$$E_S = m_m g n_{LG\ ULT\ Initial} \eta_s d_s \quad (4.13)$$

Where;

- E_S – is the energy absorbed by the shock absorbing elements [J],
 m_m – is the design maximum weight [kg],
 g – is the standard gravitational acceleration [m.s⁻²],
 d_s – is the vertical shock absorber deflection [m],
 $n_{LG\ ULT\ Initial}$ – is the selected ultimate initial landing gear load factor [-],
 η_s – is the shock absorber absorption efficiency [-].

The amount of energy that has to be absorbed by the rubber shock elements (E_S) is determined by the amount already being absorbed by the tyre from the total amount of energy calculated in *Section 4.2.5. Initial Parameters* , shown in Table 13:

Table 13 - Energy absorbed

Variable	Description	Value	Unit
E_{ULT}	The ultimate amount of energy, calculated in <i>Section 4.2.5. Initial Parameters</i>	1353.41	[J]
E_T	The energy absorbed by the tyre, calculated in <i>Section 4.3.3.</i>	745.55	[J]

The amount of energy that has to be absorbed by the rubber shock absorbing system (E_S) is calculated by using the energy conservation theory and the data in Table 13, as shown:

$$E_S = E_{ULT} - E_T$$

$$E_S = 1353.41 [J] - 745.55[J]$$

$$E_S = 607.86 [J]$$

Now that the energy that has to be absorbed is known, the required vertical stroke length of the shock absorber can be calculated by *Equation 4.13* by using data from Table 14 within *Equation 4.3* and the assumption made by Currey (1988) that the shock absorbing efficiency of rubber shock absorbers is to be assumed to be 60 [%], the required vertical stroke of the shock absorber (d_s) can be calculated.

Table 14 - Shock absorber stroke calculation data

Variable	Description	Value	Unit
$n_{LG\ ULT\ Initial}$	The selected initial ultimate landing gear load factor.	3.5	[-]
m_m	Design maximum weight, JS-2 design specification.	600	[kg]
g	The standard gravitational acceleration.	9.81	[m/s ²]
η_s	Initial assumed rubber shock absorption efficiency, (Currey, 1988).	0.6	[-]

The required vertical deflection (d_s) of the rubber shock absorbing system according to the data in Table 14 *Equation 4.13*, is calculated as follow:

$$E_S = m_m g n_{LG\ ULT\ Initial} \eta_s d_s$$

$$d_{RVS} = \frac{E_S}{m_m g n_{LG\ ULT\ Initial} \eta_s initial}$$

$$d_{RVS} = \frac{607.86 [J]}{600 [kg] \times 9.81 \left[\frac{m}{s^2} \right] \times 3.5 [-] \times 0.6 [-]}$$

$$d_{RVS} = 0.04918 [m]$$

$$\therefore d_{RVS} = 49.18 [mm].$$

The required vertical stroke will now be used with the initial kinematics scheme of the landing gear to configure the size of the polyurethane rubber elements.

4.4.2. INITIAL KINEMATICS

The landing gear's initial kinematics scheme will be developed according to the required vertical deflection needed to obtain an allowable landing gear load factor that is defined in the design parameters in *Section 4.2.5. Initial Parameters* As stated in *Section 4.2.1. Initial Layout* the landing gear structure consists of two strut arms, where the rear arm is divided into two components, the shock strut and bottom strut. The shock strut houses two shock absorbers, one within each side tube. The basic geometry of the landing gear structure and the allowable space for the landing gear within the fuselage are illustrated by means of a line graph in Figure 27.

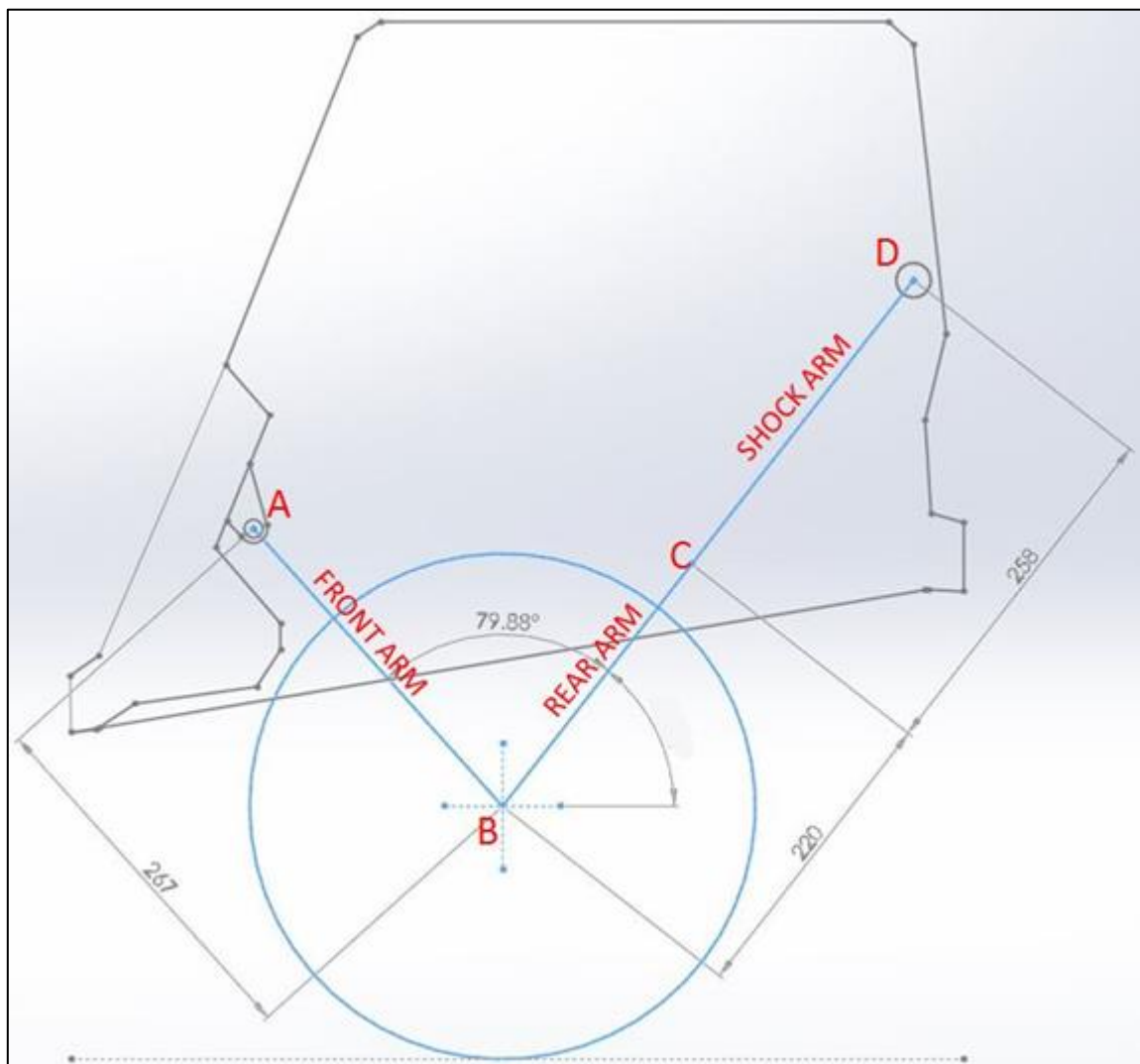


Figure 27 - Extended kinematic layout

The basic kinematic layout of the landing gear is used to determine the angle and possible length of the strut arms. In Figure 27, the grey lines illustrate the wheel box outline and the obstacles that are within the box, and the blue lines illustrate the landing gear strut arms and

outline of the main landing gear wheel with the four pivot points marked in red as A, B, C and D. There are several constraints that influence the length of the strut arms; the two constraints that are the most influential are: the main landing gear tyre has to stay clear of the wheel box and its internal obstacles while being retracted; the main landing gear must provide sufficient clearance between the ground and the fuselage when being compressed during landing. By keeping this in mind the most suitable strut lengths were determined and are shown in Table 15 and illustrated in Figure 27.

Table 15 - Strut arm lengths

Variable	Description	Value	Unit
<i>FA</i>	Front strut arm	267	[mm]
<i>RA</i>	Rear strut arm	220	[mm]
<i>SA</i>	Shock absorber strut arm	258	[mm]

The landing gear is then retracted by applying a torsional force to pivot point D in the clockwise direction. This retracted position of the landing gear is now shown in Figure 28.

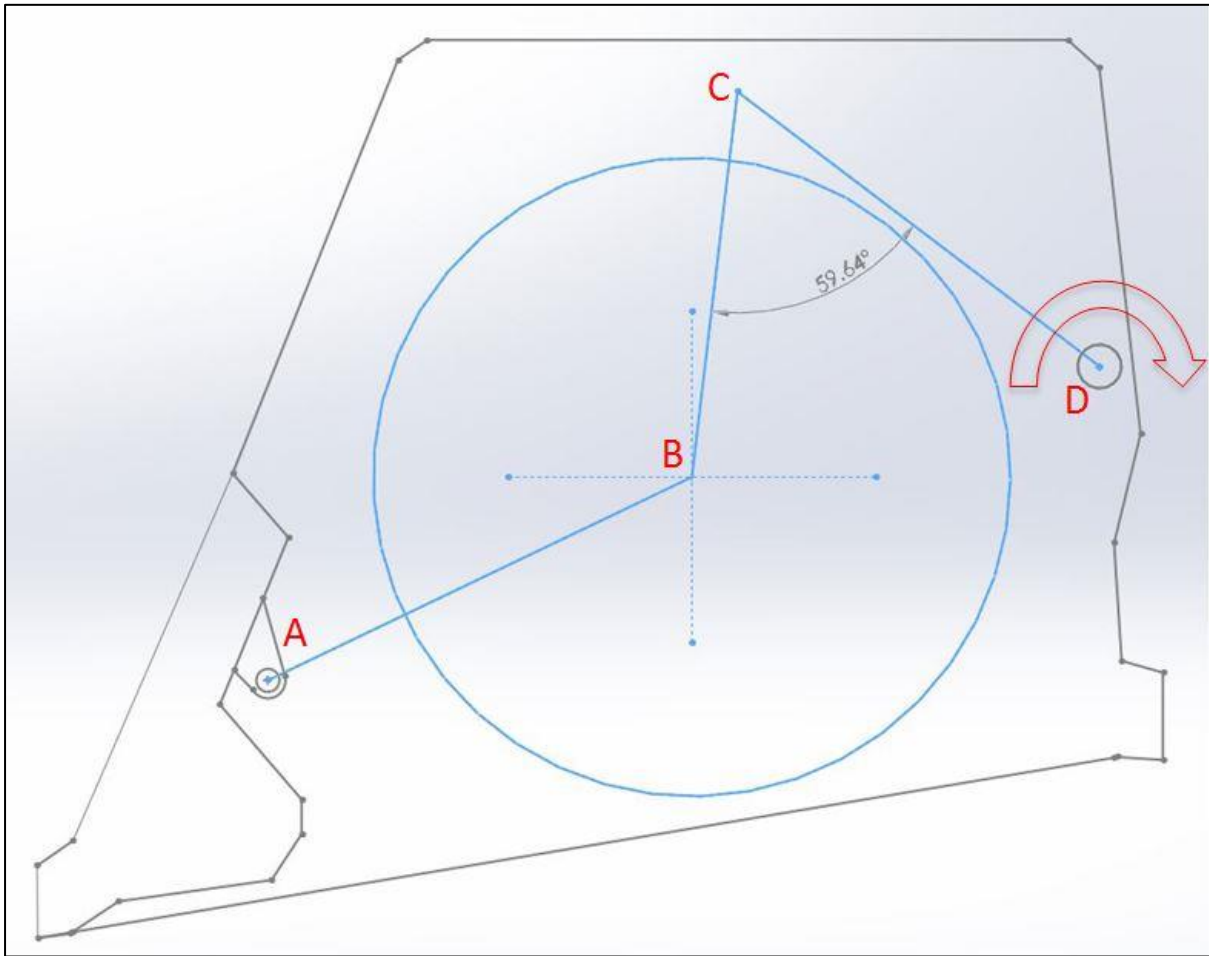


Figure 28 - Retracted kinematic layout

With the selected strut lengths the landing gear also has efficient ground clearance while being fully compressed with a vertical deflection of the shock absorber (d_{VS}) equal to 49.18 [mm] and vertical deflection of the tyre (d_t) equal to 77 [mm]. The vertical ground clearance (δ_{VC}) is equal to 119 [mm], which is above the minimum allowable vertical ground clearance (δ_{MVC}) of 100 [mm] defined in *Section 4.2.5. Initial Parameters*. The sufficient ground clearance is illustrated in Figure 29.

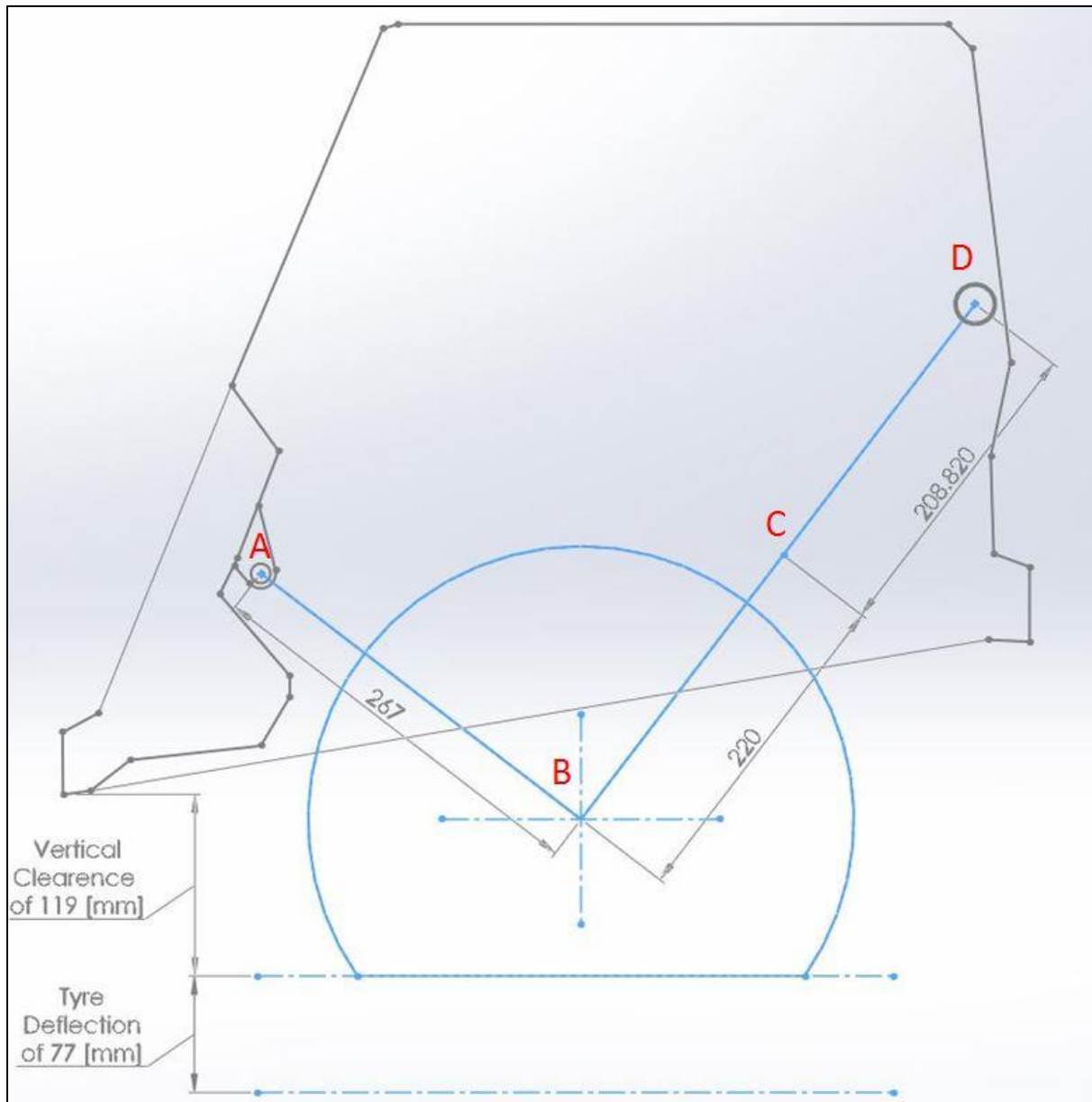


Figure 29 - Kinematic layout of ground clearance

This concludes that the selected strut lengths and the angle of the two rear strut arms in relation to the ground can now be used to develop the polyurethane rubber shock elements.

4.4.3. DEVELOPMENT OF RUBBER DISC ELEMENTS

Due to the nonlinear behaviour of polyurethane rubber it is quite difficult and complex to determine the required size and compression modulus of the rubber. Thus the method that will be used to determine the size of the rubber elements is based on the assumption that the rubber elements operation remains in the linear state. According to Gent (2012), the linear range of rubber under compression is within the 0-30 [%] strain range. Gent also concludes that this method is an adequate approximation for design purposes. The initial design parameters and the allowable strut lengths will now be used to determine the amount and size of the rubber discs used in the shock absorber strut.

STEP1: CALCULATING THE ELEMENT LENGTH

The required vertical deflection and the angle of the shock absorber strut (SA) will first be used to determine the true deflection of the shock absorber that is required to maintain a safe load factor. The values used are shown in Table 16:

Table 16 - True required deflection calculations

Variable	Description	Value	Unit
d_{RVS}	Required vertical deflection	49.18	[mm]
α	Rear strut arm angle according to the ground	51.91	[°]

$$d_{RTS} = \frac{d_{RVS}}{\sin \alpha}$$

$$d_{RTS} = \frac{49.18}{\sin(51.91)}$$

$$d_{RTS} = 62.49 [mm].$$

Due to the nearly incompressible material being used, the polyurethane rubber will not be able to be compressed to 100 [%] of its initial length. According to the uniaxial compression test data of the polyurethane rubber in *Section 3.3.1. Uniaxial Compression Test* the maximum deflection in compression of the polyurethane rubber samples was 53 [%] without losing any structural integrity. By assuming that the possible deflection is equal to 53 [%] the total length of the shock absorber element will have to be at least 117.91 [mm] to ensure that the requirements are met.

This minimum length of the landing gear shock element exceeds the possible length for a single compression disc according to Currey (1988). When designing rubber shock absorbers the rule of thumb is to design the compression discs so that they do not exceed approximately 1.5 inches (38.1 [mm]) (Currey, 1988). Thus, by keeping this rule in mind, each rubber shock element would consist of 3 rubber discs, each with a length of 40 [mm]. Now that the length of the rubber elements is determined the shape and load bearing area has to be taken into consideration.

STEP2: CALCULATING THE LOAD BEARING AREA

An interesting property of nearly incompressible rubber under compression that has to be included into the design procedure is that the rubber can become stiff or soft depending on its shape. This property is defined as the Shape factor (S), which is defined as the loaded area of the rubber piece divided by the total area of the rubber that is free to bulge (Ciesielski, 1999). As we reduce the area of the rubber that is free to bulge, the shape factor becomes greater and the rubber stiffer and this will increase the Young's modulus (E_o). The shape factor's influence on the Young's modulus (E_o) is illustrated in Figure 30.

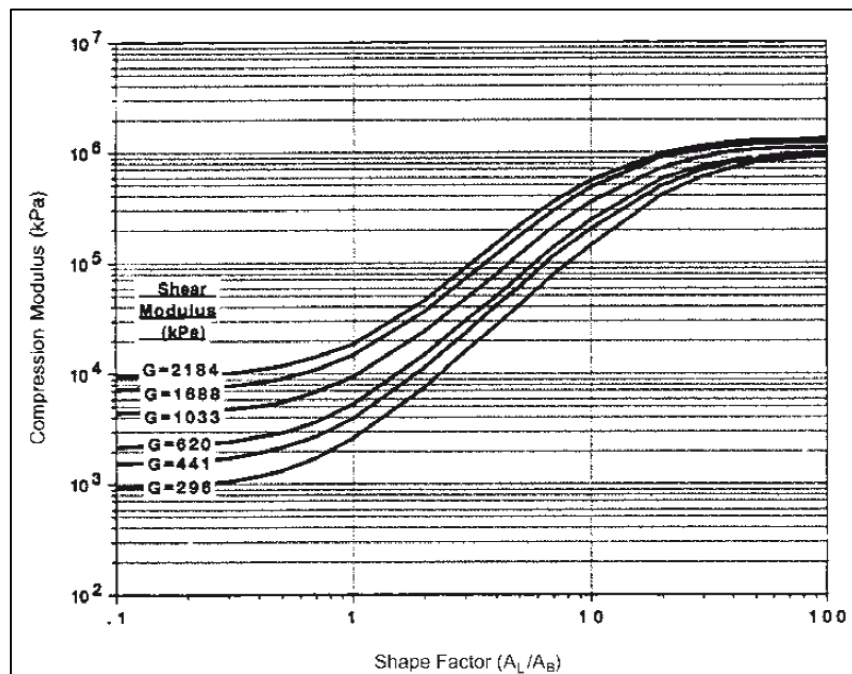


Figure 30- Compression modulus E_c versus shape factor S for various shear moduli (Gent, 2012)

In order to obtain a deflection of 53 [%] while a maximum load is applied to the rubber elements, it must be ensured that the shape of the rubber element does not influence its stiffness. Therefore, according to the comparison in Figure 30, the shape factor has to be selected between the ranges of 0.1 [-] and 0.4 [-], to ensure that the shape factor has the lowest influence on the Compression modulus (E_c). The shape factor (S) according to Gent (2012) is as follow:

$$S = \frac{A_{Load}}{A_{Free}} \quad (4.14)$$

Where;

A_{cross} — is the cross sectional area where the load is being applied to [m^2],

A_{Free} — is the area that is free to bulge [m^2].

Equation 4.14 will now be used to determine the outside diameter of the rubber elements.

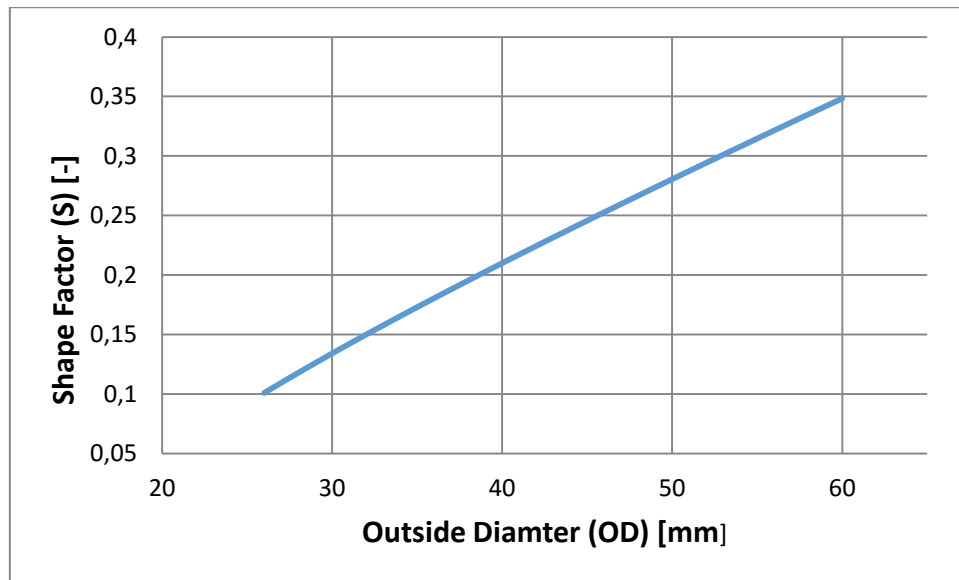


Figure 31 - Shape Factors (S) influence on outside diameter (OD)

The outside diameters that have been obtained from the shape factor ranging between 0.1 [-] and 0.4 [-] can all possibly be used for the design. The final decision on the outside diameter is made based on the space requirements of the landing gear wheel box. Thus, the outside diameter of the rubber element is selected equal to 40 [mm] with a corresponding shape factor equal to 0.21 [-]. The load bearing area of the rubber element is, therefore, calculated and is equal to 1055.5 [mm^2].

STEP3: DETERMINING THE REQUIRED COMPRESSION MODULUS

In order to further develop the polyurethane rubber elements by determining the required Compression modulus (E_c) it is necessary to assume that the deflection of the rubber element is within the linear range. By assuming that the linear state is within the 30 [%] strain range it is possible to determine the hardness of the rubber by means of spring rate. For most designs, the spring rate, also known as stiffness, is a key design parameter. The stiffness of a rubber part in terms of a function is defined by *Equation 4.15* as the amount of force required to cause a unit deflection:

$$K_c = \frac{F}{d} \quad (4.15)$$

Where;

F – is the applied load on one single element [N],

d – is the deflection [m].

The spring rate (K) can be for shear, compression, tension or a combination thereof depending on the direction of the applied load. The rubber elements being developed in this case are only subjected to compression loads. Thus, the spring rate (K) of a part subjected to compression loads can be defined by *Equation 4.16* in terms of geometry and modulus (Gent, 2012):

$$K_c = \frac{AE_c}{T} \quad (4.16)$$

Where;

A – is the effective load area [m²],

E_c – is the Compression modulus of the rubber element [MPa],

T – is the thickness of the un-deformed element [mm].

