Structural Design Project

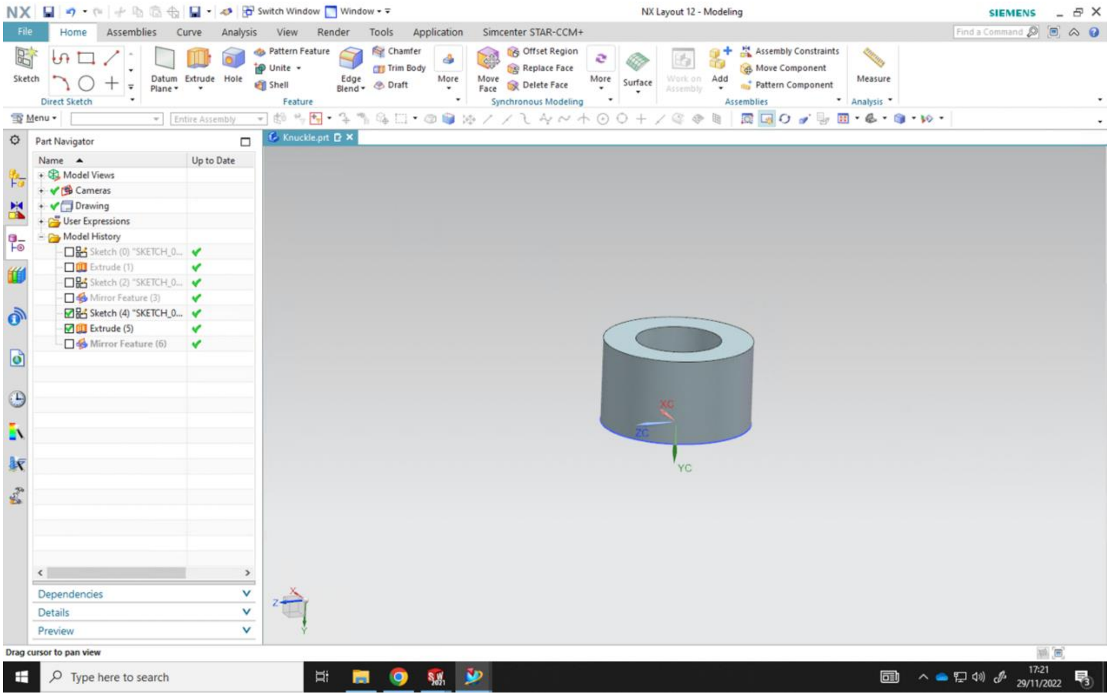
COURSEWORK 1

GROUP 30

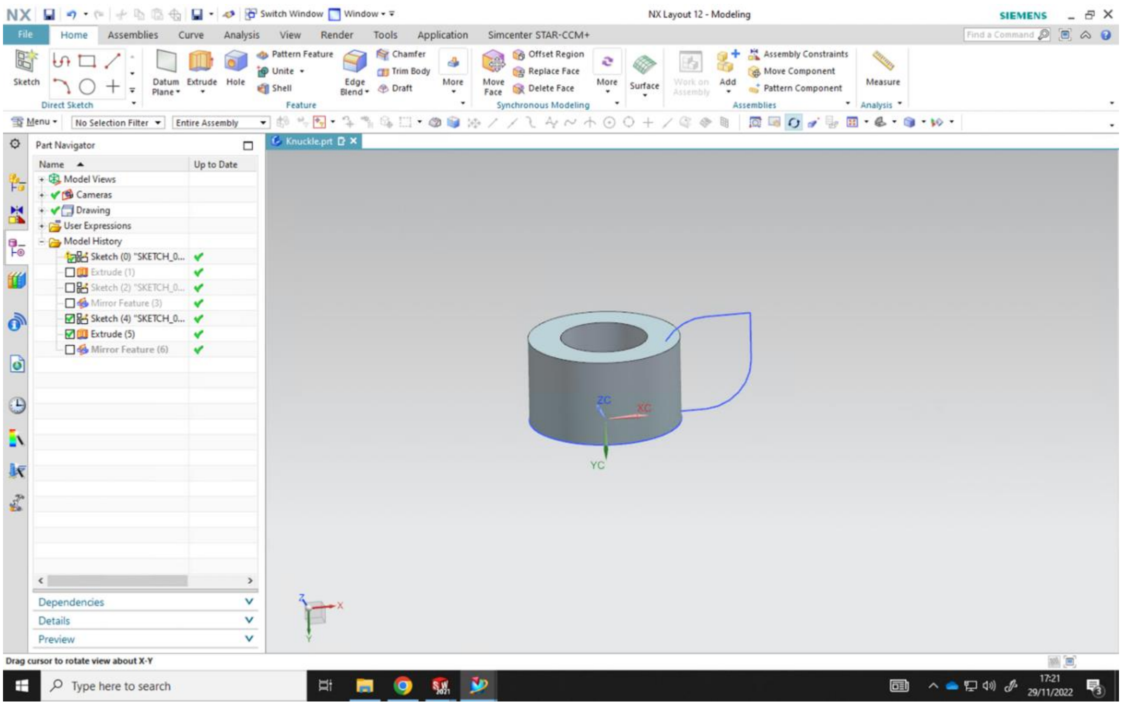
Section 1

In section 1 the construction of the double wishbone assembly is displayed, and structural analysis is carried out on a selection of the parts in section 2. This suspension system is commonly used in racing. Section 3 consists of a hollow box model to simulate the front wing of a formula one race car.

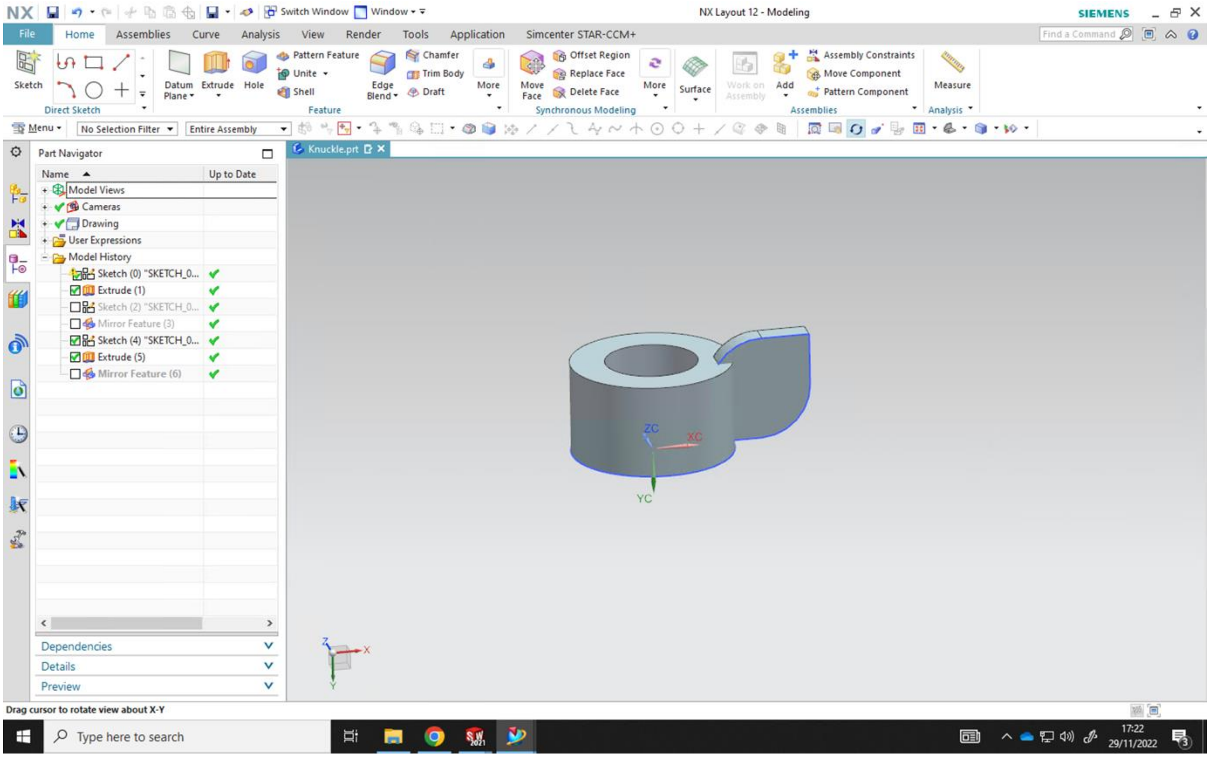
Suspension Knuckle



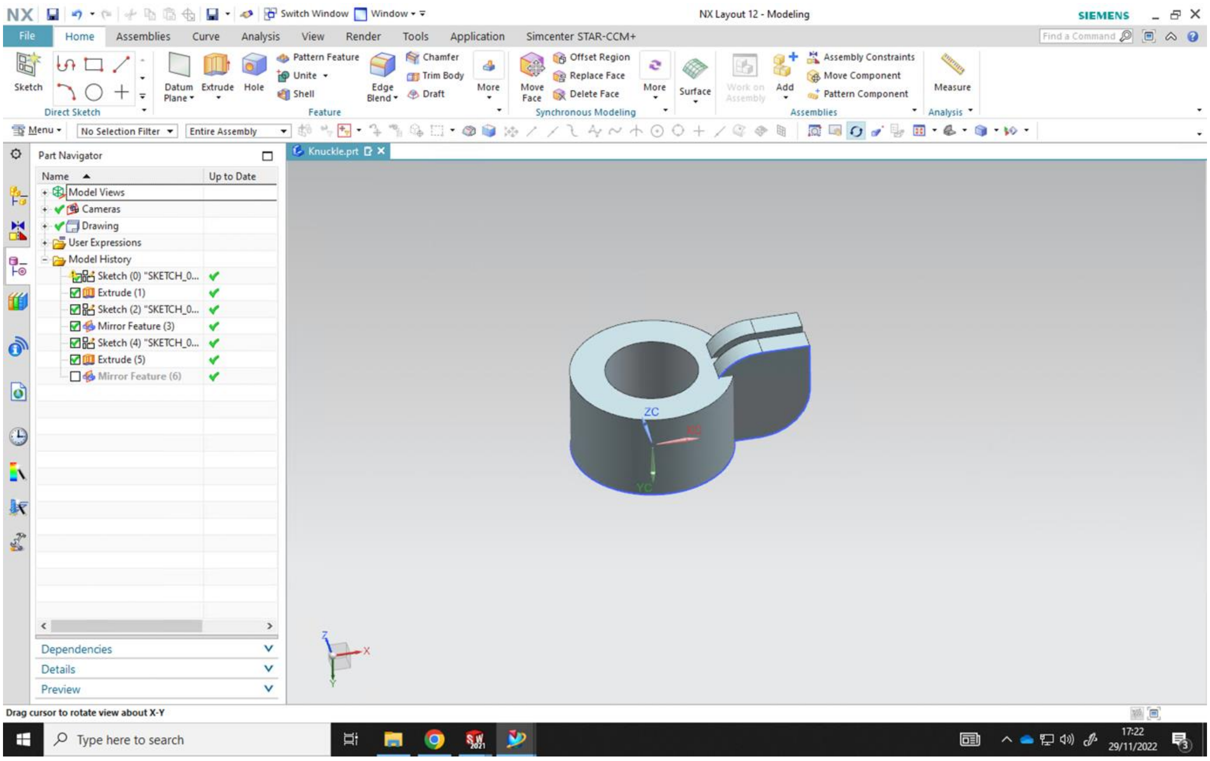
- Started by extruding a hollowed cylinder, with inner diameter of 40mm and outer diameter of 70mm, by a distance of 40mm.



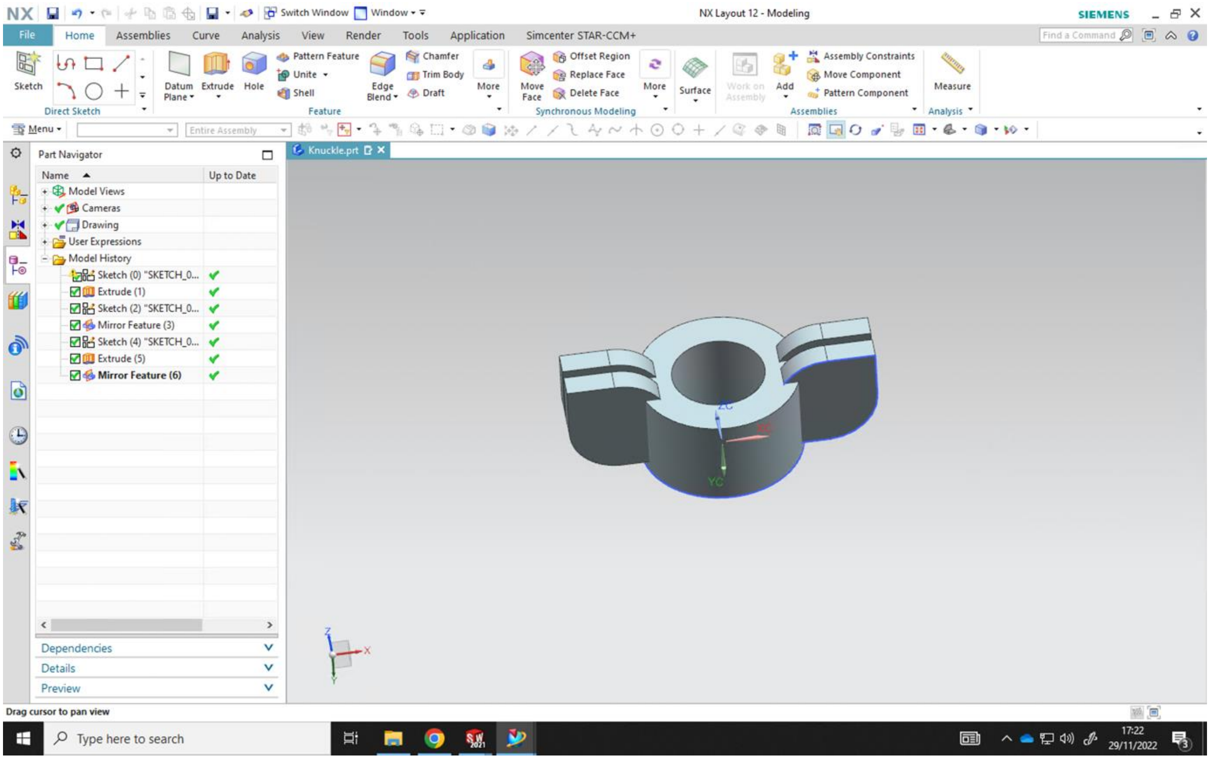
- Sketched a connectinng arm using a combination of the circle and line tool. Circle has a diameter of 44mm.



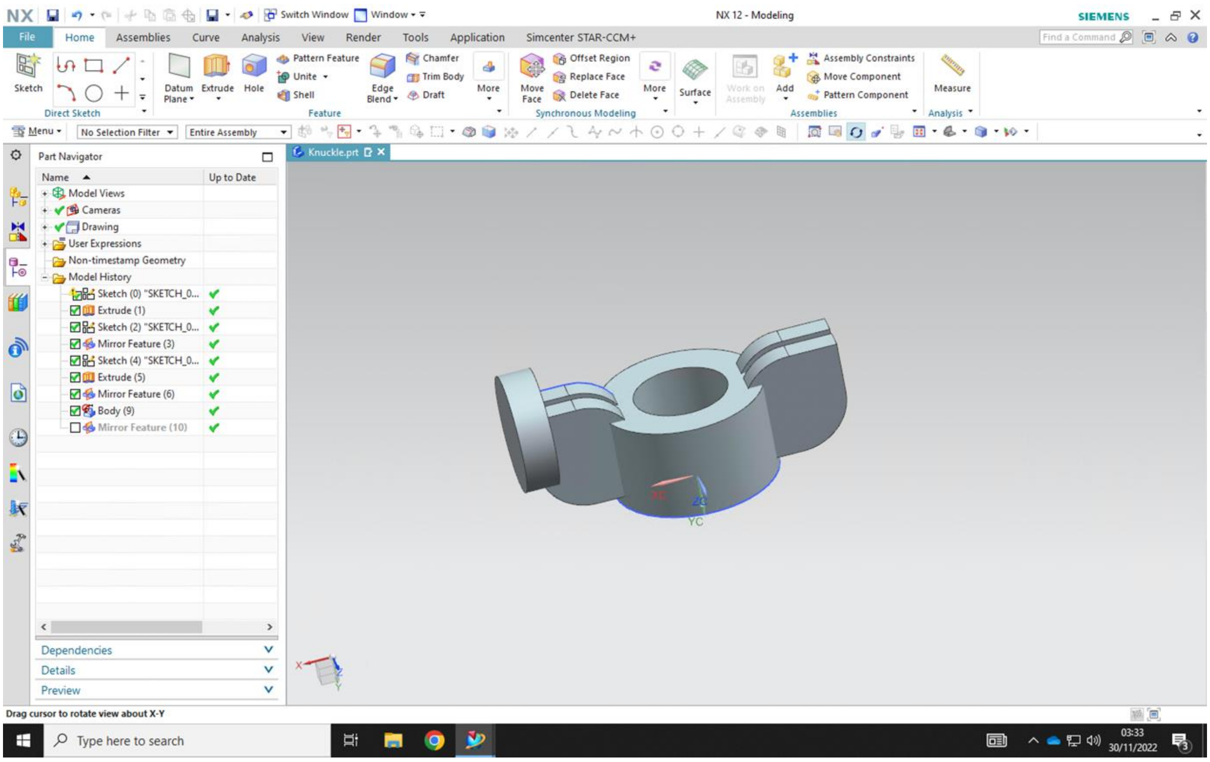
- Extruded this sketch by a distance of 10mm and shifed it in the Z direction by 2mm



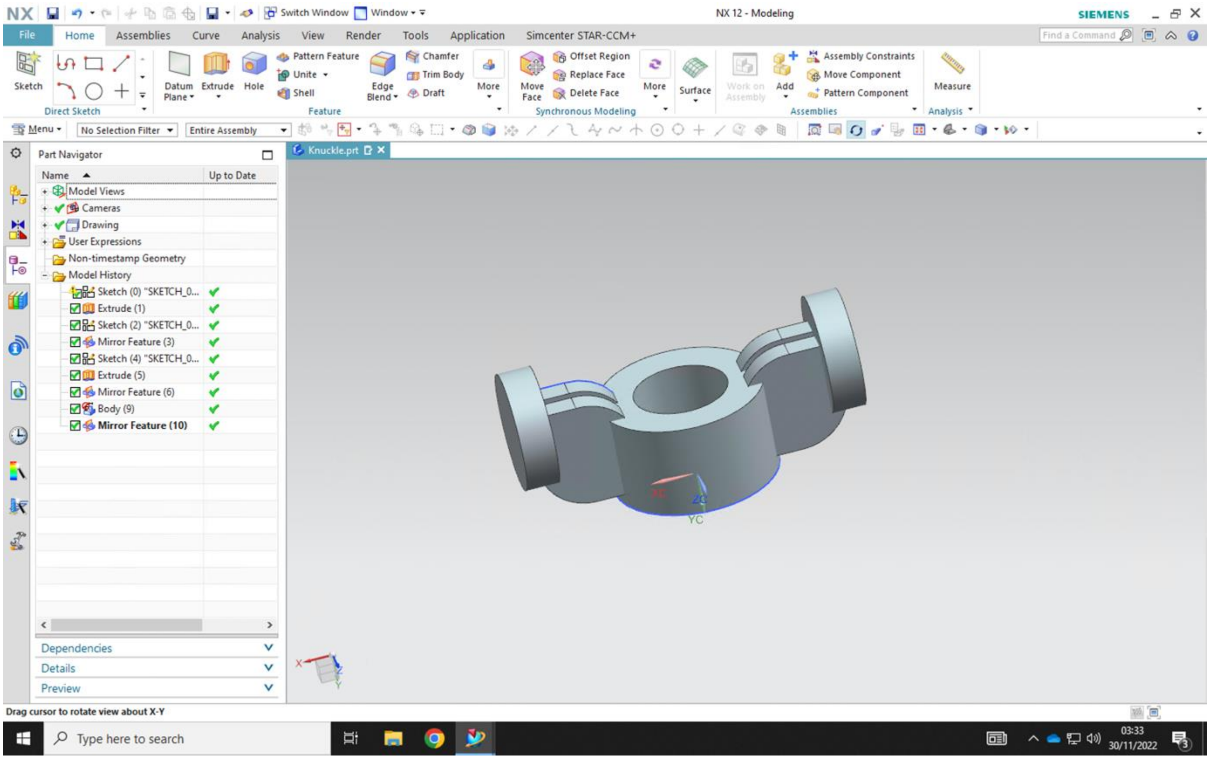
- Mirrored this extrusion at X=0 at the X plane. This provided increased strength to the part.



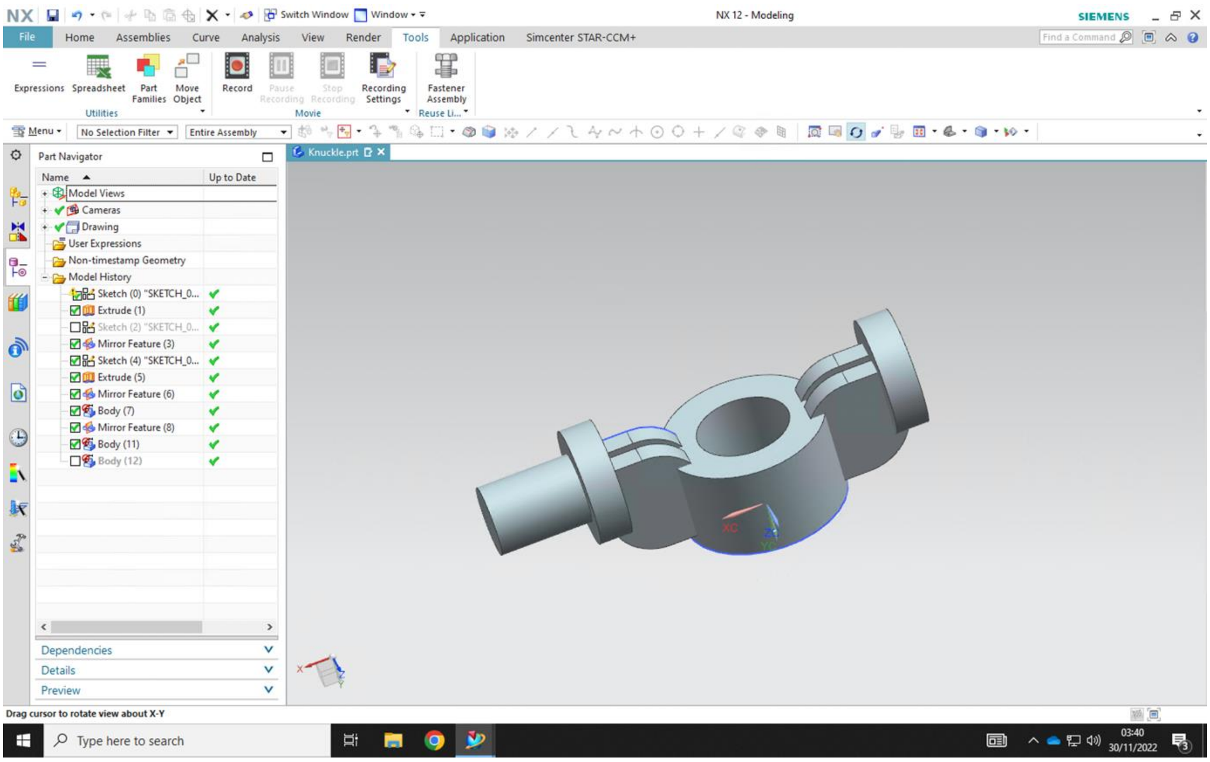
- Mirrored these again at Z=0 in the Z plane



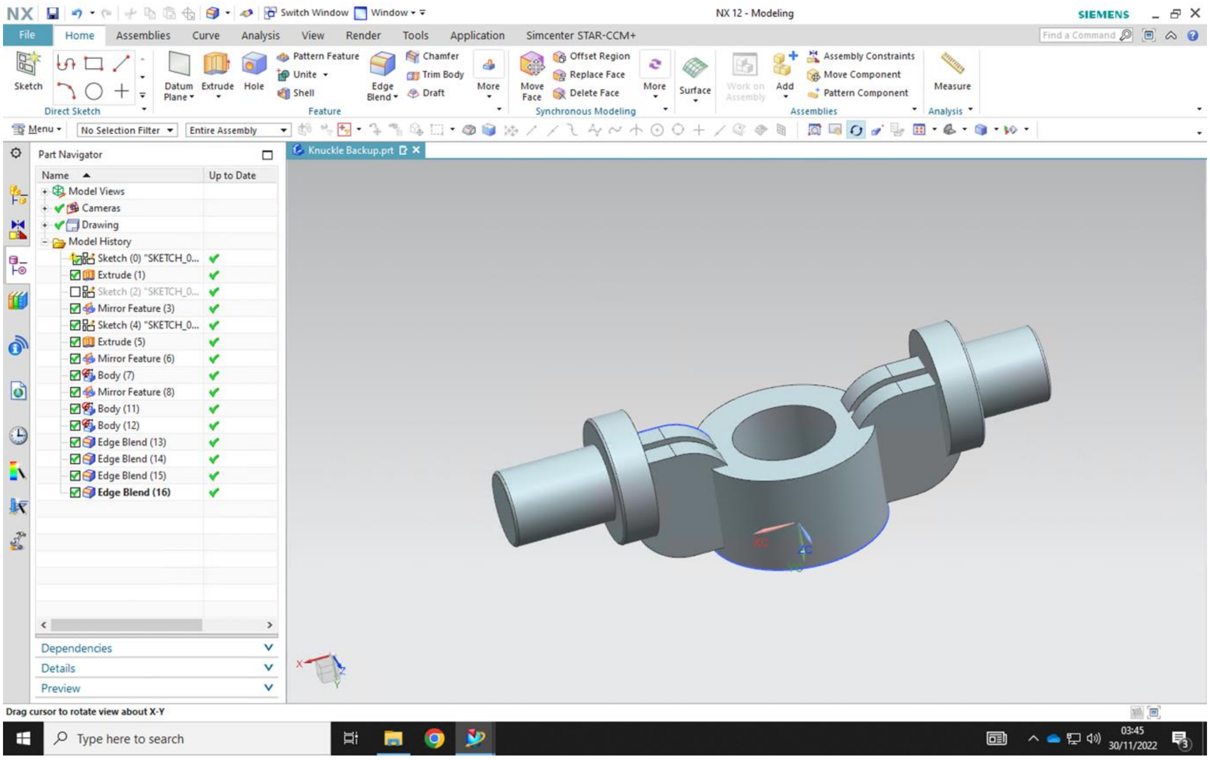
- Sketched a cylinder of diameter 50mm and extruded by 14mm in the X plane



- Mirrored this at Z=0 in the Z plane. These will be our stoppers for the two arms.

