Design for a Convertible Roof of a Fiat 595 Abarth
Design Report

Group 21B

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1. Introduction

This report aims to describe a new design of a convertible roof to be integrated into the Fiat 595 Abarth. The design is meant to be a direct competitor to the BMW convertible Mini Cooper while also following the constraints provided by the company.

The primary stakeholders in the project are the technical lead, technical designer, customer, and Fiat itself; these groups heavily affect the viability and success of the project, albeit in different ways: the technical designer is responsible for designing and delivering a functional product, the technical lead will determine the viability of the concept and decide whether to move forward with the design, the customer will be the one buying the convertible, and Fiat will be the one manufacturing the convertible if the design is used.

Other stakeholders include the companies making the standard components, which are Bosch and KHK gears. These groups do not affect the outcome of the project as much as the primary stakeholders. Bosch and KHK handle supplying the electric motors and gears, respectively. Stakeholder Analysis is shown in Table 6.1 of the Appendix.

The design followed several company constraints, with safety of passengers being the highest priority. It is also important that the roof does not change the dimensions of the existing model by an unreasonable amount, and that the cost of the new convertible model is not unreasonably more than the current model.

In order to make an effective Product Design Specification as seen in Table 2.1 of section 2 below, a few key assumptions had to be made. The first was that a typical mechanism driving the roof lasts about 10 years, and that the roof material itself, usually fabric or vinyl for soft-top convertibles, lasts around 5 years. It is also assumed that the roof would be able to operate in relatively extreme weathers, such as heavy snow or extreme heat. Another key assumption is that the convertible model will not add more than £3300 to the current model, which is also exactly how much more expensive the Mini Cooper convertible model is compared to the normal model.

2. Product Design Specification (PDS)

No.	Category	Requirement	Target/ limit	Ranking	Test Method
1	Performance	Roof deployment time	15 s	5	MATLAB modelling
2	Company Constraint	Must use existing battery to power the mechanism	12 V	5	MATLAB modelling
3	Company Constraint	Must only use rotational joints		5	Linkage Simulation
4	Company Constraint	Must use single DC motor		5	MATLAB modelling
5	Size	Maximum length when deployed	< 1640 mm	5	Linkage Simulation
6	Size	Maximum length when retracted	< 670 mm	5	Linkage Simulation
7	Size	Maximum height of 360mm when retracted	< 360 mm	5	Linkage Simulation
8	Weight	Roof mass must not add more than 100kg to original mass of	<100 kg	5	Calculate mass of the roof
9	Company Constraint	Production cost must not add more than £3300 to existing	~£19,065 + £3300	5	Costing Analysis

10	Safety	Mechanism must cease motion if power is lost		5	Integrate safety feature
11	Safety	Roof must not cause harm to passengers in the event of		5	Integrate safety feature
12	Environment	Mechanism must work in common operating conditions		5	Analysis of mechanism behaviour
13	Life in service	Mechanism must have a life span of at least 10 years	10 years	5	Materials life span
14	Life in service	Roof material must have a life span of at least 5 years	5 to 7 years	5	Materials life span
15	Materials	Must be made of recyclable materials		5	Research recyclability
16	Performance	Roof should be able to deploy and retract open to relative air	30 kph	4	Aerodynamic analysis
17	Safety	Mechanism must be able to detect and stop if there is an		5	Integrate safety feature
18	Safety	Must open at safe height above passengers' height	150 mm above headrest	5	Simulation in Linkage
19	Company Constraint	Profile of the roof must not deviate more than 10 mm to current model	10 mm	3	MATLAB Modelling
20	Materials	Roof must be a soft top made from flexible material		5	Simulation in Linkage Software

Table 2.1: Product Design Specification (comprehensive table in appendix)

Product design specification was generated after reviewing the project brief and doing market research. Requirements 1, 8, 12, 13, 14, and 16 in the PDS were generated based on the product's direct competitor (BMW Mini Cooper). Safety is the highest priority type of requirement. Concept generation and selection was based on specifications listed in the PDS.

3. Concept Development, Testing and Selection

Initially, four concepts were generated (shown deployed and retracted in figs. 3.1-3.4) and modelled in Linkage (Rector, n.d.). Market research proved a useful tool in the concept generation phase; a scissor-style mechanism was found to be common in existing convertibles, such as certain Mini Cooper and the Ford Mustang models. This was the inspiration for the 'pantograph' concept, which therefore had the advantage of being proven in practice. This was also the origin of the flip concept, with similar designs found in cars such as the Saab 9-3 or Mercedes-Benz CLK. Using these tried and tested designs as a basis, a goal was set to improve on them in key areas; the high clearance of the flip mechanism was identified as a weakness, which led to the development of the 'compact flip' concept, which reduced the necessary clearance by over 50% (table 3.1) by splitting the long front section into two. This came at the cost of much higher complexity, however. With this in mind, the aim of the fourth concept was to minimise the required number of links, and therefore cost and complexity. While both the pantograph and flip configurations used 7 links, the '6 bar' concept used only 5, although it failed to impress in other key metrics such as clearance and stowed size. Improvements were attempted in other areas, most notably stowed size, for which the flip mechanism was found to be near-optimal; as such, no significant improvement could be made.