By using *equation 4.15* and *4.16* the required compression modulus can be calculated with the assumption that the rubber elements are within the linear strain range. The final unknown variable of the rubber elements of the cylindrical discs can be determined with the following initial values and constraints:

Table 17 - Single rubber element constraints

Variable	Description	Value	Unit
T	Thickness of the un-deformed element.	40	[mm]
$F_{Element}$	Applied compression load on one single rubber element.	10300.50	[N]
$ID_{Element}$	Internal diameter of cylindrical disc, constrained due to inner alignment tube of shock absorber.	16	[mm]
$d_{Element}$	Required deflection of single rubber element.	20.83	[mm]

In order to calculate the required stiffness (K_C) of a single rubber element from the constraints in Table 17 it is important to keep the incompressible behaviour of the rubber in mind, where the rubber's stiffness will increase as the applied load increases. This means that the stiffness of the rubber element will then be much stiffer for strains larger than 30 [%] for the same rubber composition, making it necessary to assume that the load used within the linear range will be less than half of the total load applied to the rubber elements. The required linear spring rate or stiffness of a single rubber element for a deflection of 30 [%] is calculated by using *Equation 4.14*, and by assuming that the load is approximately a quarter of the total load shown in Table 17 to accommodate the linear strain section:

$$K_C = \frac{F}{d}$$

$$K_C = \frac{\frac{10300.5 [N]}{2.5}}{20.83 [mm]}$$

$$K_C = 197.80 \left[\frac{N}{mm} \right].$$

The required stiffness will now be used to determine the required Compression modulus (E_c) for selected geometry by using *Equation 4.16* as follow:

$$K_c = \frac{AE_c}{L}$$

$$197.80 \left[\frac{N}{mm} \right] = \frac{1055.58 [mm^2] \times E_c}{40 [mm]}$$

$$E_c = 7.50 [MPa].$$

This calculated Compression modulus (E_c) will now be used to determine what hardness of rubber is to be used for the shock absorber. The Compression modulus (E_c) of the five different rubbers are determined by using the linear stress-strain data of each rubber in *Section 3.3.1. Uniaxial Compression Test*. The outside diameter (*OD*) is calculated for each of the different rubbers, each with a different hardness. The results are shown in Table 18:

Table 18 - Initial compression modulus of different rubber hardness

Shore A Hardness	Compression modulus (E_c) [MPa]
40	4.97
50	7.77
60	11.28
65	12.57
80	17.89

According to Table 18 it was decided to select the polyurethane rubber with 50 [Shore A] hardness for the rubber material that will be used for the JS-2 shock absorber elements.

SUMMARY

The two rubber shock absorbers that are situated in the shock strut arm of the landing gear structure are each equipped with three rubber elements with the following dimensions and properties:

Table 19 - Initial properties of rubber shock absorber elements

Variable	Description	Value	Unit
L	Thickness of the rubber element.	40	[mm]
OD	Outer diameter of rubber element.	40	[mm]
ID	Internal diameter rubber element.	16	[mm]
$HARDNESS$	Required deflection of single rubber element.	50	[SHORE A]
E_C	Compression modulus	7.50	[MPa]

These properties and dimension will now be used to calculate numerically the nonlinear load-deflection data that can be used to determine if the shock absorber is able to absorb the total amount of energy during landing.

4.4.4. FEA OF THE JS-2 RUBBER SHOCK ABSORBER

Similarly to the JS-1 rubber shock absorber the JS-2 rubber shock absorber is primarily developed as a mechanical shock that uses polyurethane rubber as the shock absorbing medium to absorb shock during landing. It was shown in the previous section that the JS-2 shock absorber will absorb the required energy when the design consists of three 50 [Shore A] polyurethane rubber cylindrical discs, each with a height of 40 [mm] and inner and outer diameter of 16 [mm] and of 40 [mm]. The three polyurethane rubber discs are also separated by stainless steel washers that are slightly bigger than the discs. The three discs and washers are kept in line by a low alloy steel centre tube and are compressed by two medium carbon steel end caps, one on the bottom and one at the top that is similar to the JS-1 design. The JS-2 rubber shock absorber can now be modelled in a similar manner as the JS-1 was modelled in *Section 3.4.3. Modelling of JS-1 Rubber Shock Absorber* in order to ensure than an accurate results will be obtained.

SOLID MESH AND MODEL CONSTRAINTS

Due to the great difference in stiffness between the metal and the polyurethane rubber parts and because the polyurethane rubbers behaviour is the only material under investigation, it is decided to model the polyurethane rubber and stainless steel divider washers as deformable bodies and the rest of the parts as rigid surfaces.

The polyurethane rubber and stainless steel divider washers deformable bodies were modelled using 4-node quadrilateral hybrid solid elements with a global element edge length of 4 [mm], as shown in Figure 32.

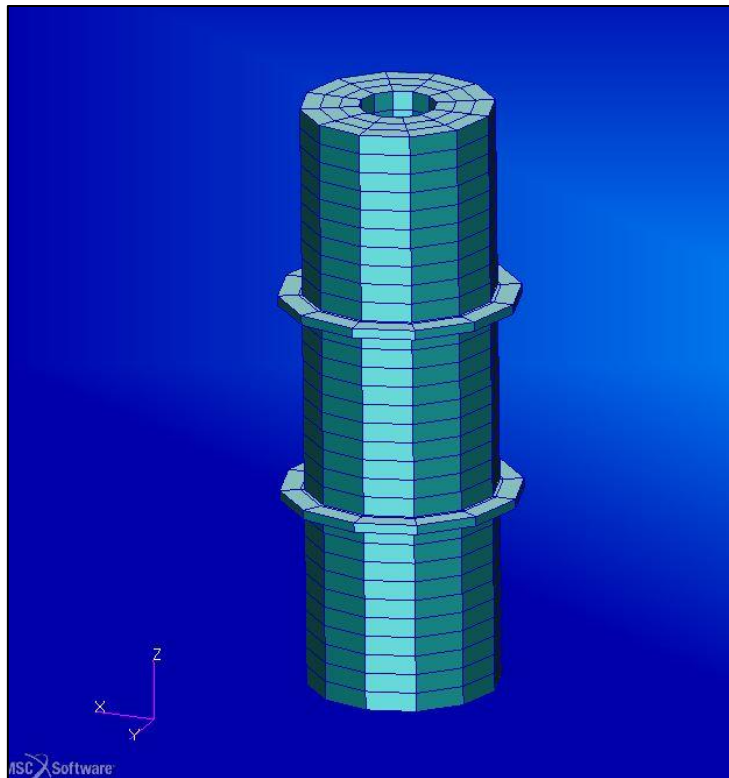


Figure 32 - JS-2 Rubber elements FE mesh

These parts were then confined between two rigid surfaces and centred by a cylindrical rigid surface situated in the middle of the discs that respectfully represent the top and bottom end caps and centre tube of the JS-2 shock absorber. In order to represent the actual behaviour of the J2 shock absorber being compressed, the bottom and cylindrical centre tube rigid surfaces were constrained in all directions. The upper rigid surface applying the load was constrained in all directions except for the vertical direction, allowing the load to be applied to the polyurethane rubber discs.

MATERIAL PROPERTIES ASSIGNMENT

The properties of the stainless steel were assigned to the numerical model of the washer by using PATRAN isotropic linear elastic material model where the Young's Modulus (E) is equal to $E = 200 \text{ [GPa]}$ and the Poisson ratio (ν) is equal to $\nu = 0.29 \text{ [-]}$.

The material properties of the polyurethane rubber were assigned to the rubber discs by using PATRAN nonlinear nearly incompressible hyperelastic material model known as MATHE. The hyperelastic material model MATHE with only one polynomial order had to be adapted in order to accommodate the Yeoh material model. This was done by inserting a zero value for coefficients C_{01} and C_{11} . In order to simulate accurately the nearly incompressible behaviour of the material, the Poisson's ratio was selected equal to $\nu = 0.499 \text{ [-]}$ and the mass density of the material was selected according to the manufacturer's data that is equal to $\rho = 1.017 \times 10^{-6} \text{ [Kg/mm}^3\text{]}$. The coefficients of the rubber with hardness value of 50 [Shore A] were used for the Yeoh material model, and are shown in Table 20:

Table 20 - Yeoh material coefficients for JS-2 rubber shock absorber

Hardness [Shore A]	Yeoh Material Coefficients		
	C_{10} [MPa]	C_{20} [MPa]	C_{30} [MPa]
50	0.958	-0.027	0.043

LOADS APPLIED TO THE MODEL

As in the JS-1 shock absorber simulation it was necessary to enable friction between the rubber and its contacting surfaces in order to simulate practical conditions. It was decided that a friction coefficient of $\mu=0.5 \text{ [-]}$ was used where the rubber disc elements came in contact with any rigid surface.

The compression load was applied to the model by assigning the loads to the top rigid surface. The load was then systematically applied in the vertical direction that compressed the rubber discs. The load being applied was equal to 12000 [N], that is higher than the required load that a single shock absorber have withstand, stipulated in *Section 4.2.5. Initial Parameters*. The simulation was then run as an implicit nonlinear simulation (SOL 600) that was able to solve the nonlinear behaviour of the hyperelastic material. Load and deflection results could then be obtained from the SOL 600 simulation that will be discussed in the following section. The maximum deflection of the JS-2 rubber elements are shown in Figure 33.

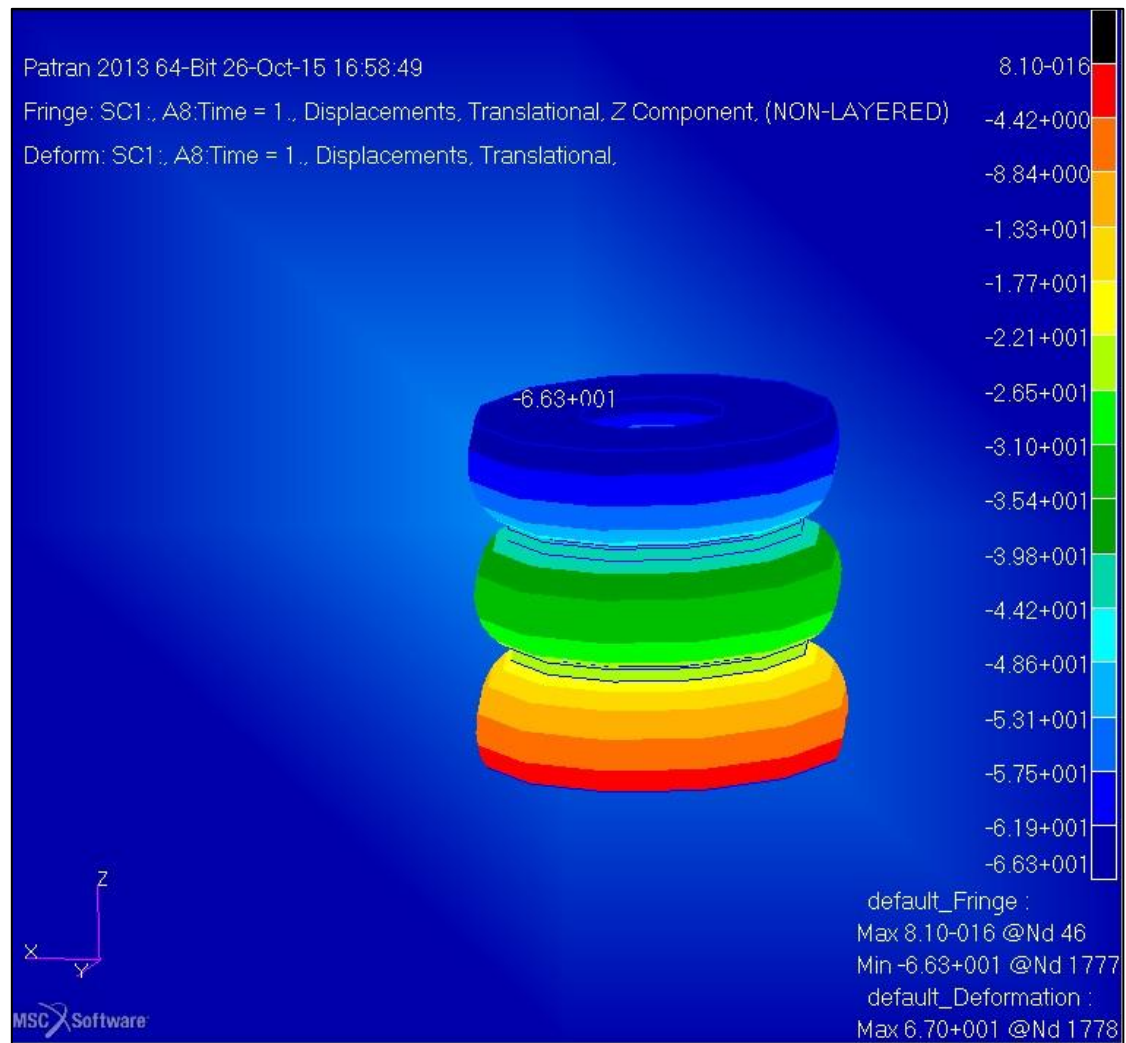


Figure 33 - JS-2 Rubber elements FEA result

LOAD-DEFLECTION RESULTS

The load-deflection data of the JS-2 shock absorber elements with rubber of 50 [Shore A] hardness were plotted in order to illustrate the nonlinear behaviour of the polyurethane rubber. The data are shown in Figure 34:

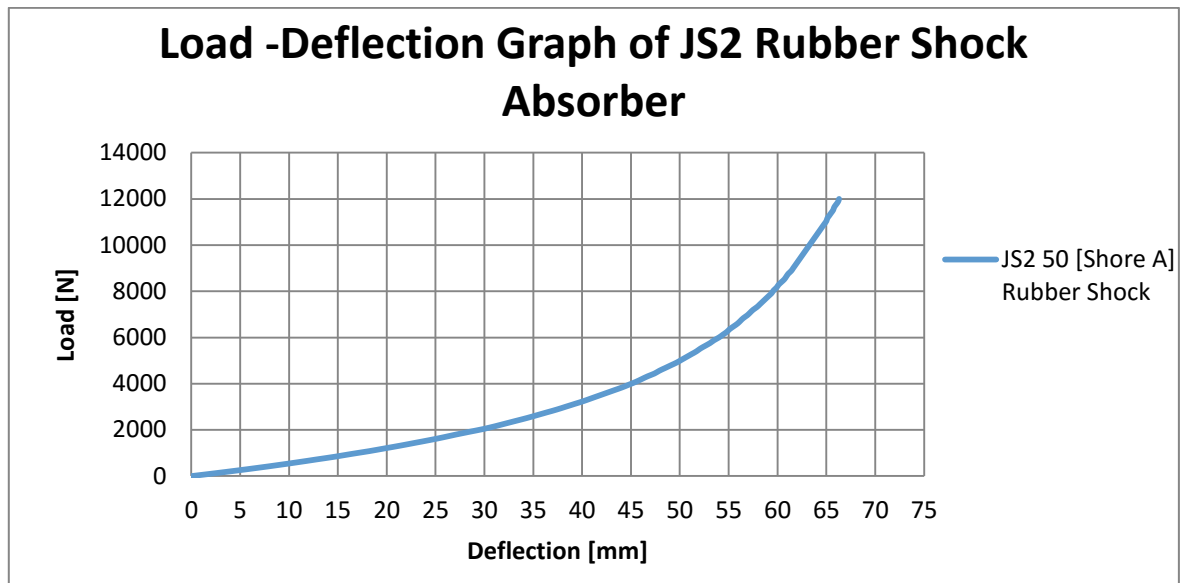


Figure 34 - Load-Deflection data of JS-2 rubber shock absorber

As stated in *Section 4.4.3. Development of Rubber Disc Elements* by Gent (2012), the load-deflection data of the JS-2 rubber shock absorber clearly show nonlinear projection for a deflection larger than 30 [%] and a linear region within the first 30 [%] deflection range of its original length, just as Gent predicted. Thus concluding that the method used to determine the initial geometry of the rubber elements is effective enough for the initial development. The maximum deflection of the rubber elements is 52.25 [%] of the rubber elements' initial length without losing any structural integrity. The deflection of the rubber elements at the required landing load was equal to 63.8 [mm]. That is more than the required deflection of 62.6 [mm].

The data obtained from the tyre and rubber shock absorber will now be used to determine if the landing gear load factor is within the CS-22 requirements.

4.5. JS-2 LANDING GEAR ENERGY ABSORPTION

Now that all of the deflection data of the tyre and rubber shock absorber are known the ultimate and limit landing gear load factor can be calculated for the amount of energy that has to be absorbed. As discussed in Section 4.2.3. Shock Absorption the JS-2 landing gear shock absorbing system that includes the tyre and the rubber shock absorber has to prove that it is able to absorb the efficient amount of energy for two different cases. The one is according to paragraph CS 22.725(a) where the shock absorbing elements, including the tyre, must be capable of absorbing the maximum amount of energy developed in an expected landing without being fully depressed keeping structural integrity.

The other is where the JS-2 landing gear shock absorbing system has to prove that it can absorb 1.44 times the energy described in CS 22.725(a); this is known as a shock absorption test. During the shock absorption test the landing gear structure is allowed to yield as long as it is able to absorb the energy without bottoming out. By using *Equation 4.4 and 4.5* the load factors for both these cases can be determined.

4.5.1. LANDING GEAR LOAD FACTOR

The limit landing gear load factor ($n_{LG\ LIM}$) will be calculated with *Equation 4.4* and the initial design parameters described in *Section 4.2.5. Initial Parameters* Due to the influence of the load factor on the load that is applied to the shock absorbing system an initial load factor has to be selected and altered until the maximum amount of energy (E_{MAX}) is absorbed. The equation, the input values as well as the calculated result are shown in Table 21:

Table 21 - Landing gear load factor results for JS-2 shock absorbing system

Equation 4.4			
$\frac{1}{2}m_m V_Z^2 = g n_{LG\ LIM} m_m (\eta_s d_s + \eta_t d_t)$			
Input Variables	Description	Value	Unit
m_m	The design maximum weight, JS requirement.	600	[Kg]
V_Z	The vertical descent velocity, CS 22 requirement.	1.77	[m/s]
g	The standard gravitational acceleration.	9.81	[m/s ²]
η_t	Main landing gear wheel absorption efficiency, (Currey, 1988).	0.47	[-]
η_s	Initial assumed rubber shock absorption efficiency, (Currey, 1988).	0.6	[-]
Results	Description	Value	Unit
$n_{LG\ LIM}$	The limit landing gear load factor, complying to CS 22.725(a).	2.794	[-]
d_s	Vertical deflection of landing gear rubber shock absorbers.	47.14	[mm]
d_t	Vertical deflection of landing gear tyre.	61.41	[mm]

4.5.2. SHOCK ABSORPTION TEST LOAD FACTOR

The ultimate landing gear load factor ($n_{LG\ ULT}$) will now be calculated in a similar manner as the limit landing gear load factor ($n_{LG\ LIM}$). However, the total amount of energy that has to be absorbed is 1.44 times more. The calculation will be done by using *Equation 4.5* and the initial design parameters described in *Section 4.2.5. Initial Parameters*. The equation, the input values as well as the calculated result are shown in Table 22:

Table 22 - The landing gear shock absorption test results for the JS-2 shock absorption system

Equation 4.5			
$1.44 \times \left(\frac{1}{2} m_m V_Z^2 \right) = g n_{LG\ ULT} m_m (\eta_s d_s + \eta_t d_t)$			
Input Variables	Description	Value	Unit
m_m	The design maximum weight, JS requirement.	600	[Kg]
V_Z	The vertical descent velocity, CS 22 requirement.	1.77	[m/s]
g	The standard gravitational acceleration.	9.81	[m/s ²]
η_t	Main landing gear wheel absorption efficiency, Equation 3.12.	0.47	[-]
η_s	Initial assumed rubber shock absorption efficiency, (Currey, 1988).	0.6	[-]
Results	Description	Value	Unit
$n_{LG\ ULT}$	The ultimate landing gear load factor according to Shock absorption test, complying with CS 22.725(a).	3.484	[-]
d_s	Vertical deflection of landing gear rubber shock absorbers.	50.06	[mm]
d_t	Vertical deflection of landing gear tyre.	76.55	[mm]

4.5.3. CONCLUSION

The results in Table 21 and Table 22 clearly show that the JS-2 landing gear shock absorption system adheres to the CS-22 shock absorption requirements discussed in *Section 4.2. Design Parameters*. The results shown in Table 21 prove that the JS-2 shock absorption system is able to absorb the maximum amount of energy during landing impact without bottoming out while obtaining a limit landing gear load factor ($n_{LG\ LIM}$) of 2.794 [-] that is below the CS-22 allowable load factor of 4.5 [-].

The results in Table 22 also prove that the JS-2 shock absorption system also provides sufficient evidence that it is able to absorb the ultimate amount of energy during a shock absorption test without bottoming out while obtaining ultimate landing gear load factor ($n_{LG\ ULT}$) of 3.484 [-] that is less than the CS-22 allowable load factor of 4.5 [-].

This concludes that the JS-2 shock absorbing system adheres to the CS-22 shock absorption requirements as discussed in *Section 4.2. Design Parameters*.

4.6. SUMMARY

In this chapter it was shown analytically and numerically that the developed shock absorption system of the JS-2 landing gear system adheres to the CS-22 certification specifications. This included the development of the shock absorber rubber elements and the development and selection of the wheel and the tyre.

The shock absorbing system was developed according to the total amount of energy that had to be absorbed by the system that was defined by the CS-22 requirements and the predetermined initial parameters. The landing gear main wheel and tyre were selected and developed by using analytical calculations. The shock absorption abilities of the tyre were then used to determine the amount of energy that it is capable of absorbing at a desired pressure. The remaining energy that was not absorbed by the tyre was then used to develop the rubber shock elements.

The rubber element geometry and material stiffness were then calculated by assuming that the deflections of the rubber elements are within the linear state. The nonlinear behaviour of the rubber elements was then predicted by modelling the elements in FEA code. The FEA results of the rubber elements were then used with the deflection data of the tyre to prove that the JS-2 shock absorbing system is able to absorb the total amount of energy without bottoming out. The landing gear load factor calculated was all within the range of the CS-22 requirements. The landing gear load factor can now be used to develop the JS-2 landing gear structure.

CHAPTER 5 : DEVELOPMENT OF THE LANDING GEAR STRUCTURE

5.1. INTRODUCTION

The focus of this chapter is to develop the JS-2 landing gear structure in order to accommodate the selected components from the previous chapters and to adhere to Jonker Sailplanes' and EASA CS 22 requirements by using the FEA code. This includes the structural analysis of the landing gear structure strut arms and the wheel axle. There are several different methods that can be followed to develop the landing gear structure. The method used in this study is based on the design requirements of Jonker Sailplanes and the CS-22 specification.

This chapter will also include the verification of the numerical FEA stress results. The results will be verified by analytical stress calculations in order to ensure that the FEA provides the expected results. This will ensure that the results that are obtained from the developed FEA model are accurate and reliable to prove that the landing gear structure adheres to the CS-22 structural safety requirements.

5.2. STRUCTURAL REQUIREMENTS

There are several structural requirements that have to be taken into consideration when developing the landing gear structure. These requirements are based on the CS-22 certification specifications. These requirements include the loads being applied to the landing gear structure during landing and the safety factors that the design has to adhere to.

5.2.1. GROUND LOAD REQUIREMENTS

According to paragraph CS 22.473(a) to (b) the landing ground loads must be calculated with the design maximum weight(m_m) as defined in *Section 4.2.5*, and a selected limit landing gear load factor ($n_{LG\ LIM}$). The load factor may not be less than 3 [-] or less than what would have been obtained when landing with a descent velocity of 1.77 [m/s]. The limit landing gear load factor calculated with a descent velocity of 1.77 [m/s] in the previous chapter was found to be equal to 2.794 [-] when the shock absorber's effect is taken into account. This is lower than the required minimum limit load factor of 3 [-]. Therefore, for the development of the landing gear

structure a limit landing gear load factor of $n_{LG\ LIM} = 3$ [-] and a maximum design weight of $m_m = 600$ [Kg] will be used to calculate the ground loads that will be acting onto the landing gear structure. The ground loads will be divided into two loading conditions that include the level landing condition and the side landing condition. The loads for these two landing conditions can be calculated with the initial values shown in Table 23:

Table 23 - Structural load requirements

Parameter	Description	Value	Unit
$n_{STRUC\ LG}$	The selected structural landing gear load factor, selected according to the CS-22 requirements.	3.0	[-]
m_m	The design maximum weight, JS requirement.	600	[Kg]
g	The standard gravitational acceleration.	9.81	[m/s ²]

During the level and side landing conditions it is assumed that the tail boom of the fuselage is parallel to the ground as shown in the following figure:

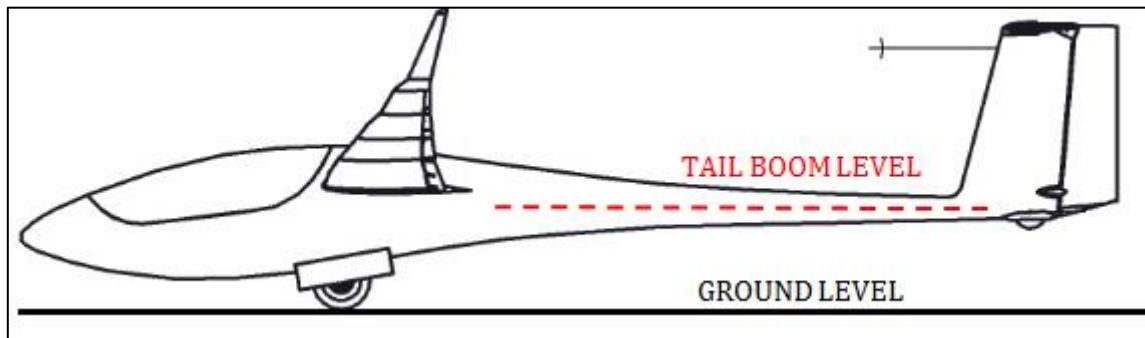


Figure 35- Level landing angle

5.2.2. LEVEL LANDING CONDITION

According to paragraph CS 22.479(c) and (d), during level landing conditions a resultant load (P_R) acts onto the main landing gear tyre at an angle of 30° to the vertical axis that can be divided into a vertical and horizontal component, as shown in Figure 36.