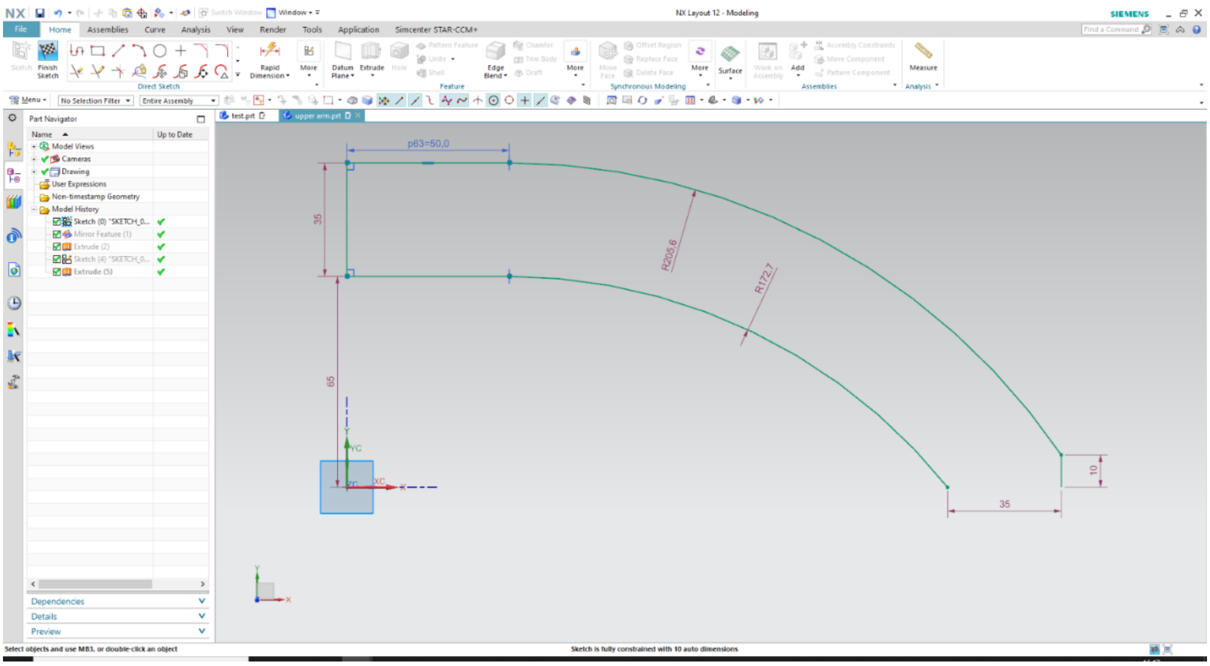


- Sketched and extruded a second cylinder to house the upper arm with a diameter of 30mm and an extrusion distance of 40mm

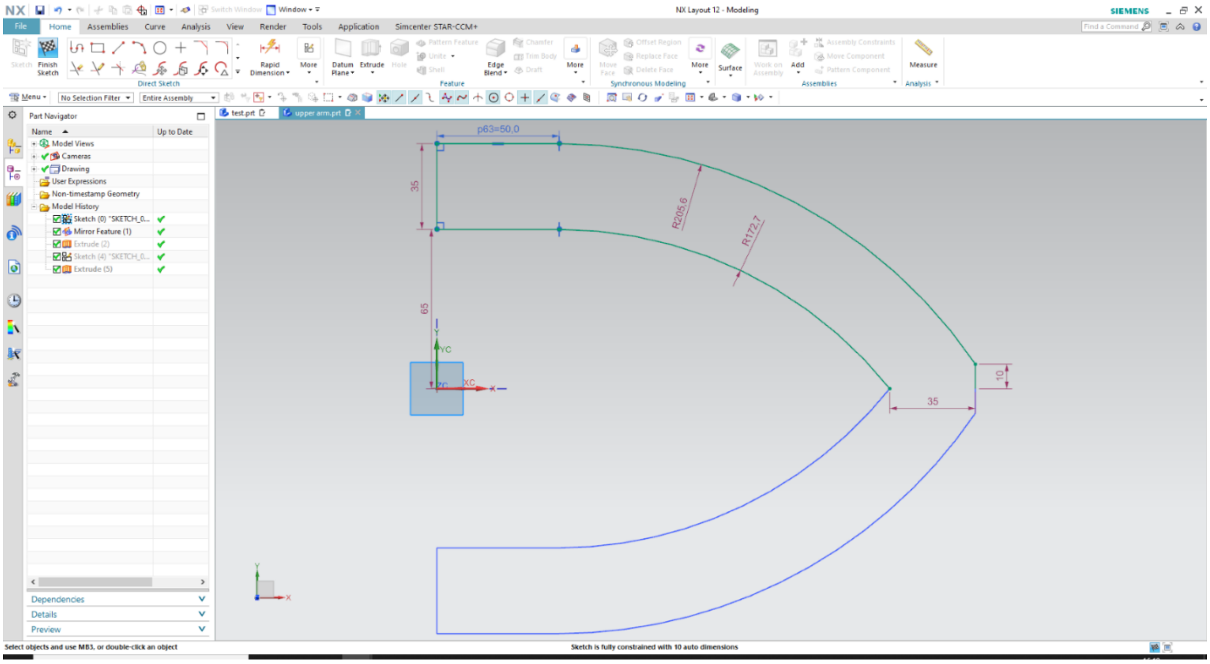


- Repeated the previous step for the lower arm but instead for an extrusion of 30mm as the lower arm had a smaller thickness at the connection. Part is finalised by using the edge blend tool to add 1mm fillets for the cylindrical edges at both ends.

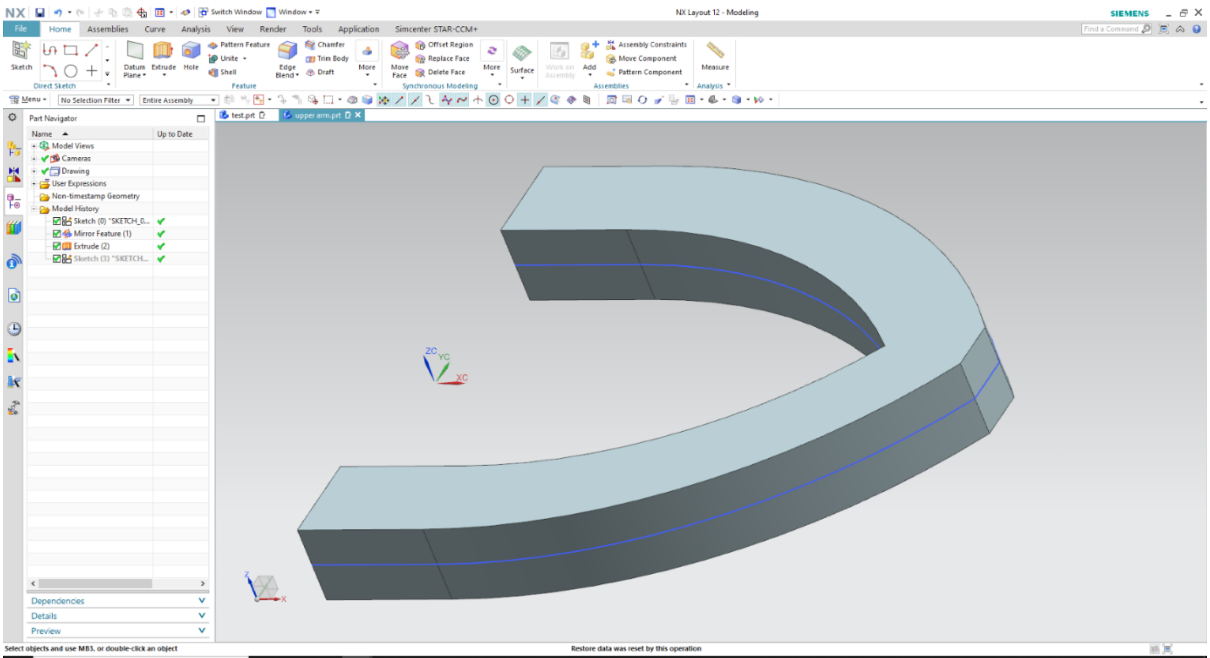
Upper Arm



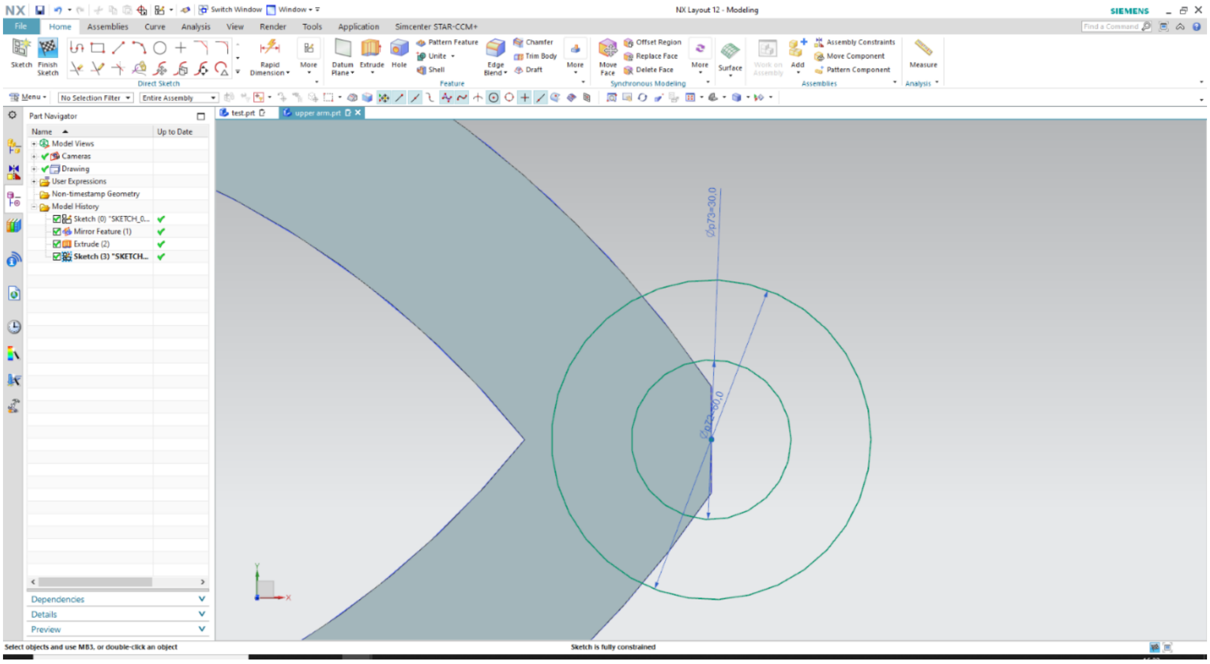
- Starting by creating the upper arm of the double wishbone suspension with a sketch of one half of the suspension arm using the line and curve tools as well as the trim tool for any unwanted overhanging sketches. The half drawn is 220mm long and 100mm wide.



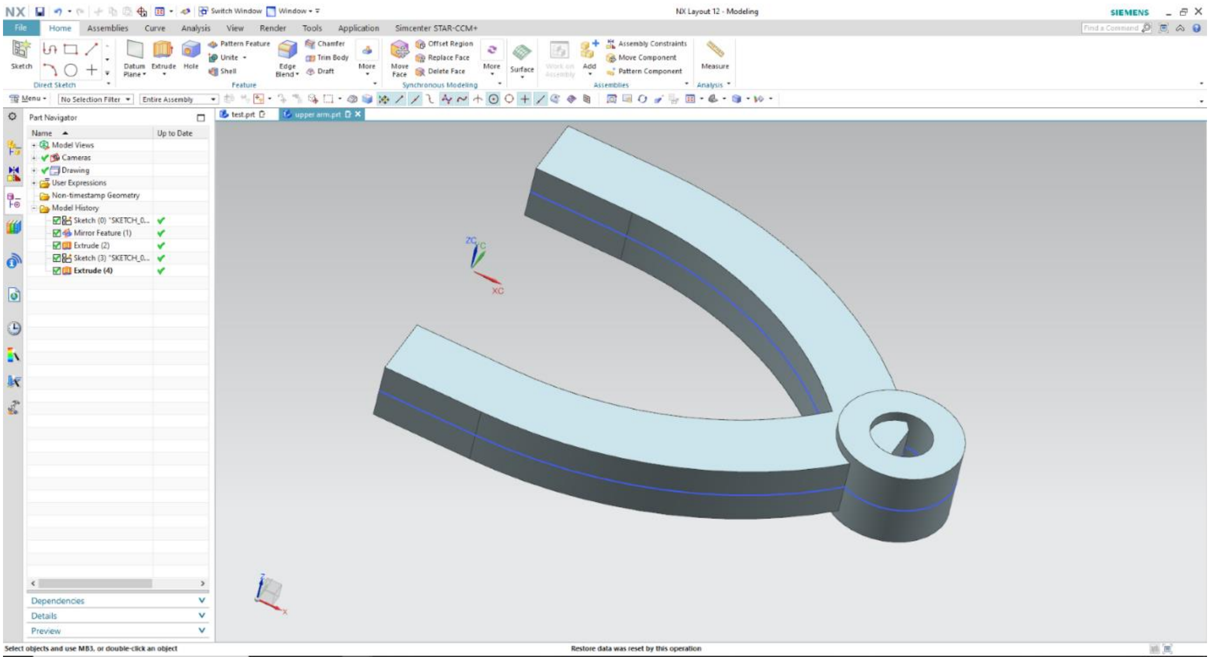
- We completed the basic shape of the upper arm and kept the dimensions for both halves identical with the use of the mirror feature, mirroring at y=0 on the y-plane.



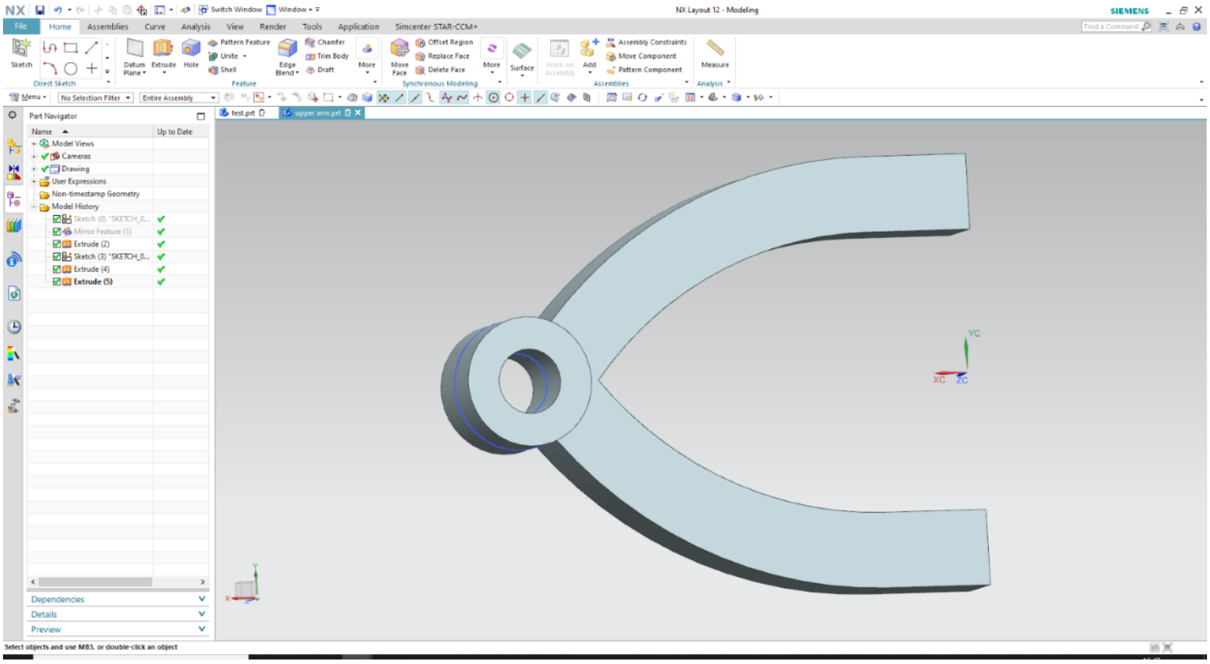
- We extruded the sketch to form the basic model that we can build on and this had a thickness of 30mm.



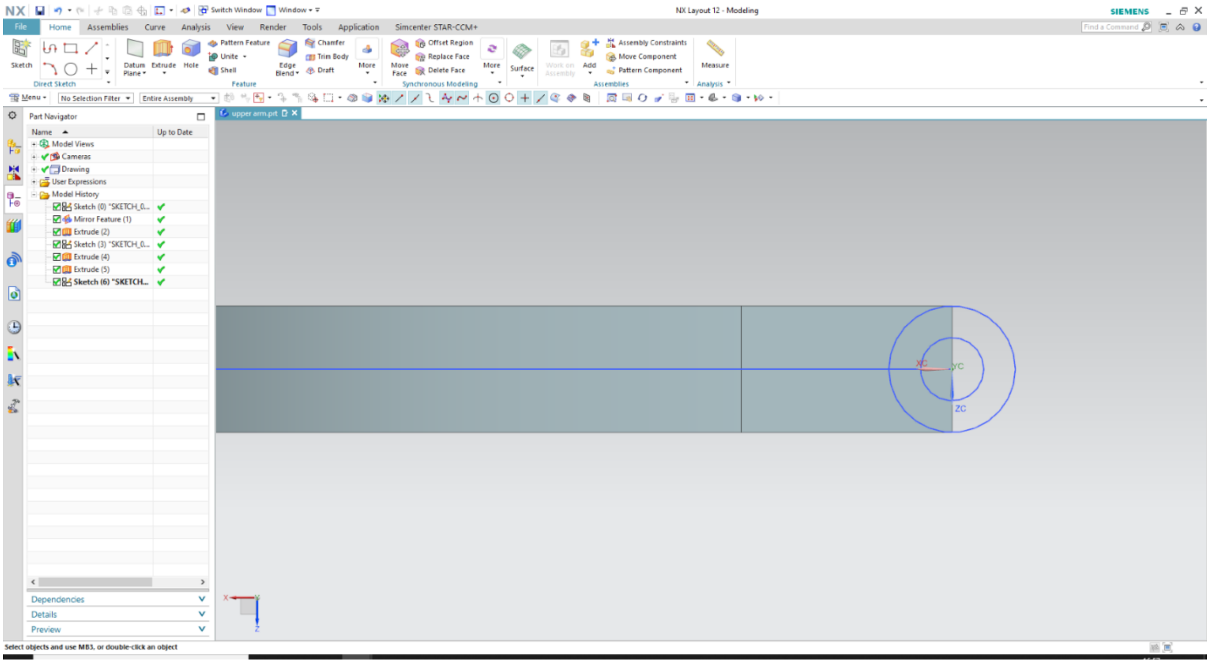
- By drawing a new sketch on the surface of the existing part created, the circle tools were used to create two circles with diameters of 60mm and 30mm.



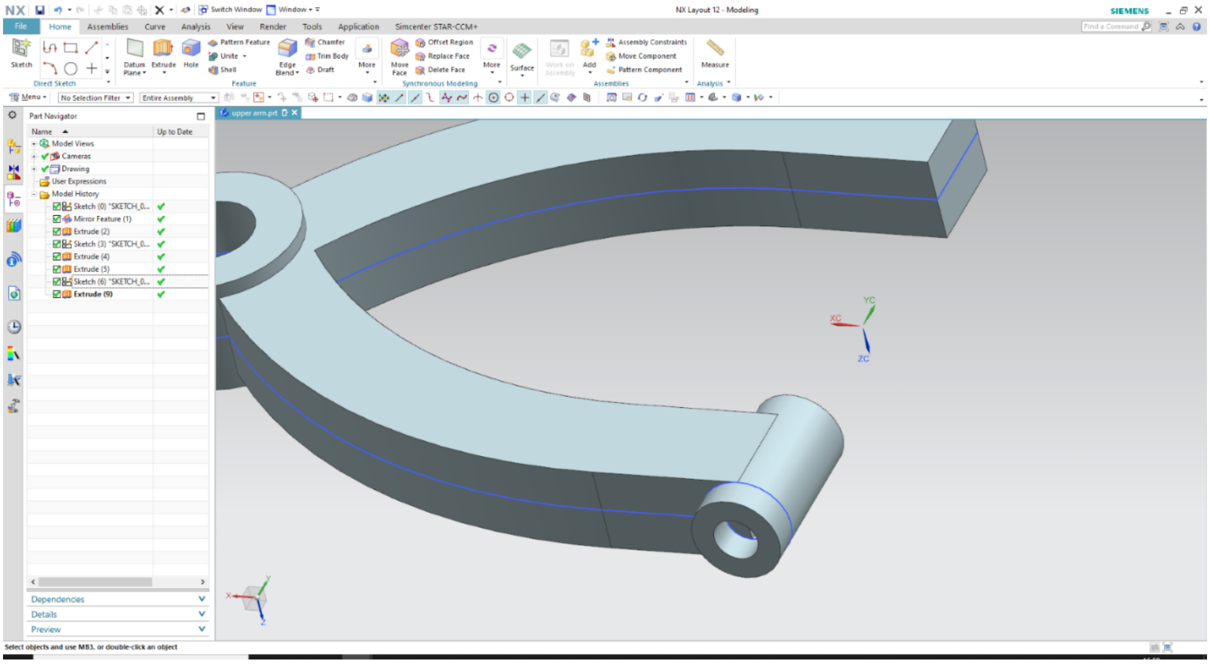
- The circle sketch was extruded to a width of 40mm, 5mm further than the extrusion for the main body of the upper arm on either side.