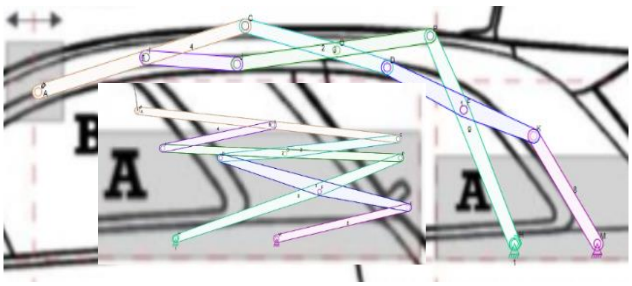

Figure 3.1 – Pantograph mechanism

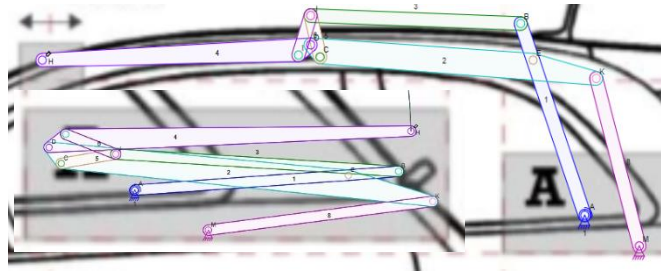

Figure 3.2 – Flip mechanism

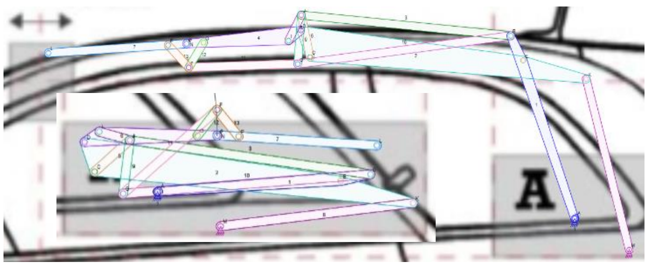

Figure 3.4 – Compact flip mechanism

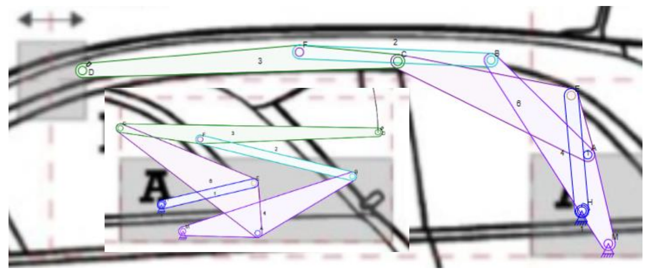

Figure 3.4 – 6 Bar linkage mechanism

With the four candidate designs finalised, four key metrics were measured from Linkage: link count, clearance (peak height of roof above initial position), stowed height, and shape deviation (max difference between original roof shape and deployed mechanism shape). These can be found in table 3.1.

Table 3.1 – Key comparison metrics

Mechanism	Link count	Joint count	Clearance (m)	Stowed height (m)	Shape deviation (m)
Pantograph	7	8	0.21	0.23	0.08
Flip	7	7	0.58	0.16	0.07
Compact flip	13	14	0.24	0.22	0.06
6 bar	5	5	0.56	0.28	0.09

In order to compare the concepts as fairly as possible, a list of criteria based on the requirements set out in the PDS was generated and weighted as shown in table 3.2.

Table 3.2 – Criterion weighting

Criteria		Roof Clearance	Manufacturability	Safety	Estimated Cost	Complexity	Stowed size	Mass	Deployed Geometry	Stowed Geometry		
	X	1	2	3	4	5	6	7	8	9	Total	%
Roof Clearance	1	X	0,33	0,11	0,33	0,33	0,33	1,00	0,11	0,33	2,89	2
Manufacturability	2	3,00	X	0,33	1,00	3,00	1,00	3,00	0,33	1,00	12,67	9
Safety	3	9,01	3,00	X	3,00	3,00	9,00	9,00	3,00	9,00	48,01	33
Estimated Cost	4	3,00	1,00	0,33	X	3,00	3,00	9,00	1,00	3,00	23,34	16
Complexity	5	3,00	0,33	0,33	0,33	X	0,33	3,00	0,33	1,00	8,67	6
Stowed size	6	3,00	1,00	0,11	0,33	3,00	X	3,00	0,33	3,00	13,78	9
Mass	7	1,00	0,33	0,11	0,11	0,33	0,33	X	0,33	0,33	2,89	2
Deployed Geometry	8	9,01	3,00	0,33	1,00	3,00	3,00	3,00	X	3,00	25,35	17
Stowed Geometry	9	3,00	1,00	0,11	0,33	1,00	0,33	3,00	0,33	X	9,12	6
		34,03	10,01	1,78	6,44	16,67	17,34	34,01	5,78	20,67	146,72	100
		23,2	6,8	1,2	4,4	11,4	11,8	23,2	3,9	14,1	100	146,72