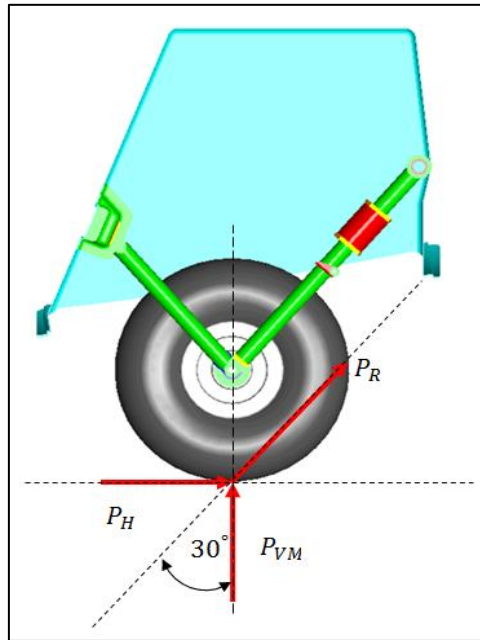


Figure 36 - Level landing loads

The vertical (P_{VM}) and horizontal (P_H) load components, according to paragraph CS 22479(b), have to be determined according to the conditions stated in *Section 5.2.1*. The level landing loads can be calculated as follow:

$$P_{VM} = n_{STRUC LG} g m_m \quad (5.1)$$

$$P_R = \frac{P_{VM}}{\cos(30^\circ)} \quad (5.2)$$

$$P_H = P_{VM} \times \tan(30^\circ) \quad (5.3)$$

The loads calculated with *Equation 5.1, 5.2 and 5.3*, and the initial values shown in Table 23 are illustrated in Table 24:

Table 24 - Calculated level landing loads

Parameter	Description	Value	Unit
P_{VM}	The vertical load component.	17658.00	[N]
P_R	The rearward acting resultant load component.	20389.70	[N]
P_H	The rearward acting horizontal load component.	10194.85	[N]

5.2.3. SIDE LANDING LOADS

According to paragraph CS 22.485, during a level landing it must be assumed that there is a side load (P_{HS}) acting on one side of the main landing gear (that is possible from both the right and left sides). It is assumed that the load is applied vertically to the centre of the tyre's contact area with the ground. This applied side load is also accompanied by a vertical component (P_{VS}), illustrated in Figure 37.

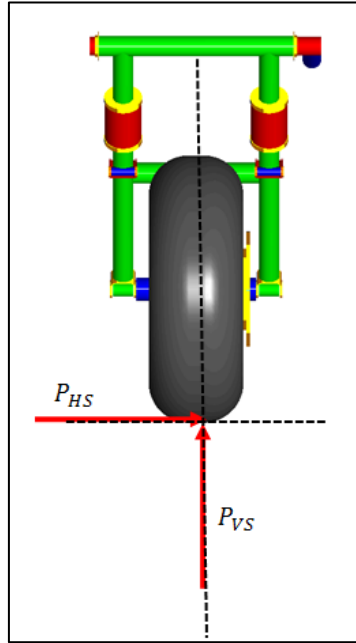


Figure 37 - Side landing loads

The vertical (P_{VS}) and horizontal (P_{HS}) side load components, according to paragraph CS 22.485, have to be determined according to the conditions stated in *Section 5.2.1*. These side landing loads can be calculated according to CS 22.485 as follow:

$$P_{HS} = 0.3P_{VM} \quad (5.4)$$

$$P_{VS} = 0.5P_{VM} \quad (5.5)$$

Where P_{VM} is the vertical load calculated in *Section 5.2.2*. Level Landing The loads calculated with *Equation 5.4* and *5.5* and the initial values shown in Figure 37 are shown in Table 25:

Table 25 - Calculated side landing loads

Parameter	Description	Value	Unit
P_{HS}	The horizontal side load component.	5297.40	[N]
P_{VS}	The vertical side load component.	8829.00	[N]

5.2.4. SAFETY FACTORS

In order to adhere to the structural requirements of the CS-22, the relevant stresses in each component of the landing gear structure have to be calculated to prove that the safety factor of each component is higher or equal to the required CS-22 safety factors. According to paragraphs CS 22.303, CS 22.623 and CS 22.625, a safety factor of 1.5 [-] has to be used throughout the design except at bolted or pinned joints and fittings where two structural parts have to be joined together. The special safety factors in each of these cases are shown in Table 26:

Table 26 - Safety Factors

Safety Factor	Description	Value	Unit
SF_N	Normal safety factor overall used, according CS 22.303.	1.5	[-]
SF_B	Bearing safety factor used for bolted and joined connections, according to CS 22.623.	3.0	[-]
SF_F	Fitting safety factor used for fittings, according to CS 22.625.	1.725	[-]

These structural requirements and landing loads were used to develop the JS-2 landing gear structure that will be described and discussed in the following section.

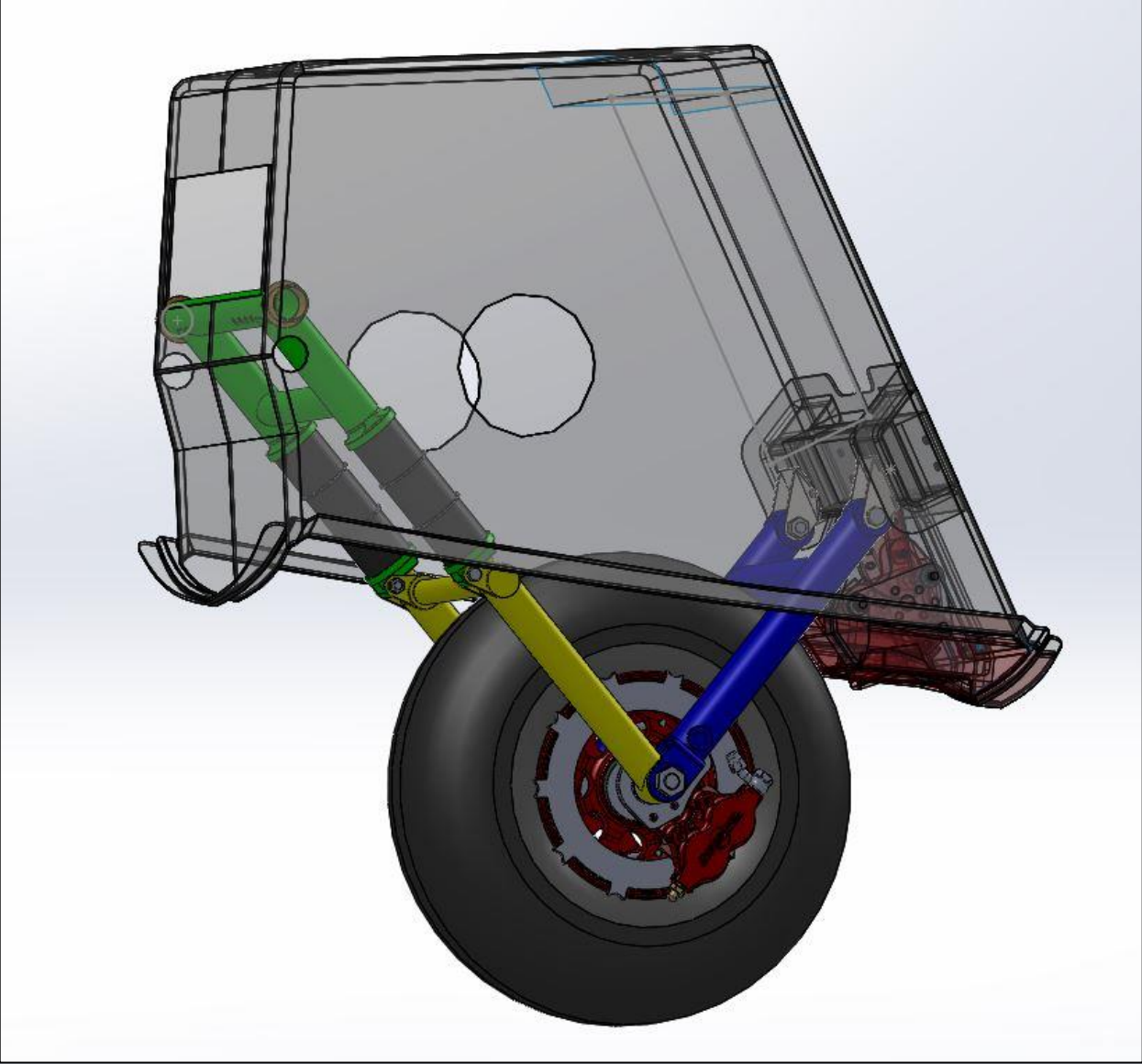
5.3. DEVELOPED JS-2 LANDING GEAR SYSTEM

The JS-2 landing gear structure was developed in order to accommodate the geometrical constraints of the JS-2 fuselage and to adhere to the CS-22 structural safety regulations. Similar to the JS-1 landing gear structure, the JS-2 structure can be divided into three strut arms known as the front arm (FA), rear arm (RA) and shock arm (SA). The three strut arm structures were developed to accommodate the landing gear components of the JS-2 glider, including the braking mechanism, wheel, tyre and rubber shock absorber that were selected and developed in the previous chapters. Although the landing gear system of the JS-2 is quite similar to the JS-1 when compared, there are several differences that can be seen. The most noticeable is that the JS-2 landing gear system is 34 [mm] narrower than the JS-1 landing gear in order to be used in the narrow JS-2 fuselage.

The JS-2 landing gear structure consists of several different materials. The main structural materials to be used in the landing gear structure are low alloy and medium carbon steel. The

pivot points are connected by stainless steel bolts and brass bushes to reduce the wearing between their contact surfaces. The main axle consists of aluminium alloy steel that is connected to the structure with a stainless steel inner axle. The basic JS-2 landing gear layout, with its components, is illustrated and described in Table 27.

Table 27 - JS-2 landing gear layout

		
Component	Description	Colour
SA	Shock absorber strut arm	Green
RA	Rear strut arm	Yellow
FA	Front strut arm	Blue
BR	Beringer wheel hub and brake mechanism	Red
SC	Rubber shock absorber elements and Michelin 5.00-5" tyre	Black

The JS-2 structure was developed in order to be able to absorb the loads during landing as discussed in *Section 5.2.2. Level Landing* and *Section 5.2.3. Side Landing loads*. In order to achieve this, it was decided that the front strut arm (*FA*) had to be developed to absorb most of the side loads, and the rear (*RA*) and shock (*SA*) strut arms developed to absorb the rearward and upward loads. This decision was based on the side load restriction of the shock strut arm (*SA*) that houses the two rubber shock absorbers. The rubber shock absorbers were developed to absorb axial compression loads, thus restricting the component's ability to absorb side loads. If the side loads are too large it may cause the shock absorber to fail. The angle of the rear (*RA*) and shock (*SA*) strut arms was selected to be as close as possible to the angle of the applied load in order to be able to absorb most of the rearward and upwards loads during level loading conditions, as illustrated in Figure 38.

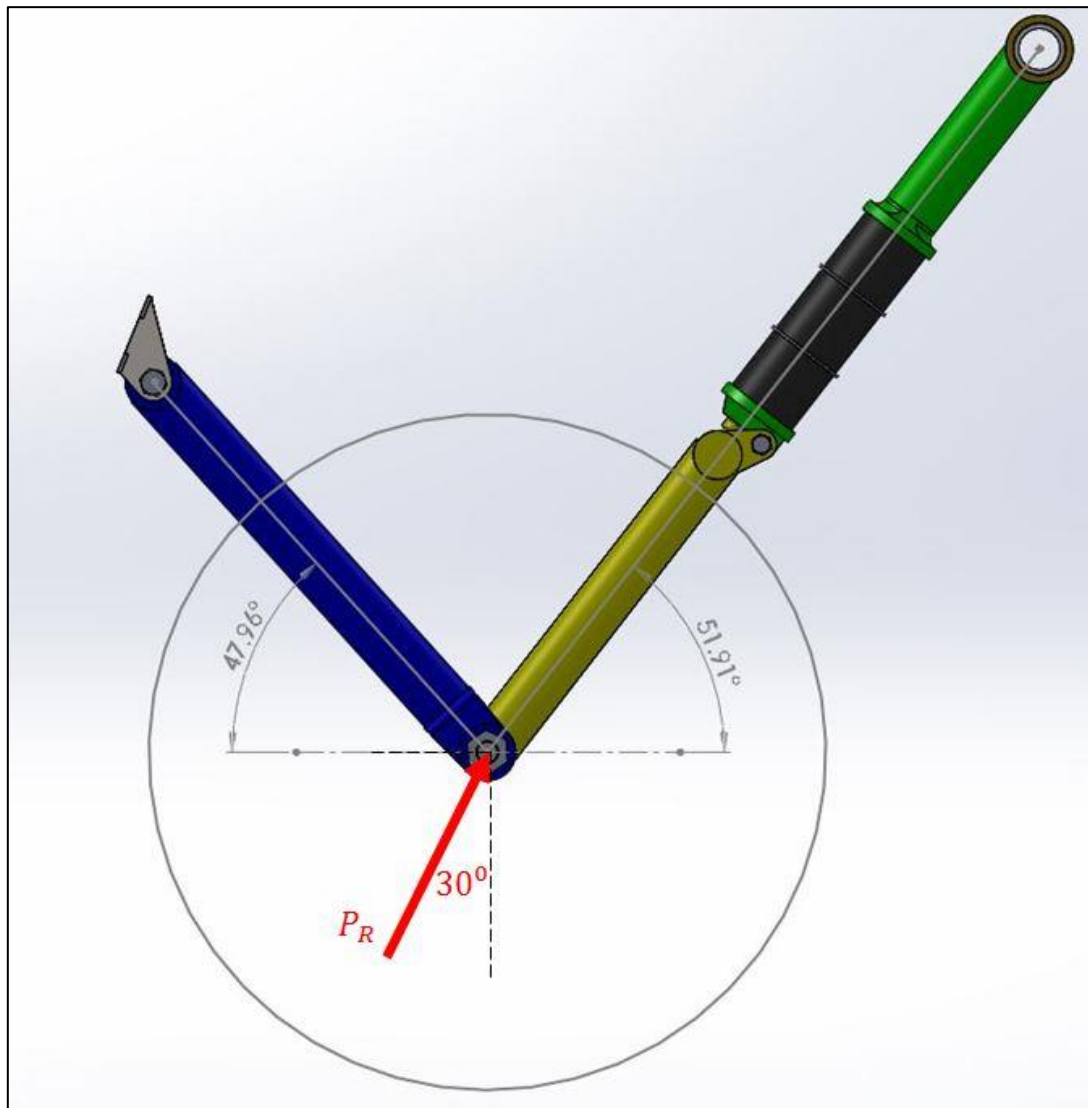


Figure 38 - JS-2 Structure load angle

In order to ensure that the front strut arm (FA) absorbs most of the side loads, it was necessary to insert reinforcements into the sidewall of the landing gear wheel box. The sidewall reinforcements support the front arm structure during side load landing conditions that ensure that no direct side loads are transferred to the shock strut arm (SA). The reinforcements pressed onto the left and right outside surfaces of the front arm structure are approximately 155 [mm] from the main wheel axle. The contact areas of the reinforcements are illustrated in red in Figure 39.

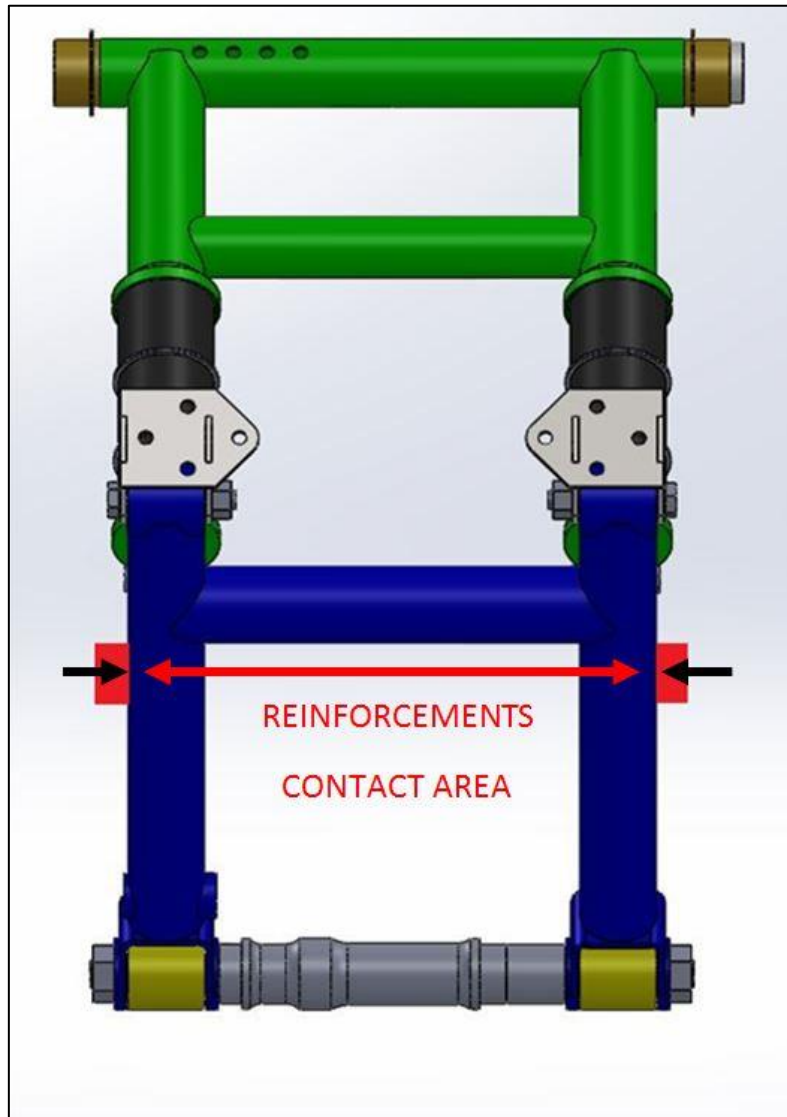


Figure 39 - Front arm side load supports

Now that the developed JS-2 landing gear system has been explained it is necessary to discuss the FEA landing gear model that was used to develop the JS-2 landing gear structure.

5.4. DEVELOPMENT OF THE JS-2 LANDING GEAR STRUCTURE FEA MODEL

For the development of the FEA JS-2 landing gear model, it was decided to use the FEA code NX NASTRAN. The decision to use the FEA NX NASTRAN code was mainly based on the FEA code's surface to surface contact function simplicity and capabilities that made it possible to model and analyse the landing structure as one model as accurately as possible.

The FEA model of the JS-2 landing gear structure mainly focused on the stress analysis of the structural components of the landing gear. Due to this reason, it was decided to model the rubber elements in the shock strut arm (SA) as a rigid body and not as a nonlinear hyperelastic material. This made it possible to accurately transfer the loads to the rest of the structure.

MODEL ELEMENT MESH

All of the parts in the landing gear structure were modelled as solid bodies by using 10-node tetrahedral 3D solid elements with an overall global element edge length of 4 [mm], as shown in Figure 40. The mesh density of the areas, where it was assumed that the surfaces will come in contact with each other, was refined to a global edge length of approximately 1 to 2 [mm].

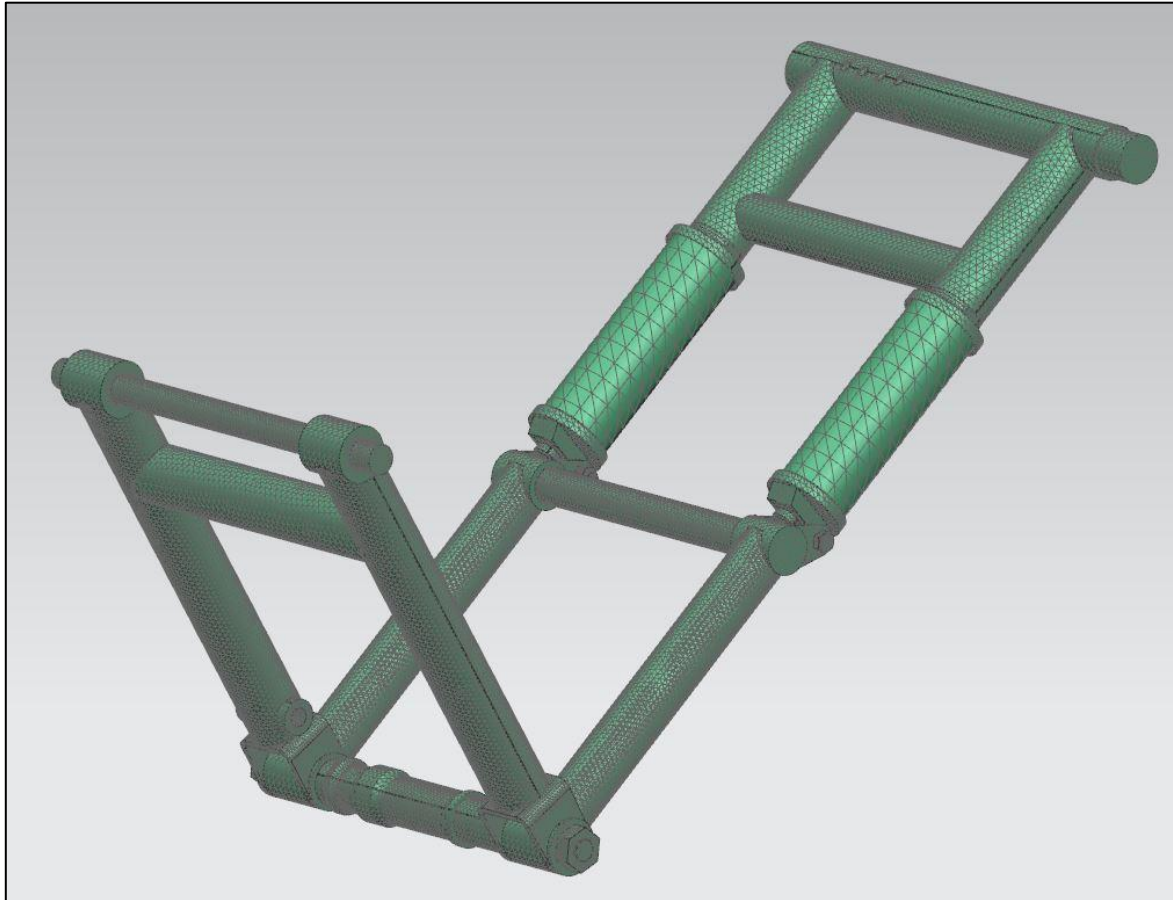


Figure 40 - JS-2 landing gear structure FEA model mesh

MODEL CONSTRAINTS

In order to represent the actual behaviour of the landing gear structure when being loaded it was decided to define the parts that were welded to each other with a surface to surface glue function. The latter represents a weld-like connection. Two rigid cylinders were then used to constrain the landing gear structure; these two rigid parts represented the actual anchor points that connected the landing gear with the fuselage. The rigid parts were constrained in all directions that allowed the load to be applied to the structure.

The second constraint that was applied to the FEA model was a nodal constraint that represented the side load reinforcements of the fuselage wheel box. This was implemented by constraining a few nodal points on the outside surface of the front arm strut tubes to ensure that no movement in the direction of the load occurred. All of these constraints are illustrated in blue in Figure 41.

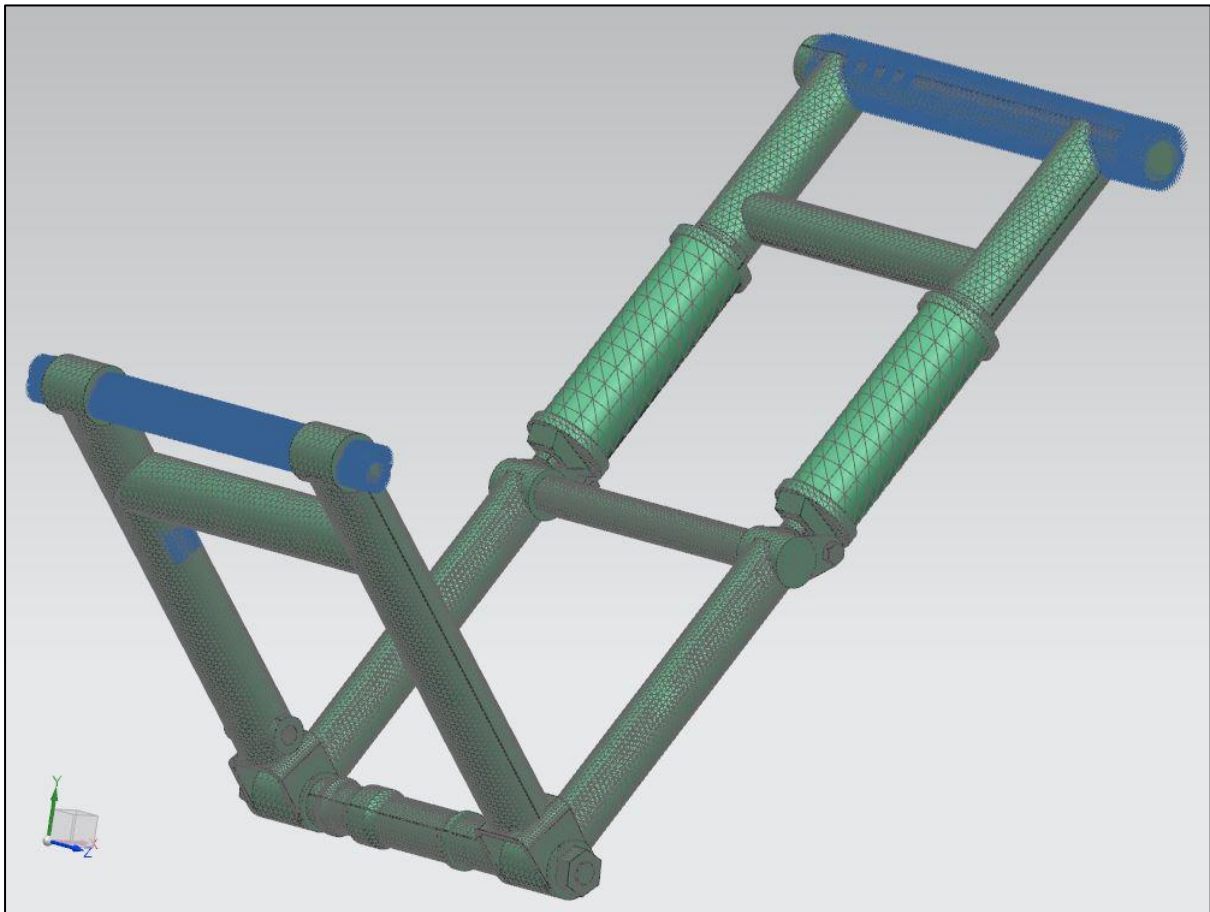


Figure 41 - JS-2 FE model constrains

MATERIAL PROPERTIES ASSIGNMENT

The properties of the different materials were assigned to the numerical model of the landing gear structure by using NX NASTRAN isotropic linear elastic material model. The mechanical properties of each material used in the FEA model are found in Table 28.