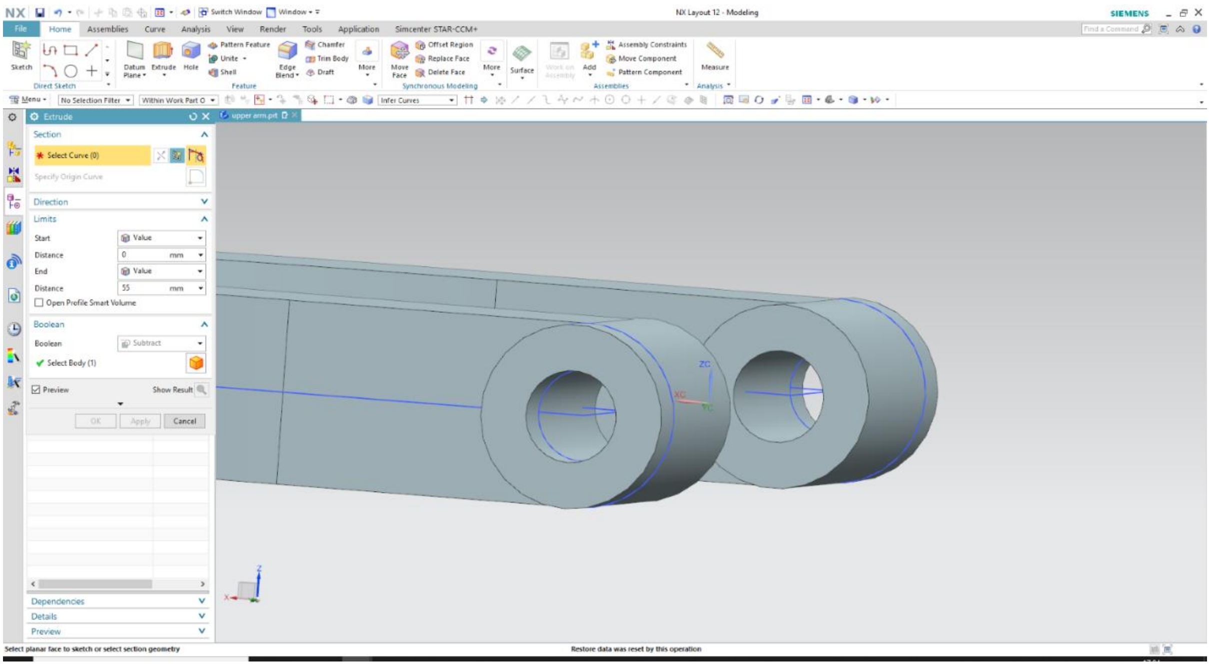
- The subtract feature of the extrude tool was then used to create the housing for the connection to the knuckle of the suspension arm by removing the piece of the arm in the centre of the hole.



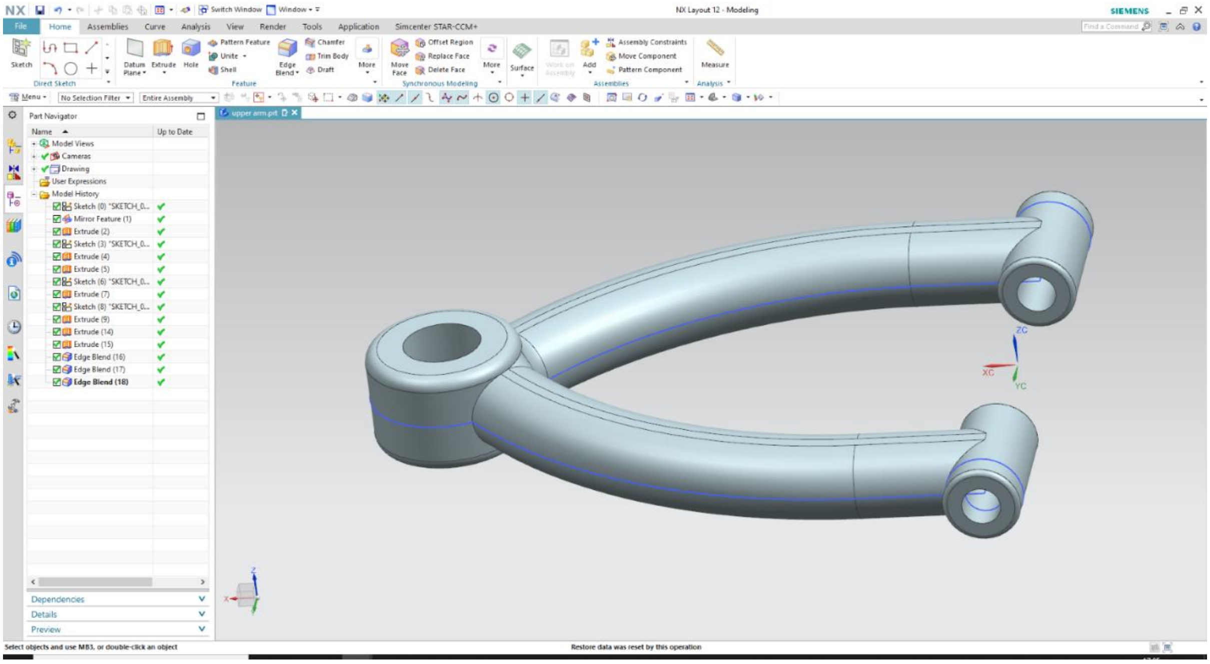
- Now choosing the side of the arm to start the sketch, another two concentric circles of diameters 30mm and 15mm were drawn at the centre of the arm so that they did not protrude beyond the arm.



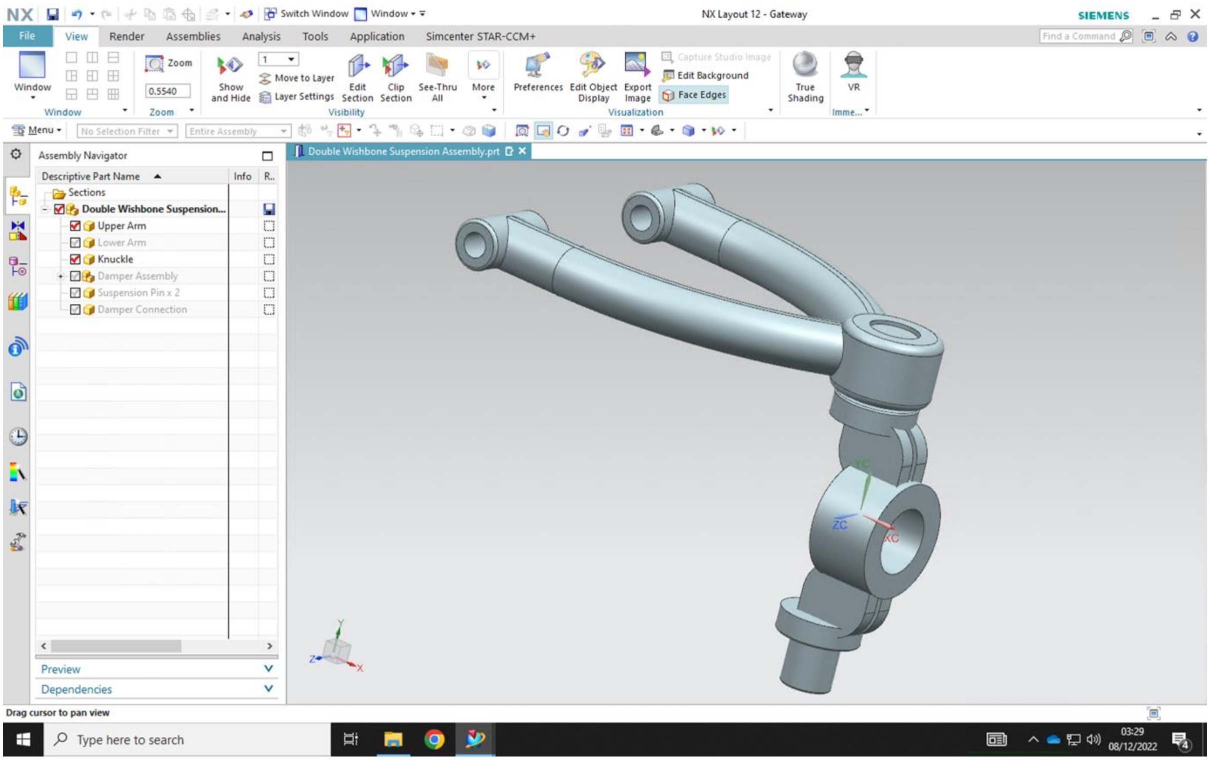
- Using the extrude tool again the circles form another hollow cylinder, this time of a length of 45mm overhanging the sides of the arm by 5mm each side.



- The same process is repeated on the other side of the body of the arm, and the subtract feature of the extrude tool is used again to remove the excess material inside the hole.

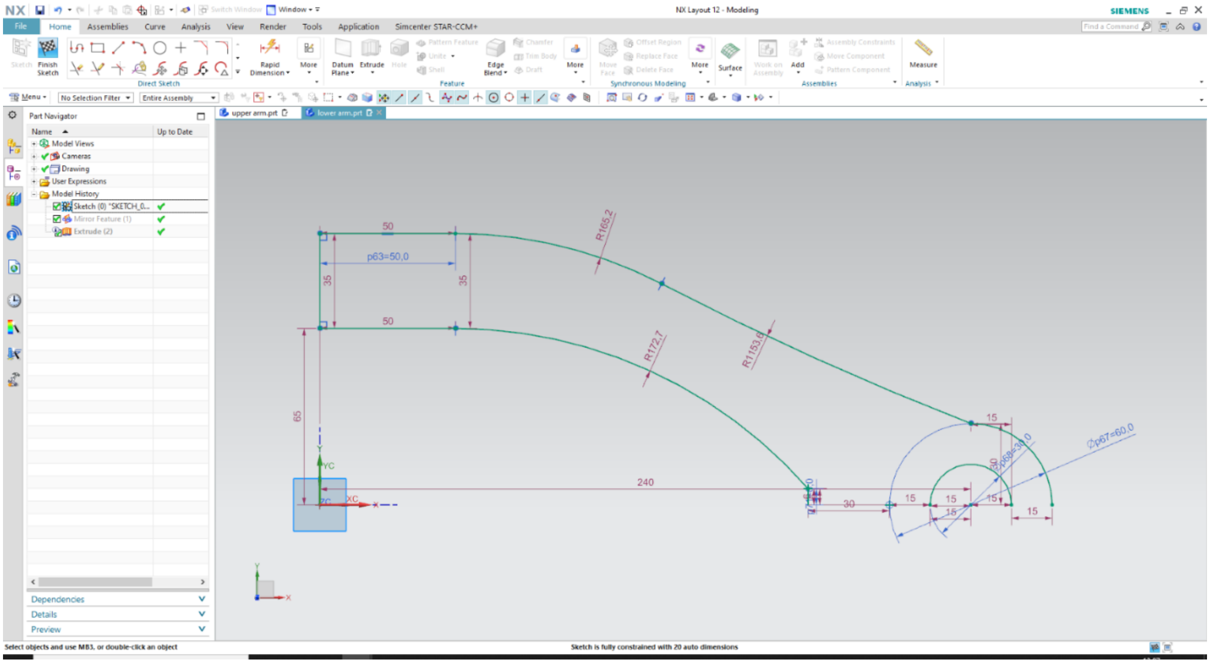


- To finish of the arm, fillets are applied in various places on the arm using the edge blend tool. The housing for the knuckle has a 5mm fillet. The main body of the arm has a fillet of 15mm. The housing for the suspension pin has a fillet of 2mm.

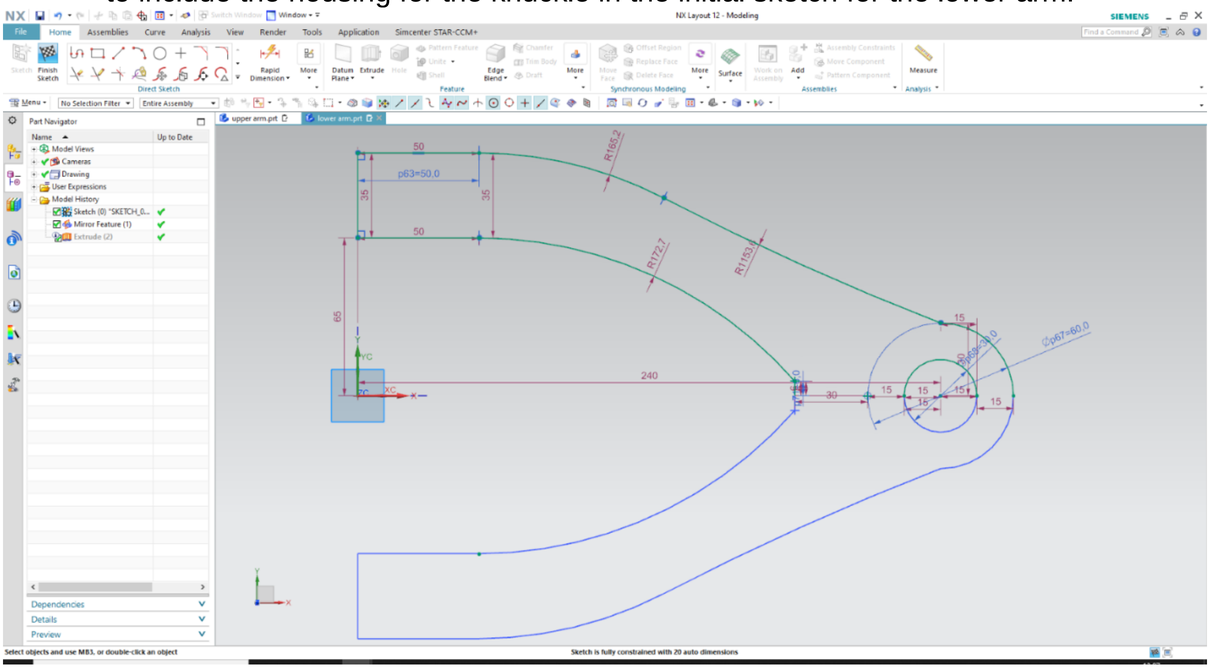


- Assembling the upper arm and the knuckle, we connect them by the larger side of the knuckle to keep the upper arm more stable.

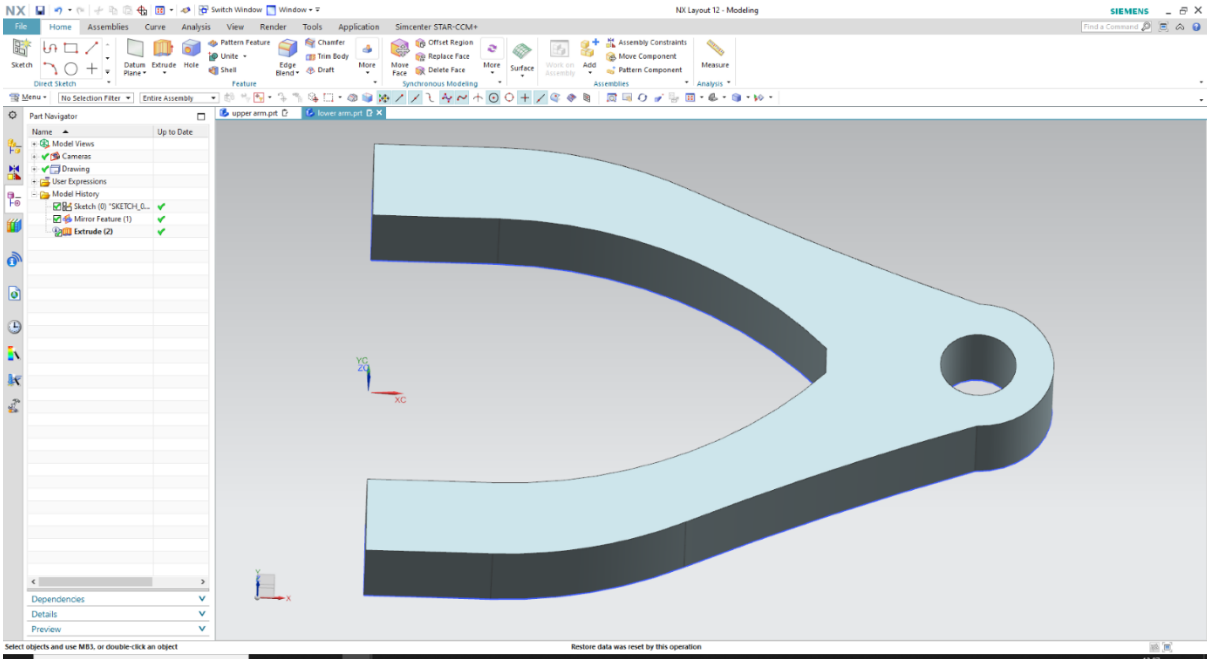
Lower Arm



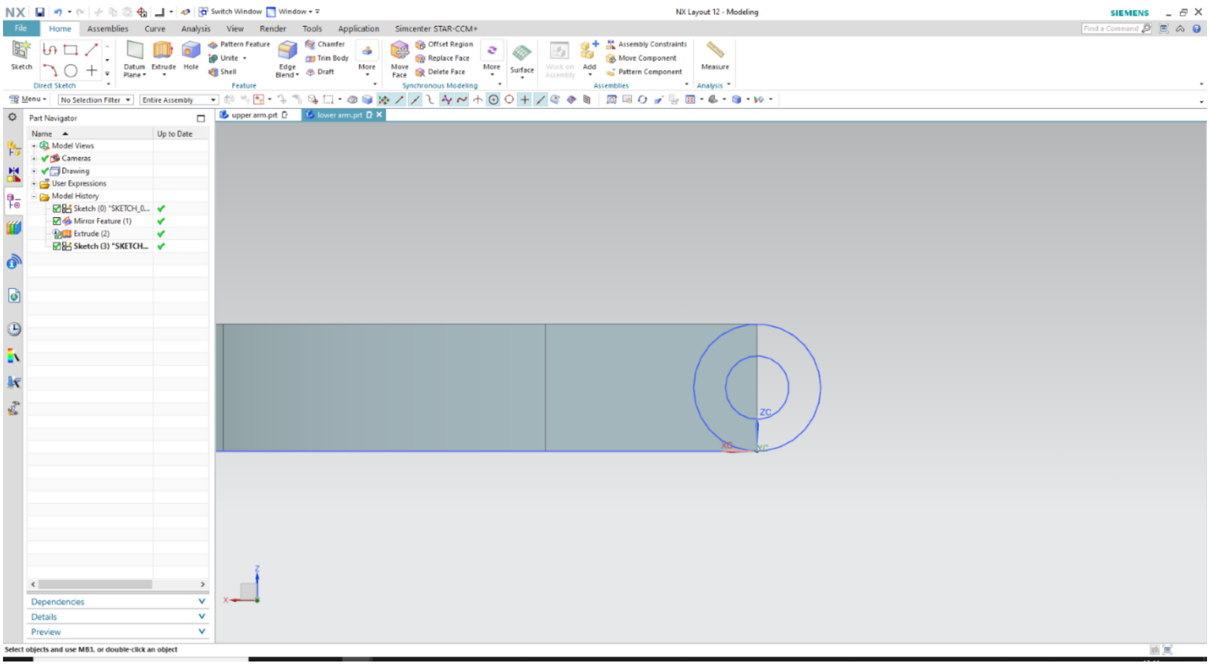
- the lower arm began with a sketch of one half of the main body of the lower arm. The half sketch is 270mm long and 100mm wide, same as the upper arm. It made sense to include the housing for the knuckle in the initial sketch for the lower arm.



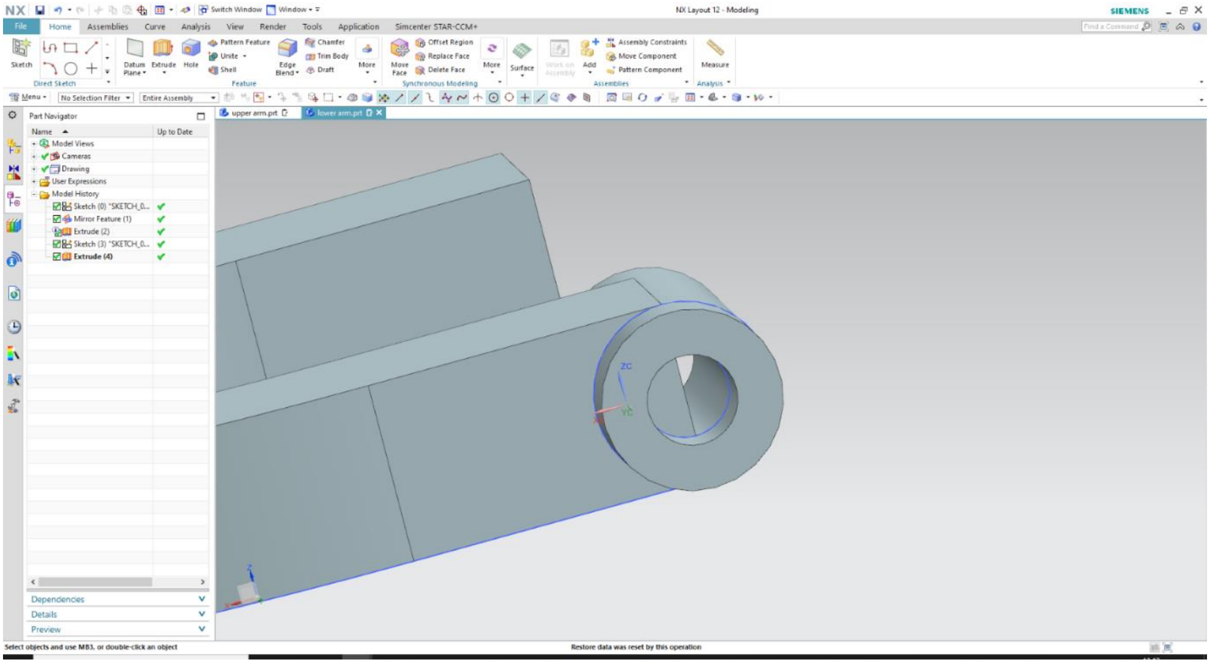
- Using the mirror feature again on the line y=0 on the y-plane to get a perfectly symmetrical sketch.



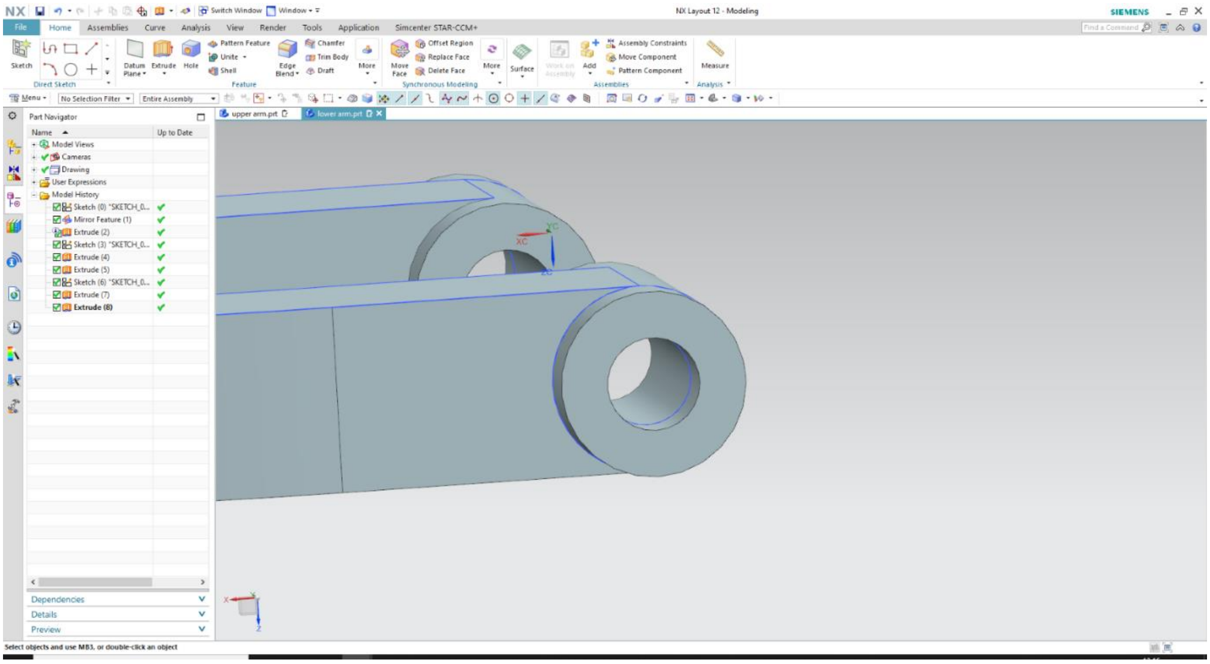
- The sketch was extruded 30mm to the same thickness of the upper arm.