Safety was assigned the heaviest weighting, as it is the highest design priority, followed by deployed geometry, estimated cost, stowed size, manufacturability, stowed geometry, and mass & roof clearance, which scored the same. The concepts were then assigned scores from 1 to 5, with 1 being poor and 5 being excellent, based on their performance in relevant metrics. Safety was scored based on the likelihood of injury in the event of an unexpected failure in either the mechanism or the safety systems; while none of the concepts are obviously unsafe, the pantograph mechanism scored slightly lower due to the danger of trapping fingers or hands (risk of scissor-like effect), and the compact scored similarly due to its extra weight above the cockpit. Deployed geometry is based on shape deviation, however compact flip's score was adjusted due to being considerably thicker than the original roof in places. Mass was calculated based on the total length of linkages (assuming a constant mass per metre). Manufacturability was approximated by total number of joints. Cost was estimated based on highest link complexity (number of joints on a single link) plus number of links. Stowed size is the same as stowed height. Stowed geometry is based on the number and severity of overlaps which may in practice damage the roof material (ranking, not numerical). Complexity is number of links.

These scores and weightings were added to a pairwise comparison matrix (table 3.3), in which the 6-bar linkage performed best.

Table 3.3 – Pairwise Comparison Matrix

Comparison Matrix												
	Criteria	**Roof Clearance**	**Manufacturability**	**Safety**	**Estimated Cost**	**Complexity**	**Stowed size**	**Mass**	**Deployed Geometry**	**Stowed Geometry**	***ERROR***	
	Weights	0.020	0.086	0.327	0.159	0.059	0.094	0.020	0.173	0.062		
Options		**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	Total	%
Pantograph mechanism	**1**	5	3	3	4	4	3	4	3	2	3.22	24
Flip mechanism	**2**	2	4	4	3	4	5	4	4	4	3.90	29
Compact flip mechanism	**3**	4	1	3	1	2	4	3	3	3	2.56	19
6-bar linkage mechanism	**4**	2	5	4	5	5	2	5	3	5	3.99	29

Hence, the 6-bar concept was selected.

4. MECHANISM DESIGN DEVELOPMENT

4.1. Damper and Spring Selection

The system was initially given a placeholder mass of 20kg as mentioned in the first paragraph of the second page of the Technical Analysis report. However, without the dampers, the system would deploy and retract far too quickly than safety would allow. This resulted in the addition of a rotational damper and two springs to slow down the mechanism and make sure the speed at which it rotates would remain constant over the course of its motion which was modelled within the MATLAB code as can be seen in pages 2 and 3 of the Technical Analysis report.

The rotational damper would be placed between the gearbox and the motor. It would damp the motion of the motor directly and reduce the overall torque of the mechanism depending on the motor's velocity. Based on the MATLAB simulation, the maximum speed of the system was 0.24 rad/s and the effective damping torque at that point was 140Nm. Converting through with the gear ratio and 315.8 and taking efficiencies into account, a maximum motor speed of 75.8 rad/s and damping torque of 0.574 Nm at was needed. Using the Kinetrol Dashpot catalogue (Kinetrol, 2022), an appropriate rotational damper with part number Q-CRD-5000 with a fluid viscosity of 5000 cSt was chosen.

The rotational damper slowed the system down significantly, but the speed throughout the motion was still accelerating. Springs were used to achieve the desired constant speed. Initially, a single, random compression spring was chosen from the Raymond Associated Springs compression springs catalogue (Compression Springs Product Catalogue, 2020) and modelled into MATLAB, with it being attached vertically to the body of the car right behind the anchor of Link EH as shown in the Assembly Drawings. However, this only produced a constant speed during the second half of the retracting motion and sped up the initial deploying motion due to the direction of the force exerted by the spring. To solve this, an extension spring was chosen from the Raymond Associated Springs Stock Precision Engineered Components catalogue (Stock Precision Engineered Components, 2020) and modelled into MATLAB, with it being attached horizontally behind Link EH, and its height about halfway from the anchor of that link as shown in the Assembly Drawings. Multiple springs from the catalogue were tested until an appropriate compression spring (C1234-162-7500M) and extension spring (part number E1750-177-5000M) which resulted in an almost constant speed throughout the motion.

4.2. Motor and Gear Ratio Selection

Based on the design brief, a motor without transmission was to be used for the design. Since the motor needed to be able to rotate in both clockwise and counter clockwise directions and come without a transmission, the selection was initially narrowed down to four motors: APM and three NSA-I motors. All the motors were modelled and simulated in MATLAB. After all the extensions were included within the model (damping, gravity, variable radius, and aerodynamics), it was found that all motors were viable provided each one used a different gear ratio. The holding ratio required for each motor, which is the minimum gear ratio needed to hold the mechanism at its start position and overcome gravity, was calculated using the equation below:

$$Holding\ Ratio = \frac{(mgrcos(\theta))}{T_S} \tag{1}$$

Where: m is the mass of the system; g is the gravitational constant; r is the radius of the centre of mass from the pivot point; θ is the angle of the centre off mass from the horizontal; and Ts is the stall torque.

