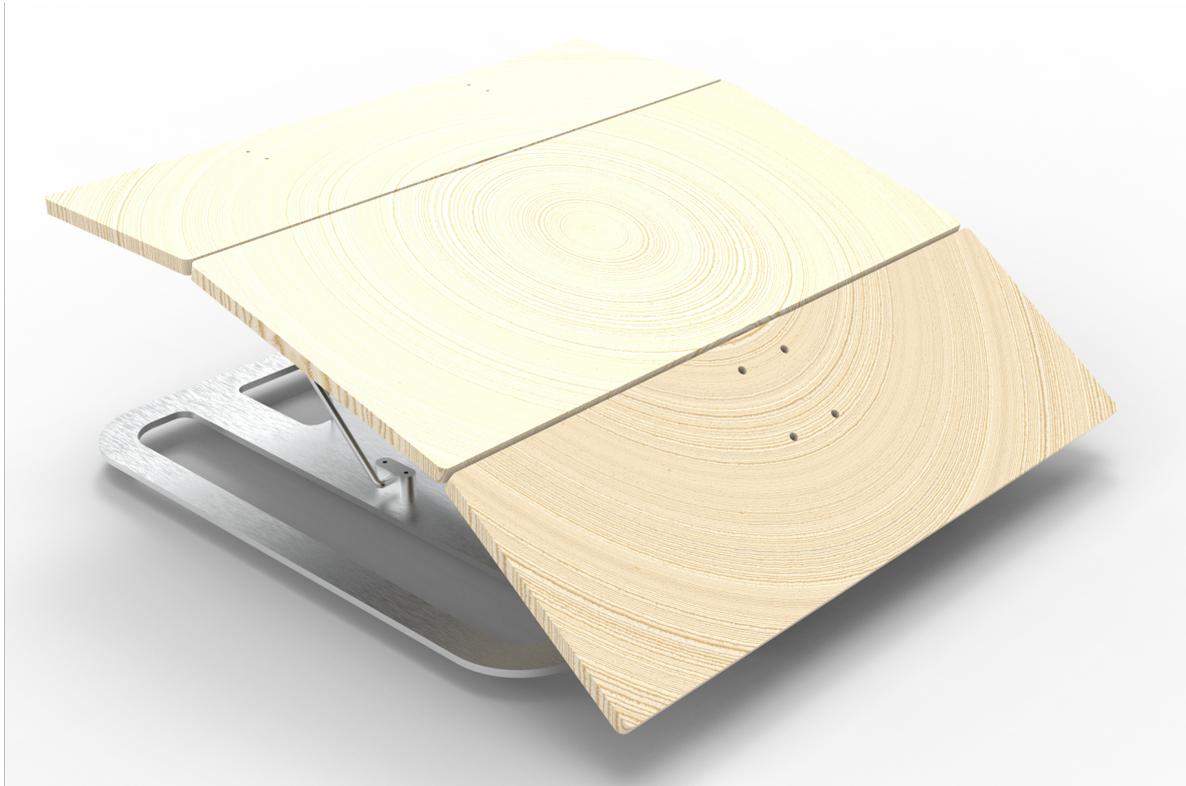

Sit-to-Stand Transition Assistance Device



EasyUp | Sit-to-Stand
Assist Device

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Executive Summary

Easy-Up is a portable sit-to-stand assist device developed to support ageing Australians with reduced lower-limb strength, aligning with United Nations SDG 3 - Good Health and Well-being. It targets elderly users, rehab patients and aged-care residents, with customers including family carers, clinics, facilities and NDIS providers.

Based on customer research, consumers want a device that fits onto any chair or surface, is reliable, safe, lightweight and compact, and is reasonably priced while providing lifting assistance without moving all load from their legs. Hydraulic, rotary mechanical, inflatable/electric, and mechanical gas-spring systems were the 4 mechanism families that were investigated. A compact gas-spring mechanism installed underneath a hinged seat was chosen as the final concept using Axiomatic Design Theory as it provides sufficient assistance without external power and is less expensive and complex.

Material and fastener selection for the base plate, hinge and seat were completed using constraints, objectives, material indices methods and bearing life calculations, confirming acceptable stresses, deflections and safety factors. The assembly for manufacturing is defined by a comprehensive CAD model and mechanical drawings. Overall, Easy-Up satisfies the essential functional requirements for a portable, safe, and reasonably priced sit-to-stand aid. However, additional mass and cost optimisation, user testing, and adjustability for varying body weights are suggested areas for future development.