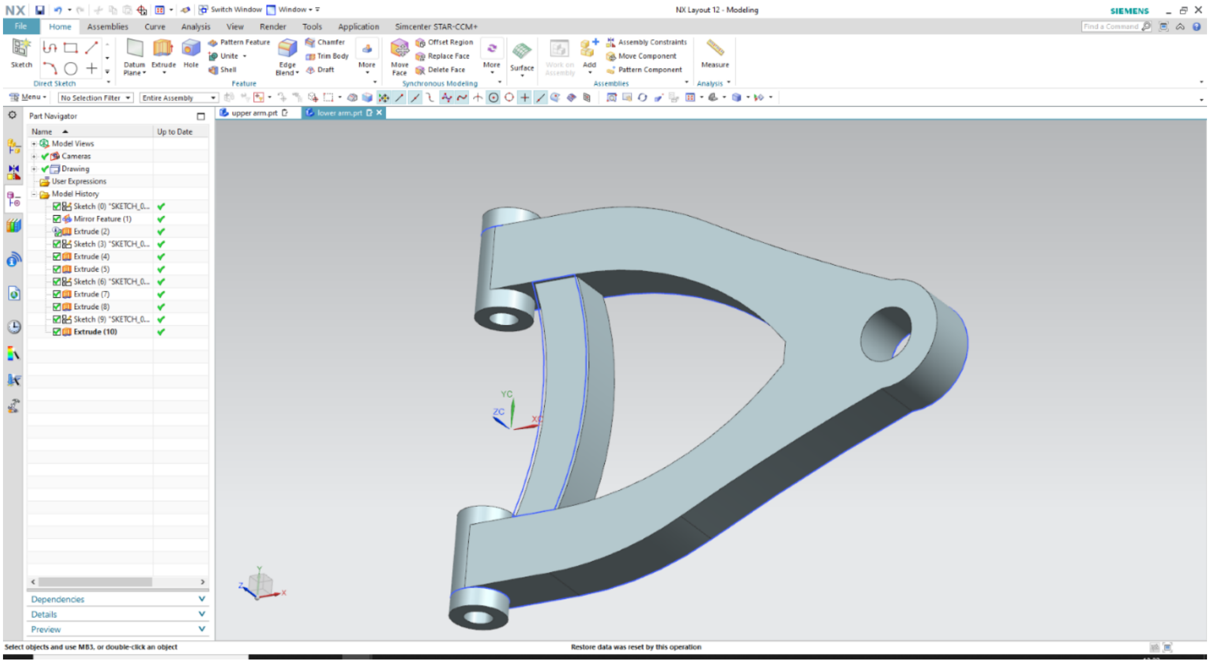
- At the rear of the main body, a new sketch on the side consisting of two concentric circles of diameter 30mm and 15mm is drawn.



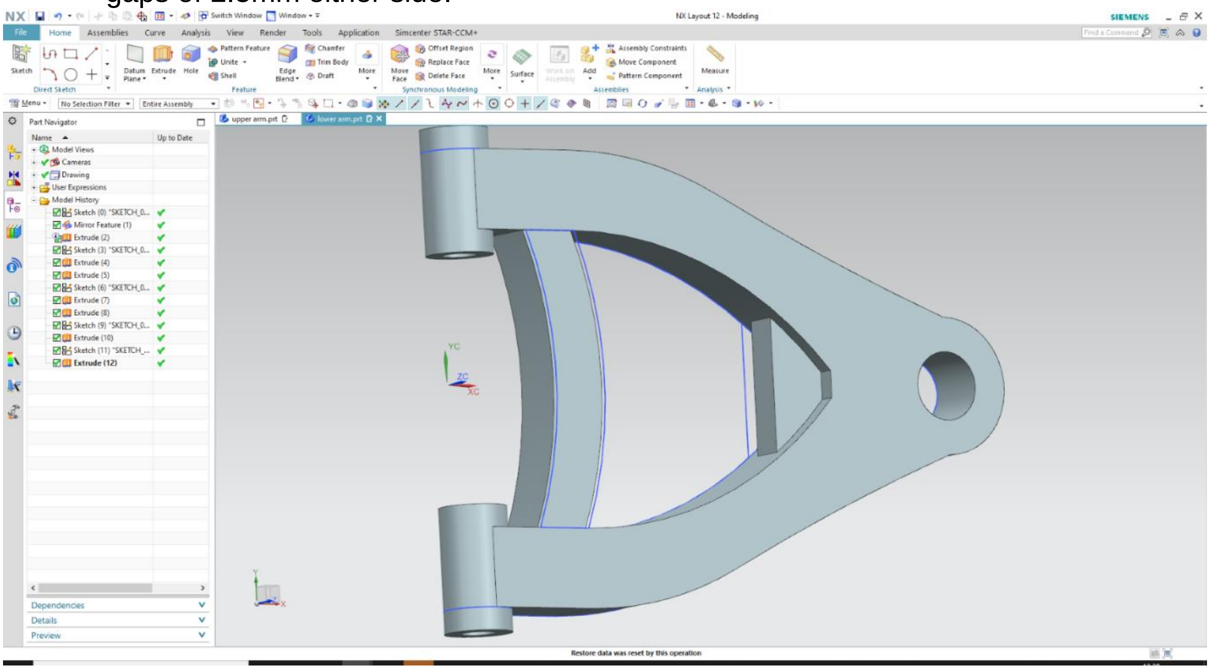
- This sketch is then extruded to a length of 45mm overhanging the sides of the arm by 5mm each side just like the upper arm. This is repeated on the other side of the other arm.



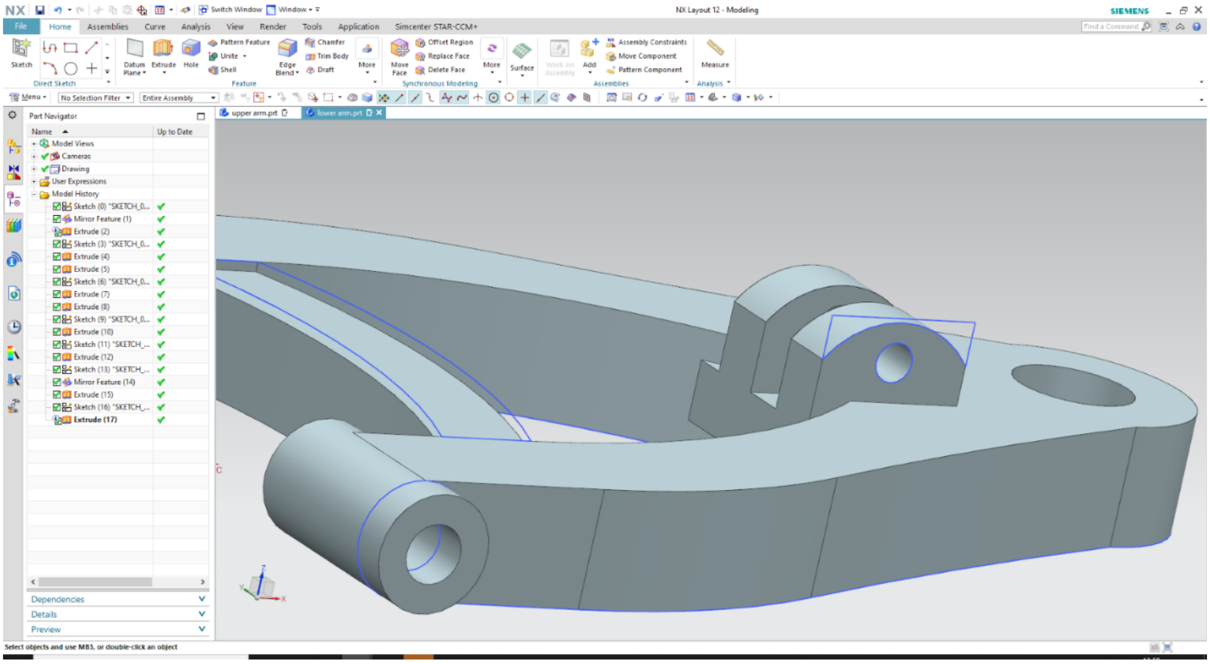
- The excess material inside the cylindrical extrusion is subtracted on both sides of the arm.



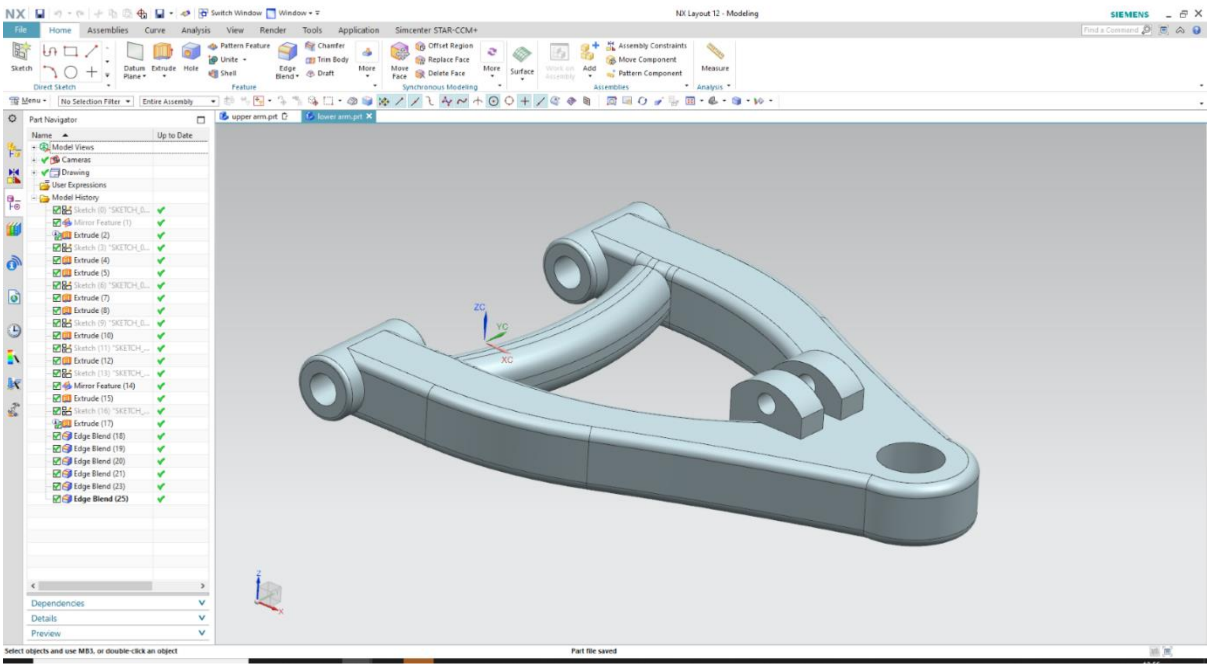
- A sketch on the same plane, with two curves of radius 180mm connecting each side of the lower arm, is drawn and extruded the sketch to be central to the arm leaving gaps of 2.5mm either side.



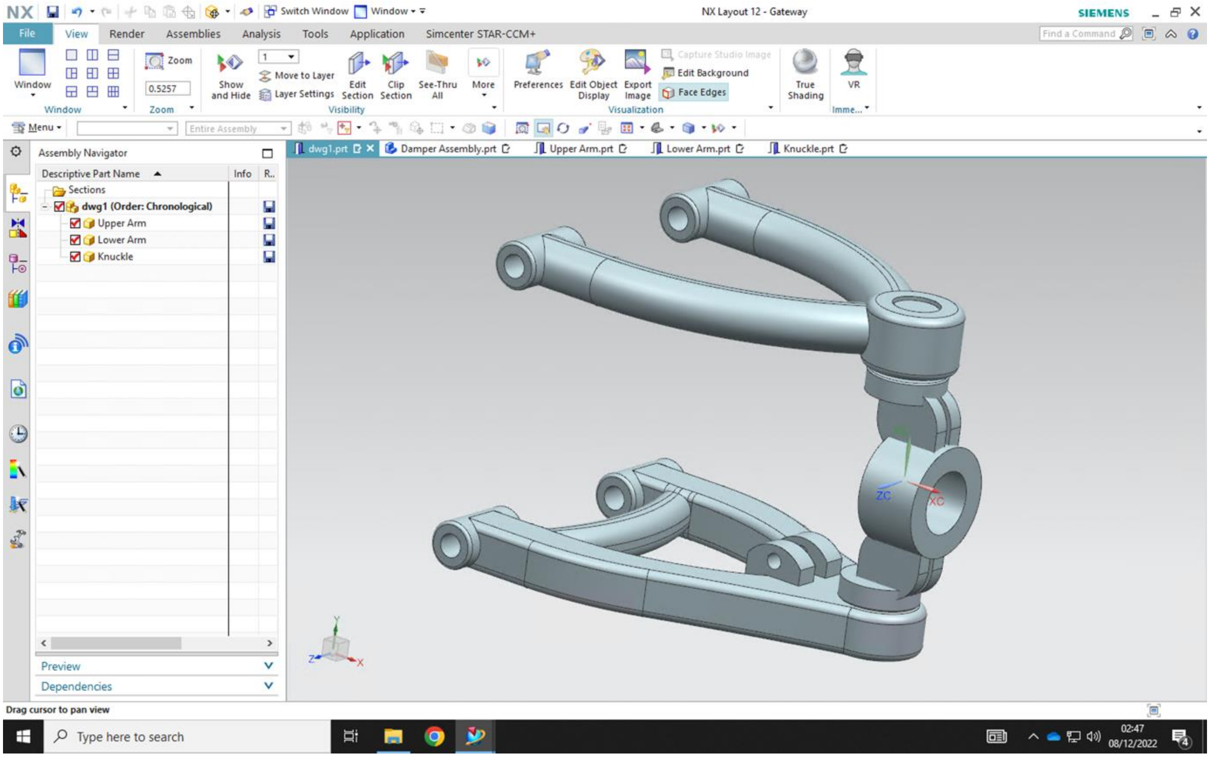
- On the same plane again, I draw a rectangle of 60mm by 25mm and extruded this sketch central to the lower arm leaving gaps of 7.5mm on either side. This is to reinforce the arm in the location where the damper will connect.



- A 40mm by 15mm rectangle was drawn on this reinforcement piece and mirrored in y=0 to make it symmetrical leaving a 12mm gap between the two rectangles. Once extruded 27.5mm above the rectangular piece, a new sketch begun on the 40mm by 20mm space available above the arms surface with a 10mm diameter circle in the centre and a curve with a 25mm radius at the top, subtracted with the extrude on both upright pieces.

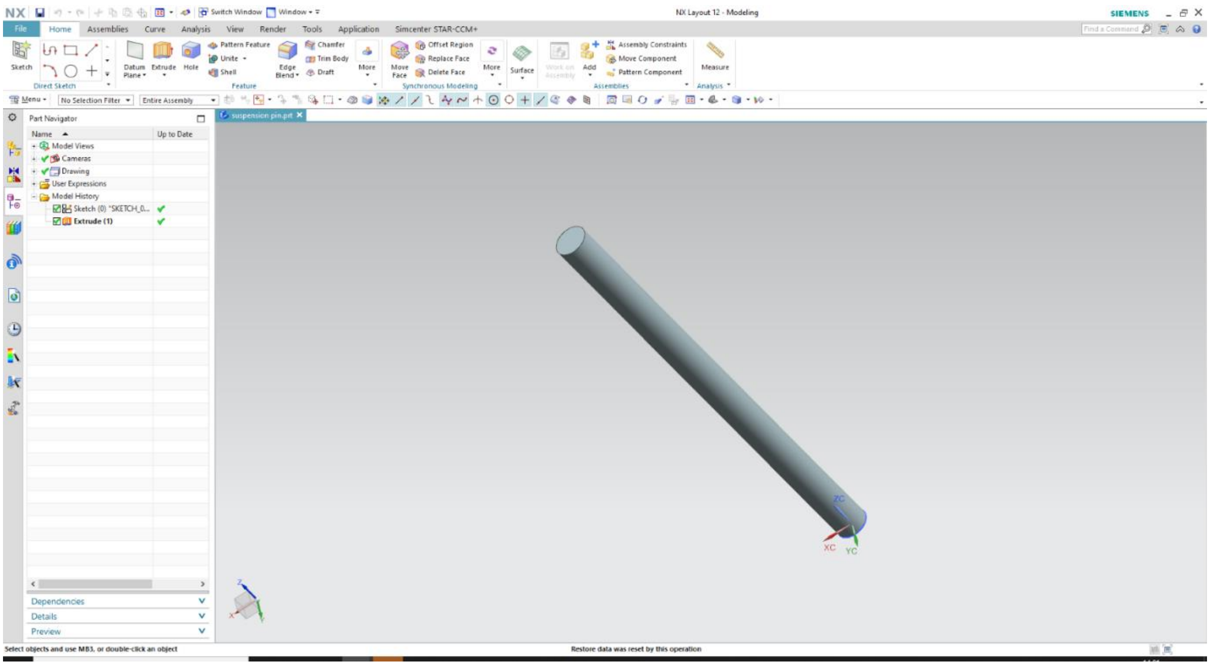


- The edge blend tool is used to fillet the edges of the part, with the housing for the suspension pin being a 2mm fillet, the bar connecting and reinforcing the two arms being a 10mm fillet, and the main body having a 5mm fillet applied.

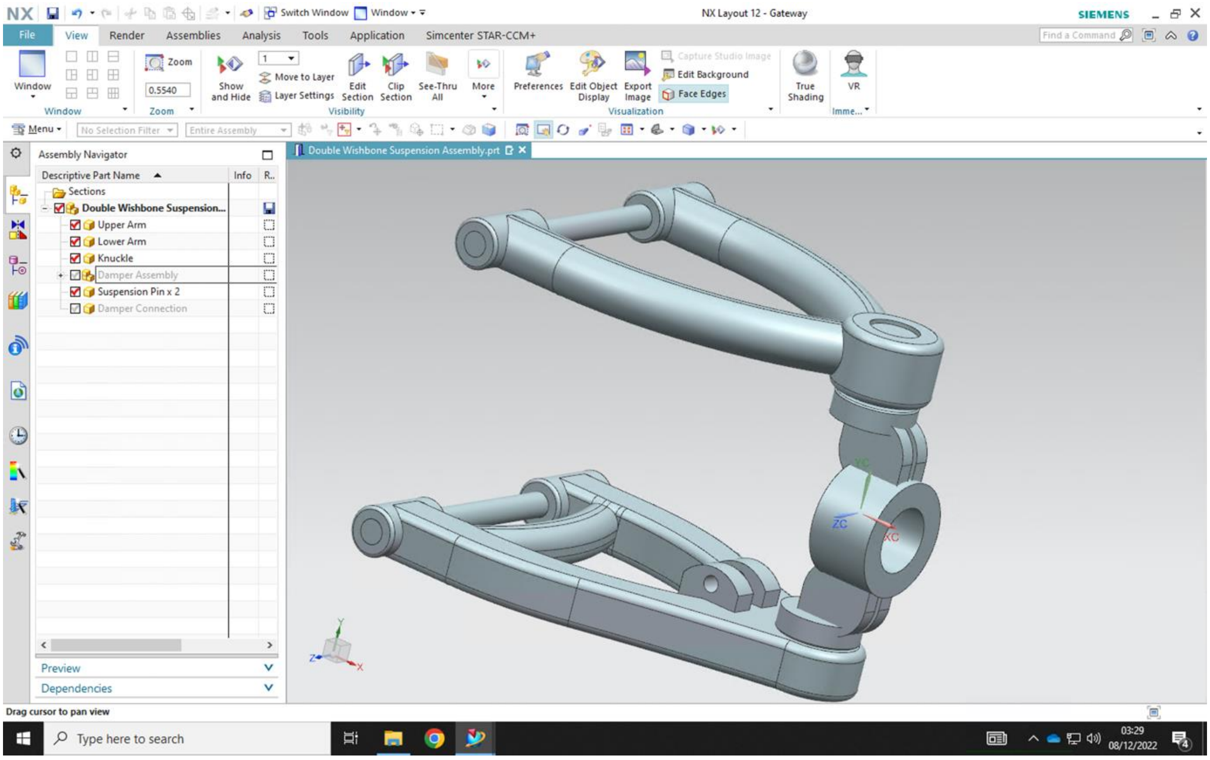


- With the lower arm added too, it is connected onto the other side of the knuckle.

Suspension Securing Pin

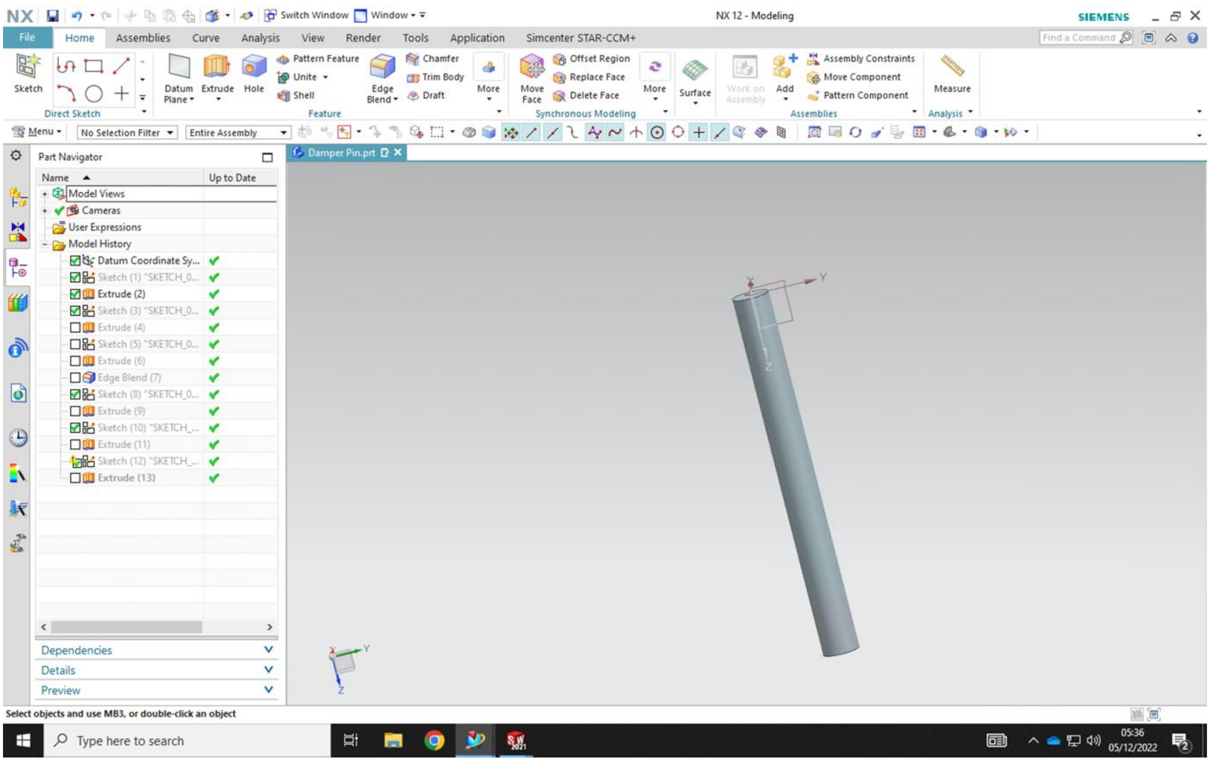


- The suspension pin is sketch of a circle with a 15mm diameter extruded by 220mm

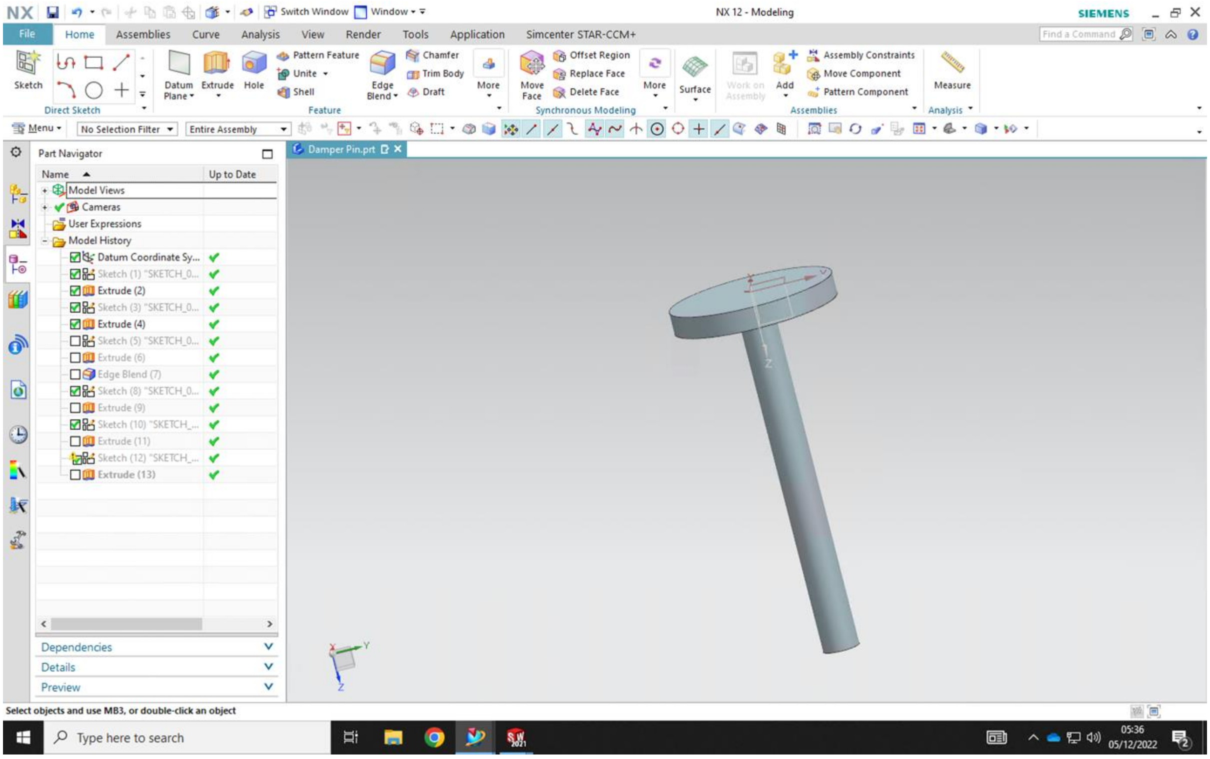


- The ‘Suspension Securing Pins’ are added to the assembly. The lower arm’s pin is more symbolic as the actual connection to the chassis may not be through a pin like this at all. The upper arm’s pin is more integral and would be a pin such as this.