Table 28 - Material properties of JS-2 landing gear structure (Jonker Sailplanes C.C., 2013)

Material Code	Material Description	Young's Modulus [GPa]	Poisson ratio [-]
316SS	Stainless steel	193	0.275
2024T3	Aluminium alloy steel	73.1	0.32
4130	Low alloy steel	205	0.29
EN8	Medium carbon steel	200	0.3
CZ121	Brass	97	0.32

APPLIED LOADS

The loads from the two different landing conditions that act onto the landing gear structure are applied to the numerical model of the landing gear structure. This is done with the assumption that the main wheel tyre and hub are rigid bodies in order to transfer the load directly onto the main axle without any force being lost due to the tyre's deflection. By assuming that the main wheel and tyre are rigid it is possible to assign the loads to the main axle surface where the wheel and axle are connected. The surfaces where the loads will be assigned to the main axle are illustrated in Figure 42 and Figure 43 for both load cases. The values of the applied loads are found in Table 29.

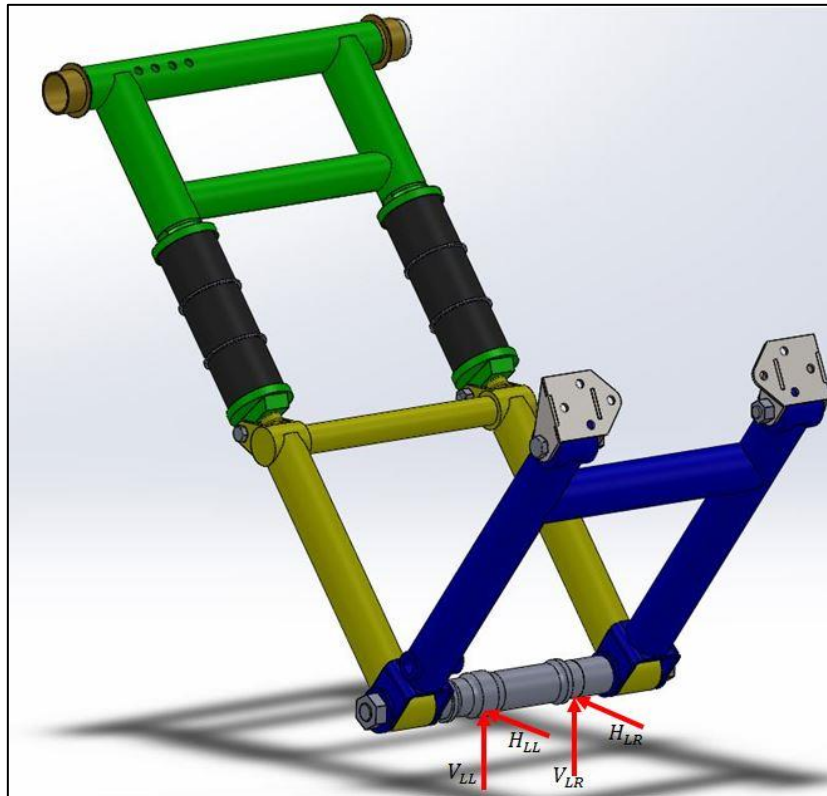


Figure 42 - Loads applied to main axle during level landing condition

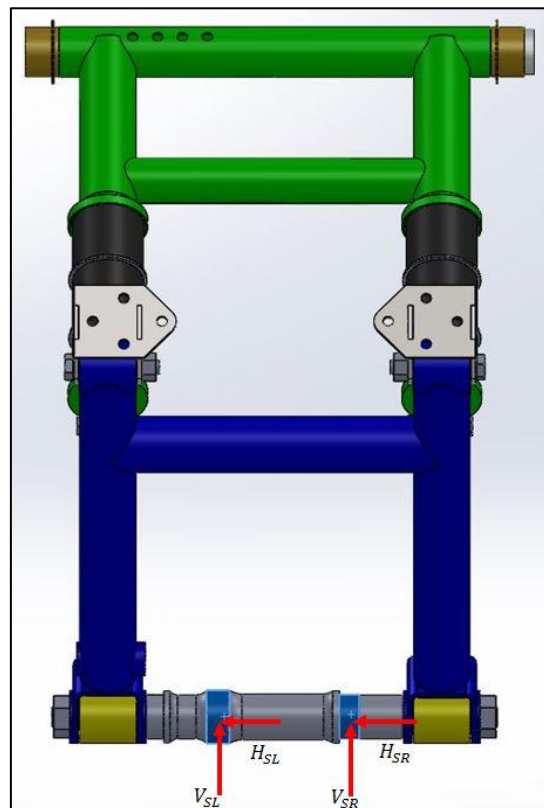


Figure 43 - Loads applied to main axle during side landing condition

The side load that acts onto the wheel resulted in a moment load at the middle of the main axle. Due to the assumption that the wheel is assumed to be rigid it is possible to transform the moment load into a vertical load that acts onto the left and right contact surfaces, as shown in Figure 43. The loads that act onto the axle for the two different landing conditions are shown in Table 29, and are illustrated on the FEA model shown in Figure 44 and Figure 45.

Table 29 - FEA Applied loads summary

Level landing condition loads			
Load	Description	Value	Unit
V_{LL}	Left side vertical load.	8829	[N]
V_{LR}	Right side vertical load.	8829	[N]
H_{LL}	Left side horizontal load rearward acting load.	5097.42	[N]
H_{LR}	Right side horizontal load rearward acting load.	5092.42	[N]
Side landing condition loads			
Load	Description	Value	Unit
V_{SL}	Left side vertical load	17206.22	[N]
V_{SR}	Right side vertical load	-8377.21	[N]
H_{SL}	Left side horizontal load side-wards acting load.	2648.7	[N]
H_{SR}	Right side horizontal load side-wards acting load.	2648.7	[N]

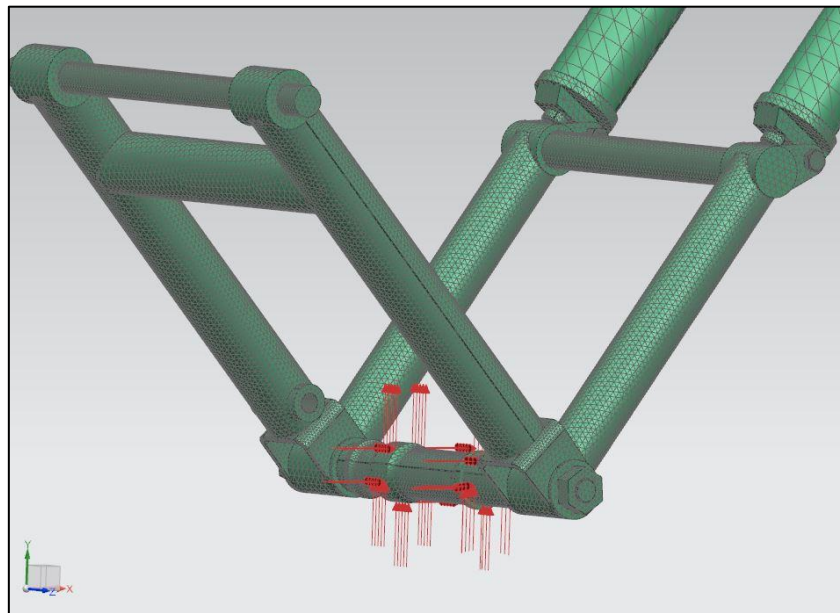


Figure 44 - FEA applied level landing loads

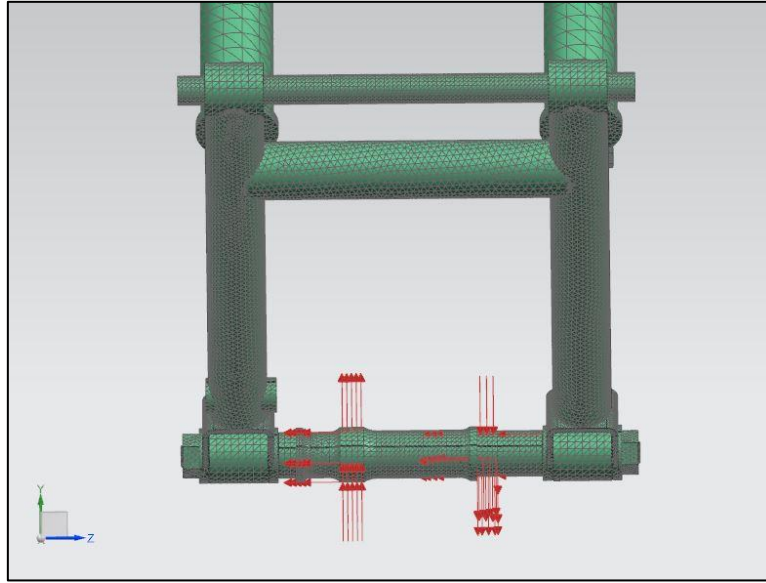


Figure 45 - FEA applied side landing loads

5.5. FEA OF THE JS-2 LANDING GEAR STRUCTURE

The numerical FEA model of the JS-2 landing gear structure discussed in the previous section was analysed as a linear simulation (SOL 101) that was able to solve the contact constraints of touching and glued parts that underwent small displacements. In order to represent a side and level landing condition separately, it was necessary to run two separate simulations. The first simulation represented the level landing conditions and loads and the second represented the side landing conditions and loads. The results of the two FEA simulations will now be discussed.

LEVEL LANDING CONDITION FEA RESULTS

The stress that the landing gear structure experiences when loads of a level landing condition is applied is shown in Figure 46 and Figure 47. Figure 46 and Figure 47 are both from the same simulation, each representing the stress that the structure experiences in a different way. In Figure 46, the stress that each node experiences, is represented and in Figure 47 the stress that each element experiences, is illustrated. The element stress is calculated by taking the average stress of all the node points that are connected to the element. This representation of the stress is much more accurate and realistic than the nodal representation when contact is present in the analysis. This is due to the reason that the element stress can accommodate FEA models with coarse meshes by excluding the possibility to create an unrealistic high stress result that is concentrated on one single node point.

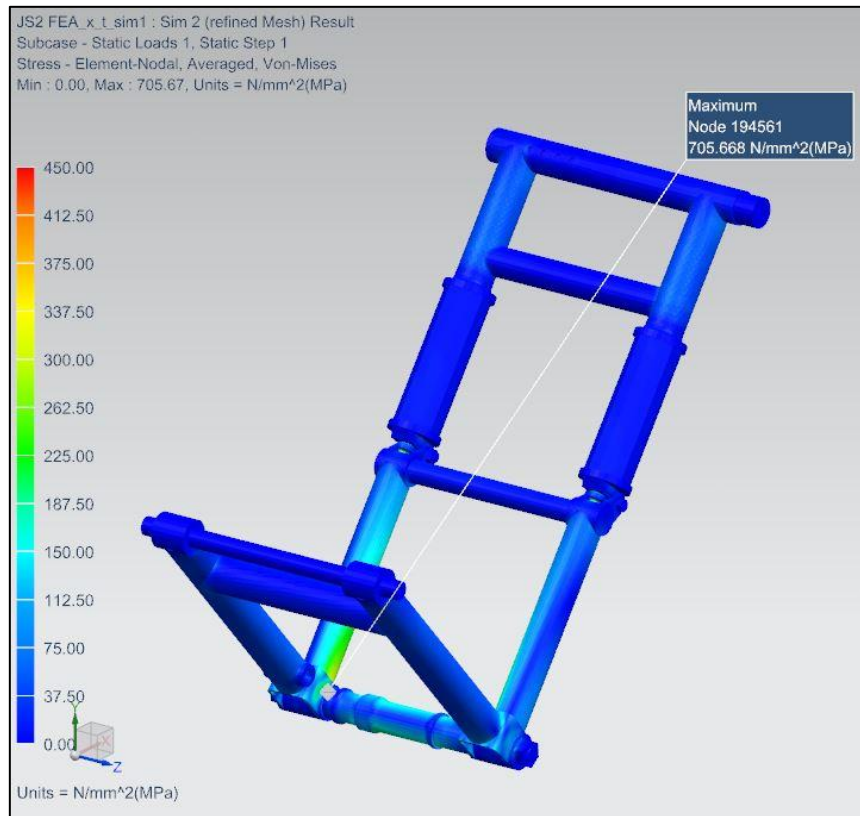


Figure 46 - Level landing condition nodal stress results

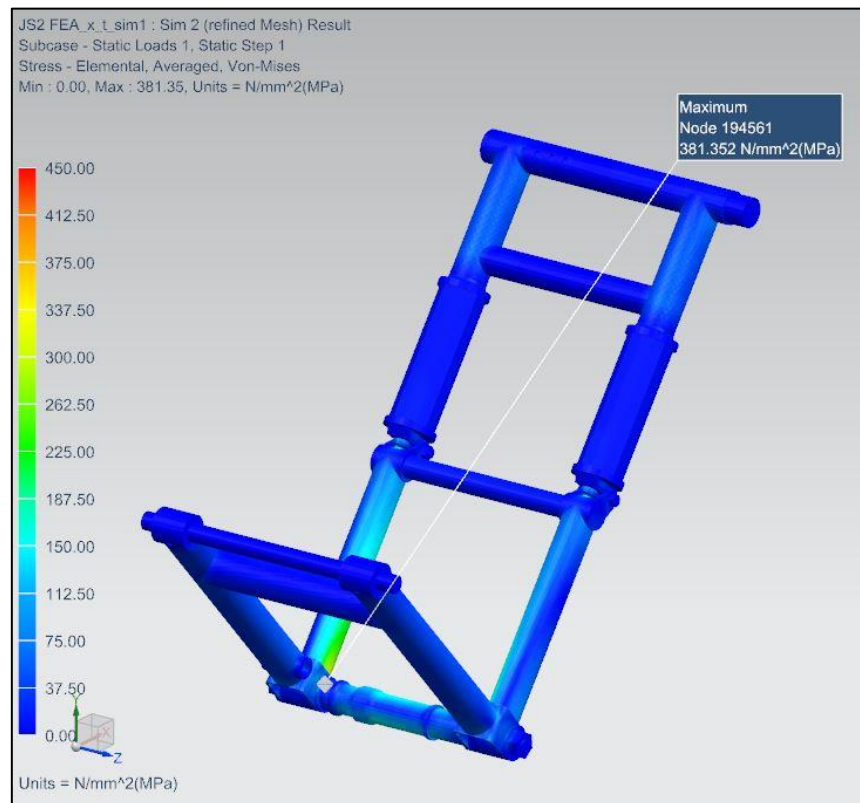


Figure 47 - Level landing condition average element stress results

In the results shown in Figure 46 and Figure 47, it is observed that during a level landing condition the loads are mostly absorbed by the rear and shock strut arm (RA), as they were designed to do. It is also clear when looking at the stress result in Figure 46 and Figure 47 that an unrealistic high stress result is found in the nodal stress representation. The maximum stress for the level landing condition taken from the element stress representation observed in Figure 47 occurs in the left bottom hinge section of the rear strut arm with a value of $\sigma_{max} = 381.35 \text{ [MPa]}$, as illustrated in Figure 48.

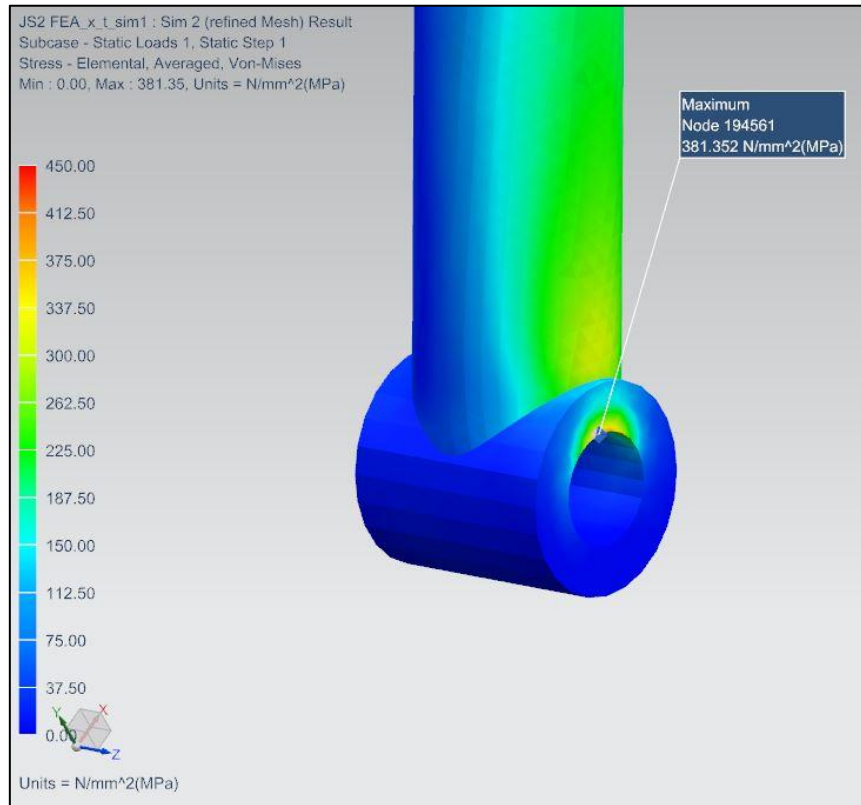


Figure 48 - Maximum stress during level landing condition

As seen in Figure 48, the stress is concentrated on a single element, therefore, making the stress result unrealistically high. This unrealistically high stress occurs due to a coarse surface mesh that can only be solved by refining the mesh between the two contact surfaces or by using the average stress of several elements. For this FEA model, it would not be feasible to refine the mesh of specific contact areas that has a global edge length of 1 [mm]. In order to get a more realistic maximum stress result, the average stress of several elements surrounding this element is used as the maximum stress with a value equal to $\sigma_{max} = 300.00 \text{ [MPa]}$.

SIDE LANDING CONDITION FEA RESULTS

The stress that the landing gear structure experiences when loads of a side landing condition are applied, is shown in Figure 49 and Figure 50. As in the level landing condition, the stress results are represented by two different methods. In Figure 49, the stress that each node experiences, is represented and in Figure 50 the stress that each element is experiencing, is illustrated. As in the level landing condition, unrealistic high stress results are obtained in the nodal representation. Due to the size of the FEA model and the associated long runtimes for a multiple contact analysis, it was not feasible to refine the mesh lower than a global element size of 1 [mm], in order to solve the problem. Consequently, it was decided to use the stress result represented as an average element stress instead of a nodal stress for further stress investigations.

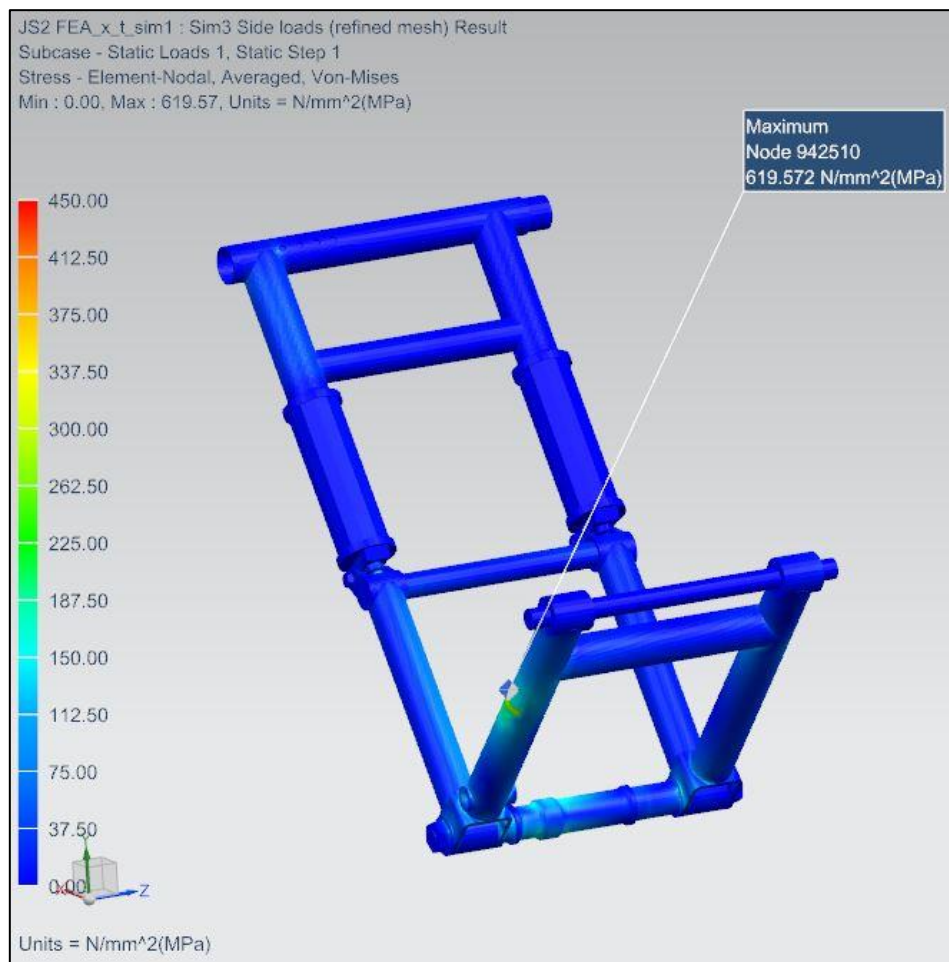


Figure 49 - Side landing condition nodal stress results

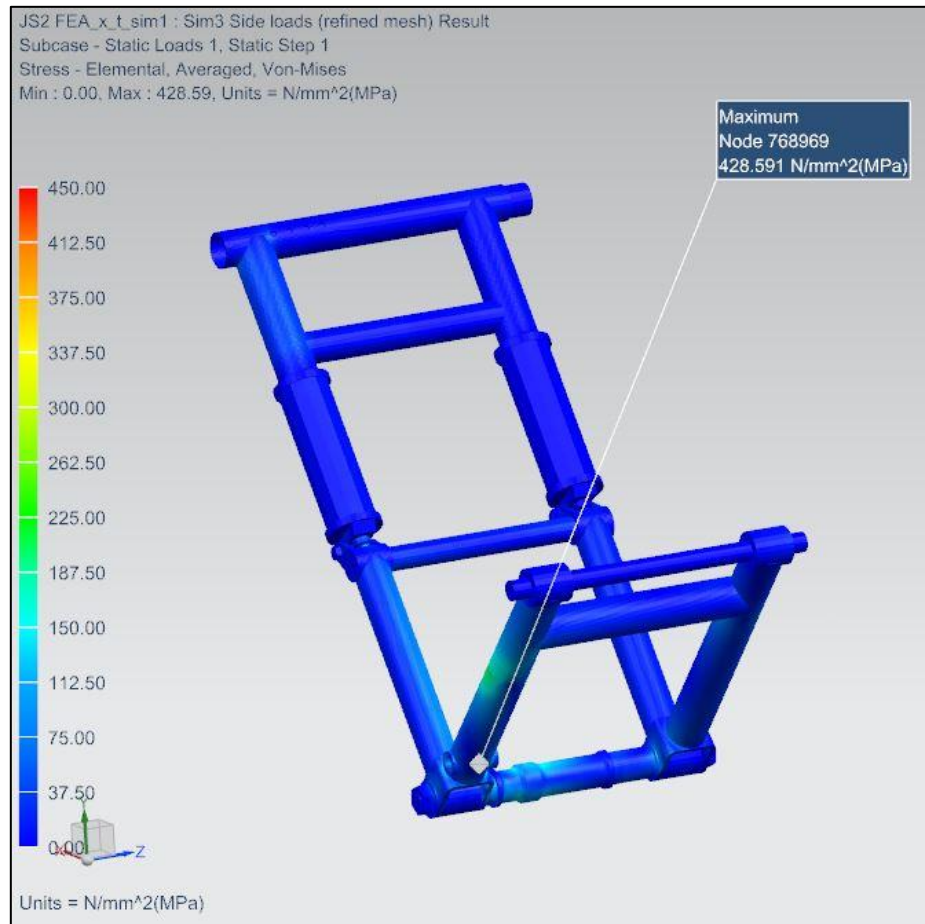


Figure 50 - Side landing condition average element stress results

In the results shown in Figure 49 and Figure 50, it is observed that during the side landing condition the side loads are mostly absorbed by the front strut arm, as it was designed to do. It is shown how the front strut arm (*FA*) tube is pressed against the fuselage wheel box reinforcement, which gives the front strut arm the necessary support to absorb all of the side loads. It is also clear from the stress result in Figure 49 and Figure 50 that an unrealistic high stress result is found in the nodal stress representation. The maximum stress for the side landing condition taken from the element stress representation in Figure 50 occurs in the left bottom hinge section of the front strut arm with a value of $\sigma_{max} = 428.59 \text{ [MPa]}$, as illustrated in Figure 51.