From this, the APM motor was screened out due to having a relatively high holding ratio of 397.5 due to a low stall torque of 0.19 Nm. Taking all the extensions into account, the motor would require an even higher gear ratio to drive the mechanism in 10 seconds, which would increase the stages and cost of the gearbox. The three NSA-I motors had reasonably low holding ratios below 200 due to its relatively higher stall torques compared to the APM motor. However, the first motor (part number 0 390 204 092) was selected due to it having the lowest nominal current, which means it would use the least power out of the three. It is also the smallest out of the three motors, which would save space in the final design of the mechanism.

From this, a holding ratio of 175.6 was needed for the selected motor. A placeholder gear ratio range of 200 – 400 was initially used as can be seen in the figures within the Technical Analysis report. A target of 10 seconds was set for the mechanism to be able to deploy and retract. Through the modelling of the extensions mentioned above, the range was narrowed down to around 300 – 320, which was a small enough range to be able to start designing the gearbox.

4.3. Gearbox Design

When designing the gearbox, three main factors were considered: cost, size, and performance. To achieve a velocity ratio of 300-320, either 5 spur pairs or 1 worm & wheel and 1 spur pair are needed (spur pairs can reach ratios of up to 6:1, worm & wheel can reach 300:1). While the spur-only configuration has an edge in performance, transmitting torque with an estimated efficiency of 90% versus 77% (based on 0.98 for each spur pair and 0.78 for the necessary worm & wheel, from chart (Beardmore, n.d.)), this is outweighed by the bulk and cost of such an arrangement. Additionally, the increased complexity of a spur-only gearbox may lead to complications during manufacture and repair. So, a ‘worm and wheel plus one’ arrangement was selected.

Parts were selected from the HPC catalogue because it provided concrete pricing and a wide range of options. A spreadsheet was created using the manufacturer’s technical selection documents (HPC Gears, 2022) (referred to as TSD henceforth) to streamline the selection process. Note that the following gear formulas are designed to be used with imperial measurements (ins/lbs). For the worm and wheel, the limiting factor is allowable torque. This was calculated for both strength and wear, according to HPC’s recommendations, using equations (2) and (3) respectively.

$$T_S = 1.8S_bX_bL_rMDcos\alpha \tag{2}$$

Where T_S is allowable torque for strength, $S_b X_b$ are table values for material and speed respectively from TSD, L_r is the length at the root of the wheel's teeth, M is the addendum, D is the wheel's pitch diameter, and α is the worm's lead angle.

$$T_W = 0.18 S_c X_c efw D^{1.8} \tag{3}$$

Where T_W is allowable torque for wear, $S_c X_c$ are table values from TSD, efw is effective face width of wheel, and D is the wheel pitch diameter.

These values were compared with the actual torque, found using equation (4), to find an effective safety factor for the worm and wheel.

$$T_A = T_m R_r \eta \tag{4}$$

Where T_A is estimated peak torque, T_m is peak motor torque (from model), R_r is reduction ratio, and η is the worm and wheel efficiency.

A target safety factor of 2 was selected, and after some preliminary tests (sampling promising combinations from the module 1, 1.25, 1.5, and 2 catalogues) a module of 1.5 with 1 start on the worm was found to provide the best balance between size, cost, and strength. The allowable torque due to wear was found to be a bottleneck; this was negated by selecting the hardened version of the worm (~30% cost increase for worm/wheel, alternative solution, going up to next wheel mod available, would increase cost by ~240% and add size). Then, the reduction ratio which provided the closest safety factor to 2 was chosen, in order to optimise cost (price increases exponentially with wheel size). This gave a reduction ratio of 50, at a safety factor of 1.97 (RR 52 gave SF 2.03, final decision based on low cost option). The chosen parts were SWH1.5-1 and M1.5-50 for worm and wheel respectively.

To select the spur pair, a target gear ratio of 6 was set based on the worm/wheel combination. Instead of torque, the manufacturer recommended basing selection on allowable tangential load, again for strength and wear, calculated with equations D and E respectively. Actual load was calculated using equation (5) (TSD).

$$F_S = \frac{X_b S_b Y F}{DP} \tag{5}$$

Where F_S is allowable tangential load for strength, $X_b S_b$ are table values from TSD, Y is strength factor (also table value), F is face width, and DP is $25.4 * MOD$.

$$F_W = \frac{X_c S_c Z F}{K} \tag{6}$$

Where F_W is allowable tangential load for wear, $X_c S_c$ are table values from TSD, K is pitch factor ($DP^{0.8}$), F is face width, and Z is zone factor (table value).

$$F_A = \frac{2 T_{in} R_r}{pcd} \tag{7}$$

Where F_A is estimated tangential load, T_{in} is the torque in the pinion, R_r is the gear ratio (1 if calculating for pinion), and pcd is pitch diameter.