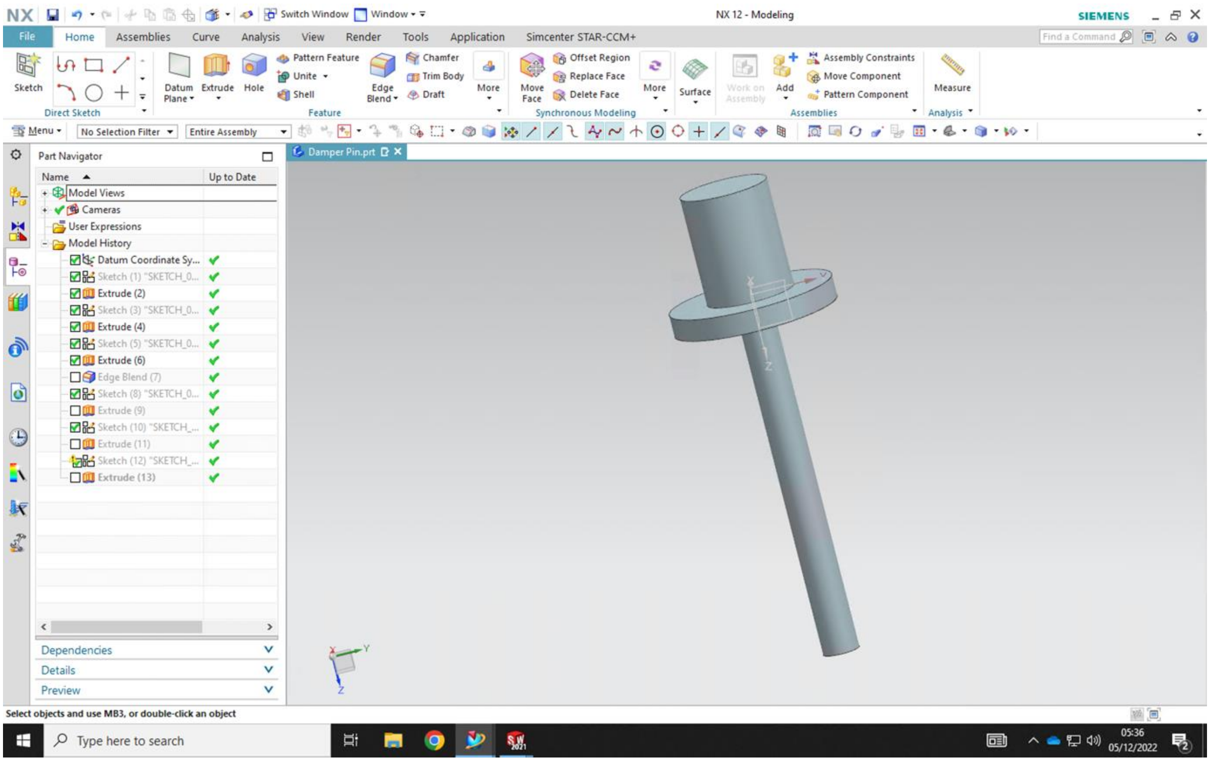
Damping Pin



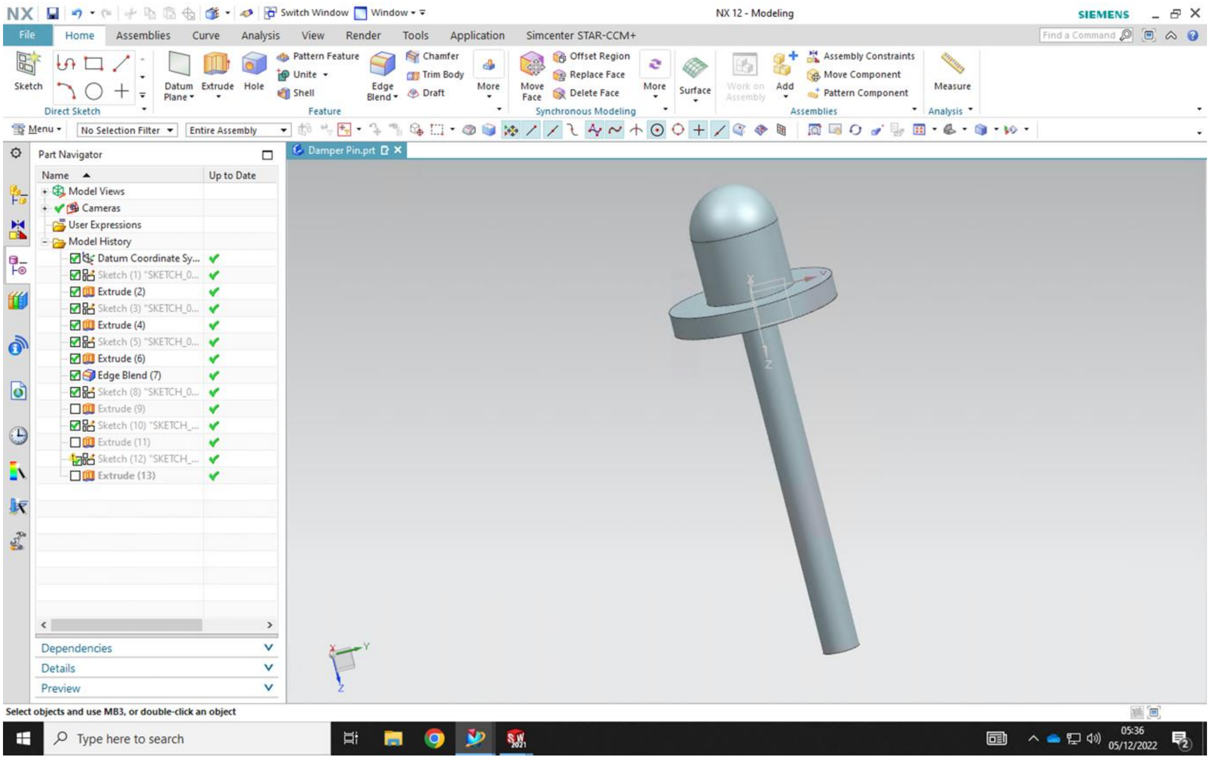
- Cylinder of 10mm is extruded to a value of 100mm. This part will be surrounded by our spring and further moves into our damper.



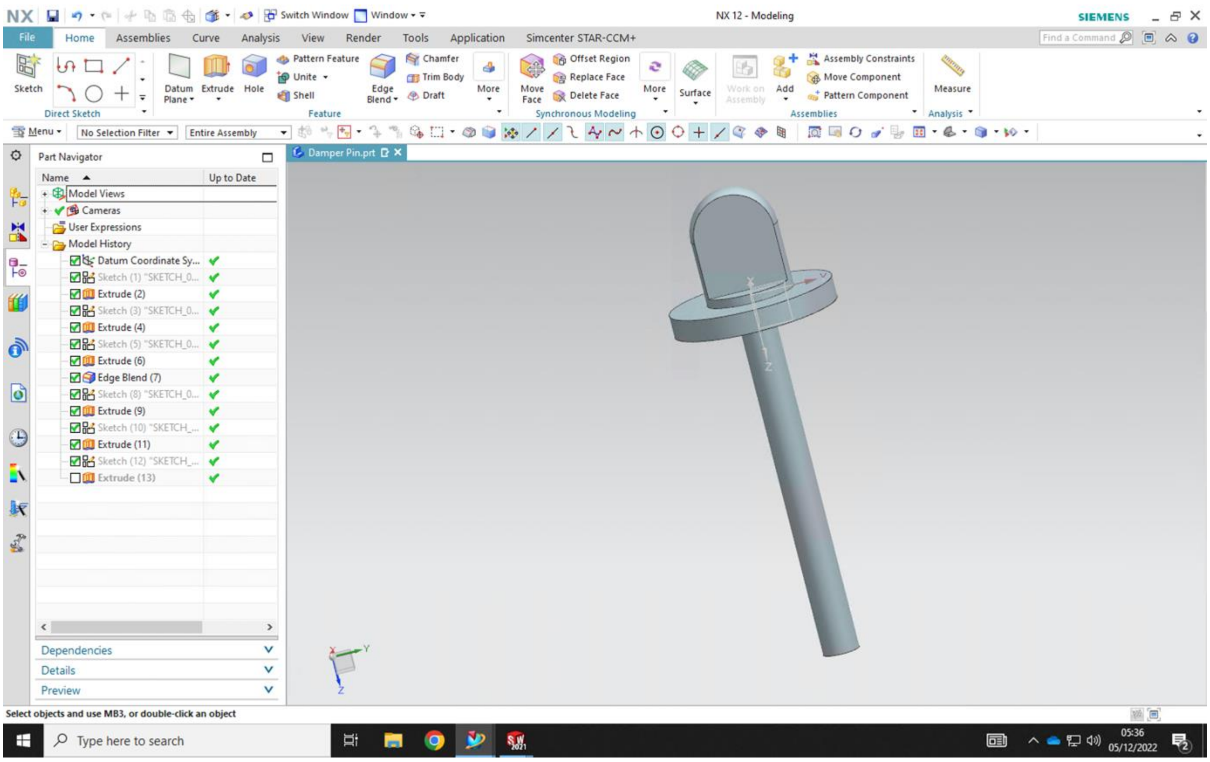
- Secondary cylinder of diameter 45mm is added and extruded to 6mm



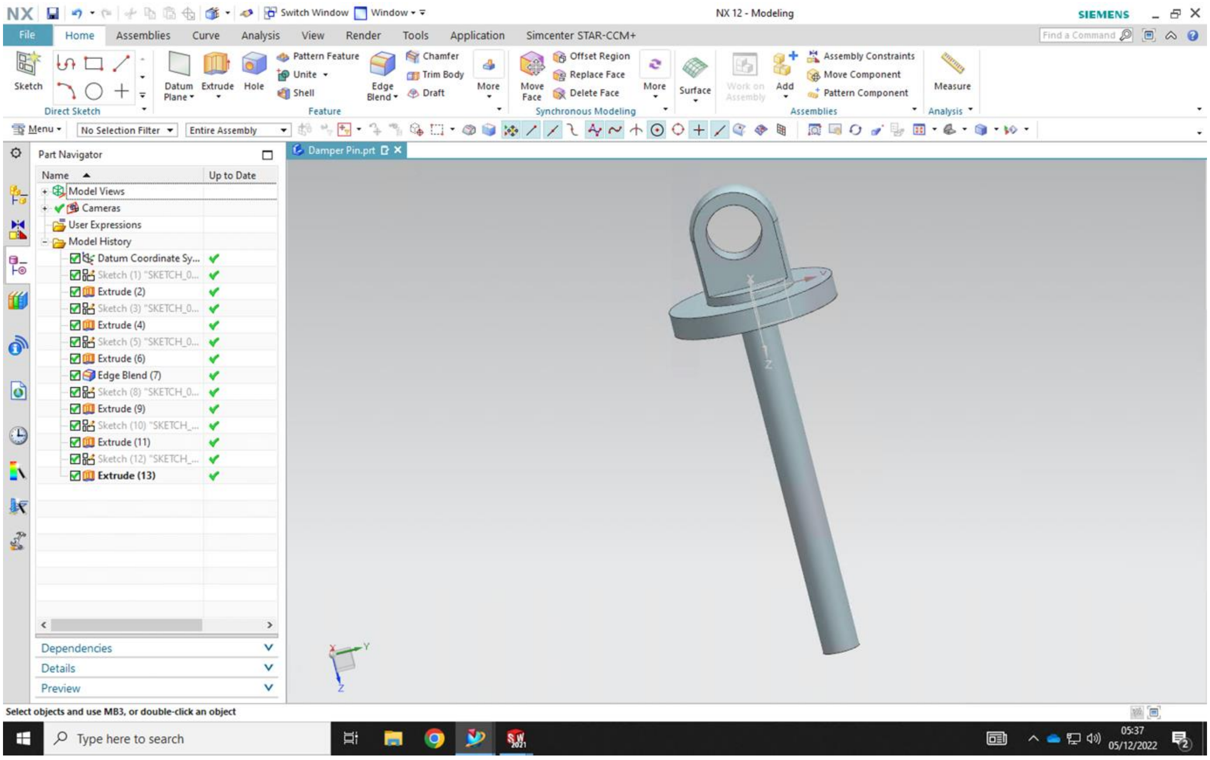
- Third and final cylinder is added with a diameter of 24mm and is extruded to 30mm



- An edge blend is used at this edge with the maximum possible value (12mm) applied to give us this semi circle shame at the end.

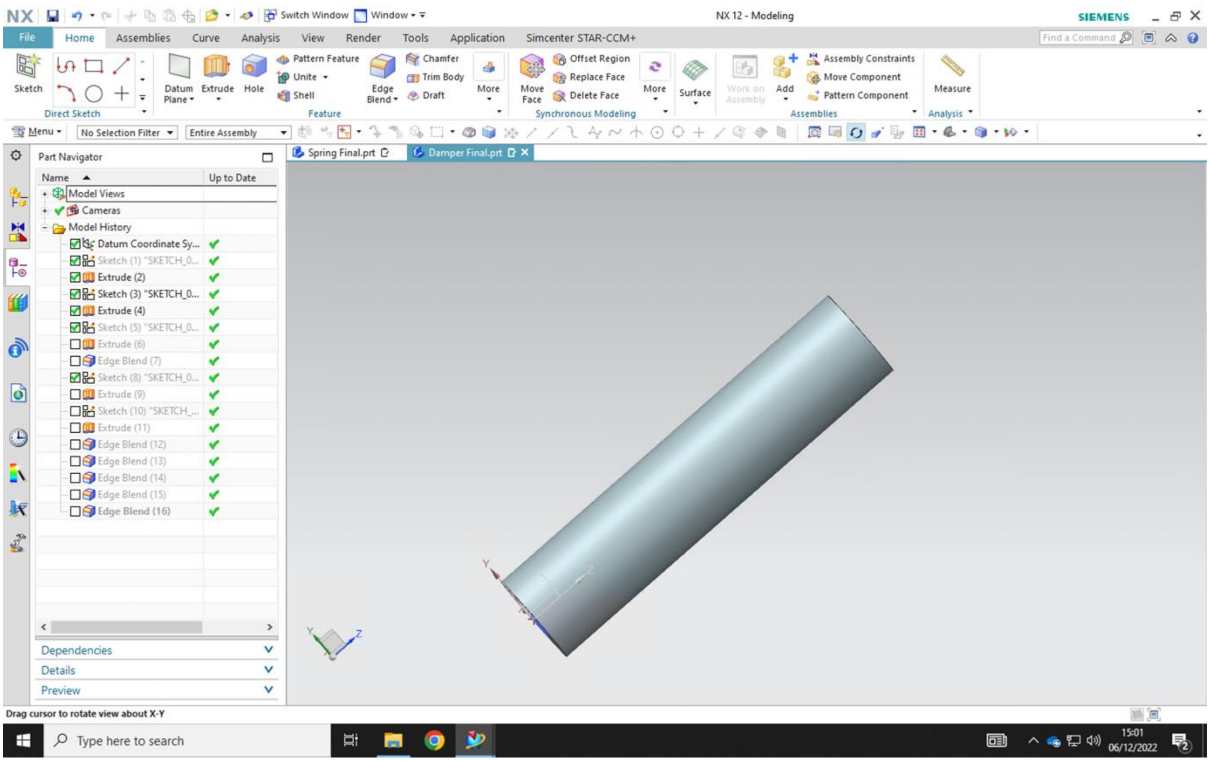


- Two rectangles are sketched and using the subtract tool within the extrude feature we can remove two sections on each side of the pin

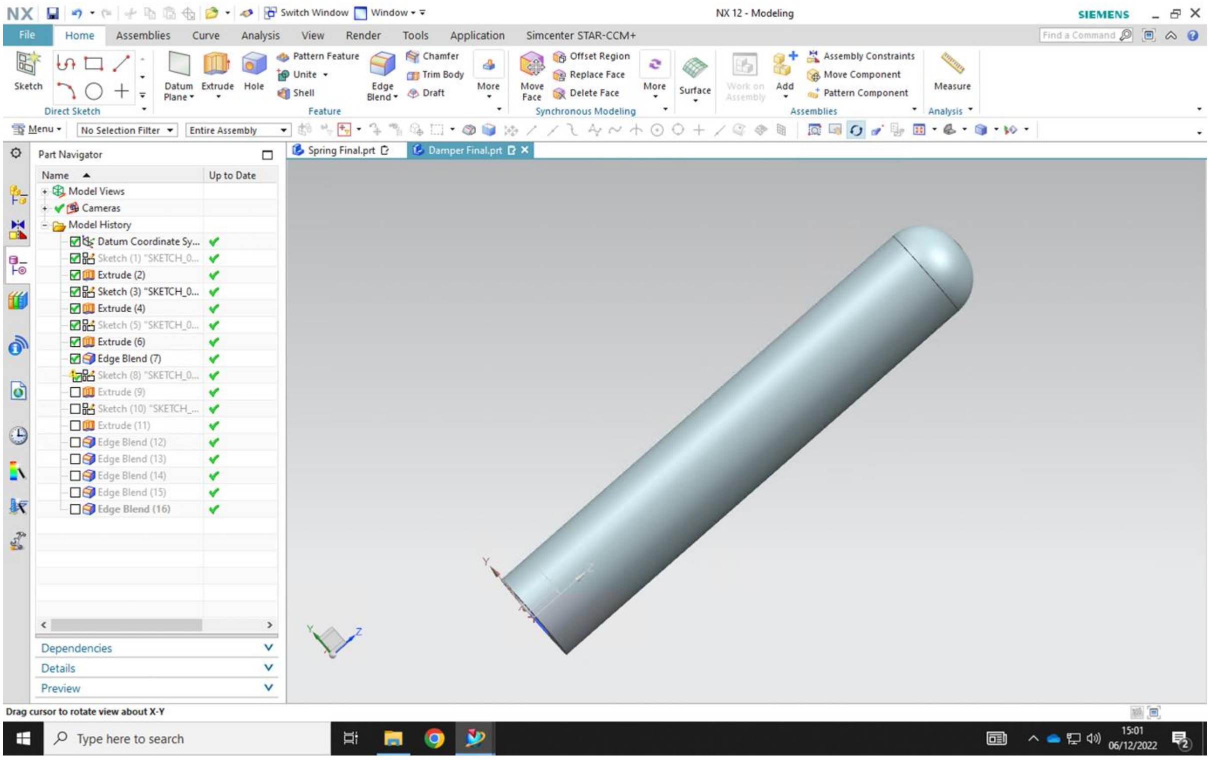


- The subtract tool is then used again on a circle of 10mm diameter which has been sketched on the pin surface. This will be the part that loops through the suspension securing pin.

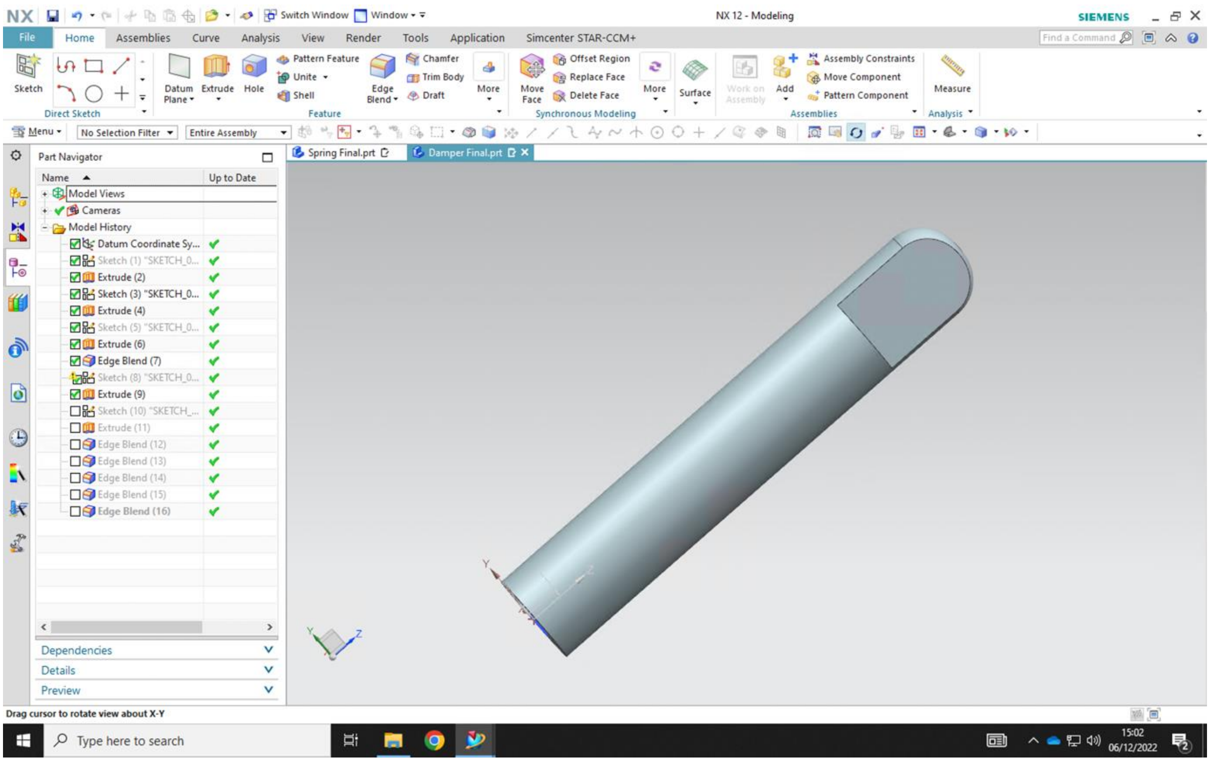
Damper



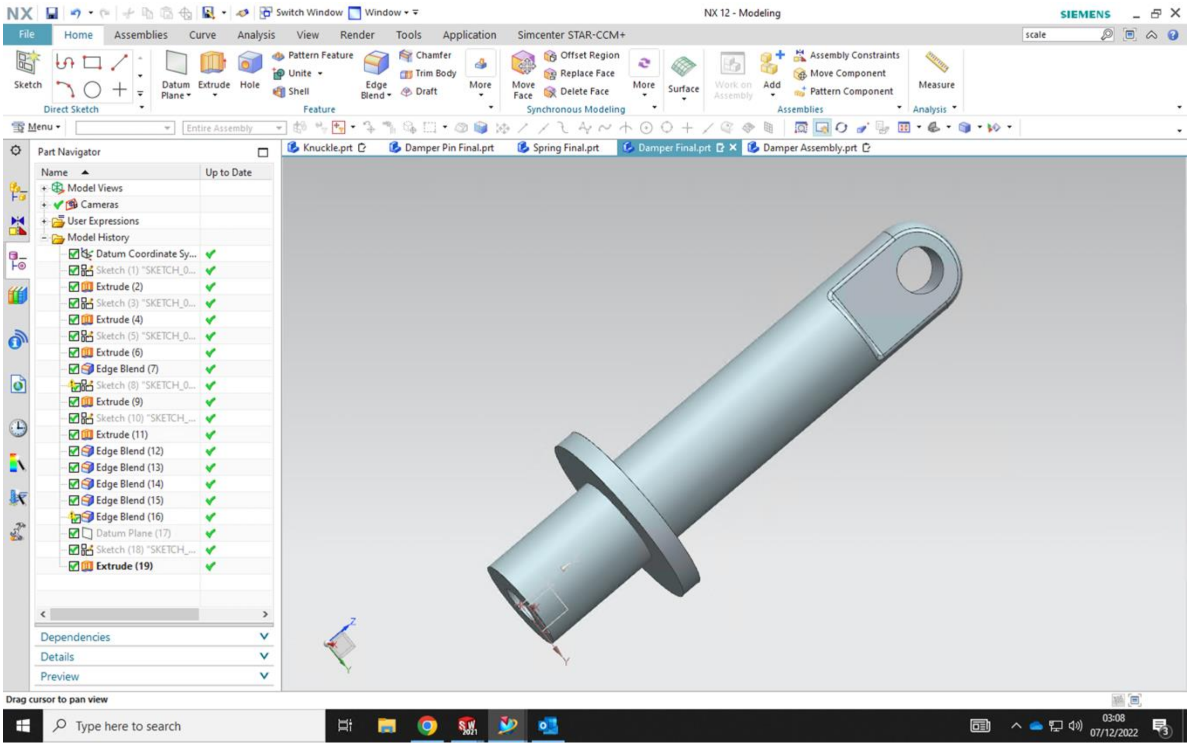
- A hollowed cylinder is designed with an inner diameter of 10mm and outer diameter 46mm. The inner diameter is the same as out damper pin and will allow it to move freely inside within the Z plane. This part is extruded by 100mm