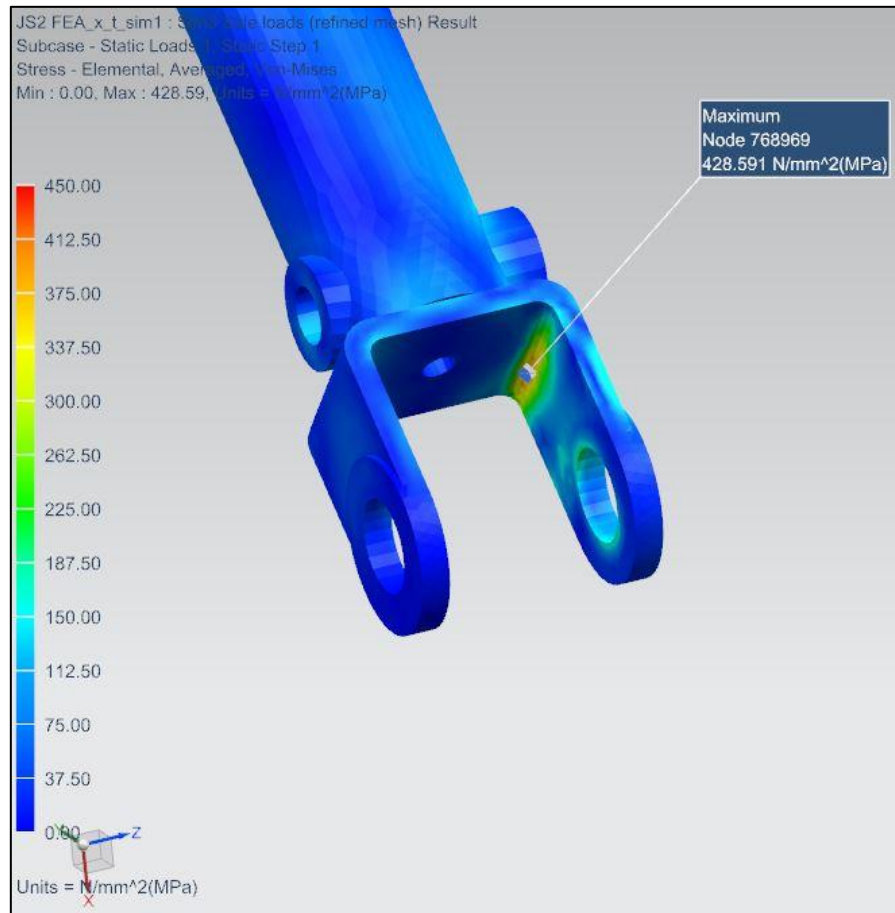


Figure 51 -Maximum stress during side landing condition

Figure 51 shows that the stress is concentrated on a few elements making the stress result unrealistically high. During real-life loading conditions this would not have occurred. The bottom rear strut arm (*RA*) hinge will provide support on the inside of the front strut arm (*FA*) bottom hinge, in order to prevent concentrated stress from occurring. The FEA solution is modelled with gaps to accommodate the coarse meshes. In order to get a more realistic maximum stress result, the average stress of several elements surrounding this element is used as the maximum stress with a value equal to $\sigma_{max} = 310.00 \text{ [MPa]}$.

SUMMARY

The FEA results of the landing gear structure during both landing conditions were investigated and it was found that the structure will experience a maximum stress of $\sigma_{max} = 300.00 \text{ [MPa]}$ during the level landing condition and $\sigma_{max} = 310.00 \text{ [MPa]}$ during the side landing condition. To get a better understanding of the total amount of stress that the components experience, each of the components were investigated individually for both the loading conditions. The stress result that was obtained during the individual investigations was then used to determine if each of these components meet the safety requirements of the CS-22 as discussed in *Section 5.2.4. Safety Factors*

The stress that each component experienced, was represented as an average element stress instead of a nodal stress in order to ensure that realistic stress results were obtained for each component. The element stress of each component was then used to determine each component's safety factor.

5.6. COMPONENTS STRESS ANALYSIS

5.6.1. SAFETY FACTOR CALCULATION

Each component of the landing gear structure will now be investigated individually. This is done to establish what stresses each component experiences during the side and level landing conditions. These relevant stresses will then be used with the materials' Yield strength (σ_{YIELD}) to calculate the components' safety factors (SF) as follow (Hibbeler, 2005):

$$SF = \frac{\sigma_{YIELD}}{\sigma} \quad (5.6)$$

Where;

σ_{YIELD} – is the yield strength of the material under investigation [MPa],

σ – is the relevant stress that the component experiences [MPa].

This is done to establish if the component adheres to the safety factor requirements of the CS-22.

Table 30 - Structural properties of the materials used in JS-2 landing gear structure (Jonker Sailplanes C.C., 2013)

Material Code	Material Description	Ultimate Strength [MPa]	Yield Strength [MPa]
316SS	Stainless steel	637	347
2024T3	Aluminium alloy steel	450	318
4130	Low alloy steel	500	475
EN8	Medium carbon steel	640	419
CZ121	Brass	415	140

5.6.2. FRONT STRUT ARM

The maximum stress that the front strut arm (FA) experiences during the level and side landing conditions is shown and compared in Figure 52 and Figure 53 .The maximum stress experienced by the component between the two landing conditions is equal to $\sigma_{max} = 428.591 \text{ [MPa]}$ during side landing conditions. The maximum stress is concentrated in one small area as shown in Figure 54 and as discussed in *Section 5.5*. During real-life loading conditions this would not have occurred. The bottom rear strut arm (RA) hinge will provide support on the inside of the front strut arm (SA) bottom hinge, in order to prevent this concentrated stress. The FEA solution is modelled with gaps to accommodate the coarse meshes. Thus, for the calculation of the safety factor (SF) at that specific point, the average stress of several elements in that area will be used.

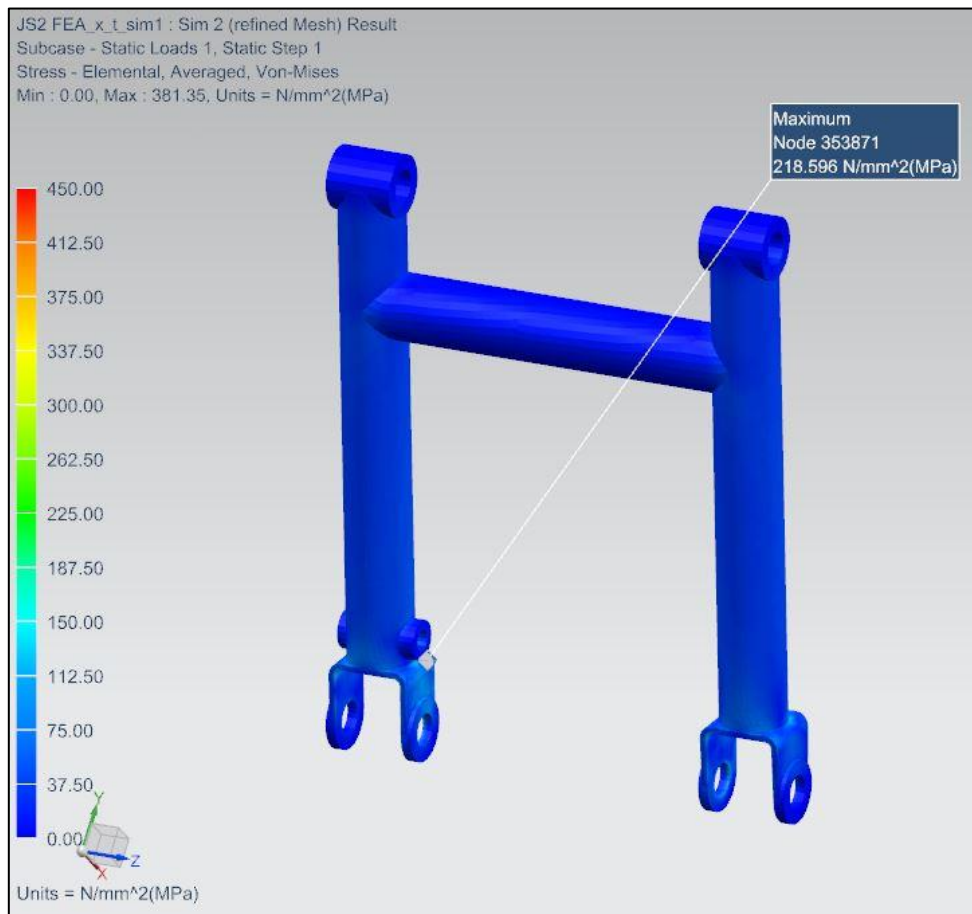


Figure 52 –Element stress results of the front strut arm during level landing condition

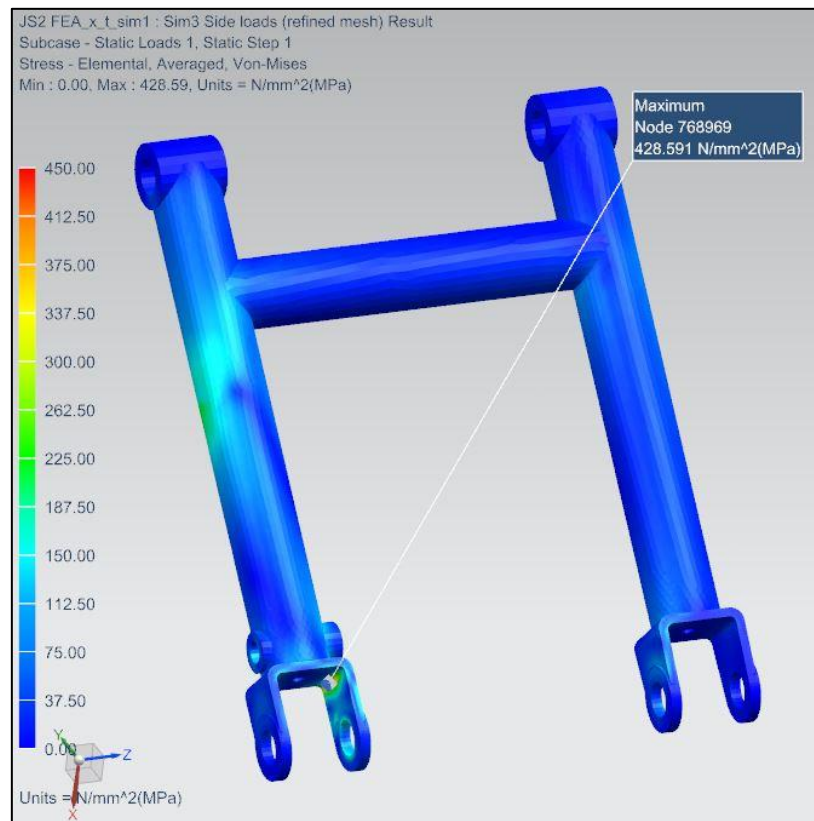


Figure 53 –Element stress results of the front strut arm during side landing condition

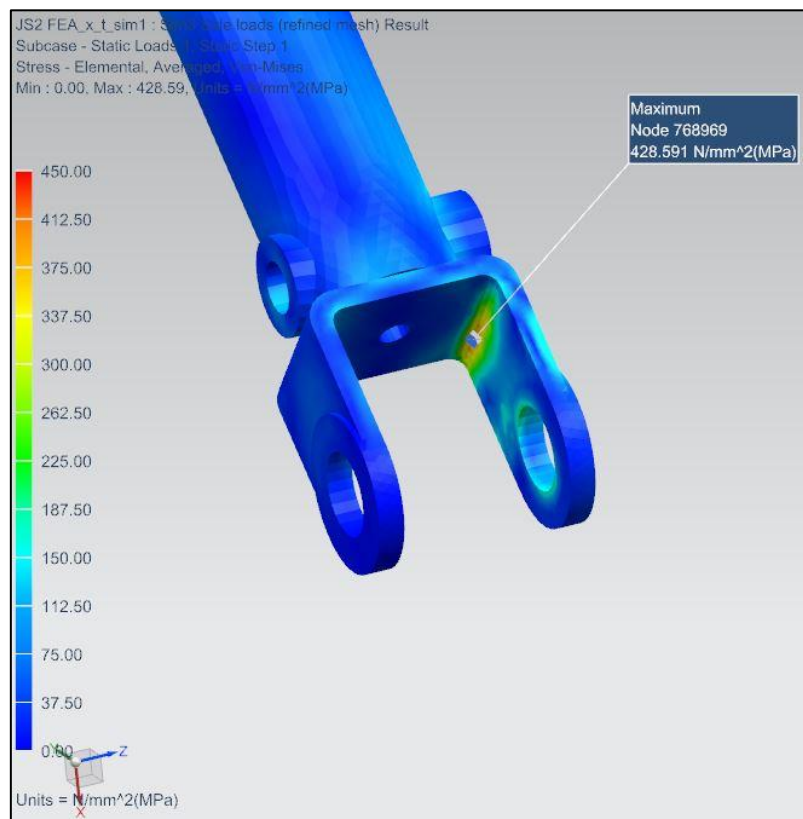


Figure 54 – Focused element stress results of front strut arm during side landing condition

The safety factors (SF) of the front strut arm (FA) are calculated with the highest stresses that are obtained from the side landing condition results. Three safety factors are calculated at three different positions on the component. Two of these are the normal safety factors calculated for the bottom hinge and the side tube shown in Figure 53. The third is a bearing safety factor that is calculated for the bottom hinge at the area where the hinge comes in contact with the axle, as shown in Figure 53. The calculated safety factors (SF) and the values used to determine them are shown in Table 31.

Table 31 – Front strut arm safety factors calculated from side landing condition stress results

Safety Factor	Description	Component Material	σ_{MAX} [MPa]	σ_{YIELD} [MPa]	SF [-]
SF_N	Normal safety factor for the bottom hinge (Figure 54).	4130	310	475	1.53
SF_B	Bearing safety factor for the bottom hinge.	4130	133	475	3.57
SF_N	Normal safety factor for the side tube.	4130	266	475	1.79

5.6.3. REAR STRUT ARM

The maximum stresses that the rear strut arm (RA) experiences during the two load cases are shown in Figure 55 and Figure 56. When compared to each other it can be seen that the stress results of the two different load cases are quite similar, except that the stress experience in the side landing condition is slightly lower. The maximum stress that is experienced by the component between the two loading conditions is found to be equal to $\sigma_{max} = 381.35 \text{ [MPa]}$ during the level landing condition.

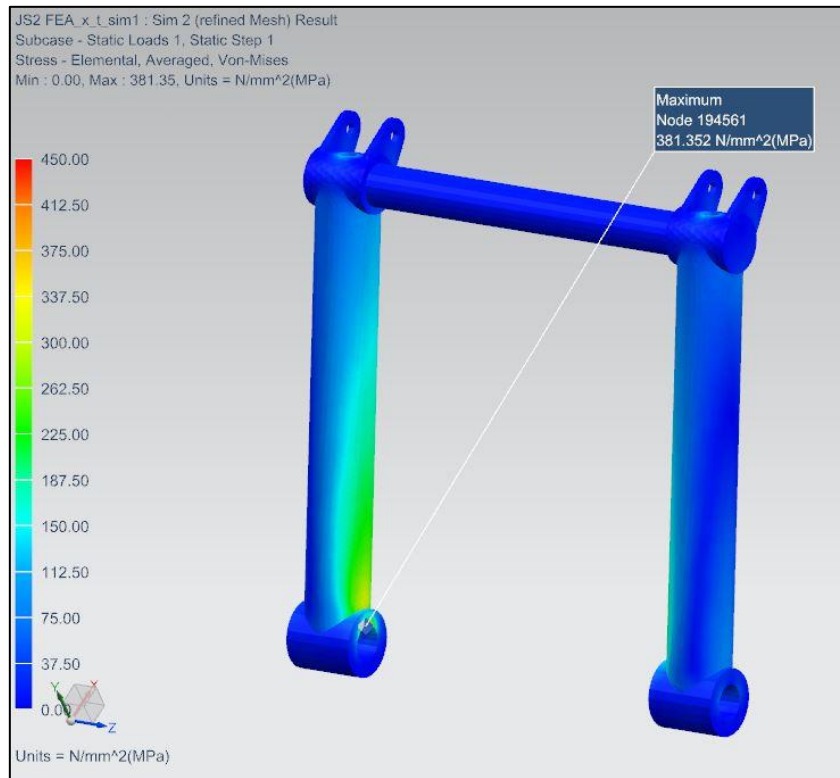


Figure 55 –Element stress results of rear strut arm during level landing condition

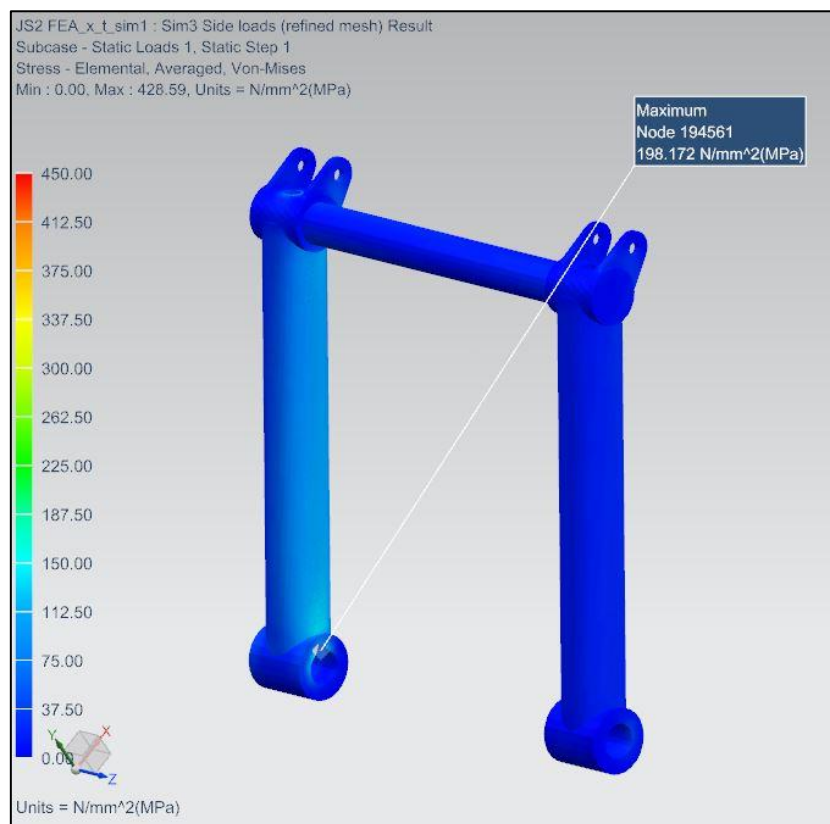


Figure 56 - Element stress results of the rear strut arm during side landing condition

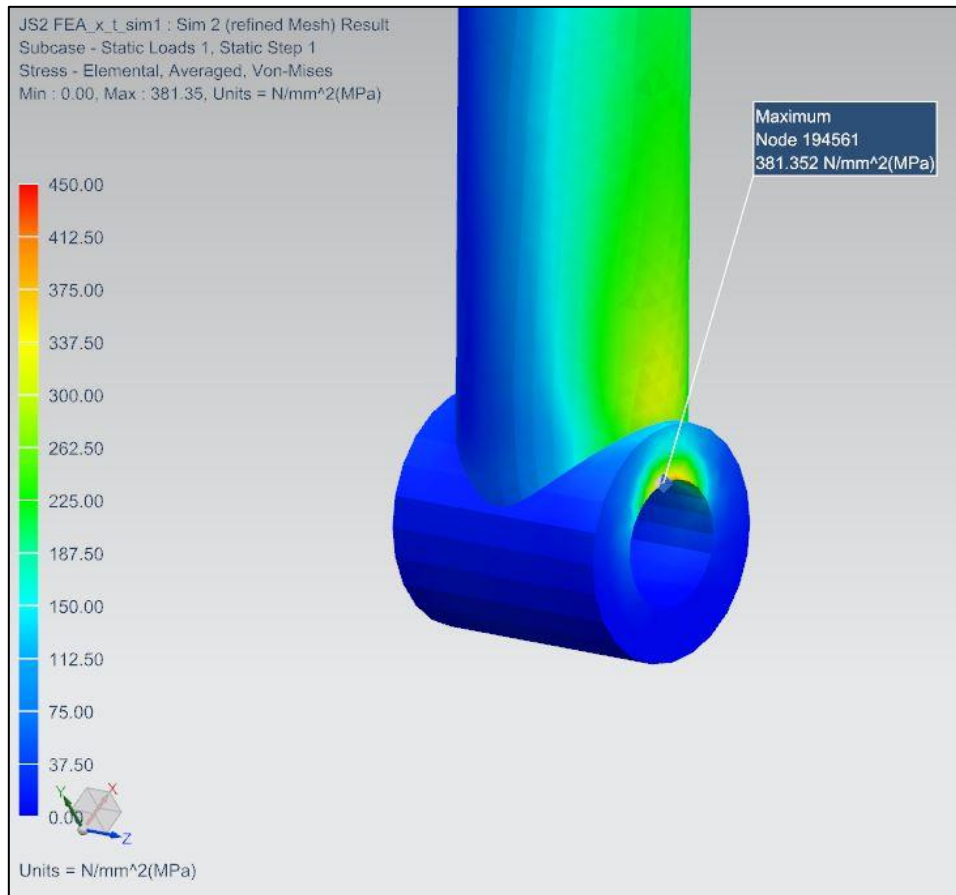


Figure 57 – Focussed element stress results of the rear strut arm during level landing condition

The area, where the maximum stress during level landing condition occurs, is enlarged in Figure 57. In this figure, it is clear that the maximum stress of $\sigma_{max} = 381.35 \text{ [MPa]}$ is concentrated on a single element, therefore, the stress result is unrealistically high. Thus, for the calculation of the safety factor (SF) at that specific point, the average stress of several elements in that area will be used.

The safety factors (SF) of the rear strut arm (RA) are calculated with the highest relevant stresses that are obtained from the stress result during a side landing condition. Only two safety factors are calculated at two different positions on the component. Two of these are the normal safety factors calculated for the bottom hinge and the side tube shown in Figure 55. The calculated safety factors and the values used to determine them are shown in Table 32.

Table 32 – Rear strut arm safety factors calculated from the side landing condition stress results

Safety Factor	Description	Component Material	σ_{MAX} [MPa]	σ_{YIELD} [MPa]	SF [-]
SF_N	Normal safety factor for the bottom hinge.	4130	300	475	1.58
SF_N	Normal safety factor for the side tube.	4130	262.50	475	1.81

5.6.4. SHOCK ABSORBER STRUT ARM

The stress that the shock absorber strut arm (SA) experiences during the level and side landing conditions is quite similar when compared to each other. The maximum stress experienced by the component when loads are applied from the level landing condition is slightly higher than the maximum stress experienced when the loads of the side landing condition are applied, as shown in Figure 58 and Figure 59. The maximum stress in both the loading conditions is found at different areas. For the level landing condition, a maximum stress is observed at the top tube connection as shown in Figure 58, and in the side loading condition, a maximum stress is observed at the bottom left hinge connection point as shown in Figure 59. The maximum stress value of $\sigma_{max} = 117.76 \text{ [MPa]}$ is obtained from the load being applied during side landing conditions, as shown in Figure 60.