Achieving a gear ratio of 6 under relatively high loads was more challenging than anticipated, requiring a module of at least 1.75 even with hardened steel in both gears, to reach an appropriate safety factor (from sampling pairs in spreadsheet). In the interest of keeping size within reasonable limits, gear pairs were sampled primarily within the 1.75 module catalogue. It was found that the allowable wear load on the pinion was the limiting factor, with 18 teeth needed to reach a safety factor of 2. Given this constraint, the two nearest eligible combinations (must be over 6 VR, closer is better) were 18 and 110 teeth, giving a gear ratio of 6.1, and 19 and 120 teeth, giving a ratio of 6.3. The former offers slightly lower cost and size at the expense of lower safety factor and poor hunting, which proved the deciding factor. Reliability was deemed the highest priority, as the two options were otherwise very similar in practice (6% safety factor difference, 14% cost difference, 11% size difference). The selected parts were YG1.75-19 and YG1.75-120. Assuming 100-200 units this brings the total cost of catalogue-bought parts per gearbox to £163.09 (£34.59 for worm, £34.71 for wheel, £10.90 for pinion, £82.89 for big spur).

With the components selected, the layout design process was straightforward: the worm and wheel were connected to the pinion of the spur pair by way of a small shaft by necessity, as any other arrangement would add bulk and cost for no real reason. This left only two major design decisions: how the gears should be fixed, and at what angle the final spur gear should be placed. The selected worm is already on a shaft, so only the wheel and spurs need to be fixed. For the spur gears, a simple key with set screws is adequate, but the wheel is also under axial load, so a pair of circlips are needed. The reasoning behind the positioning of the spur gear is to allow the motor plenty of space to attach to the gearbox from the side, and to minimise the already large (245x240.15mm) side profile. The size advantage of the worm and wheel configuration is found in its depth; the entire gear and shaft layout is only 64mm thick (see gearbox technical drawing). The orientation of the gearbox has been left open-ended – should development continue and the chassis be modified it can be fixed at whatever angle suits the space available.

4.4. Material Selection

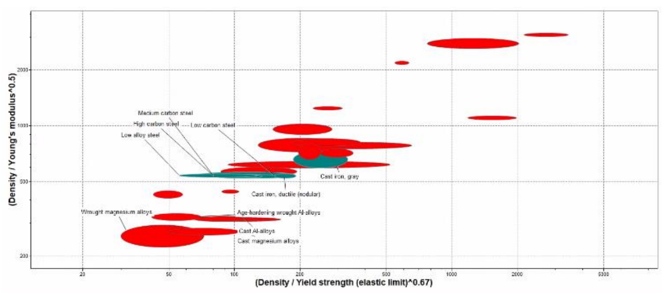

Figure 4.1(a): Minimising Mass Objective

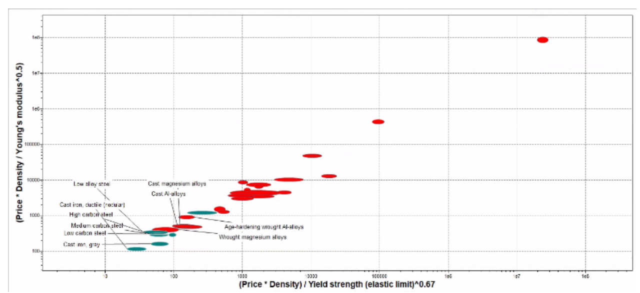

Figure 4.1(b): Minimising Cost Objective

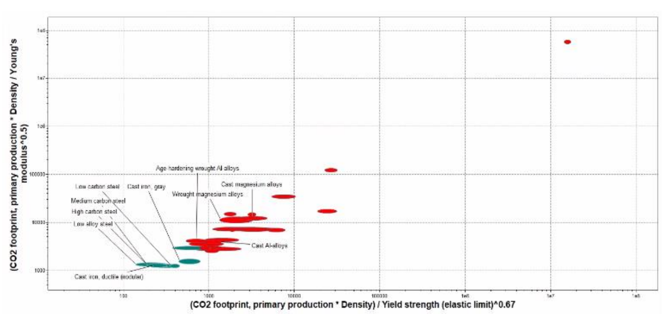

Figure 4.1(c): Minimising CO2 Footprint Objective

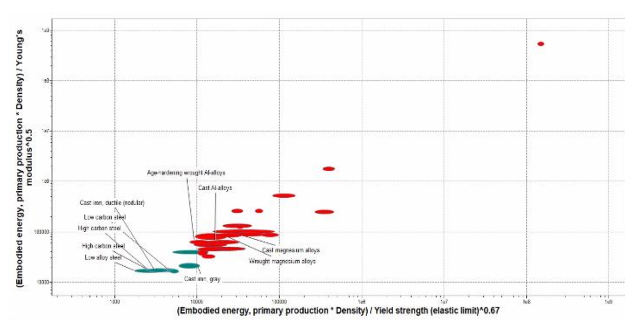

Figure 4.1(d): Minimising Embodied Energy Objective

Edupack (Ansys, 2022) was used to compare the different metals, coloured blue, and alloys, coloured red, against each objective and constraint. Graphs showing how each material performs against the two constraints for each one of the four objectives can be seen in Figure 4.1 above. The better materials can be found towards the bottom left corner.