Team Statement

Student Name/zID	Individual Contribution
Sophia Li - z5591925	<ul style="list-style-type: none"> - Material selection (Translation, Equations, Selection): <ul style="list-style-type: none"> - Base Plate - Seat Hinge - Assist with Editing/ Proofreading - Acknowledgments
Danah Almane - z5613149	<ul style="list-style-type: none"> - Material selection for Cushion to Base fasteners <ul style="list-style-type: none"> - Base - Piston Bolts - Base - Mechanic Bolts - Stabiliser Joints
Bao Long Tran - z5436663	<ul style="list-style-type: none"> - Transfer/writing report in L^AT_EX - Material selection (Translation, Equations, Selection): <ul style="list-style-type: none"> - Seat - Fastener selection: <ul style="list-style-type: none"> - Bearing selection - Formatting and referencing most of the equations, figures and references - 3D rendering
Muneel Naqvi - z5480559	<ul style="list-style-type: none"> - CAD modelling - CAD drawings (All parts) - Our Design - Piston - Bill of Materials
Quang Nguyen - z5416177	<ul style="list-style-type: none"> - Executive Summary - Introduction - Description of Design Concept - Customer needs - Final design selection - Our Design - Piston force calculations
Wilson Zhao - z5591831	<ul style="list-style-type: none"> - Assist CAD modelling - Assist CAD drawings (All parts) - Conclusion Editing - Our Design - Living hinge
Jaturapat Jiravichai - z5515965	<ul style="list-style-type: none"> - Conclusion

1 Introduction

Globally, populations are ageing and mobility limitations are becoming a major public health concern. For many older adults, the simple sit-to-stand (STS) motion is one of the most frequent and demanding daily tasks, performed approximately 30 to 70 times per day [1]. When this motion is repeated against weakened muscles and degenerated joints, it places high cyclic loads on the hips, knees and lower back, accelerating musculoskeletal wear and increasing the risk of falls and secondary injuries. In fact, around 65% of older adults report chronic musculoskeletal pain, and 1 in 5 experience pain from the lower back to the knee when standing up from a chair [2]. Most of the elderly population rely on either armrests or another person to help them stand up which erodes independence and inconvenience multiple times a day [3]

To solve this problem, a variety of powered lifting seats and commercial sit-to-stand cushions have been developed. Vertical lift cushions and other battery-powered devices offer substantial support, but are costly, heavy, and require frequent maintenance and charging. Although non-electric pneumatic and manual lift cushions are less expensive, they are typically heavy, difficult to operate, and may feel unsteady while in use. This undermines user confidence and reduces long-term adoption. There is a clear gap for an assistive device that is safe, compact, intuitive and low-cost, while still delivering significant load reduction on the user's joints.

The Easy-Up sit-to-stand assist device has been developed to address this gap and to support the United Nations Sustainable Development Goal (SDG) 3: Good Health and Well-being [4]. Easy-Up is a portable rehabilitation device that mounts on an existing chair and uses a gas-spring mechanism to generate an upward torque on the seat, reducing the effort the typical user must exert by approximately 40 to 50%. The product can be transported between settings such as homes, clinics, and community centres thanks to its design. This includes a lightweight frame, a living-hinge seat, and a detachable cushion. Easy-Up seeks to lessen joint pain, reduce the risk of overuse injuries, and help older users stand more confidently and independently by lowering the peak loads in their quadriceps and hips during STS.

This report documents the engineering development of Easy-Up. First, the background research, user needs and functional requirements are summarised, with reference to existing sit-to-stand solutions and relevant rehabilitation principles. The conceptual design process is then outlined, followed by detailed mechanical analysis of the gas-spring assisted lifting mechanism, including force, torque and safety-factor calculations. Subsequent sections address materials selection, structural stiffness of the seat and frame, and fastener analysis, leading to an integrated CAD model of the final concept. The report concludes with an evaluation of Easy-Up against the design requirements, limitations of the current prototype, and recommendations for future refinement and testing.

2 Description of Design Concept

The conceptual design phase for Easy-Up began with preliminary research and interviews into elderly mobility, market scanning of existing sit-to-stand aids, and the problem framing completed in the interim report. This study showed that many older adults find it difficult to sit-to-stand, but they still want to be physically active. They value lightweight, portable, reasonably priced devices that feel secure and reliable. These needs are partially met by current market devices like bulky walkers and powered lift cushions, but they are frequently heavy, visually medical, and too expensive for many users and carers to afford.