- A secondary cylinder is added to cut of this centre hole for further part strengthening. The edge blend function is then used to smoothen this top using a diameter of 10mm

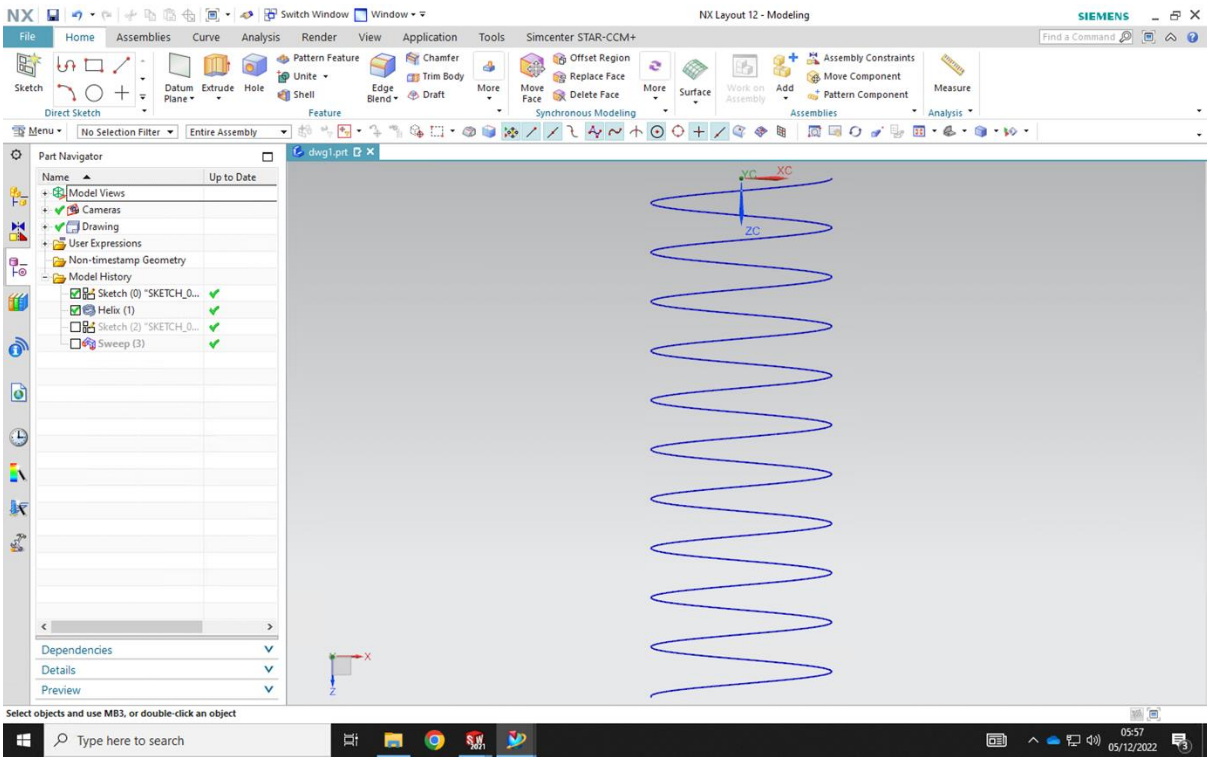


- Similarly, to the damper pin we use the subtract feature within the extrude function on two sketched rectangles to remove part of the shape on both sides of the damper.

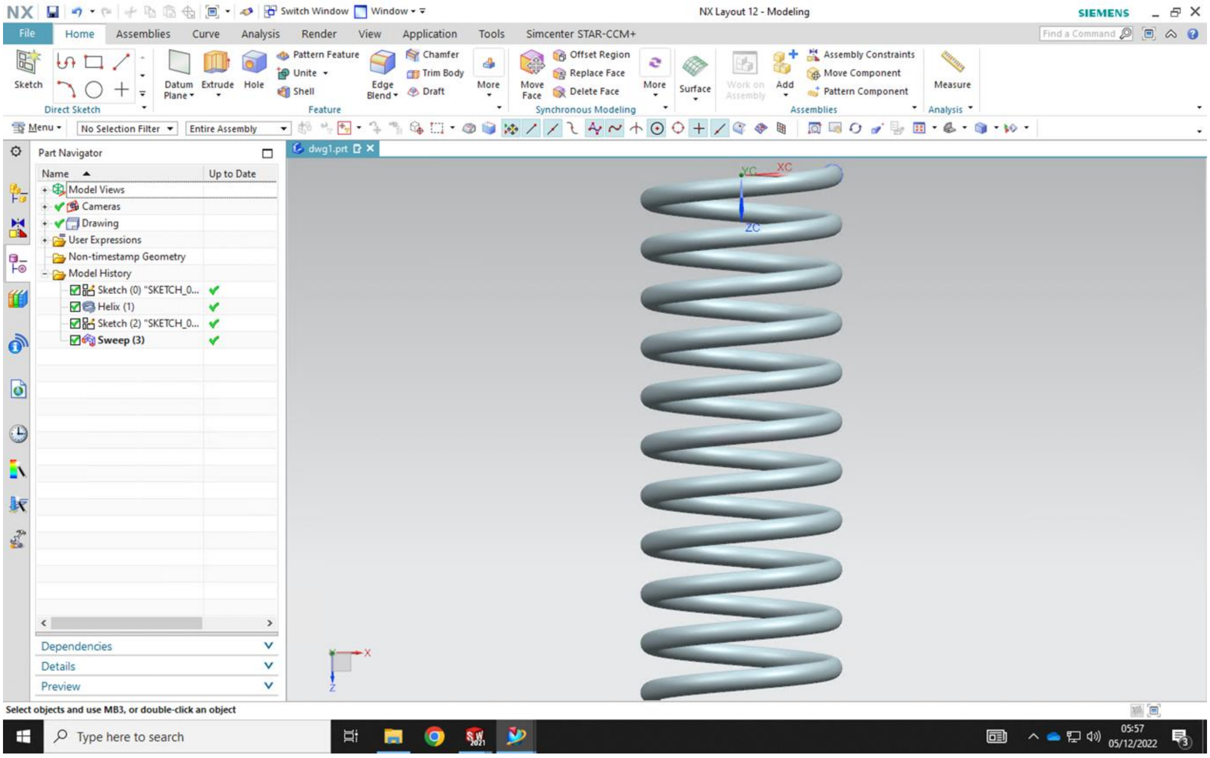


- Finally, we use the subtraction function on a circle shape same as on the pin but of diameter 11mm to allow the damper securing pin to fit through. A cylinder is added towards the base to act as a stopper for the spring part. We finish off with some 1mm edge blends to make the part smoother.

Spring

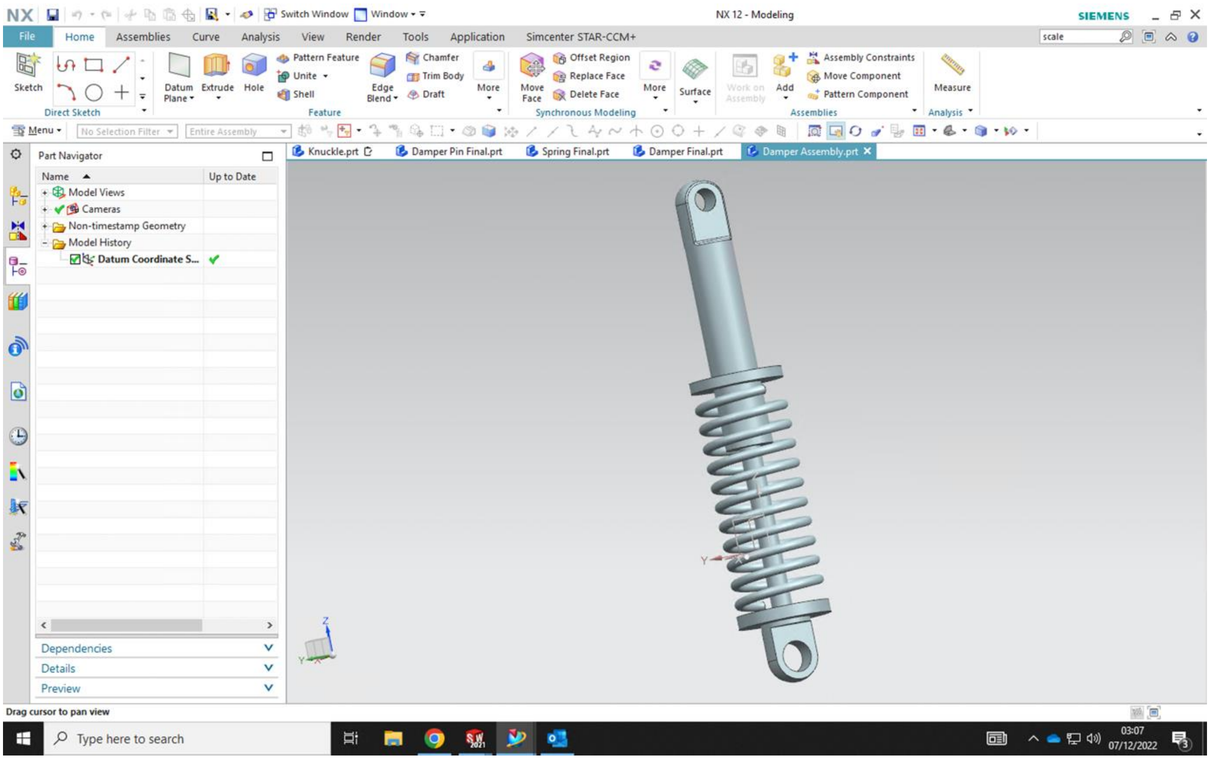


- The spring is made by simply selecting the Helix function and selecting a pitch of 9.5mm and a vertical distance of 100mm and a diameter of 40mm



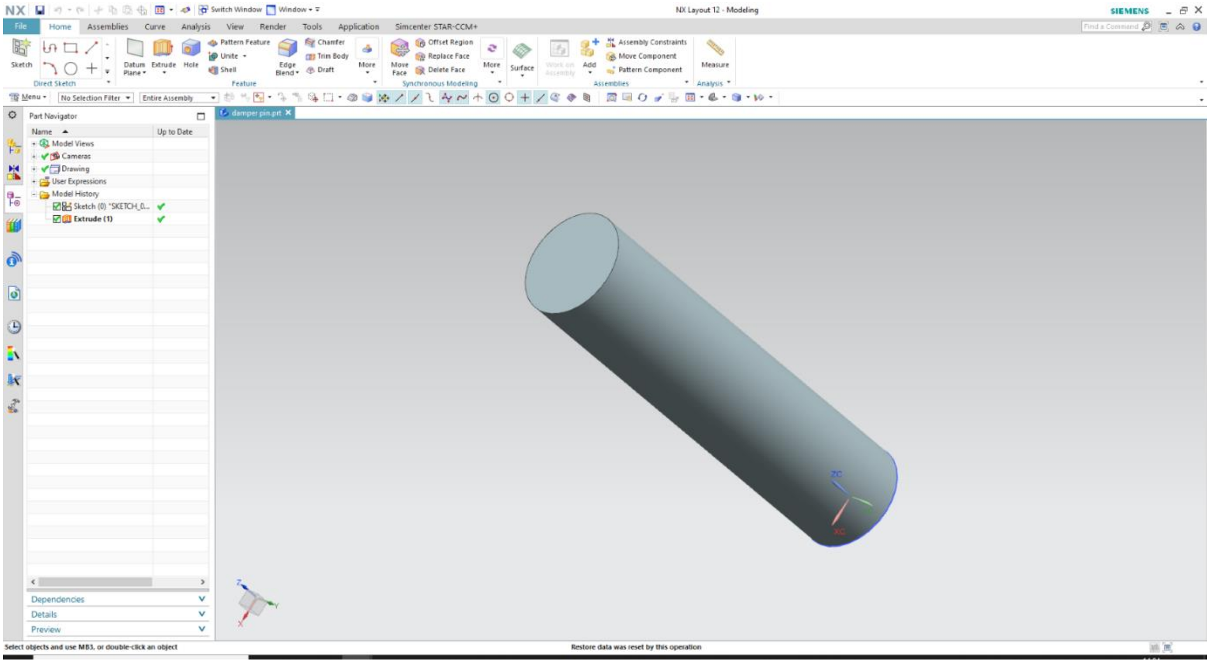
- A sketch of a circle of diameter 4.5mm is then added in the X-Z plane where the centre is the top of the helix sketch. Finally, the 'sweep along a guide' tool is used where the object is the circle, and the guide is the helix.

Damper Assembly



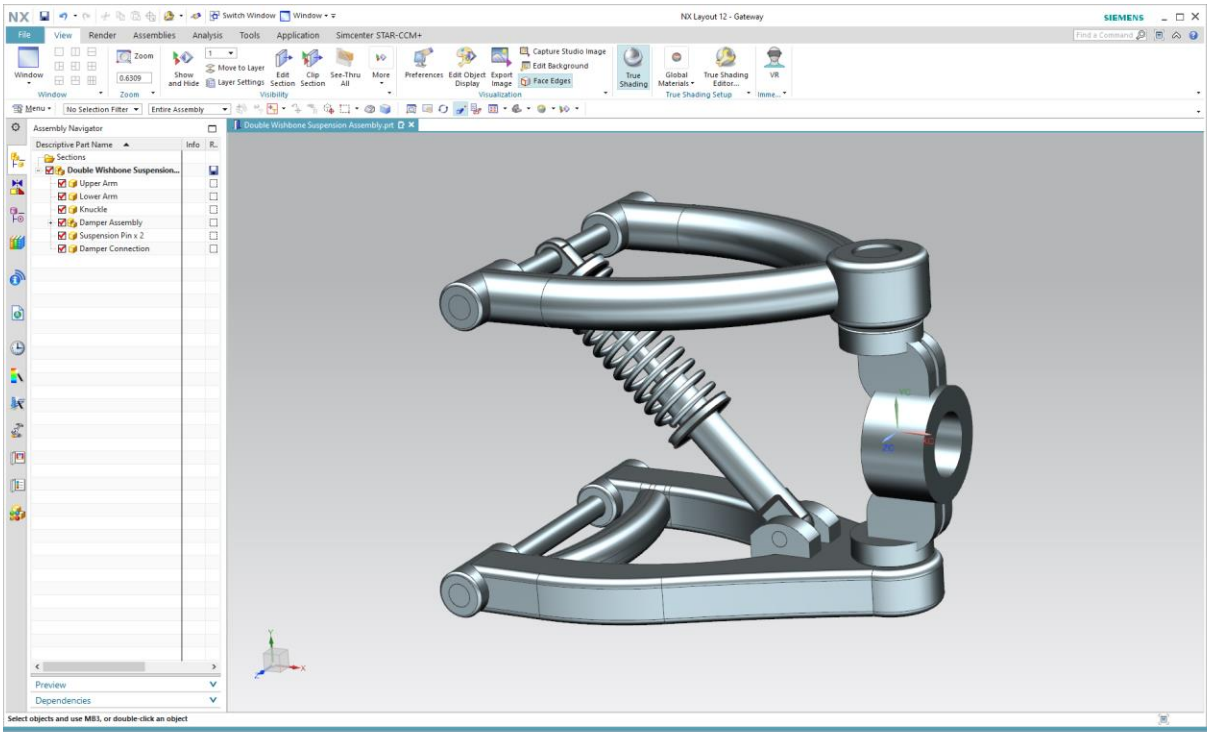
- Damper is fully assembled as shown above

Damper Securing Pin



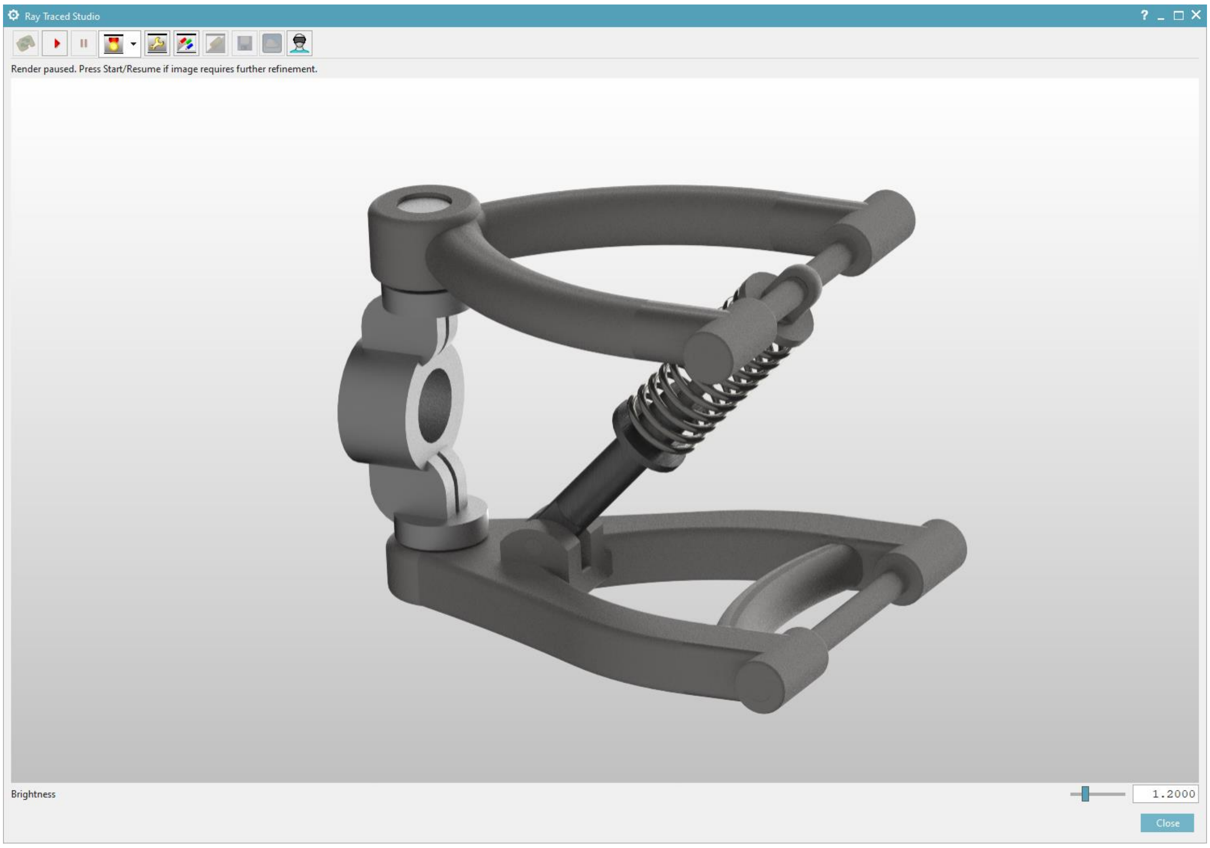
- This damper pin is a 10mm diameter sketch extruded to 42mm in length

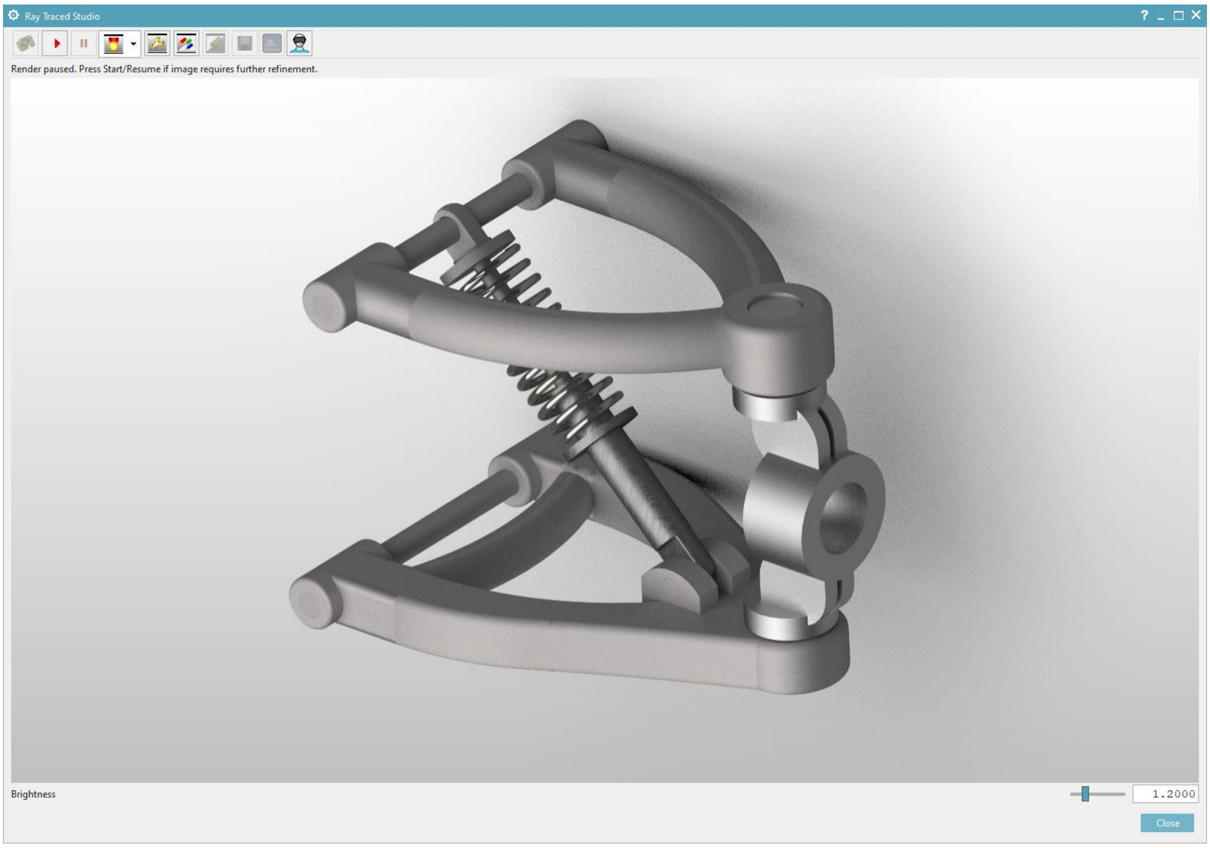
Final Assembly



- The damper components are combined as they are added to the assembly and the damper securing pin is added to secure it to the lower arm. As seen, the damper is secured to the upper arm by the suspension pin. We also render the assembly adding a steel material to all parts.