From Figure 4.1(a), it is seen that magnesium and aluminium alloys are the lightest available materials. Several convertible roof frames are supposedly built using aluminium alloys (European Aluminium Association, 2013). However, looking at Figure 4.1(b), those alloys may not be the cheapest option and the materials coloured in blue cost less. Figures 4.1(c) and (d) also support this as steel and iron produce a much lower CO2 footprint and have less overall embodied energy compared to the alloys. Although the alloys are lighter compared to steel and iron, minimising cost is of higher priority as the cost of the proposed model must not be too far greater compared to the cost of the current Fiat 595 Abarth model as mentioned in the Product Design Specification.

The remaining materials were then screened by their typical applications and general properties. Gray cast iron was screened out due to its relatively low yield strength and elongation, and being more brittle compared to the other remaining materials. Low alloy steels were also screened out due to it mostly being used for very high-strength components, much higher than what is usually needed in structural components such as the frame of this convertible (Ansys, 2022). Out of the three types of carbon steels, low carbon steel is the one mainly used for structural frames (Aluminium Warehouse, 2022) while the higher carbon steels are usually utilised for its higher strength, hardness and wear-resistance such as in cutting tools and wires (Reliance Foundry, 2022).

From this, low carbon steel and ductile iron are the two remaining materials. Although low carbon steel has a slightly higher yield strength and stiffness, ductile iron is a much cheaper material, costing on average £ 0.219 per kg, while low carbon steel can cost anywhere from £ 0.537 to £ 0.566 (Ansys, 2022). In some cases, low grade carbon steel can be replaced by ductile iron due to the latter's lower overall cost, provided their properties are still within the required constraints (Dandong Foundry, 2022).

To ensure that the mechanism will not structurally fail when using either material, two FEA simulations were carried out in Fusion 360 for the roof frame; one for each material. The simulations included external loads, loads due to self-weight, and bending moments due to the rotating motion of the mechanism. From these, a minimum safety factor of 10 was obtained for both simulations, meaning that both materials are suitable to be used for the roof frame

4.5. Fixings and Refining Geometry

All the links were first assumed to have a thickness of 12mm and width of 40mm with 25mm holes to attach the connecting shaft as shown in the Assembly Drawings. To ensure the links could rotate relative to the shaft as modelled on the Linkage software, bushings were chosen from the SKF Bushing catalogue (Svenska Kullagerfabriken (SKF), 2010). A flanged, sintered bronze bushing (part number PSMF 182418 A51) was used due to its good corrosion and vibration-resistant properties according to the catalogue. The flanges provided extra fixing supports at the links.

This bushing had an inner and outer diameter of 18mm and 24mm respectively with 3mm thick flanges. From this, the 25mm holes on the links were resized to 24mm with a h6 clearance fit to allow some rotation relative to the shaft. Stainless steel shafts with an inner and outer diameter of 15mm and 18mm respectively were used to connect links as well as both sides of the mechanism together with steel rods as shown in the Assembly Drawings.

4.6. Final Design Model Output

The final MATLAB simulation outputs can be seen in Figures 4.2, 4.3 and 4.4. The mechanism takes 11 seconds and 9 seconds to retract and deploy respectively as shown in Figures 4.2(a) and 4.2(b). A maximum power output of 25.6W is required, which is achieved during the deploying motion of the mechanism, as seen in Figure 4.3(b). The centre of mass of the mechanism rotates through an angle of 2.08 rad or just under 120 degrees shown in Figure 4.4.