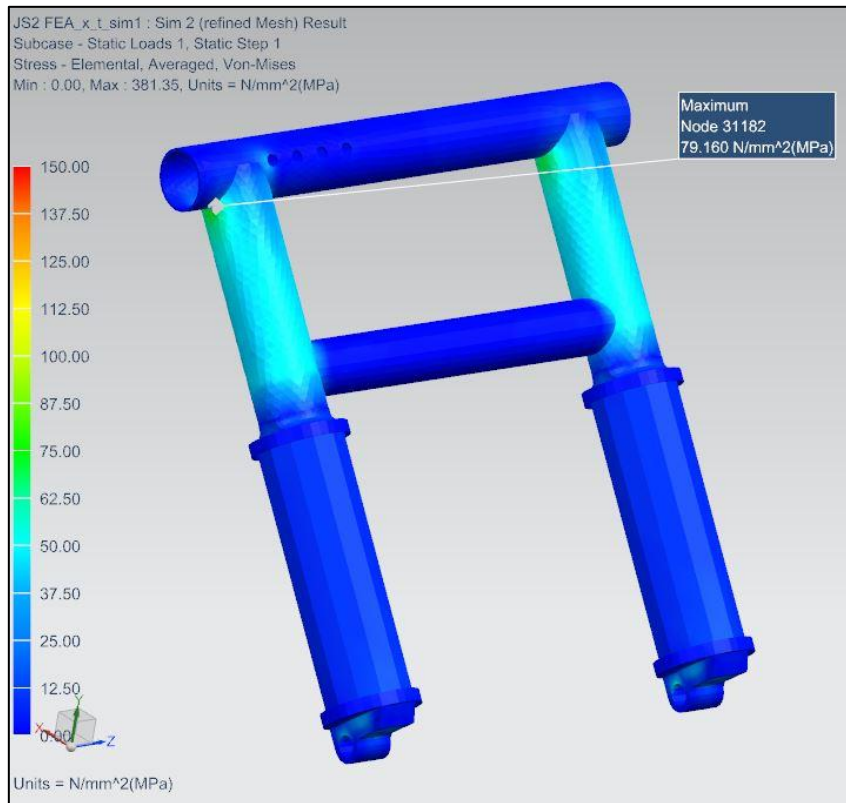


Figure 58 – Element stress results of the shock absorber strut arm during level landing condition

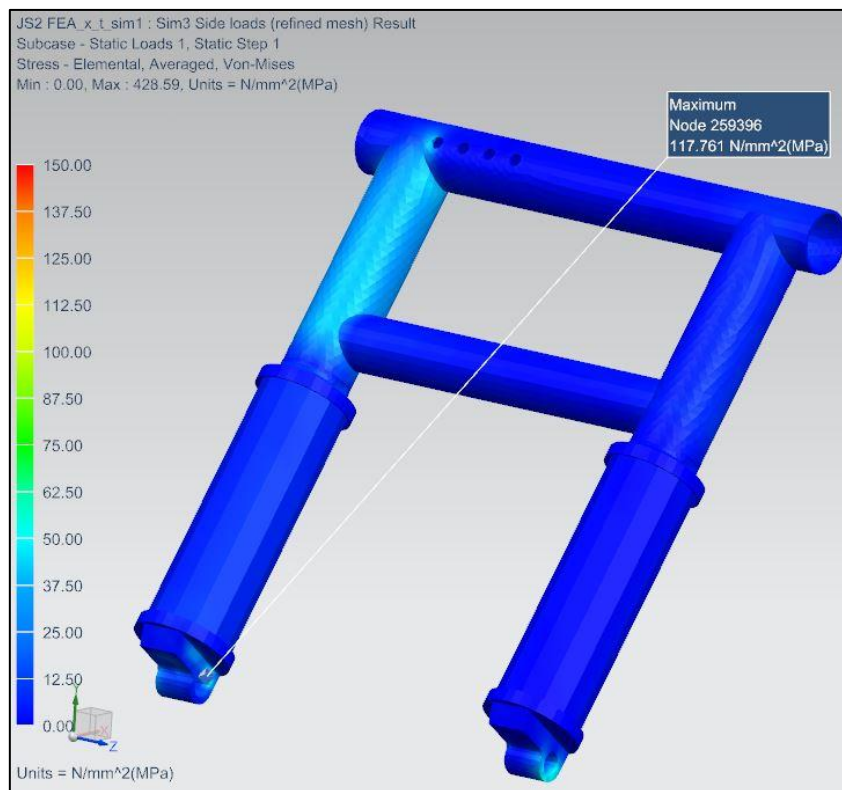


Figure 59 – Element stress results of the shock absorber strut arm during side landing condition

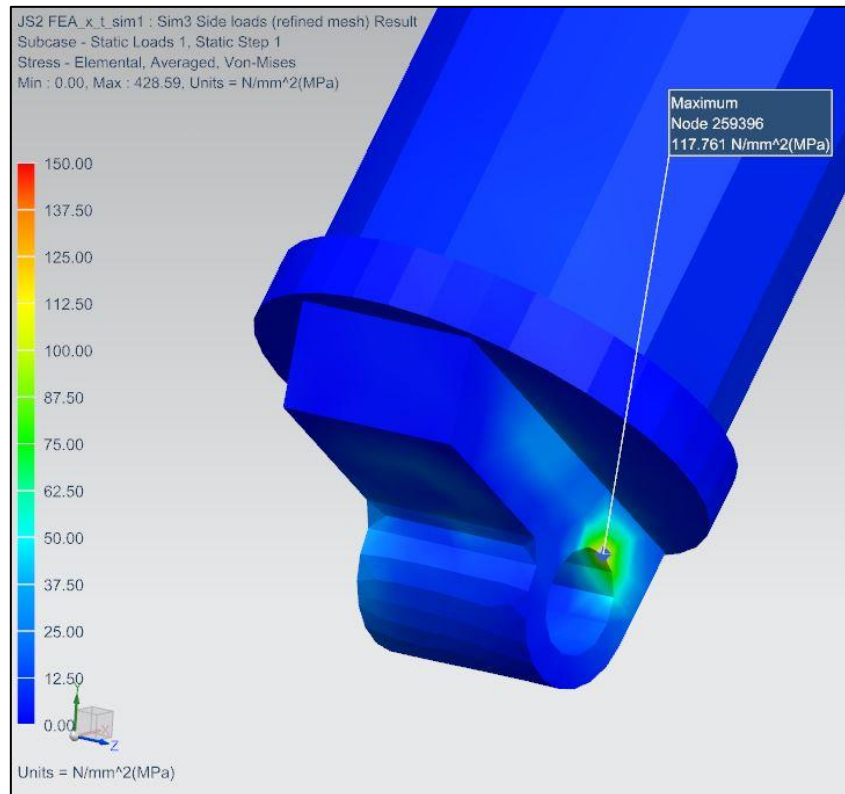


Figure 60 - Maximum bearing stress experienced by the shock absorber strut arm during side landing condition

The safety factors (SF) of the shock absorber strut arm (SA) are calculated with the maximum stress results obtained in both landing conditions. Three safety factors are calculated at three different positions on the component. Two of these are the normal safety factors calculated for the bottom shock holder cap (shown in Figure 60) and the other is calculated for the side tube (shown in Figure 58). The third calculated safety factor is a bearing safety factor calculated for the top tube at the point where it is connected to the mechanical retraction arm, as shown in Figure 59. The calculated safety factors (SF) and the values used to determine them are shown in Table 33.

Table 33 - Shock strut arm safety factors calculated from side landing condition stress results

Safety Factor	Description	Component Material	σ_{MAX} [MPa]	σ_{YIELD} [MPa]	SF [-]
SF_N	Normal safety factor at the bottom shock holder (Figure 60).	EN8	112.50	419	3.72
SF_B	Bearing safety factor at the top connection point.	4130	79.94	475	8.44
SF_N	Normal safety factor at the side tube.	4130	75	475	6.33

5.6.5. MAIN AXLE

The relevant stress that the main axle experiences during the level and side landing conditions is shown in Figure 61 and Figure 61.

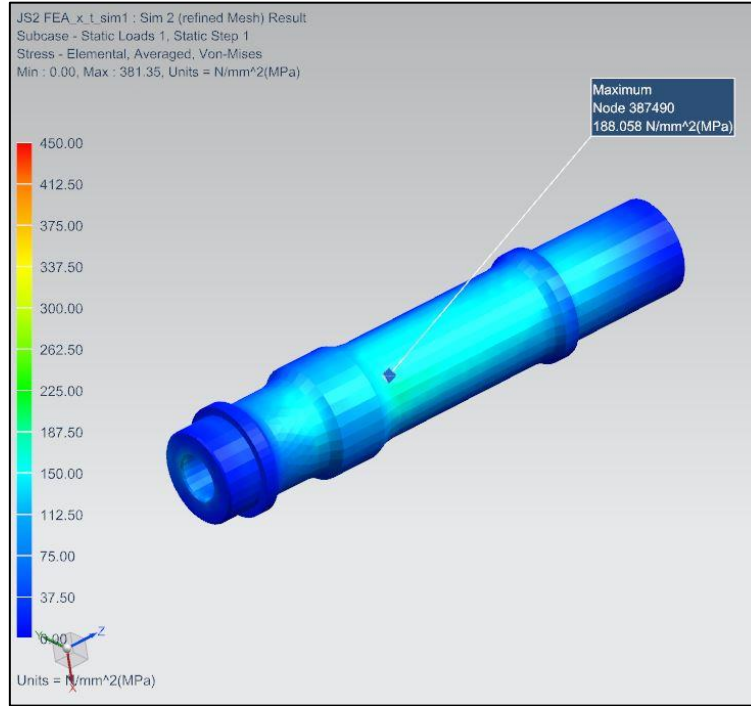


Figure 61 - Element stress results of main axle during level landing condition

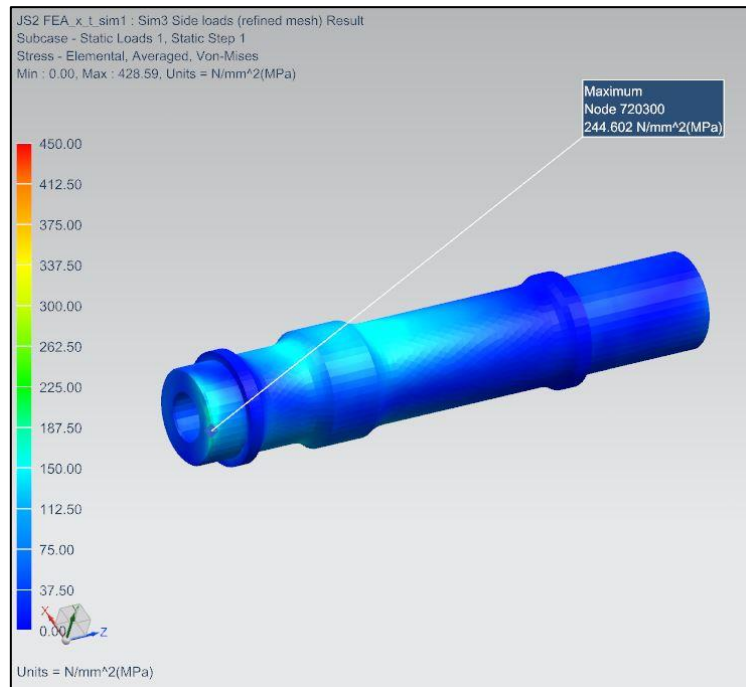


Figure 62 - Element stress results of the main axle during side landing condition

The maximum stress experienced by the component is equal to $\sigma_{max} = 244.60 \text{ [MPa]}$, which is a realistic value, but still higher than anticipated. The reason may be that the contact area is smaller than in a real-life application. Thus, for the calculation of the component's safety factor (SF) the maximum stress will be ignored and the stress being applied due to the bearings in the centre of the axle will be used.

The safety factor (SF) of the main axle is calculated from the maximum stress results obtained from the level landing condition. Only one safety factor will be calculated for the main axle. The reason, therefore, is that the main axle consists only of one material. The calculated safety factor (SF) and the values used to determine them are shown in Table 34.

Table 34 - Main axle safety factors calculated from level landing condition stress results

Safety Factor	Description	Component Material	σ_{MAX} [MPa]	σ_{YIELD} [MPa]	SF [-]
SF_N	Normal safety factor for the main axle.	2024T3	188.05	318	1.69

5.6.6. SUMMARY OF SAFETY FACTOR RESULTS

The safety factor results of each component show that the JS-2 landing gear structure adheres to the CS-22 structural safety requirements as discussed in *Section 5.2.4. Safety Factors*. The results prove that the landing gear structure may be loaded with side and level landing loads and still obtain a safety factor equal to 1.5 [-]. The minimum safety factor obtained is found at the front strut arm and is equal to 1.53 [-].

In order to prove that the stress results of the numerical simulation are reliable to prove that the landing gear structure complies with the structural safety requirements of the CS-22, it is necessary to verify the numerical results with analytical results.

5.7. VERIFICATION OF THE FEA RESULTS

In order to ensure that the results and data obtained from the FEA are accurate enough to be used to develop the JS-2 landing gear structure, it is necessary to compare the numerical result to the results obtained from analytical calculations. As the same FEA model was used throughout the landing gear structure analysis, only one component will be used in the verification. For ease of calculation the landing gear front arm was selected. The maximum load that this component experiences is during the side load condition and, therefore, this load condition will be used for comparison purposes.

5.7.1. SIMPLIFIED ANALYTICAL LOADS

The side landing condition, as discussed in *Section 5.2.3. Side Landing loads*, assumes that a horizontal side load and vertical upward load are applied to the wheel at the point where the tyre makes contact with the ground. By assuming that the tyre and the wheel are rigid bodies, the applied loads will result in vertical loads at the end of the main axle, as shown in Figure 63.

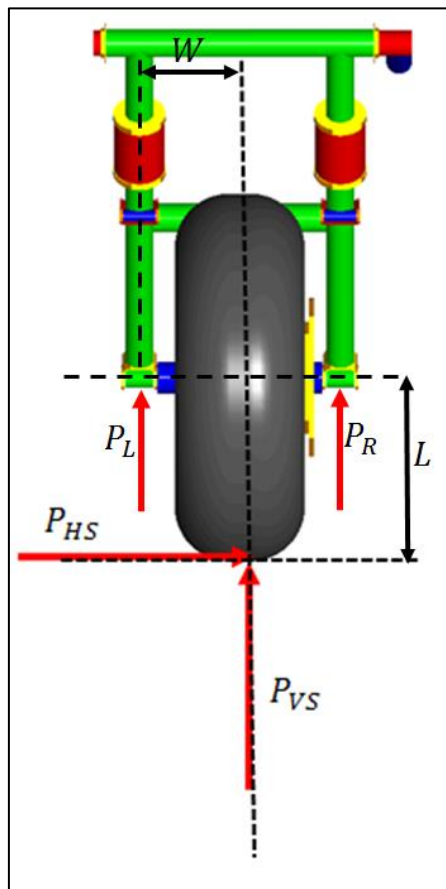


Figure 63 - Reaction loads calculated from side load

As shown previously, the side load condition consists of a vertical and side force component. In order to establish the total load being applied to the front strut arm (FA), the resultant load has to be calculated.

HORIZONTAL REACTION LOADS

The reaction loads due to the horizontal side load are calculated with *Equation 5.6* and the input variable in Table 35 as follow:

$$+\circlearrowleft \sum M_{Middle} = M_{Side\ load}: P_{HS}L = P_{HSL}W + P_{HSR}W \quad (5.6)$$

Due to the equal distance between the two reaction forces, it may be assumed that they are equal to each other. Thus, *Equation 5.6* is simplified as follow:

$$+\circlearrowleft \sum M_{Middle} = M_{Side\ load}: P_{HS}L = 2P_{HSReaction}W \quad (5.7)$$

By inserting the input variables in Table 35 into *Equation 5.7*, the reaction forces can be calculated as follow:

$$\begin{aligned} P_{HS}L &= 2P_{HSReaction}W \\ 5297.4 \text{ [N]} \times 180.5 \text{ [mm]} &= 2 \times P_{HSReaction} \times 94 \text{ [mm]} \\ \therefore P_{HSReaction} &= P_{HSL} = P_{HSR} = 5086.07 \text{ [N]} \end{aligned}$$

VERTICAL REACTION LOADS

Now that the reaction loads due to the horizontal side load are determined, the reaction loads due to the vertical side load may be calculated with the *Equation 5.8* as follow:

$$+\uparrow \sum P = P_{VS}: P_{VS} = P_{VSL} + P_{VSR} \quad (5.8)$$

Due to the equal distance between the two reaction forces, it may be assumed that they are equal to each other. Thus, *Equation 5.8* is simplified as follow:

$$+\uparrow \sum P = P_{VS}: P_{VS} = 2P_{VSReaction} \quad (5.9)$$

By inserting the input variables in Table 35 into *Equation 5.9*, the reaction forces can be calculated as follow:

$$P_{VS} = 2P_{VSReaction}$$

$$8829 = 2P_{VSReaction}$$

$$\therefore P_{VSReaction} = P_{VSL} = P_{VSR} = 4414.5 [N]$$

TOTAL REACTION LOADS

The calculated reaction loads of the side and level landing load case are now combined to calculate the total reaction load that will be acting onto the front and rear strut arms:

RIGHT REACTIONS LOAD

$$P_R = P_{VSR} + P_{HSR}$$

$$P_R = 4414.5 [N] + 5086.07[N]$$

$$\therefore P_R = 9500.57 [N]$$

LEFT REACTION LOAD

$$P_L = P_{VSL} + P_{HSL}$$

$$P_L = 4414.5 [N] + (-5086.07[N])$$

$$\therefore P_L = -671.57 [N]$$

The calculated vertical reaction loads that act onto the main axle fixture points, with the input variables used to calculate them, are summarized and shown in Table 35.

Table 35 - Vertical reaction loads

Input Variables			
Parameter	Description	Value	Unit
P_{HS}	The side load component.	5297.4	[N]
P_{VS}	The vertical load component.	8829	[N]
L	Vertical distance to the load contact point.	180.5	[mm]
W	Horizontal distance to the load contact point.	94	[mm]
Results			
Parameter	Description	Value	Unit
P_R	Right vertical reaction load	9500.57	[N]
P_L	Left vertical reaction load	-671.57	[N]

5.7.2. FRONT STRUT ARM LOADS

Due to the landing gear geometry angles, shown in Figure 64, the maximum vertical reaction load can further be divided into the loads acting onto the rear and front strut arms. The reaction loads that act onto the front strut arm will now be calculated with *Equation 5.10* and *5.11*, and the maximum reaction load (P_{MAX}) between the right (P_R) and left (P_L) vertical reaction loads from the previous section, as shown in

Table 36.

$$+\uparrow \Sigma P_y = 0: \quad 0 = P_{MAX} - P_{FRONT} \sin(47.96) - P_{REAR} \sin(51.91) \quad (5.10)$$

$$+\rightarrow \Sigma P_x = 0: \quad 0 = P_{FRONT} \cos(47.96) - P_{REAR} \cos(51.91) \quad (5.11)$$

The results obtained from these two equations are as follow:

$$P_{FRONT} = 5948.94 \text{ [N]}$$

$$P_{REAR} = 6457.63 \text{ [N]}$$

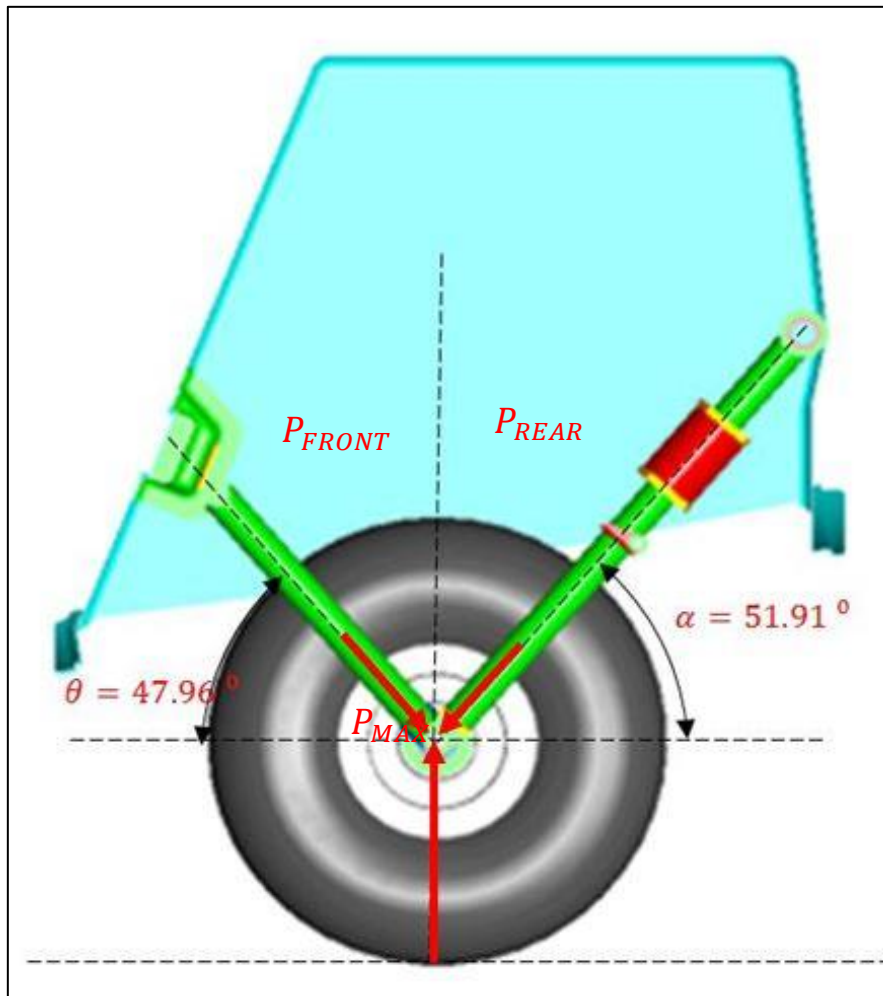


Figure 64 - Front and rear strut arm reaction forces during side landing condition

Table 36 – Input variables for the calculation of the loads acting onto the front strut arm

Input Variables			
Parameter	Description	Value	Unit
θ	Front strut arm angle	47.96	[°]
α	Rear strut arm angle	51.91	[°]
P_{MAX}	Maximum reaction load	9500.57	[N]

The loads that act onto the front strut arm (FA), which will be used to calculate the stress in the component, is summarised in Table 37.

Table 37 - Reaction loads acting onto the front strut arm

Results			
Parameter	Description	Value	Unit
P_{FRONT}	Load acting on front strut arm, perpendicular to the front arm tube axis.	5948.94	[N]
P_{SIDE}	Side load acting on front strut arm. Calculated by dividing the horizontal side load component, found in <i>Section 5.2.3</i> . Side Landing loads, by the number of strut arms ($P_{HS}/2$).	2648.70	[N]

Now that the loads that are acting onto the front strut arm (FA) are known the relevant stress can be calculated analytically in order to compare it to the numerical results.

5.7.2. ANALYTICAL STRESS RESULTS

The stress will be calculated at two different areas. The first stress is a bending stress that occurs at the position where the wheel box side reinforcement presses against the front strut arm tube surface. The second stress is a normal stress on the cross-sectional area of the front strut arm tube.

BENDING STRESS

The side load being applied to the front arms structure, as shown in Figure 65, will create a bending load to the side of the front strut arm tube where the fuselage reinforcement presses against the structure. The analytical bending stress due to this side load will be calculated in order to be compared to the numerical stress results. The equations and input variables used to calculate the bending stress, together with the results, are shown in Table 38.

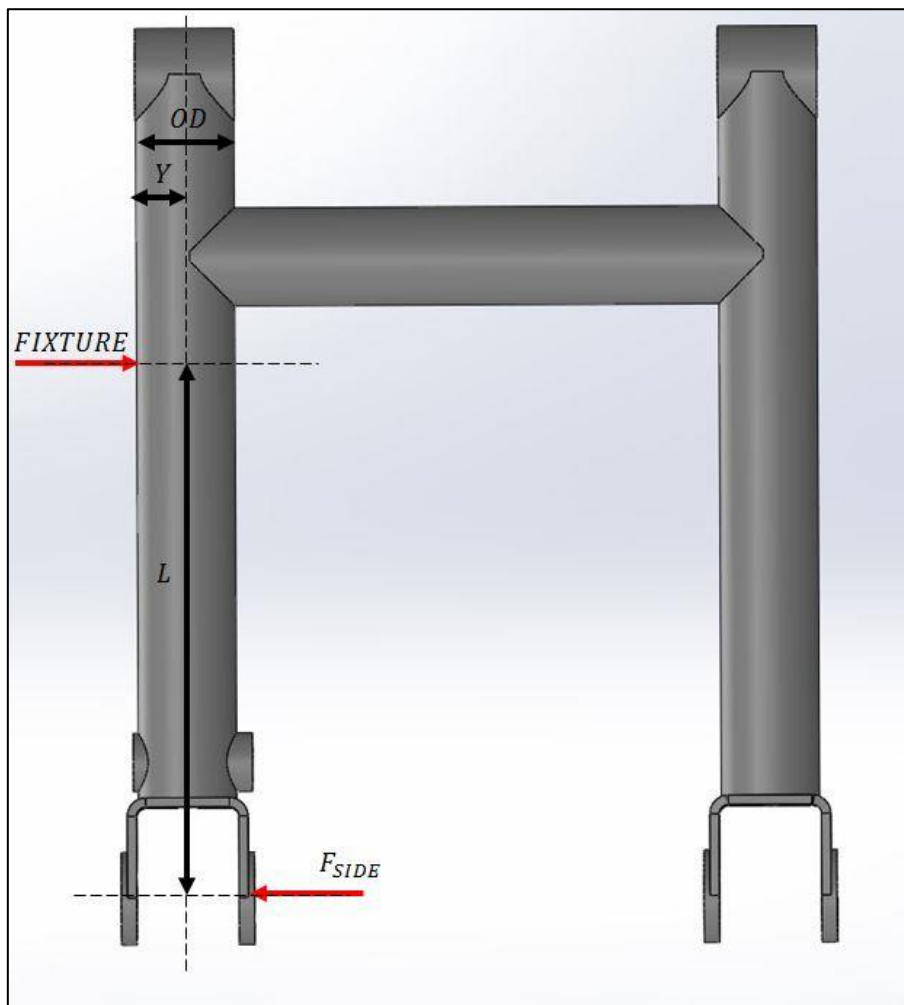


Figure 65- Loads creating bending stress on the front strut arm

Table 38 - Analytical bending stress Result

Equations (Hibbeler, 2005)			
$\sigma_{Bend} = \frac{M Y}{I_x} \quad (5.12)$			
Where;			
$M = F_{SIDE} L \quad (5.13)$			
$I_x = \frac{\pi}{4} (r_{OD}^4 - r_{ID}^4) \quad (5.14)$			
Input Variables	Description	Value	Unit
r_{OD}	Outer radius of side tube.	15.86	[mm]
r_{ID}	Inner radius of side tube.	13.46	[mm]
Y	Perpendicular distance to the inertia axis.	13.46	[mm]
L	Distance between side load and side support contact points.	155	[mm]
F_{SIDE}	Side load acting onto one tube of the front strut arm.	2648.70	[N]
Results	Description	Value	Unit
I_x	Area moment of inertia.	23914.61	[mm ⁴]
M	Bending moment at the side reinforcement contact area.	410.55	[N/m]
σ_{Bend}	The maximum bending stress.	231.07	[MPa]

This calculated bending stress will be compared to the numerical results in *Section 5.7.3*.

COMPRESSION STRESS

The side load being applied to the front arms structure will also create an upward compression stress onto the cross-sectional area of the front strut arm tube, as shown in Figure 66. The analytical compression stress due to the compression load will also be calculated in order to be compared to the numerical stress results. The equations and input variables used to calculate the bending stress, together with the results, are shown in Table 39.