- Ray tracing images are shown below


Ray Traced Studio
Render paused. Press Start/Resume if image requires further refinement.
Brightness
1.2000
Close


Ray Traced Studio
Render paused. Press Start/Resume if image requires further refinement.
Brightness
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Close

Section 2

Bending Analysis

The part chosen to perform bending analysis on is the upper arm of the suspension assembly. This part would logically have the most chance of being under bending stress

The arm had a length from the knuckle end to the vehicle end of 220mm and the arms had a width of 30mm and a depth of 35mm.

Vehicle weight was estimated at roughly 1600kg. Because the suspension is the gateway from the unsprung mass to the sprung mass, it only feels the weight of the sprung mass which we approximated to be 1400kg. The unsprung mass that isn't considered is therefore 200kg.

The chosen material for the upper arm is steel.

Normal Stress:

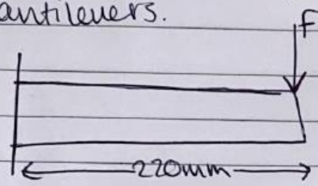
For the force calculated, due to the weight estimate, we approximated the value of g as 10.

Force $= 1400 \times 10 = 14000$ N

$\div 4$ gives 3500 N at each tyre

$\div 2$ gives 1750 N for the upper suspension arm

the connection point with the knuckle acts as the wall to the arms of the upper arm which are simplified as cantilevers.



Force, F on one arm $= \frac{1750}{2}$

on both is 1750N

Moment, $M = 1750 \times 220$

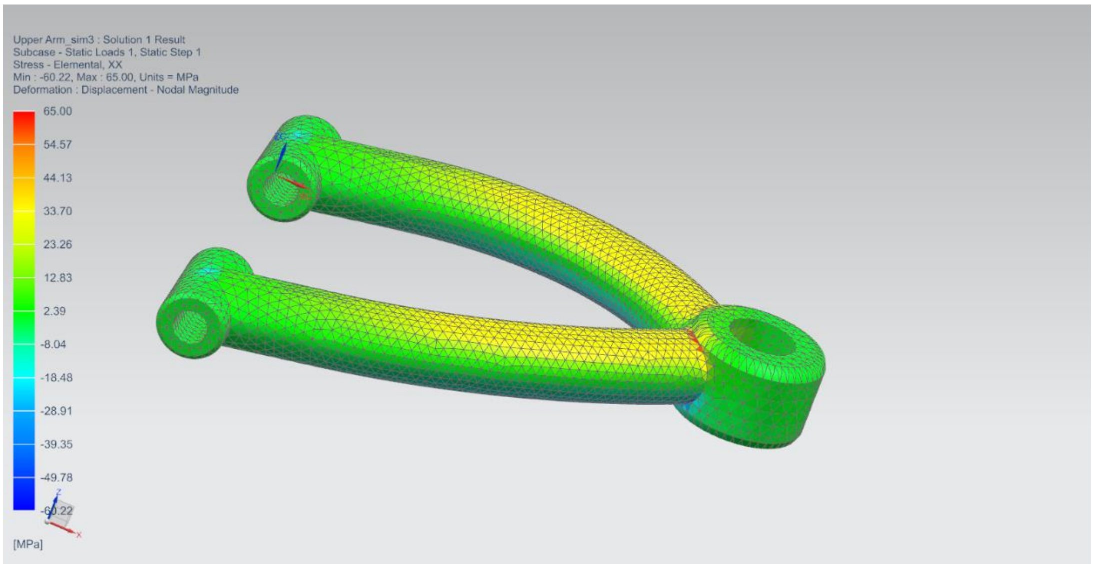
$= 385000$ Nmm

$I = \frac{bd^3}{12}$ where $b = 30$mm and $d = 35$mm

(and therefore $y = \frac{35}{2}$ mm)

$$I = \frac{30 \times 35^3}{12} = 107187.5$$

$$\sigma = \frac{385{,}000 \times \frac{35}{2}}{107187.5} = 62.86 \text{ MPa}$$



In NX the force was applied on each side of the arms with the total force being spread across the two, as per the hand calculations. The point at which the arm is connected with the knuckle was constrained so we could see the bending of the suspension arm.

The analysis in NX gave us a maximum normal stress = 65.00 MPa

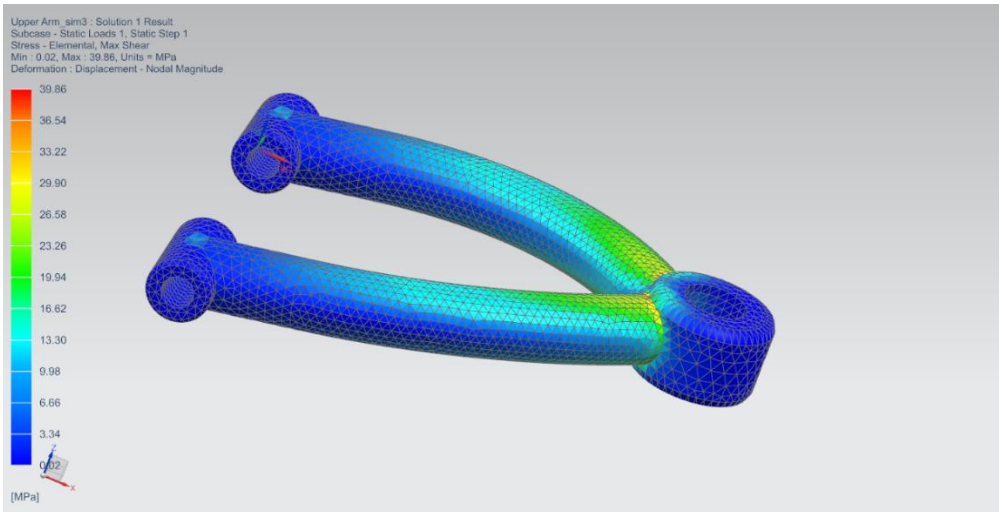
In the hand calculations due to the complexity of the suspension shape we simplified the model so that each arm of the upper arm was a cantilever beam, and the 'wall' was the knuckle which was constrained. This resulted in a theoretical maximum value slightly below what NX calculated which showed that approximating each arm as cantilever beams was fairly accurate for the bending considered.

Shear Stress:

$$\tau_{max} = \frac{V_{max}\, Q_{max}}{Ib} \qquad V_{max} = \frac{FL}{2} = 192500$$

$$Q_{max} = \frac{bd^2}{8} = 4593.75$$

$$\tau_{max} = \frac{192500 \times 4593.75}{107187.5 \times 30} = 275 \text{ MPa}$$



In NX the shear stress was evaluated and is visibly at its greatest where the arms extend off the knuckle. The maximum shear stress given from NX was 39.86MPa

When considering shear stress however, the simplification of cantilever beams in the hand calculations did not produce a similar result. The hand calculations produced a maximum shear stress of 275MPa which made sense as the cross section of a rectangle gives a greater cross section than the actual filleted arms on the suspension. Therefore, we can assume that the shear stress of the suspension arm should be much lower than this value. The arm modelled shows a greater stiffness than it has strength due to the maximum normal stress being much higher than the maximum shear stress.

Buckling Analysis

A Buckling Analysis was completed to the damper pin which encounters a compressive force due to movement from the upper arm from vehicular weight.

The Damper Pin has an overall length of 100mm and a diameter of 10mm. Using the previously found unsprung mass of 1400N, we can calculate a force acting at 45° to be 919.31N.

The material used for this part was steel which has a Youngs Modulus of 210 GPa.

Knowing these values, we can calculate the minimum critical force acting on the damper pin to cause global buckling. This is shown below:

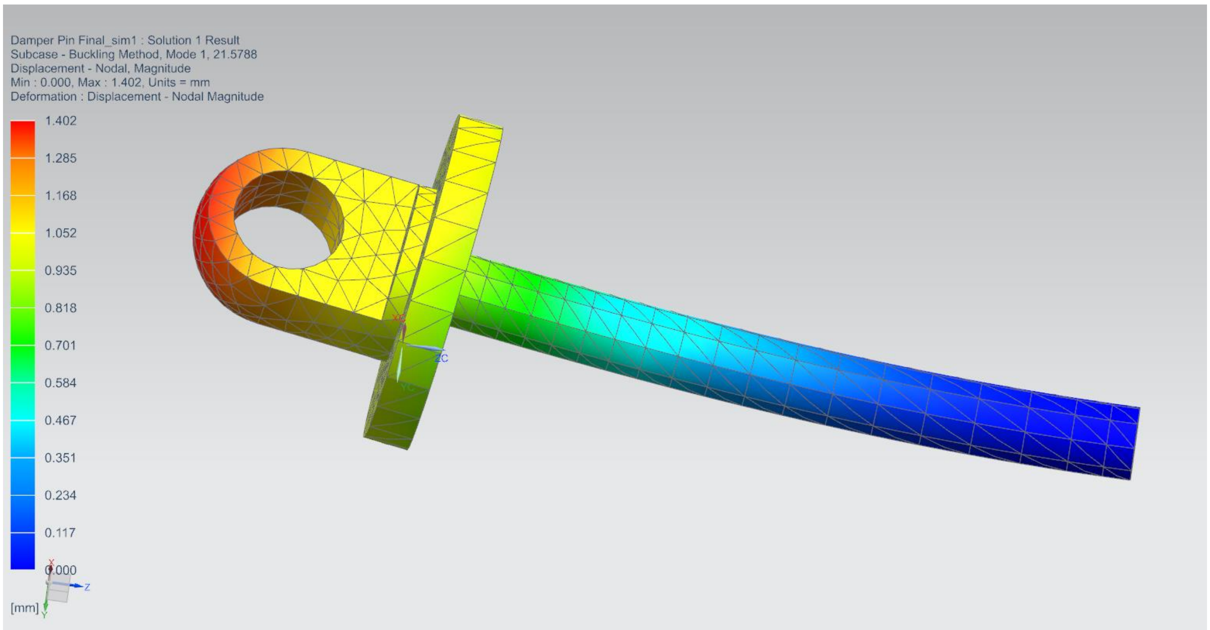
Eulers Buckling Equation: Pinned ends

$$P_{cr} = \frac{\pi^2 EI}{L^2} \qquad P_{cr} = \frac{(\pi^2)(210\times10^9)(107187.5\times10^{-12})}{(100)^2}$$

$I = 107187.5\ mm^4$

$E = 210\ GPA$

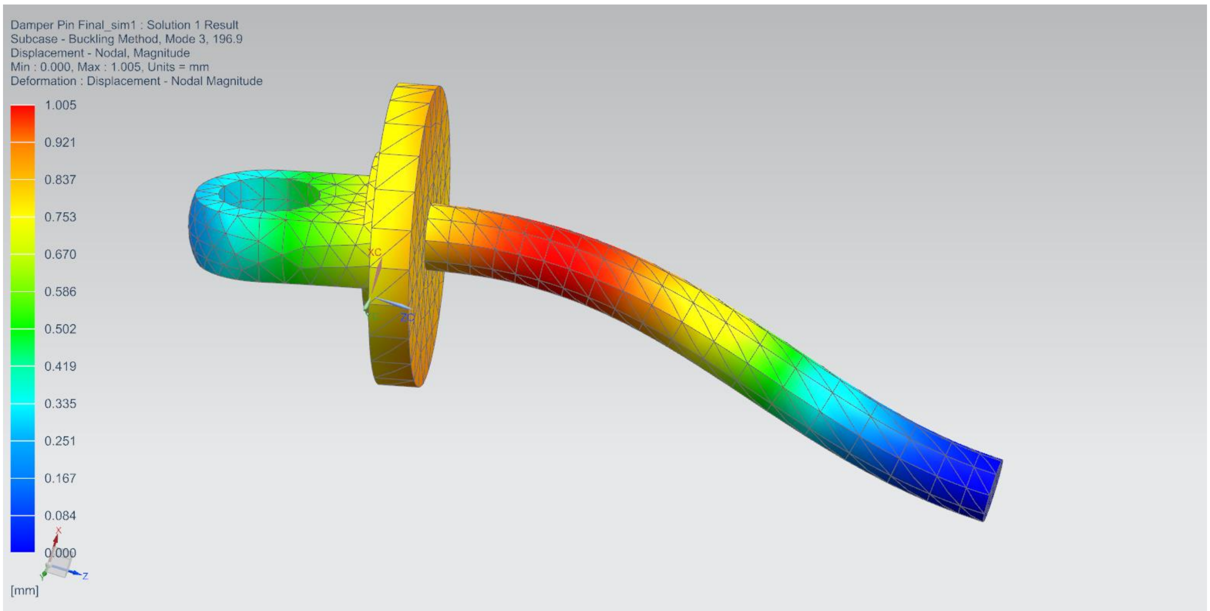
$P_{cr} = 22.2\ N$



The result received is close to the critical value in Mode 1 (21.57N) to cause global buckling. Potential errors could be due to rounding or calculations. The pin was treated as a regular cylindrical beam with constraints on the end (which would enter the damper casing) and at the centre circle where a suspension pin would usually be placed. Because these parts are in constraints, we wouldn't experience any buckling in these positions and the therefore the results seem slightly exaggerated towards the centre due to the restrictions.

It is also important to note that effects from the spring is not included in this analysis so we are seeing the worst-case scenario where this part may fail due to fracture or other damage.

The critical force to receive local buckling is shown below using NX at Mode 3.



Torsion Analysis

Unfortunately, due to a lack of a twisting force occurring in a double wishbone suspension and no torsion bar in our design we must use the spring in our damper system as an example of torsion. It is important to know that the following calculations below are done for a solid circular bar of diameter 4.5mm, this diameter is equal to the spring wire width.

Firstly, we calculate the length of the helix using the formula shown where P represents the pitch, which is known to be 9.5mm, and the H is height also known to be 100mm. Plugging these values in with diameter gives a helical length of 1300mm.

Now that the length is known the angle of twist must be calculated and is done be rearranging the equation given. A torque is also calculated using our found force at this angle (See Buckling Analysis) and this was used to give an assumed torque of 36.8Nm. These overall outputs an angle of twist of 15 rad.

Finally, these can be added to the last equation to give us the shear force is the solid bar.

The Material of the spring is steel which has a shear modulus of 77GPa

$$L = \frac{H}{P}\sqrt{(\pi D)^2 + P^2} \qquad H = 100mm$$

$$L = 1330mm \qquad D = 40mm$$

$$P = 9.5mm$$

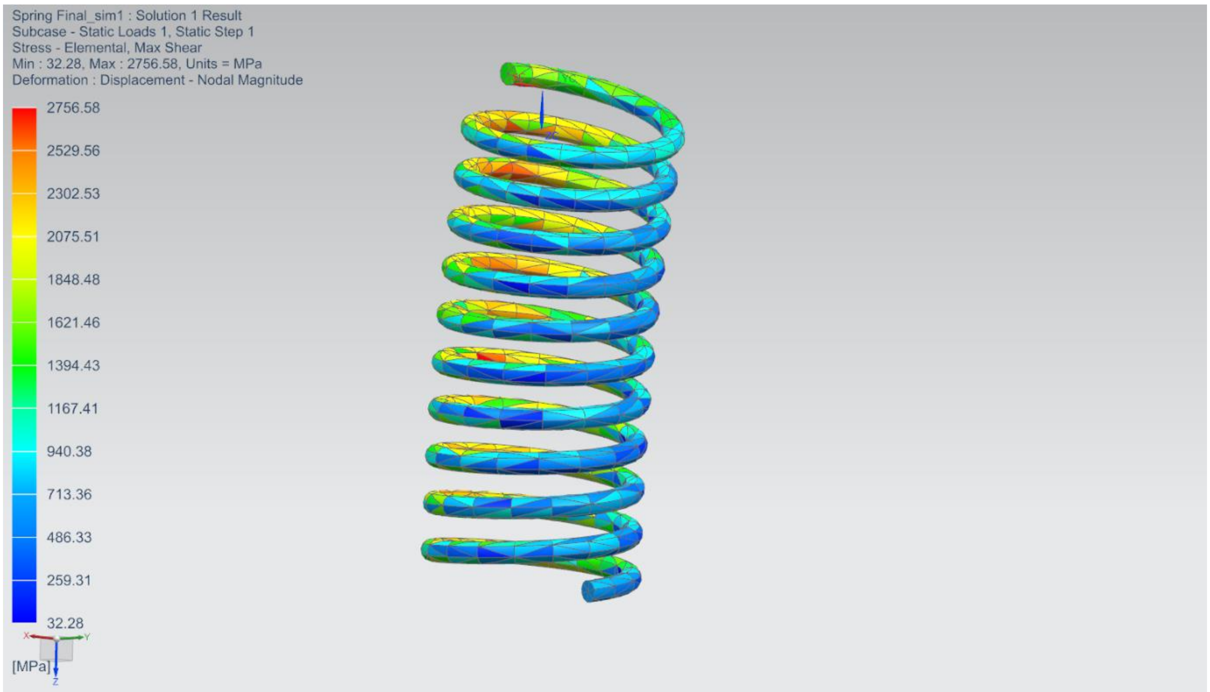
$$\theta = \frac{TL}{JG} = \underline{15 \text{ rad}} \qquad T = 36.8 \text{ Nm}$$

$$L = 1330mm$$

$$G = 77Gpa$$

$$J = \frac{\pi D^4}{32} \text{ or } 40.257 \text{ mm}^4$$

$$\tau = \frac{G\theta r}{L} = \frac{(77\times10^9)(15)(2.25)}{1330} = 1.95\times10^9 \text{ Pa}$$

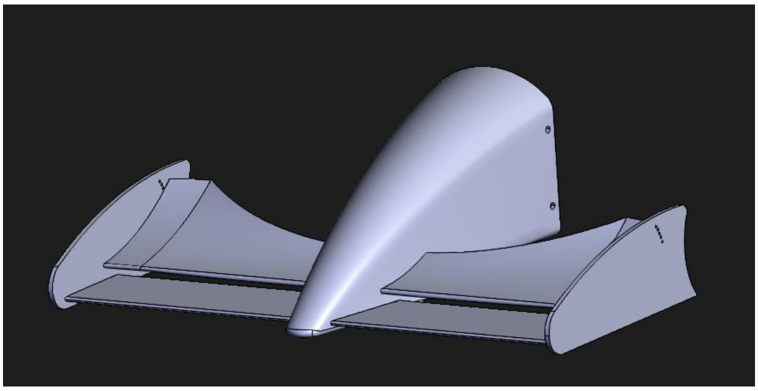


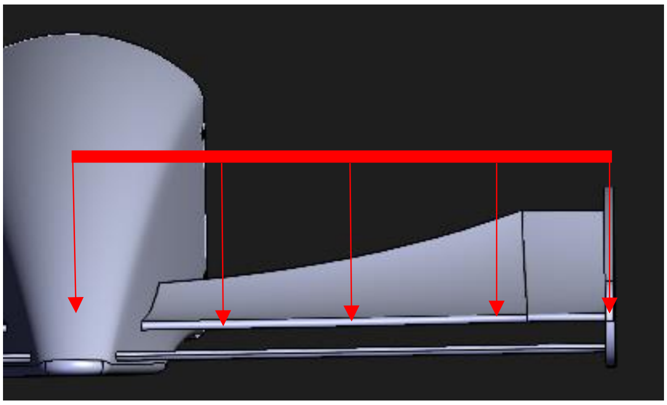
Notice the major change the simulated and calculated max shear stress. Reasoning for this would mainly be due to a difficulty in calculating and generating an accurate shear using the same torsion in the spring. Constraints would be at the base of the spring where the damper would stop it and a vertical force would be acting evenly on the top from the damper pin as a result of the vehicles weight. This cannot be simulated through NX despite uniform forces being implemented downwards in the above image.

Section 3

We have decided to base our hollow box model on a Formula 1 front wing. We will use a cantilever beam design to analyse the forces.

The wing is fixed to the rest of the chassis at the nose making the sides ideal for cantilever beam analysis. Most Formula 1 wings are very complex, however for simplicity we will assume that downforce is equally distriburted either side of the nose towards the endplate.



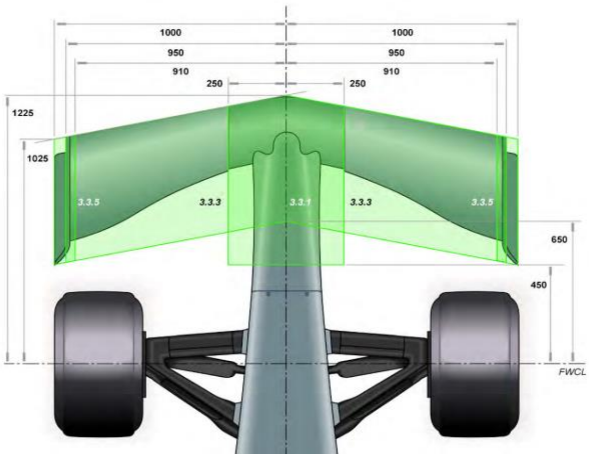


'The front wing assembly, including the endplates, runs to the full width of the car – 1000mm each side of the centreline.' (the-race.com) This is the length we will use in the model.