Based on these results, the team came up with a number of concept families that could help with some lifting while still enabling the user to use their lower limbs. During this phase, the mapping between customer needs, functional requirements, and design parameters was structured using Axiomatic Design Theory (ADT), and an Agile design approach made sure that concepts were iteratively refined through brief "sprints" of CAD modelling, straightforward calculations, and feedback the tutors. The Easy-Up device, a small, gas-spring assisted mechanism that mounts on a light base to offer controlled sit-to-stand assistance for elderly users in clinical, home, and aged-care settings.

2.1 Customer needs

The majority of Easy-Up's users are elderly people who live in the community, patients undergoing rehabilitation, and residents of assisted living facilities who need assistance with sit-to-stand transfers. These people, their family carers, and institutional buyers like clinics, assisted living facilities, and NDIS providers are among the clients. Restricted mobility, fear of falling, and pain during sit-to-stand are the main reasons people buy mobility assistive devices, according to preliminary market research and a review of mobility-aid literature. Cost and portability were found to be the main obstacles to adoption at the same time.

Insights from primary interviews and Miro synthesis are summarised as follows. Elderly users "want something that helps them move faster" but do not want to be completely passive; they value products that support their movement while still allowing their legs to work. Many existing devices, such as walkers and powered seats, are heavy and cumbersome, making them difficult for frail users and carers to lift into cars or move between rooms. For Easy-Up, this led to customer needs around lightweight construction, compact geometry and assisted portability (for example, with the help of a carer). Cost sensitivity was another recurring theme, with customers prioritising price-effectiveness relative to perceived benefit.

Table 1: Outlines the key customer needs devised for Easy-Up and links each to the main stakeholder groups

Customer Need	Summary	Stakeholder(s)
CN1: Reduced effort during sit-to-stand	The device should reduce the physical effort required to stand while still engaging the user's leg muscles.	Elderly users, Rehab patients, Physiotherapists / OTs
CN2: Safe and stable assistance	The device must feel stable, control the lifting motion, and not increase the risk of falls or tipping.	Elderly users, Family carers, Clinics, Aged-care facilities
CN3: Lightweight and compact	The system should be light and compact so it can be moved and transported by a carer or individual.	Elderly users, Family carers, NDIS providers
CN4: Affordable / cost-effective	The product should be as cheap as possible so that individuals and facilities can justify purchase within limited budgets.	Individuals, Aged-care facilities, NDIS providers
CN5: Compatible with existing chairs	Easy-Up should fit standard dining and arm chairs without permanent modification.	Family carers, Aged-care facilities, Clinics
CN6: Simple, low-maintenance operation	The device is easy to use, requires minimal training, and avoids complex electronics or frequent charging.	Elderly users, Family carers, Staff in clinics and facilities
CN7: Supports rehabilitation and independence	The assistance level should allow progressive strengthening, supporting rehabilitation goals and maintaining user independence.	Rehab patients, Physiotherapists / OTs, Clinics

2.2 Functional requirements

For Easy-Up, the functional requirements were kept intentionally broad during the initial stages of development. They are listed as such:

The device must

- Provide meaningful sit-to-stand assistance while keeping the user's legs active,
- Remain safe and stable,
- Be lightweight and portable,
- Fit typical household and facility chairs, and
- Stay affordable and easy to maintain.

These requirements were then explored further through four mechanism families [5] [6] [7] [8] identified in Figure 1 below:

- A small hydraulic cylinder installed beneath the seat provides lift assistance via fluid pressure in a hydraulic assist seat.
- A mechanism known as a rotating seat with rear support frame pushes the user forward and upward by rotating the seat around a high pivot.
- An air inflatable device that raises the user vertically is known as an inflatable or electrically raised cushion.
- The Easy-Up gas-spring piston mechanism generates torque around the rear pivot by means of a passive gas-spring attached to a hinged seat base.