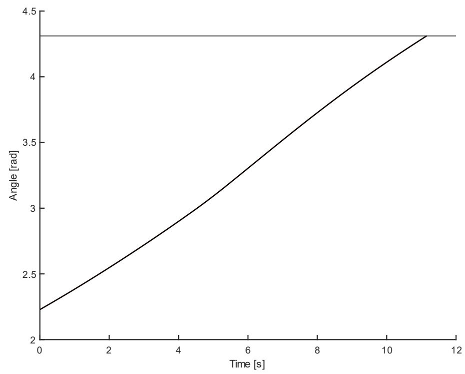

Figure 4.2(a): Angle vs Time for Retracting

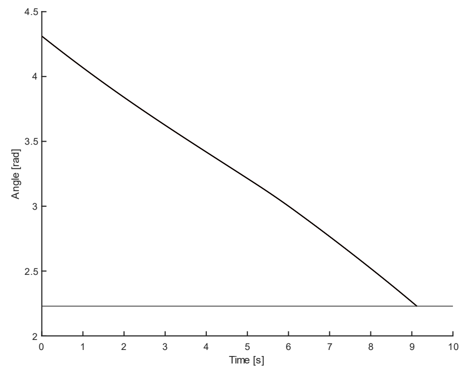

Figure 4.2(b): Angle vs Time for Deploying

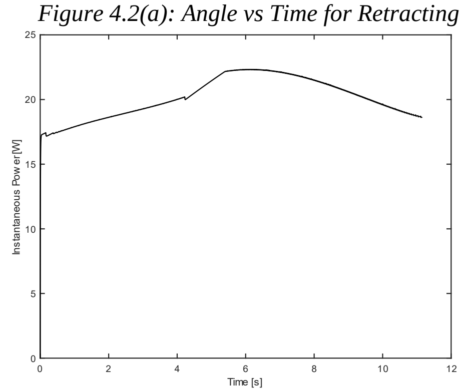

Figure 4.3(a): Power vs Time for Retracting

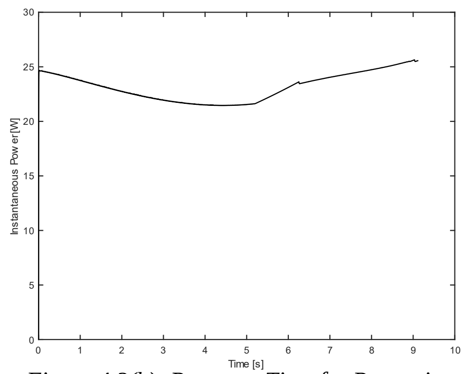

Figure 4.3(b): Power vs Time for Retracting

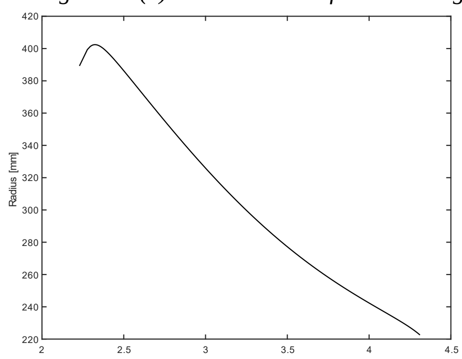

Figure 4.4: Radius of Centre of Mass vs Time

5. Design Evaluation

For the most part, the final design met its requirements (as set out in the PDS) adequately. There were, however, several outliers; on the one hand, the mass requirement (8) was met easily: the gearbox and mechanism combined weighed less than 40kg, possibly suggesting that this requirement was set poorly (estimated based on difference between real-world hardtop and soft top versions of matching models). Similarly, based on the model results the roof could operate in winds exceeding 50km/h, far higher than required (17). However, the basic aerodynamic approximation used in the model was unlikely to be particularly accurate, especially at high wind speeds; this margin was cause for confidence in its ability to perform at the required level, but not sufficient evidence to spec it higher. On the other hand, while no requirements were failed outright, the work done was not sufficient to prove that the cost (9) or lifespan (14) of the mechanism fall within acceptable bounds. The former was largely a result of a lack of data; without an idea of the number of units planned, for example, estimating cost per unit is difficult. Using the crude metric that materials account for around 20% of cost per unit and standard raw material (from EduPack (Ansys, 2022)) and component costs, a very rough figure of £1000-2000 was found (broken down as £170 for gearbox, £50 for mechanism, £40 for motor), but this result was deemed effectively meaningless. The latter was found to be problematic due to an overwhelming number of potential failure vectors within the linkage. The modelling process assumed that the mechanism could be approximated by a simple mass and pivot, which meant it could not be used to find the loads and stresses within the linkage during operation, and the complexity of the system's motion made manual calculation unwieldy. A more advanced modelling process capable of fatigue analysis and physical testing are possible methods to adequately assess this metric, but neither was found to be practical.

Regarding the gearbox design, while the individual design methods and rationale were mostly sound, the overall selection process could have been conducted more thoroughly. On reflection, selecting one gear pair first then basing the selection of the second on these results likely missed more compact solutions. Ideally, either some code or a spreadsheet would be used to perform a four-way comparison of every possible gear combination and arrangement, outputting the most compact and cheapest viable combinations. This would have been considerably more time consuming than the method used here, especially as table values for every gear would have to have been input manually, possibly detracting from other aspects of the project as a result, but a more optimal solution would have been found.

On the topic of optimisation, the initial concept selection was limited in its scope – limited time and, potentially, a lack of a clear and full understanding of the project's goals during the early stages may have hampered the quality of selection, especially decisions which were made based on fine margins (e.g., choice between flip & 6-bar concepts). More specifically, useful metrics were missed which could, in retrospect, have improved the quality of the final design, such as the relative torques in each joint or the estimated position of the roof's centre of mass (keeping centre of mass close to the pivot reduces necessary torque output). A more thorough consideration of the project's route and especially an appreciation for factors which are likely to become relevant later at an early stage would have helped here.