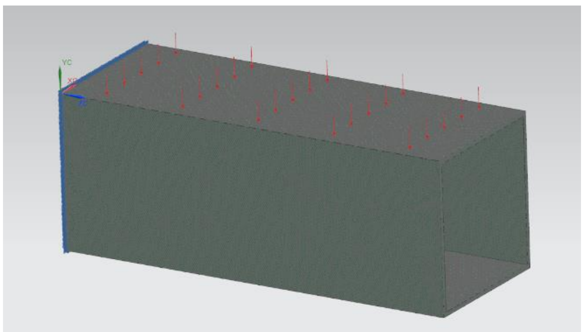
'The front wings produce around 25 per cent of the down force.' (f1chronicle.com)

'Modern Formula One cars can typically generate around 750 kg — or 1,653 pounds — of downforce at speeds of 100 mph.' (continentalautosports.com)

Using this information, we can estimate that half of an F1 front wing produces approximately 920N of downforce at 100 mph. The measurement guidelines provided by the FIA allow us to estimate the top surface area of half of the wing to be 370,000mm^2 or 0.375m^2

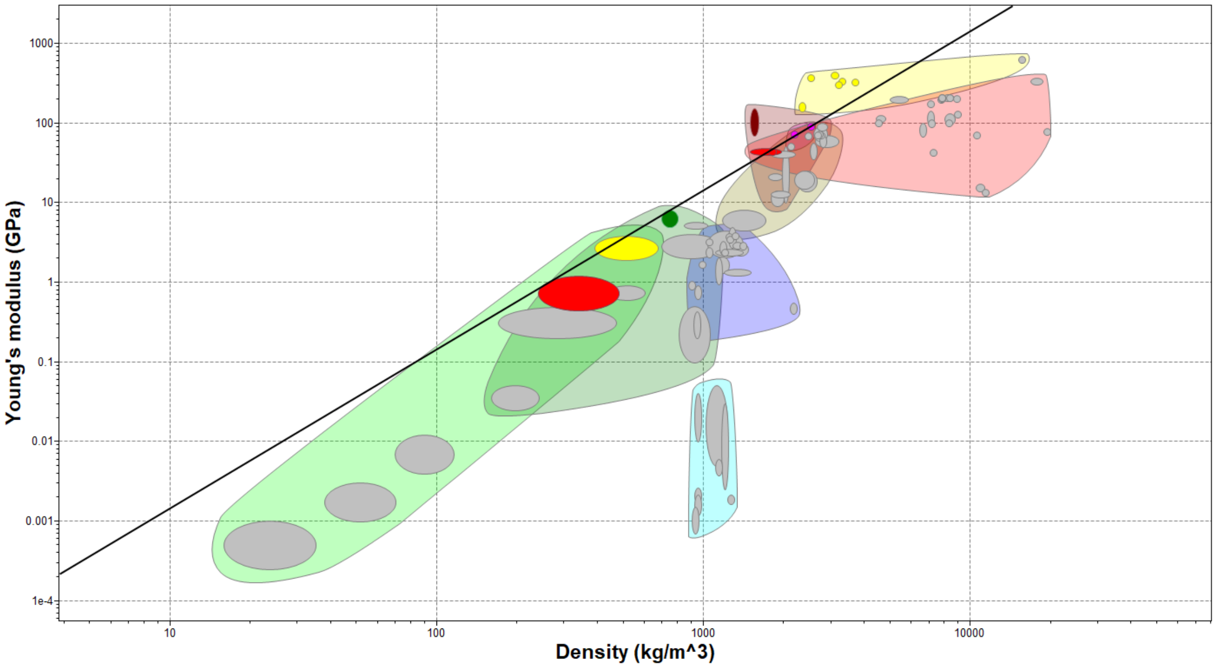


Here is an example of how the hollow box will be simulated. To obtain a deflection estimate we used a wall thickness of 15mm and carbon fibre for the young's modulus. The deflection estimate came out to be 1.25mm



$$\delta_{max} = \frac{wL^4}{8EI}$$

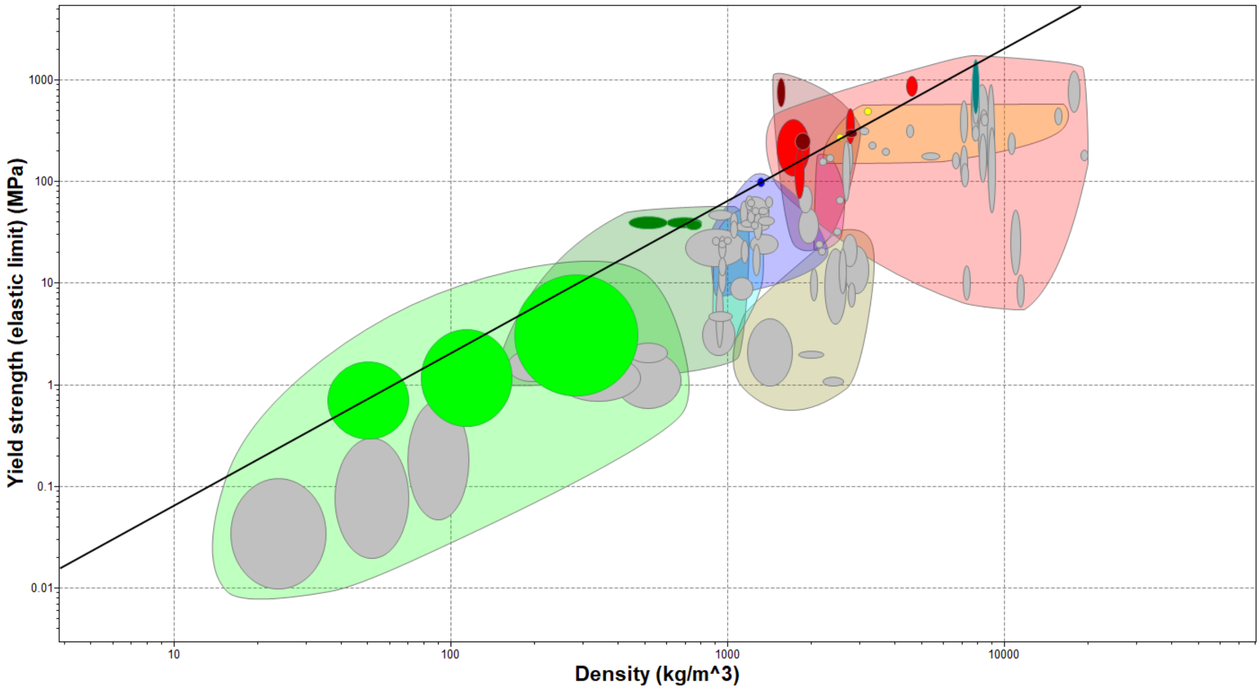
Material Index for Stiffness



Using a plot of young's modulus against density; we can narrow down the most suitable materials to create the index. We can remove any materials with a young's modulus of 10 as it is evident that these materials will not suffice.

Material	Young's Modulus GPa	Density kg/m^3	$k^U \times 10^{-3}$	Embodied Energy MJ/kg	CO2 Emissions kg/kg
CFRP	109.5	1550	6.8	689	48.2
Boron carbide	371	2520	7.6	161	8.7
Aluminium nitride	335.5	3300	5.5	232.5	12.6
Alumina	330	3700	4.9	51.1	2.8
Silicon	160	2330	5.4	129.5	10.9
Silicon carbide	400	3100	6.5	120	7.1
Silicon Nitride	310	3195	5.5	167.5	7.2
Glass ceramic	91	2530	3.8	39.4	2.3
Silica glass	73	2195	3.9	37.6	2.1
Cast magnesium alloys	44.5	1810	3.7	322.5	45.6
Wrought magnesium alloys	44.5	1725	3.9	317.5	44.7

Material Index for Strength

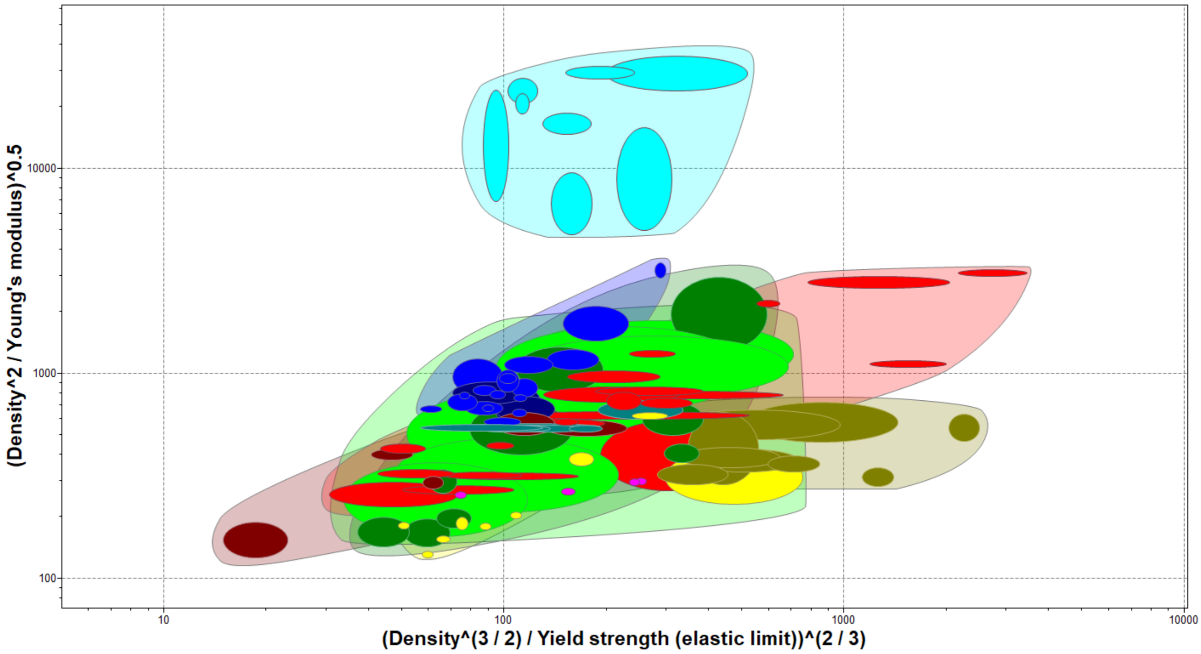


By using a plot of yield strength against density, we can select the most appropriate materials.

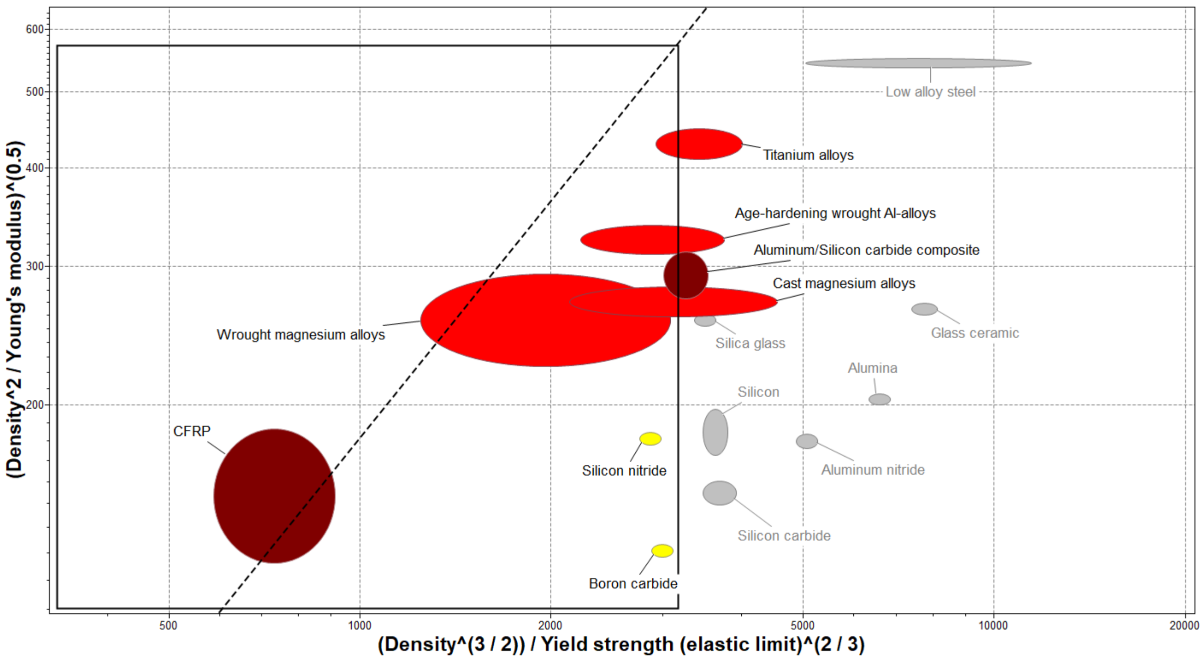
Material	Yield Strength MPa	Density kg/m^3	K^U x 10^2	Embodied Energy MJ/kg	CO2 Emissions kg/kg
CFRP	800	1550	5.6	689	48.2
Boron carbide	275	2520	1.7	161	8.7
Silicon nitride	500.5	3195	2	167.5	7.2
Titanium alloys	895.5	4610	2	621.5	35.9
Wrought magnesium alloys	262.5	1725	2.4	317.5	44.7
Cast magnesium alloys	142.5	1810	1.5	322.5	45.6
Aluminium/Silicon carbide composite	302	2780	1.6	199.5	12.2
Age-hardening wrought Al-alloys	380.5	2755	1.9	196.5	13.1
Low alloy steel	1034.5	7800	1.3	31.1	2.5

There are 11 materials used in the stiffness index graph, and 9 materials used in the strength index graph, with 5 of these materials overlapping. This gives us a total of 15 materials.

Coupling the Material Indices



We can then plot our 15 selected materials on a separate graph.

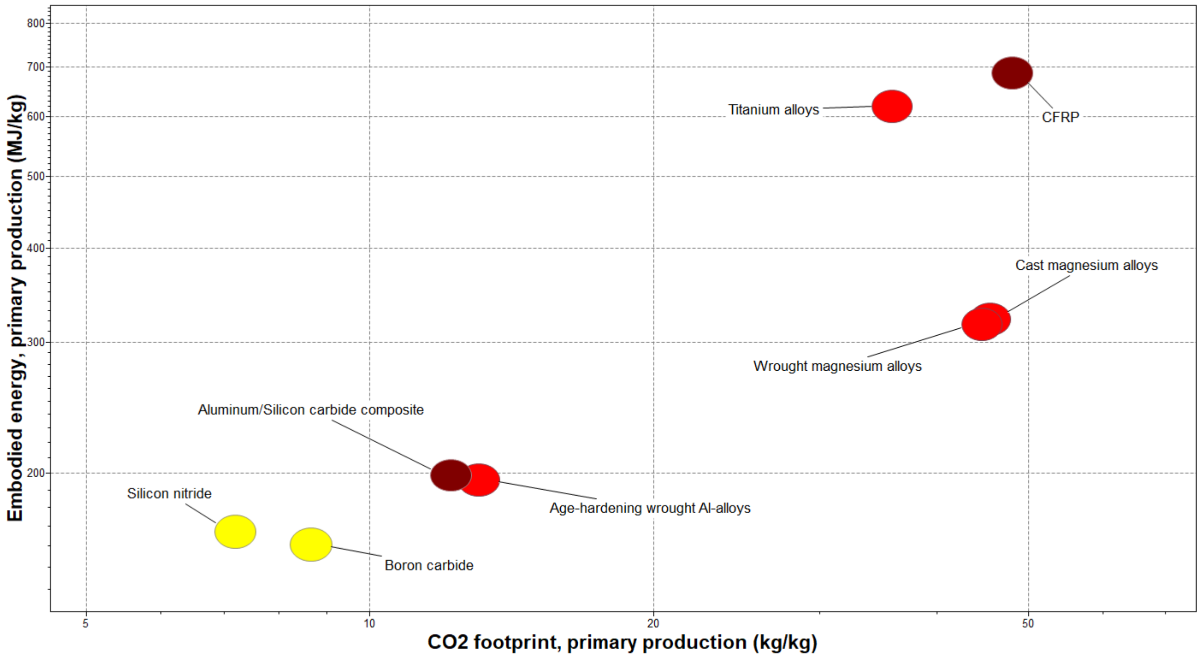


Using our deflection estimates from the first step, we can generate an equation for the coupling line

$$M_1 = \frac{\left(L^{\frac{5}{3}}\right)(6P)^{\frac{2}{3}}}{2\left(\frac{P}{\delta}\right)^{\frac{1}{2}}\left(L^5\right)^{\frac{1}{2}}} M_2 = (4.5)^{\frac{1}{3}}(L)^{-\frac{5}{6}}(P)^{\frac{1}{6}}(\delta)^{\frac{1}{2}} M_2$$

$M_1 = 0.182 M_2$

Using the selection box allows us to further narrow down our list of sufficient materials. To consider the environmental impact, we plotted embodied energy against CO2 emissions.



To choose the three most suitable engineering materials we will consider the environmental impact along with the combined material indices.

CFRP is the first material we will choose to model. It is the least environmentally friendly out of our eight remaining materials, however the strength and stiffness properties are significantly higher that any of the other materials. This is what makes it a suitable choice.

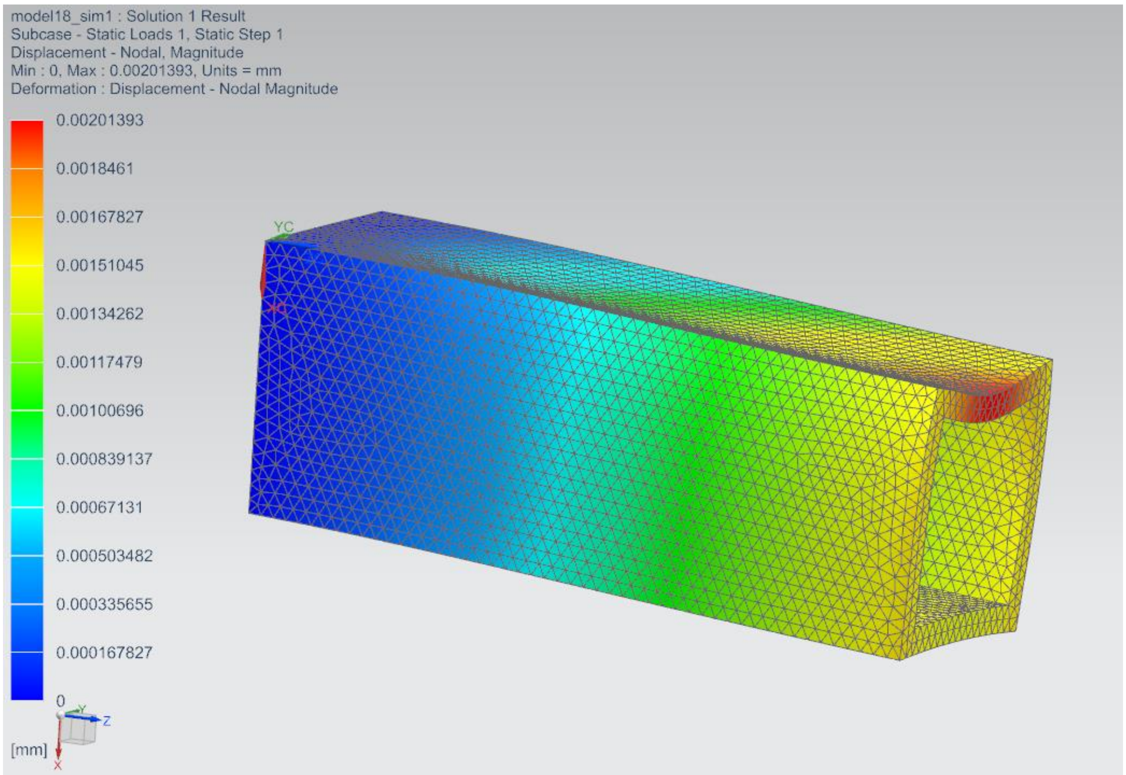
Our next selected material is boron carbide. It performs well overall and has much less environmental impact than the other materials due to its low embodied energy and CO2 emissions. What may hold it back is its relatively low yield strength compared to our other selected materials.

Our final chosen material is silicon nitride. It has arguably the least environmental impact out of all our materials, while performing well overall when considering strength and stiffness. It does however have the highest density of the three. This could hold the material back depending on the necessary wall thickness.

NX Calculations

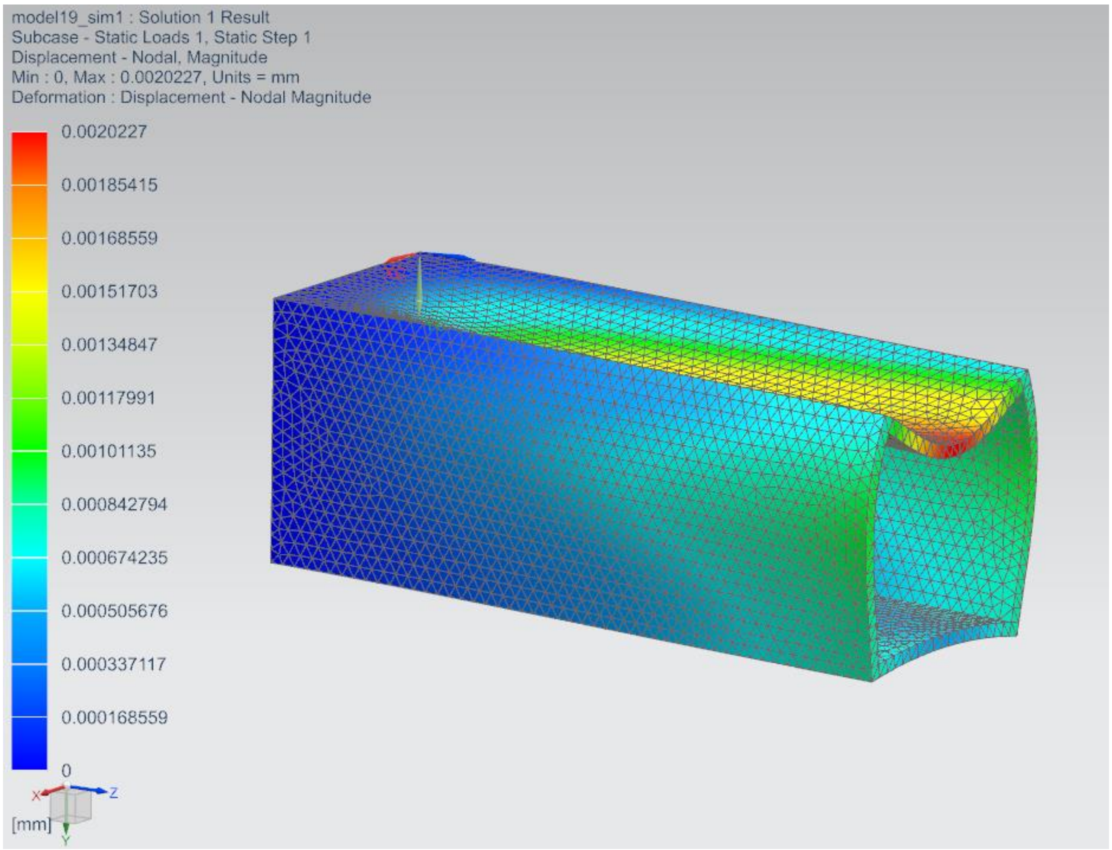
To determine whether each individual hollow box is suitable for our specific loading scenario, we will set criteria that each model must meet. We decided for a material to meet our criteria; it must undergo 0.002mm (+/- 0.00005mm), of maximum nodal displacement. This will allow us to directly compare each material. The only variable will be wall thickness. To keep our results consistent, we will use a 3D tetrahedral mesh with 20mm elements for each simulation. The length, width, and height of the beam, along with the loading will also remain consistent.

CFRP Simulation



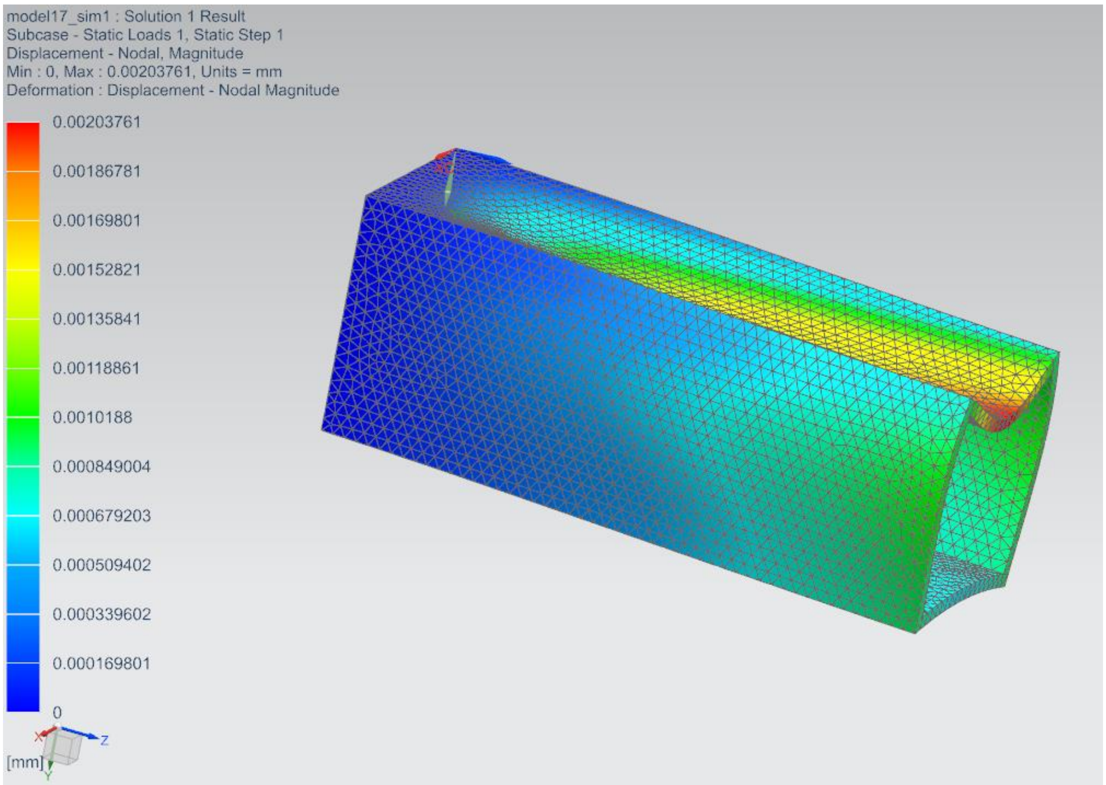
Chosen wall thickness – 37.5mm

Boron Carbide simulation



Chosen Wall thickness – 19mm

Silicon Nitride Simulation



Chosen wall thickness – 20.6mm

Results Table

Material	Wall Thickness (mm)	Box Volume (m^3)	Box Mass (kg)	Embodied Energy per Box (MJ)	CO2 Emissions per Box (kg)
CFRP	37.5	0.0506	78.47	54065	3782
Boron carbide	19	0.0271	68.18	10977	593
Silicon nitride	20.6	0.0292	93.30	15628	1176

It is evident that boron carbide is the best material for our specific scenario. It is best in all categories and has a significantly lower carbon footprint than the other materials, particularly CFRP. However, in real- world scenarios this may not be the case. Boron carbide is relatively brittle and is very difficult to machine making it less ideal for complex designs. The material has specific uses, such as high-speed body armour, but is likely to not be used in a complex loading design.

CFRP is the material most likely to be chosen for a real-world loading scenario. It is commonly used for its high strength-to-weight ratio and exceptional fatigue resistance. However, it also has its drawbacks as shown in our results table it is not environmentally friendly to manufacture. It is also expensive to produce at £28.6 - £31.8 / kg.