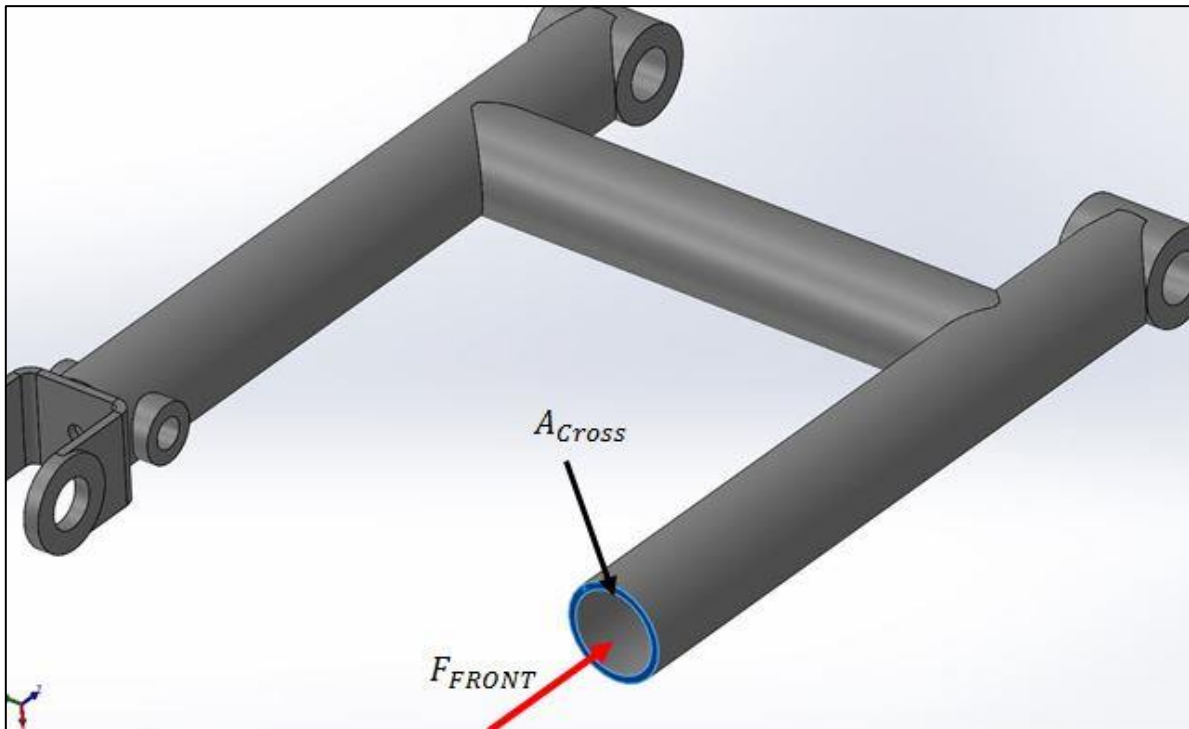


Figure 66 - Loads creating a compression stress on the front strut arm

Table 39 - Analytical compression stress results

Equations (Hibbeler, 2005)			
$\sigma_{Comp} = \frac{F}{A_{Cross}} \quad (5.13)$ Where; $A_{Cross} = \frac{\pi}{4} (OD^2 - ID^2) \quad (5.15)$			
Input Variables	Description	Value	Unit
OD	Outer diameter of side tube.	15.86	[mm]
ID	Inner diameter of side tube.	13.46	[mm]
F_{FRONT}	Load acting on front strut arm, perpendicular to the front arm tube axis.	5948.9	[N]
Results	Description	Value	Unit
A_{Cross}	Cross-sectional area where load is being applied.	221.07	[mm ²]
σ_{Comp}	The maximum compression stress.	26.91	[MPa]

This calculated compression stress will be compared to the numerical results in *Section 5.7.3*.

5.7.3. STRESS RESULT COMPARISON

The two calculated stress results will now be compared with the FEA results. The maximum bending and compression stress obtained from the numerical FEA results is equal to $\sigma_{Bending\ N} = 225\ [MPa]$ and $\sigma_{Compression\ N} = 25\ [MPa]$, and is shown in Figure 67. This correlates with the analytical stress when compared to each other, with a maximum bending stress of $\sigma_{Bending\ A} = 231.07\ [MPa]$ and $\sigma_{Comprssion\ A} = 26.91\ [MPa]$.

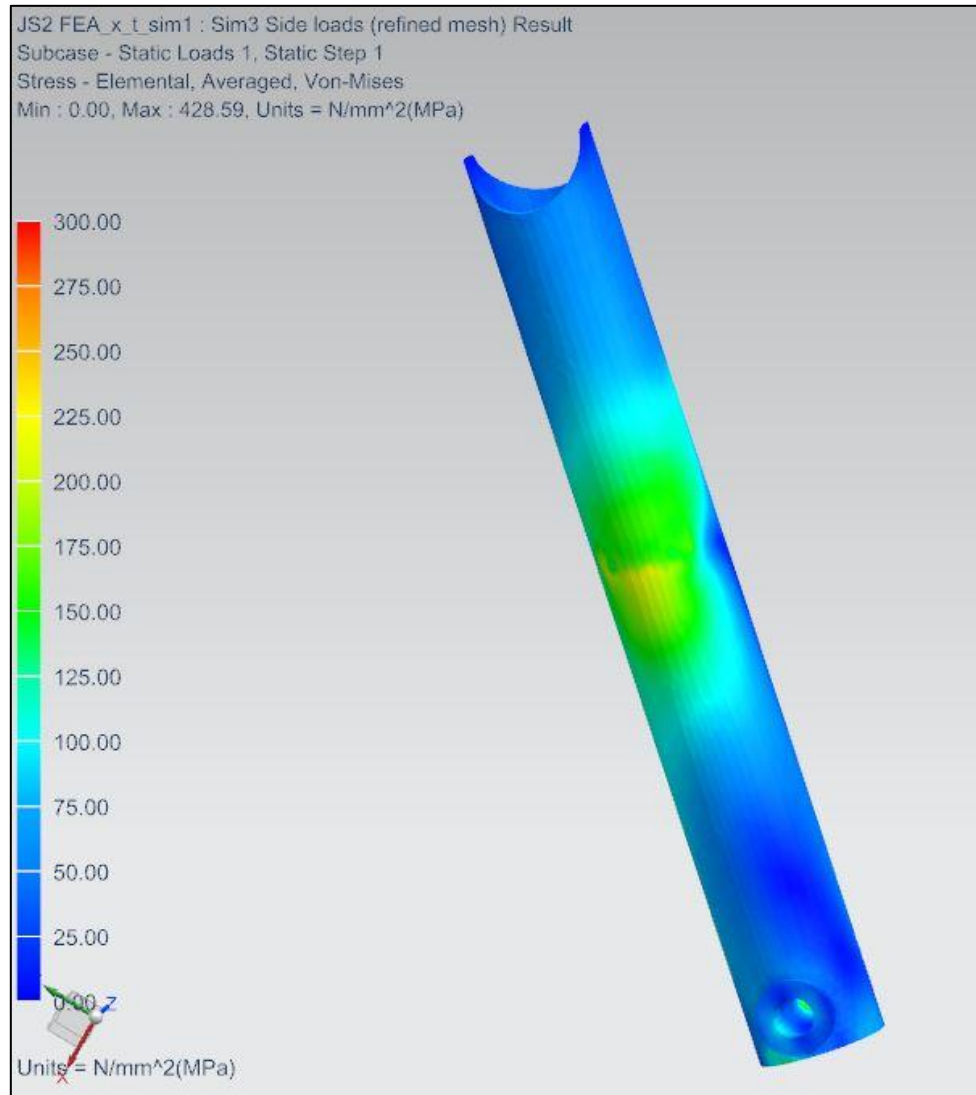


Figure 67 - Maximum bending stress due to side loads

This proves that the JS-2 landing gear structure is correctly modelled and that it is able to predict accurate results. Therefore, it is possible to use the numerical stress results to prove that the JS-2 landing gear structure adheres to the CS-22 safety requirements.

5.8. SUMMARY

In this chapter it was shown analytically and numerically that the developed JS-2 landing gear structure adheres to the CS-22 structural safety requirements. This included the discussion of the final JS-2 landing gear structure and the FEA model used to develop it.

The FEA code, NX NASTRAN, was used to investigate the stress that was experienced by each structural component of the landing gear during side and level landing conditions. It was found that the element stress represented more realistic results than the nodal stress distribution. It was then decided to use the average element stress results to calculate the safety factor of each individual component in order to ensure that the structure adheres to the CS-22 requirements. It was found by analysing each component that the landing gear structure adheres to the CS-22 requirements with a minimum load safety factor of 1.57 [-].

The stress results that were obtained from numerical calculations were then compared to stress results that were calculated analytically in order to establish if the results of the numerical calculations were accurate. The results of the two different calculation methods correlated with each other. Thus, it was verified that the JS-2 landing gear structure was modelled correctly and that the numerical results are reliable and accurate enough to be used to prove that the JS-2 landing gear structure adheres to the CS-22 safety requirements.

CHAPTER 6 : CONCLUSION AND RECOMMENDATIONS

6.1. CONCLUSION

The main objective of the study was to redevelop the JS-1 landing gear to fit the JS-2 fuselage with specific focus on the shock absorption system so that the CS-22 requirements were fully met. In order to achieve this objective, it was found by conducting an accurate literature study that the rubber shock absorber used within the JS-1 landing gear could be redesigned to be much smaller to fit within the narrower JS-2 fuselage.

Several different constitutive hyperelastic material models used to describe the behaviour of elastomers were investigated in order to determine the most applicable model for the rubber that will be used in the JS-2 shock absorber. The Yeoh model was selected because of its ability to predict accurately the behaviour for large strain applications in different deformation modes. The Yeoh material model coefficients were then derived by a single deformation mode, uniaxial compression test.

The hyperelastic Yeoh material model was then validated by modelling the JS-1 rubber shock in the FEA codes NASTRAN, MARC and PATRAN and by comparing the numerical FEA results to actual experimental results. The validation results indicated a correlation between the experimental and numerical results that concluded that it was possible to use the Yeoh material model for complex geometries subjected to compressional load, where the material coefficients were determined by a single uniaxial compression test.

Having validated that the polyurethane rubber had been modelled correctly with the Yeoh material model in the commercially available FEA PATRAN code, the Yeoh material model was used to develop the JS-2 rubber shock absorber. This included the development of the shock absorber rubber elements and the development and selection of the wheel and the tyre. It was then proven analytically and numerically that the developed shock absorption system of the JS-2 landing gear system adhered to the CS-22 certification specifications, with a limit landing gear load factor ($n_{LG\ LIM}$) of 2.794 [-] and an ultimate landing gear load factor ($n_{LG\ ULT}$) of 3.484 [-]. These load factors with the developed JS-2 rubber shock absorber were then used to develop the JS-2 landing gear structure with FEA code.

The FEA code, NX NASTRAN, was used to develop the JS-2 landing gear structure. This code was also used to investigate the stress that was experienced by each structural component of the landing gear during side and level landing conditions in order to ensure that the structure adheres to the CS-22 requirements. It was found by analysing each component that the landing gear structure adheres to the CS-22 requirements with a minimum load safety factor of 1.57 [-].

The stress results that had been obtained from numerical calculations were then compared to a stress result that was calculated analytically in order to establish if the results of the numerical calculations were accurate. The results of the two different calculation methods correlated with each other. Thus, verifying that the JS-2 landing gear structure was modelled accurately and that the numerical results are reliable and accurate to prove that the JS-2 landing gear structure adheres to the CS-22 safety requirements.

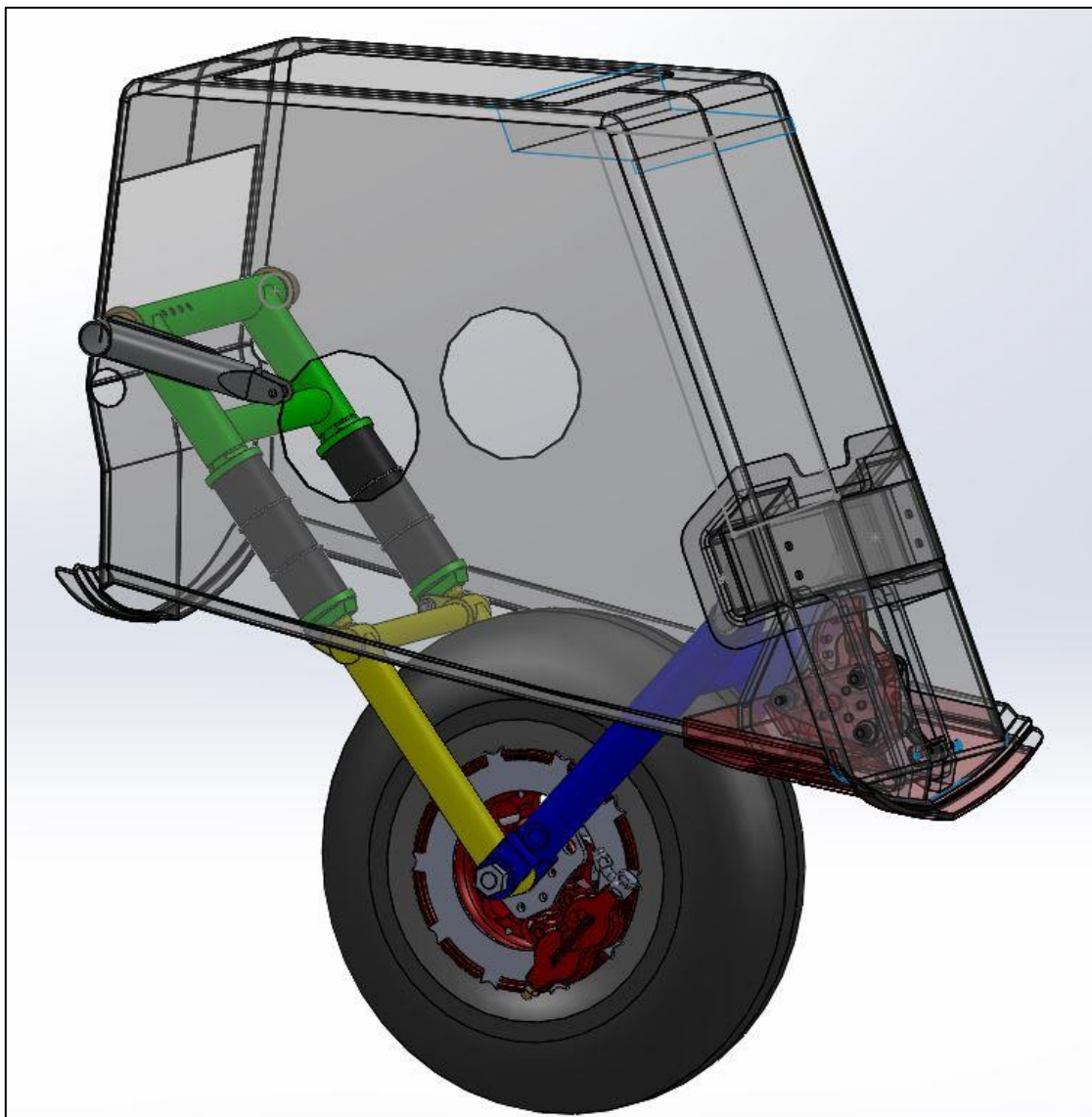


Figure 68 - Develop JS-2 landing gear system

It was concluded that the landing gear system, as shown in Figure 68, which was developed in this study adhered to the JS-2 fuselage geometrical constraints as well as to the certification specifications of the EASA CS-22. The developed JS-2 landing gear system is 34 [mm] narrower compared to the JS-1 landing gear system and it is able to absorb the total amount of energy that is developed with a maximum all up weight of 600 [Kg] and a decent velocity of 1,77 [m/s], without bottoming out and exceeding the CS-22 load factor limit of 4.5 g's [-].

6.2. RECOMMENDATIONS

Although it was specified by the CS-22 certification specifications that it is only necessary to prove by static calculations that the landing gear system adheres to its safety and structural requirements, it is recommended that for future investigations, the landing gear system should be developed by static and dynamic loads.

It is also recommended that a test rig should be developed that is able to experimentally determine the main landing gear tyre's behaviour for several different internal pressures, instead of using the manufacturer's data. It may also be useful to develop a 3D FEA model of the selected tyre to be used with the FEA model of the rubber elements to get a more accurate result of the landing gear energy absorption capabilities.

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APPENDIX A – Uniaxial Compression Test

The compression test is based on ASTM D575-91 that is the standard test method for rubber properties in compression. The standard covers the test procedure for determining the compression-deflection characteristics of rubber compounds.

APPARATUS

The apparatus used for the uniaxial compression test was the MTS Landmark Servo-hydraulic testing machine of the North-West University's Material Laboratory, as shown in Figure 69. The testing machine is fully automated and is capable of recording the Force- displacement results through a data logger. The testing machine's calibration documents are shown in *APPENDIX C*.



Figure 69 - MTS Landmark Servo-Hydraulic test system (MTS Systems Corporation, 2015)

TEST SPECIMENS

The specimens were made to the requirements of ASTM D575. They are cylindrical in shape with an outside diameter of 28.6 [mm] and a height of 12 [mm]. The specimens were made of polyurethane rubber, which is a synthetic rubber that is manufactured by mixing two chemical parts and pouring the mixture into a mould of the required shape. Five different polyurethane specimens were casted for the test, each with a different hardness [Shore A], as shown in Table 40. The hardness of the five different specimens was verified by a durometer, as shown in Table 40.

Table 40 - Specimen hardness comparison

Theoretical Hardness [Shore A]	Measured Hardness [Shore A]
40	40
50	51
60	61
65	65
80	80

TEST CONDITIONS

The test was done at the Material Laboratory of the North-West University at a temperature of 25 [°C]. The specimens that were tested were stored in the material laboratory, three hours prior to testing as required by the test standard.

PROCEDURE

The test specimens were placed between the two parallel compression plates. Sheets of sandpaper were placed on the two parallel compression plates where the rubber specimens came in contact with them. This is done to ensure that no lateral slippage occurs at the specimens' contact surfaces.

A small force was then applied to the specimen in order to adjust the displacement gauge. After the displacement gauge had been set to zero the force was applied with a strain rate of 0.1

[mm/s] until a maximum force of 4500 [N] was achieved. The force and deflection was recorder in 1 second increments until the desired load was achieved.

RESULTS

The load deflection results of the five different samples are graphically represented in Figure 70.

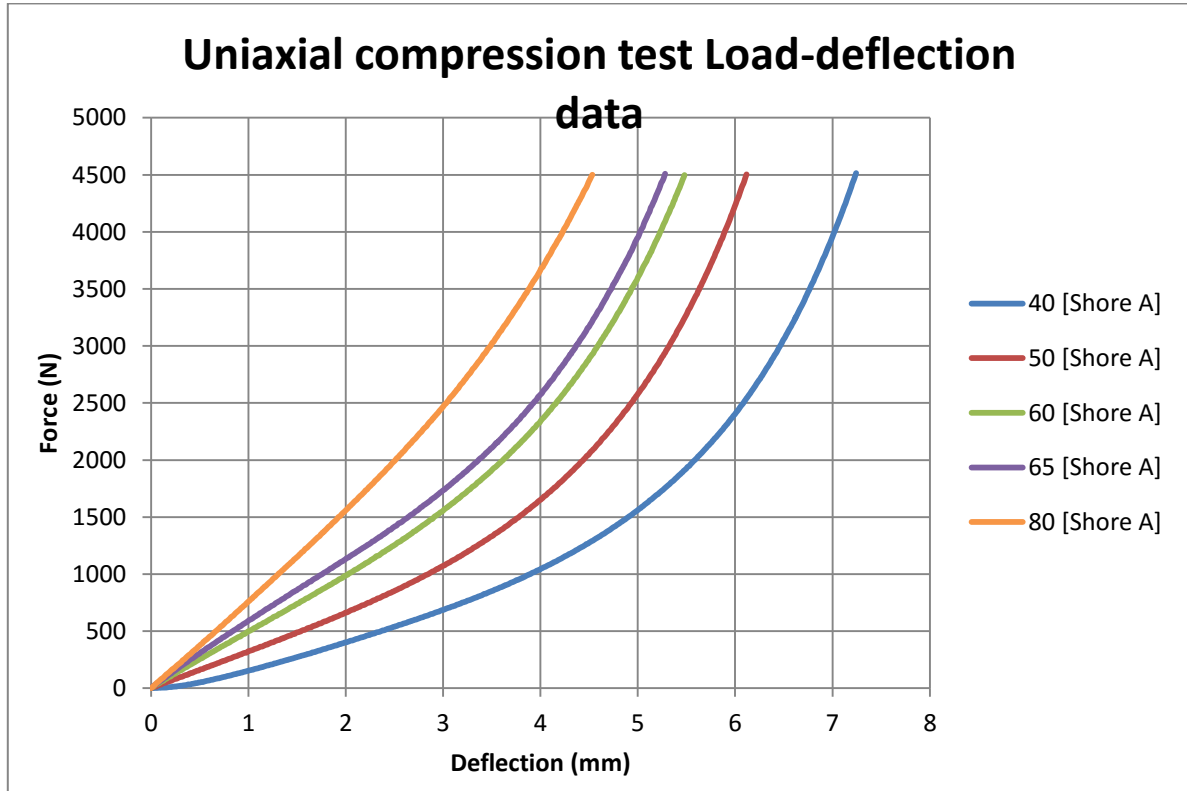


Figure 70 - Uniaxial compression test load-deflection data

In order to calculate the five different specimens' Yeoh material coefficient, it was necessary to represent the data in terms of stress and strain. This was done by using the initial dimensions of the specimens, and the following two equations:

The stress (σ) was calculated by using the applied load (F) and the diameter of the sample (Hibbeler, 2005):

$$\sigma = \frac{F}{A} \quad (A.1)$$

Where;

F – is the applied load in [N],

A – is the cross-sectional contact area of the specimen [mm²].

The strain (ε) calculated by using the initial height (L_o) of the specimen and the height of the specimen when compressed (Hibbeler, 2005):

$$\varepsilon = \frac{L_o - L_f}{L_o} \quad (A.2)$$

Where;

L_o – is the initial height of the specimen [mm],

L_f – is the height of the specimen when compressed [mm].

The force displacement results represented as stress strain data are shown in Figure 71.

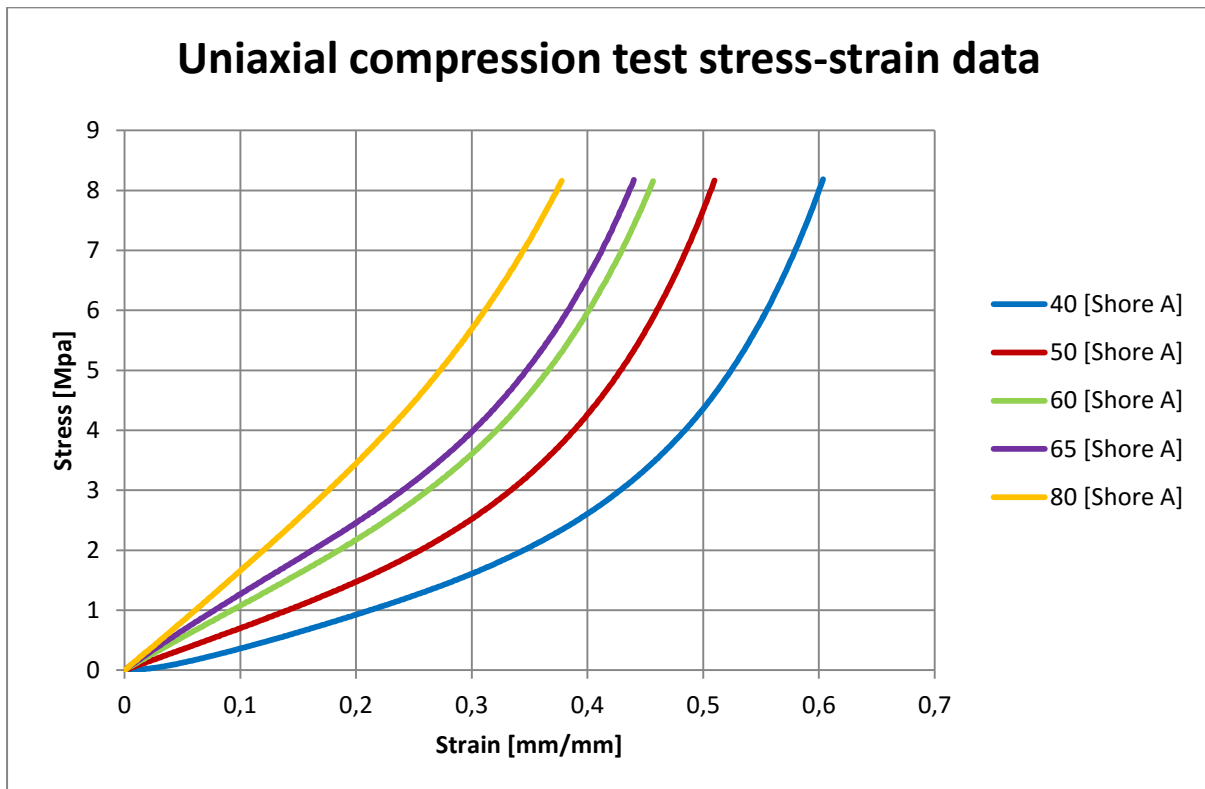


Figure 71 - Uniaxial compression test stress-strain data

These results were used to determine the coefficients of the Yeoh material model.