Overall, despite occasional instances of short-sightedness, the project accomplished its goal: based on the acceptable modelled performance and meeting all top-priority requirements, this design for a convertible roof for the Fiat 595 Abarth is likely to be viable. If further research is undertaken, it is highly recommended that a more thorough cost analysis be conducted, preferably with the aid of data on realistic levels of demand. Furthermore, fatigue testing and analysis, ideally including physical prototyping, are required in order to determine the service lifetime of the mechanism.

6. **Appendix**

No.	Stakeholder Name	Impact How much does this project impact them (Low, Medium, High)	Influence How much influence do they have over the project (Low, Medium, High)	Stakeholder Priorities
1	Technical Lead	High	High	Proposed design comply with given requirements
2	Technical Designer	High	High	Design and deliver a functional product
3	Customer	Medium	Medium	Affordability and reliability
4	Fiat	High	High	Increase sales margins
5	Bosch	Medium	Medium	Sales and profit
6	KHK Gears	Medium	Medium	Sales and profit

Table 6.1: Stakeholder analysis

No.	Category	Requirement	Target / I	Ranking	Test Method
1	Performance	Roof must be able to deploy and retract within 15 seconds	15 s	5	Modelling and simulation
2	Company Constraints	Must use existing 12V battery to power the mechanism	12 V	5	Analysis of mechanism behaviour by modelling
3	Company Constraints	Must only use rotational joints		5	
4	Company Constraints	Mechanism must use a single DC motor		5	Analysis of mechanism behaviour by modelling
5	Size	Roof must have a maximum length of 1640mm when deployed	< 1640 mm	5	Simulation in Linkage Software
6	Size	Roof must have a maximum length of 670mm when retracted	< 670 mm	5	Simulation in Linkage Software
7	Size	Roof must have a maximum height of 360mm when retracted	< 360 mm	5	Simulation in Linkage Software
8	Weight	Mass of the roof must not add more than 100kg to original mass of the car	100 kg	5	Calculate mass of the roof
9	Product Cost/Company Constraints	Cost of new car model must not add more than £3300 to production cost compared with that of equivalent hard top (from equivalent Mini)	~£19,065 + £3300	5	Calculate number of joints and the materials used, and cost of other components
10	Safety	Mechanism must cease motion if power is lost		5	Integrate safety features
11	Safety	Convertible roof must not cause harm to passengers in the event of failure		5	Integrate safety features
12	Safety	Roof must not obstruct rear view mirror when it is retracted		5	Analysis of mechanism behaviour by modelling
13	Environment	Roof mechanism must work in common operating weather		5	Analysis of mechanism behaviour by modelling
14	Life in service	Mechanism must have a life span of at least 10 years	10 years	5	Calculations based on materials and components used
15	Life in service	Roof material must have a life span of at least 5 years	5 to 7 years	5	Calculations based on materials and components used
16	End of Life/Material	Must be made of a material that can easily be recycled and reused for other products		5	Research recyclability of materials used
17	Performance	Roof should be able to deploy and retract open up to relative air speeds of 30kph (mini)	30 kph	4	Aerodynamic analysis
18	Safety	Mechanism must be able to detect and stop if there are any obstructions		5	Integrate safety features
19	Service	Should be compatible with standard maintanance tools		3	Structural analysis, testing with simulated loads
20	Service	Mechanism must operate properly under normal conditions for 12 months between services	12 months	5	Calculations based on materials and components used
21	Materials	Roof must be a soft top made out of flexible material		5	Simulation in Linkage Software
22	Company Constraints	Profile of the roof must not deviate by more than 10mm from current design	10mm	5	Simulation in Linkage Software
23	Manufacture	Should be possible to manufacture consistently & precisely		4	Analyse manufacture process for complexities beforehand, order test batch
24	Manufacture	Manufacturing processes should be simplified where possible		4	Analyse design for overcomplication and model assembly in CAD
25	Size	Minimise profile when stowed to minimise drag		3	Simulation in Linkage Software and measure profile surface area
26	Safety	Must open at safe height above passengers' height	150 mm above headrest	5	Simulation in Linkage Software
27	Ergonomics	Roof must not cause any extra discomfort to passengers, especially to the driver		5	Real-life testing
29	Size	Height must not vary from current roof design by more than 50mm	50mm	5	Simulation in Linkage Software
30	Legal	Must not infringe on the intellectual property of any other company (patents, copyrights, etc)		5	Research
31	Politics	Roof must not contain design elements which may cause offense		5	Research
32	Product Cost/Company Constraints	New car model should cost as little as possible extra to produce compared with equivalent hardtop	~<£3300	4	Calculate number of joints and the materials used, and cost of other components
33	Environment	Must operate in hot conditions	Up to 100 degrees C	5	Ensure tolerances allow for heat expansion
34	Performance	Roof should need minimum clearance to open	Smaller heights is better	3	Modelling and simulation in Linkage
35	Performance	Roof should have minimal overlaps when stowed	Less overlaps is better	3	Modelling and simulation
36	Performance	Roof must not be able to open when car is travelling over 30 km/h	30 km/h	5	Integrate safety features

Table 6.1: Product Design Specification

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