APPENDIX B – JS-1 Shock Absorber Load- Deflection Test Data

Test:	Compression	JS-1 Shock Absorber
Test	Run:	Test
Date:	6/9/2014	
_Extension	_Load	_Time
mm	kN	sec
0.123842532048002	0.126615882873535	0.1650390625
0.21023461886216	0.153959915161133	0.2646484375
0.294720026431605	0.160552734375	0.3642578125
0.378983764676377	0.184645111083984	0.4638671875
0.464582029962912	0.209236099243164	0.5634765625
0.54884230485186	0.23434049987793	0.6630859375
0.63456071075052	0.256212646484375	0.7626953125
0.718754774425179	0.279149017333984	0.8623046875
0.803120085038245	0.301526947021484	0.9619140625
0.887653965037316	0.328081329345703	1.0615234375
0.973317888565362	0.352270629882813	1.1611328125
1.05778977740556	0.373894989013672	1.2607421875
1.14306714385748	0.3954677734375	1.3603515625
1.22673669829965	0.420438659667969	1.4599609375
1.31198309827596	0.444277648925781	1.5595703125
1.39558152295649	0.468831115722656	1.6591796875
1.48136727511883	0.494139129638672	1.7587890625
1.56586396042258	0.5189677734375	1.8583984375
1.64954783394933	0.541685180664063	1.9580078125
1.73390307463706	0.559481872558594	2.0576171875
1.81978824548423	0.58698046875	2.1572265625
1.89962412696332	0.613510314941406	2.2568359375
1.98727613314986	0.633141174316406	2.3564453125
2.07176874391735	0.652468566894531	2.4560546875
2.15634540654719	0.675888732910156	2.5556640625
2.24075163714588	0.696909484863281	2.6552734375
2.3258610162884	0.719052062988281	2.7548828125
2.40988912992179	0.738068908691406	2.8544921875
2.49427813105285	0.761452819824219	2.9541015625
2.57815117947757	0.781102966308594	3.0537109375
2.6620514690876	0.795792541503906	3.1533203125

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2.74762883782387	0.820731628417969	3.2529296875
2.83081014640629	0.850189270019531	3.3525390625
2.91580753400922	0.869773132324219	3.4521484375
2.99937557429075	0.887062316894531	3.5517578125
3.0850253533572	0.908709045410156	3.6513671875
3.16828140057623	0.928511901855469	3.7509765625
3.25209717266262	0.952462097167969	3.8505859375
3.33652505651116	0.972895446777344	3.9501953125
3.42229660600424	0.992052917480469	4.0498046875
3.50673403590918	1.009724609375	4.1494140625
3.59118799678981	1.03581359863281	4.2490234375
3.67439980618656	1.05764538574219	4.3486328125
3.76052432693541	1.07758642578125	4.4482421875
3.84332798421383	1.09806750488281	4.5478515625
3.92816821113229	1.11901965332031	4.6474609375
4.01249108836055	1.14167468261719	4.7470703125
4.09617461264133	1.16564306640625	4.8466796875
4.18149260804057	1.18584008789063	4.9462890625
4.26523759961128	1.20389831542969	5.0458984375
4.3500242754817	1.22496032714844	5.1455078125
4.43476578220725	1.24608178710938	5.2451171875
4.51946957036853	1.26594226074219	5.3447265625
4.60321642458439	1.28672314453125	5.4443359375
4.68637142330408	1.30401440429688	5.5439453125
4.7700172290206	1.32863488769531	5.6435546875
4.85609844326973	1.35138989257813	5.7431640625
4.94027929380536	1.36514733886719	5.8427734375
5.02455700188875	1.38812536621094	5.9423828125
5.10994391515851	1.40971801757813	6.0419921875
5.19421836361289	1.43123400878906	6.1416015625
5.27771934866905	1.45083422851563	6.2412109375
5.36223640665412	1.47341149902344	6.3408203125
5.44657232239842	1.49231640625	6.4404296875
5.53101813420653	1.516279296875	6.5400390625
5.61437057331204	1.53767407226563	6.6396484375
5.6985174305737	1.55549877929688	6.7392578125
5.78321423381567	1.57785192871094	6.8388671875
5.86855644360185	1.59819665527344	6.9384765625
5.95117034390569	1.61809887695313	7.0380859375
6.03611161932349	1.63880053710938	7.1376953125
6.11984077841043	1.65988732910156	7.2373046875
6.20471918955445	1.67602429199219	7.3369140625
6.28929166123271	1.69257482910156	7.4365234375
6.37308834120631	1.71924938964844	7.5361328125
6.4582759514451	1.74399084472656	7.6357421875

Appendix B

6.5415482968092	1.76492041015625	7.7353515625
6.62633683532476	1.783666015625	7.8349609375
6.71056983992457	1.80235021972656	7.9345703125
6.79292622953653	1.8228525390625	8.0341796875
6.88023632392287	1.84494689941406	8.1337890625
6.96490192785859	1.86658129882813	8.2333984375
7.04928580671549	1.88840002441406	8.3330078125
7.13107315823436	1.91070935058594	8.4326171875
7.21766380593181	1.92802099609375	8.5322265625
7.30187445878983	1.94321044921875	8.6318359375
7.3866187594831	1.96384252929688	8.7314453125
7.47017376124859	1.99005932617188	8.8310546875
7.55498604848981	2.00815844726563	8.9306640625
7.63984303921461	2.03468371582031	9.0302734375
7.72416172549129	2.05035913085938	9.1298828125
7.80795002356172	2.06720703125	9.2294921875
7.89150130003691	2.089072265625	9.3291015625
7.97627214342356	2.11326147460938	9.4287109375
8.06059595197439	2.1326328125	9.5283203125
8.14621243625879	2.15264672851563	9.6279296875
8.22891015559435	2.17424658203125	9.7275390625
8.31031054258347	2.19426025390625	9.8271484375
8.39807186275721	2.21325439453125	9.9267578125
8.48175678402185	2.23638549804688	10.0263671875
8.56582075357437	2.25989379882813	10.1259765625
8.65024421364069	2.27905029296875	10.2255859375
8.73488001525402	2.29792553710938	10.3251953125
8.81933141499758	2.31734301757813	10.4248046875
8.90494044870138	2.3413203125	10.5244140625
8.98902025073767	2.36293408203125	10.6240234375
9.07301343977451	2.38003198242188	10.7236328125
9.1572143137455	2.39417993164063	10.8232421875
9.24053974449635	2.41624755859375	10.9228515625
9.32695250958204	2.42750122070313	11.0224609375
9.41077060997486	2.44806127929688	11.1220703125
9.49453096836805	2.4733056640625	11.2216796875
9.57795139402151	2.49762329101563	11.3212890625
9.66327171772718	2.51817456054688	11.4208984375
9.7469761967659	2.55241674804688	11.5205078125
9.83092933893204	2.57089282226563	11.6201171875
9.91499703377485	2.595431640625	11.7197265625
10.0003518164158	2.62444580078125	11.8193359375
10.0844400003552	2.64714770507813	11.9189453125
10.1690413430333	2.67070874023438	12.0185546875
10.2536305785179	2.69123583984375	12.1181640625

Appendix B

10.3375157341361	2.72196459960938	12.2177734375
10.421278886497	2.74059326171875	12.3173828125
10.507014580071	2.76713427734375	12.4169921875
10.5904079973698	2.7886474609375	12.5166015625
10.6751536950469	2.81134716796875	12.6162109375
10.7583869248629	2.83692602539063	12.7158203125
10.8441999182105	2.85780737304688	12.8154296875
10.9274312853813	2.879771484375	12.9150390625
11.013307608664	2.89698876953125	13.0146484375
11.0955638810992	2.91248461914063	13.1142578125
11.1815901473165	2.9418798828125	13.2138671875
11.2660452723503	2.96749877929688	13.3134765625
11.3493846729398	2.98902514648438	13.4130859375
11.4350486546755	3.01032592773438	13.5126953125
11.5191144868732	3.03548681640625	13.6123046875
11.6033433005214	3.05533081054688	13.7119140625
11.6869229823351	3.07503637695313	13.8115234375
11.7713883519173	3.10718725585938	13.9111328125
11.8570318445563	3.1270537109375	14.0107421875
11.9403786957264	3.15125048828125	14.1103515625
12.0240952819586	3.17002319335938	14.2099609375
12.108001857996	3.1987958984375	14.3095703125
12.1933510527015	3.22069409179688	14.4091796875
12.2774634510279	3.24402661132813	14.5087890625
12.3577201738954	3.27033251953125	14.6083984375
12.4464370310307	3.2852392578125	14.7080078125
12.5296944752336	3.30917846679688	14.8076171875
12.6152206212282	3.32822998046875	14.9072265625
12.6987965777516	3.35984155273438	15.0068359375
12.783432379365	3.37643505859375	15.1064453125
12.8668220713735	3.39561376953125	15.2060546875
12.9503346979618	3.42206665039063	15.3056640625
13.0365900695324	3.44300390625	15.4052734375
13.1214382126927	3.47217797851563	15.5048828125
13.2062193006277	3.4963798828125	15.6044921875
13.2902273908257	3.514876953125	15.7041015625
13.3747709915042	3.53384228515625	15.8037109375
13.4590733796358	3.56313110351563	15.9033203125
13.5426446795464	3.59066723632813	16.0029296875
13.6273512616754	3.60895141601563	16.1025390625
13.7095069512725	3.63207592773438	16.2021484375
13.7939499691129	3.66367211914063	16.3017578125
13.8790598139167	3.68139086914063	16.4013671875
13.9649407938123	3.69776831054688	16.5009765625
14.0480203554034	3.72244458007813	16.6005859375

Appendix B

14.1333062201738	3.74324169921875	16.7001953125
14.2177119851112	3.76114404296875	16.7998046875
14.3014285713434	3.79104614257813	16.8994140625
14.3857877701521	3.8154111328125	16.9990234375
14.4697045907378	3.832171875	17.0986328125
14.5542714744806	3.85505615234375	17.1982421875
14.6376984193921	3.88304541015625	17.2978515625
14.7234918549657	3.91708227539063	17.3974609375
14.8084554821253	3.92852270507813	17.4970703125
14.8905562236905	3.95047924804688	17.5966796875
14.9758094921708	3.987787109375	17.6962890625
15.0608560070395	4.00419482421875	17.7958984375
15.145349316299	4.01760571289063	17.8955078125
15.2286579832435	4.05051049804688	17.9951171875
15.3134260326624	4.0713466796875	18.0947265625
15.399094671011	4.1007333984375	18.1943359375
15.4825942590833	4.11345556640625	18.2939453125
15.5671248212457	4.13568505859375	18.3935546875
15.6502779573202	4.17298681640625	18.4931640625
15.7356020063162	4.18682275390625	18.5927734375
15.8201847225428	4.20229150390625	18.6923828125
15.9034859389067	4.233072265625	18.7919921875
15.9893967211246	4.25365478515625	18.8916015625
16.0715952515602	4.2869638671875	18.9912109375
16.1558575928211	4.307138671875	19.0908203125
16.2401143461466	4.31814599609375	19.1904296875
16.324371099472	4.34209033203125	19.2900390625
16.4110641926527	4.3638251953125	19.3896484375
16.4939537644386	4.40239208984375	19.4892578125
16.5781937539577	4.41817822265625	19.5888671875
16.6624616831541	4.4207197265625	19.6884765625
16.7477559298277	4.46379931640625	19.7880859375
16.8312750756741	4.496583984375	19.8876953125
16.9167146086693	4.504916015625	19.9873046875
17.0013718307018	4.52232470703125	20.0869140625
17.0847121626139	4.548240234375	20.1865234375
17.169201746583	4.58499609375	20.2861328125
17.2529593110085	4.6109248046875	20.3857421875
17.3372812569141	4.61962109375	20.4853515625
17.4208208918571	4.652880859375	20.5849609375
17.5059214234352	4.66556982421875	20.6845703125
17.591604962945	4.6918388671875	20.7841796875
17.6764596253634	4.72010595703125	20.8837890625
17.755750566721	4.73042822265625	20.9833984375
17.8433060646057	4.7626484375	21.0830078125

Appendix B

17.9277714341879	4.78627783203125	21.1826171875
18.0118400603533	4.802123046875	21.2822265625
18.0961452424526	4.840306640625	21.3818359375
18.1826669722795	4.8519755859375	21.4814453125
18.265375867486	4.8656796875	21.5810546875
18.3504074811935	4.90731591796875	21.6806640625
18.434064462781	4.9337822265625	21.7802734375
18.5186024755239	4.9573564453125	21.8798828125
18.6029616743326	4.97050927734375	21.9794921875
18.6881814152002	4.982642578125	22.0791015625
18.7716446816921	5.03114404296875	22.1787109375
18.8555177301168	5.06139404296875	22.2783203125
18.9418029040098	5.06786328125	22.3779296875
19.0248265862465	5.09747607421875	22.4775390625
19.1084798425436	5.114166015625	22.5771484375
19.1932860761881	5.14469140625	22.6767578125
19.2786976695061	5.1556337890625	22.7763671875
19.3617455661297	5.17515185546875	22.8759765625
19.4451548159122	5.21552978515625	22.9755859375
19.5314511656761	5.23218115234375	23.0751953125
19.6156464517117	5.26702001953125	23.1748046875
19.6986440569162	5.29973974609375	23.2744140625
19.7827946394682	5.30227294921875	23.3740234375
19.8674574494362	5.334650390625	23.4736328125
19.9525225907564	5.37210595703125	23.5732421875
20.0376622378826	5.3986845703125	23.6728515625
20.1197024434805	5.41351806640625	23.7724609375
20.2049054205418	5.42598876953125	23.8720703125
20.2889703214169	5.46965673828125	23.9716796875
20.3719027340412	5.4939931640625	24.0712890625
20.4586070030928	5.50970849609375	24.1708984375
20.5423384904861	5.5415283203125	24.2705078125
20.6248722970486	5.56670849609375	24.3701171875
20.7103341817856	5.58180859375	24.4697265625
20.7964964210987	5.60108935546875	24.5693359375
20.8797808736563	5.6233427734375	24.6689453125
20.9648422896862	5.6459541015625	24.7685546875
21.0488215088844	5.6785126953125	24.8681640625
21.1335439234972	5.696505859375	24.9677734375
21.2184060364962	5.71736572265625	25.0673828125
21.3008988648653	5.74028271484375	25.1669921875
21.386781707406	5.76828466796875	25.2666015625
21.4708801358938	5.8063369140625	25.3662109375
21.5518120676279	5.8211005859375	25.4658203125
21.639870479703	5.84383154296875	25.5654296875

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21.7208992689848	5.87766650390625	25.6650390625
21.8084696680307	5.8907099609375	25.7646484375
21.8923632055521	5.92347509765625	25.8642578125
21.9769645482302	5.9624130859375	25.9638671875
22.0604538917542	5.97839599609375	26.0634765625
22.1461895853281	5.99676416015625	26.1630859375
22.2294796258211	6.01484033203125	26.2626953125
22.3110280930996	6.02522607421875	26.3623046875
22.3978627473116	6.067298828125	26.4619140625
22.4818903952837	6.09373974609375	26.5615234375
22.5673355162144	6.1015	26.6611328125
22.6514711976051	6.12742822265625	26.7607421875
22.7348357439041	6.15202099609375	26.8603515625
22.819684818387	6.16847119140625	26.9599609375
22.9031760245562	6.19503466796875	27.0595703125
22.9898057878017	6.2209873046875	27.1591796875
23.0724345892668	6.2462099609375	27.2587890625
23.1572594493628	6.27098828125	27.3583984375
23.2409257441759	6.28946337890625	27.4580078125
23.326151072979	6.339544921875	27.5576171875
23.4099850058556	6.35064013671875	27.6572265625
23.4933625906706	6.36310595703125	27.7568359375
23.5787499696016	6.4080400390625	27.8564453125
23.6628092825413	6.42727099609375	27.9560546875
23.7473286688328	6.44770751953125	28.0556640625
23.8312780857086	6.4718935546875	28.1552734375
23.915521800518	6.4959541015625	28.2548828125
24.0004733204842	6.52777490234375	28.3544921875
24.0844115614891	6.548056640625	28.4541015625
24.1684801876545	6.5507890625	28.5537109375
24.2525096982718	6.5865791015625	28.6533203125
24.3381094187498	6.6128623046875	28.7529296875
24.4222041219473	6.6291064453125	28.8525390625
24.5074965059757	6.66367041015625	28.9521484375
24.5914589613676	6.68148046875	29.0517578125
24.6757380664349	6.6935126953125	29.1513671875
24.7600711882114	6.7348662109375	29.2509765625
24.8450338840485	6.75517431640625	29.3505859375
24.9271057546139	6.771501953125	29.4501953125
25.0137839466333	6.8059892578125	29.5498046875
25.096345692873	6.8357041015625	29.6494140625
25.1827221363783	6.855865234375	29.7490234375
25.264136493206	6.877814453125	29.8486328125
25.3496151417494	6.908751953125	29.9482421875
25.4329927265644	6.9234755859375	30.0478515625

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25.5172532051802	6.94528564453125	30.1474609375
25.6020613014698	6.9600625	30.2470703125
25.6869457662106	6.98982861328125	30.3466796875
25.7716737687588	7.03137451171875	30.4462890625
25.8547980338335	7.04532763671875	30.5458984375
25.9405486285686	7.05455615234375	30.6455078125
26.0234940797091	7.09845751953125	30.7451171875
26.1074732989073	7.10988623046875	30.8447265625
26.1929649859667	7.12399853515625	30.9443359375
26.276171207428	7.170927734375	31.0439453125
26.3617560267448	7.19170166015625	31.1435546875
26.4460220932961	7.1990908203125	31.2431640625
26.5309978276491	7.228373046875	31.3427734375
26.6153514385223	7.2584775390625	31.4423828125
26.6996640712023	7.29029443359375	31.5419921875
26.7835333943367	7.31210693359375	31.6416015625
26.868661865592	7.34630712890625	31.7412109375
26.9518364220858	7.3589521484375	31.8408203125
27.037438005209	7.37923681640625	31.9404296875
27.1197333931923	7.38876953125	32.0400390625
27.2056013345718	7.4244541015625	32.1396484375
27.28739567101	7.462154296875	32.2392578125
27.3749865591526	7.4784912109375	32.3388671875
27.4594761431217	7.49482421875	32.4384765625
27.5430157780647	7.52591259765625	32.5380859375
27.6261996477842	7.54353759765625	32.6376953125
27.7093667536974	7.55187939453125	32.7373046875
27.7958195656538	7.58980810546875	32.8369140625
27.8801675885916	7.6273349609375	32.9365234375
27.96327508986	7.635783203125	33.0361328125
28.0490033328533	7.66173193359375	33.1357421875
28.1329602003098	7.6894033203125	33.2353515625
28.2188057899475	7.70510693359375	33.3349609375
28.3016972243786	7.732791015625	33.4345703125
28.3868778496981	7.75805322265625	33.5341796875
28.4673720598221	7.7891796875	33.6337890625
28.5549201071262	7.79981201171875	33.7333984375
28.6383386701345	7.81850244140625	33.8330078125
28.7232473492622	7.86466357421875	33.9326171875
28.8075562566519	7.87790576171875	34.0322265625
28.8883857429028	7.90086328125	34.1318359375
28.9750061929226	7.92722705078125	34.2314453125
29.0606934577227	7.94826513671875	34.3310546875
29.1461851447821	7.96711181640625	34.4306640625
29.2301252484322	7.9907587890625	34.5302734375

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29.3131601065397	8.01394580078125	34.6298828125
29.3978117406368	8.0331708984375	34.7294921875
29.4828452169895	8.07050537109375	34.8291015625
29.566328972578	8.0771435546875	34.9287109375
29.6496208757162	8.09503466796875	35.0283203125
29.7345984727144	8.1243125	35.1279296875
29.8192966729403	8.14659619140625	35.2275390625
29.9029070883989	8.1776875	35.3271484375
29.9887303262949	8.2110859375	35.4267578125
30.0724003463984	8.211203125	35.5263671875
30.1570612937212	8.2327822265625	35.6259765625
30.2409511059523	8.2723681640625	35.7255859375
30.3253289312124	8.2965244140625	35.8251953125
30.4110422730446	8.3026259765625	35.9248046875
30.4952394217253	8.320974609375	36.0244140625
30.5781122297049	8.361234375	36.1240234375
30.6626036763191	8.380373046875	36.2236328125
30.7481102645397	8.393578125	36.3232421875
30.8317989110947	8.39910546875	36.4228515625
30.9163443744183	8.4223359375	36.5224609375
31.000092625618	8.4592998046875	36.6220703125
31.0850590467453	8.475291015625	36.7216796875
31.1662703752518	8.473736328125	36.8212890625
31.2541052699089	8.4949033203125	36.9208984375

APPENDIX C – MTS Landmark Servo-hydraulic Test System Calibration Documentation

IMP Calibration Services (Pty) Ltd HEAD OFFICE Tel: +27 (11) 916 5000 Fax: +27 (11) 914 4509 DUBAN • Tel: +27 (31) 764 2821 Fax: +27 (31) 764 4288 WESTERN CAPE Tel: +27 (21) 852 6133 Fax: +27 (21) 852 6129 EASTERN CAPE • Tel: +27 (41) 364 0159 Fax: +27 (41) 364 0231 FREE STATE • Tel +27 (18) 293 3333 Fax: +27 (18) 293 3333 POSTAL ADDRESS PO Box 1110, Boksburg, 1460, South Africa Email: info@imp.co.za • Website: www.imp.co.za	 
Company Reg No: 2003/024009/07 VAT Reg. No: 4230138614 ISO/IEC17025	
ON SITE CALIBRATION CERTIFICATE FOR EXTENSION MEASUREMENT SANAS ACCREDITED LABORATORY FOR FORCE - NO: 816	
CALIBRATION CERTIFICATE NO: NHQ-18595 A	
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COMPANY: CONTACT PERSON: PHYSICAL ADDRESS: LOCATION OF CALIBRATION: ITEM CALIBRATED: DATE CALIBRATED: CALIBRATED BY: MANUFACTURER: MODEL NO: EXTENSION RANGE: UNIVERSAL TESTER SERIAL NO: RANGE CALIBRATED: MACHINE RESOLUTION: TEMPERATURE: STANDARD USED: TRACEABILITY NO: PROCEDURE FOLLOWED: CALCULATED UNCERTAINTY:	NWU MR. SAREL NAUDE STEVE BIKO STR, POTCHEFSTROOM LABORATORY DYNAMIC TESTER 2012/06/18 F. CLASSEN / M. BARNARD MTS LANDMARK 0 TO 75mm & 0 TO -75mm 10359599 0 TO 75mm & 0 TO -75mm 0.01mm 18°C / 18°C / 18°C NEWALL DP700 (HA022CM/MG88KM) E0413 IMP 1042 ±0.05mm
The accuracy's of the standards used are, by way of regular inter-comparison, traceable to the National Standards, within the limitations of the services available. The values in the certificate are correct at the time of calibration. Subsequently the accuracy will depend on such factors as the care exercised in handling as well as the use of instruments, equipment, and the frequency of use. It is recommended that recalibration be undertaken at an interval that will ensure that the instrument remains within the desired limits. IMP Calibration services cannot be held responsible for values changes. No long term stability checks were carried out. A SANAS certificate automatically expires if the instrument is repaired or if the seal is broken.	
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DIRECTORS: B.F. HOHENSTEIN N.D. Eng Met (Phys.), A.L. HOHENSTEIN, Dr R.H. BUSH M.B. B.Ch.D.C.H (S.A.) MBA(cum laude), W. DELPORT (SERVICE MANAGER), MJS MATTHYSEN (NDCA)	

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Company Reg No: 2003/024009/07		VAT Reg. No: 4230138614 ISO/IEC17025

**ON SITE CALIBRATION CERTIFICATE
OF A DYNAMIC TESTER**

SANAS ACCREDITED LABORATORY FOR FORCE - NO: 816

CALIBRATION CERTIFICATE NO: NHQ-18595 B

This certificate is issued in accordance with the conditions of the South African National Accredited System. It is a correct record of the measurements made. Copyright of this certificate is owned by the issuing laboratory. This certificate may not be reproduced other than in full, except with the prior written approval of IMP Calibration Services (Pty) Ltd.

COMPANY: CONTACT PERSON: PHYSICAL ADDRESS: LOCATION OF CALIBRATION: DATE CALIBRATED: CALIBRATED BY: ITEM CALIBRATED: MANUFACTURER: MODEL NO: LOADCELL SERIAL NO: UNIVERSAL TESTER SERIAL NO: CAPACITY: RESOLUTION: TEMPERATURE: RANGE CALIBRATED: STANDARD USED: TRACEABILITY NO: PROCEDURE FOLLOWED: CALCULATED UNCERTAINTY:	NWU MR. SAREL NAUDE STEVE BIKO STR, POTCHEFSTROOM LABORATORY 2012/06/18 F. CLASSEN / M. BARNARD DYNAMIC TESTER MTS LANDMARK 400803 10359599 100kN 0.01kN 18 °C / 18 °C / 18 °C 0 TO 100kN 100kN LOADCELL (115696/2498) IHQ-08409 B IMP 1001 ±0.3% OF READING
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The accuracy's of the standards used are, by way of regular inter-comparison, traceable to the National Standards, within the limitations of the services available. The values in the certificate are correct at the time of calibration. Subsequently the accuracy will depend on such factors as the care exercised in handling as well as the use of instruments, equipment, and the frequency of use. IMP Calibration services cannot be held responsible for values changing. No long term stability checks were carried out. A SANAS certificate automatically expires if the instrument is repaired or if the seal is broken.

- Page 1 of 3 -

DIRECTORS: B.F. HOHENSTEIN N.D. Eng Met (Phys.), A.L. HOHENSTEIN, Dr R.H. BUSH M.B. B.Ch.D.C.H (S.A.) MBA(cum laude), W. DELPORT (SERVICE MANAGER), MJS MATTHYSEN (NDCA)