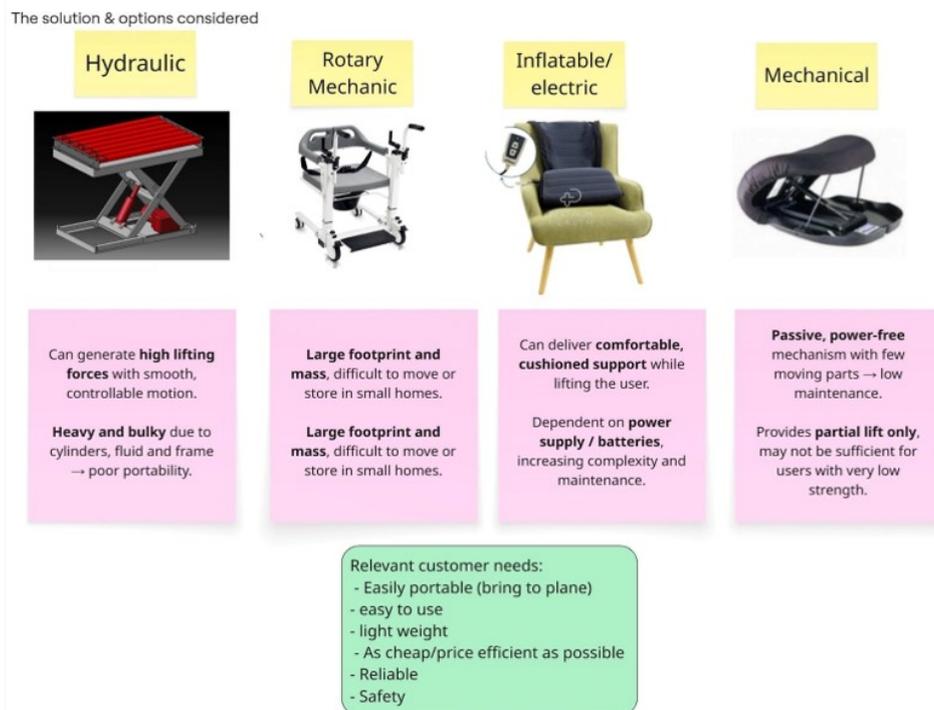


Figure 1: Concept Consideration of four mechanism families

2.3 Axiomatic design theory

Axiomatic design theory leverages two design axioms; the Independence axiom and Information axiom, to sequentially compare and select the least complex design out of many, by utilising the design matrix A:

$$\begin{bmatrix} FR_1 \\ FR_2 \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \cdot \begin{bmatrix} DP_1 \\ DP_2 \end{bmatrix}$$

We first defined our functional requirements (FR), before evaluating the design parameters (DP) of our various concepts. Listed below are said FR's, and the DP's of our settled upon design.

Functional Requirements:

- FR1:** Supports user comfortably
- FR2:** Elevates user to standing height
- FR3:** Integrates with many surfaces
- FR4:** Minimal risk of accidents
- FR5:** Convenience of use
- FR6:** Cost effective

Design Parameters:

- DP1:** Large seat cushion
- DP2:** Supports for user balance
- DP3:** Automatic mechanical lift system
- DP4:** Flat, textured base
- DP5:** Simple components
- DP6:** Adjustable height setting
- DP7:** Compact frame

Of the 4 designs, only the gas-spring powered seat, processing the above DP's, could feasibly satisfy the independence axiom.

Hence, we deemed the gas-spring mechanism most capable of solving our problem. It did not rely on complicated, difficult to repair/ replace electronics or fluid systems, offered adjustable assistance via the gas-spring, and had a small, under-seat layout that could be retrofitted to accommodate all manner of sitting surfaces. This idea best suited the team's mechanical engineering expertise, ambitions, as well as the project's limitations (limited mass, cost, and external power) whilst simultaneously satisfying the customer needs we set out to meet as seen in Table 2.

Table 2: Independence axiom matrix for gas-spring powered concept, demonstrating a decoupled design.

FR	Design Parameters						
	1	2	3	4	5	6	7
1	x	x					
2			x				
3				x			
4					x		
5						x	
6							x

2.4 Final design selection

A decision matrix that compared the four primary concepts to weighted criteria derived from the functional requirements and customer needs was used to select the final concept. Safety and perceived stability, degree of assistance while maintaining leg activity, portability, ease of use, compatibility with standard chairs, affordability, and alignment with SDG 3 outcomes (lower risk of injury, increased independence, and potential for rehabilitation) were among the criteria.

The gas-spring Easy-Up concept routinely received the highest scores, especially in the categories of portability, simplicity, and safety. The maintenance burden and perceived unreliability of powered systems could be avoided by implementing passive mechanical components, which would also enable assistance to be adjusted by choosing suitable gas-spring specifications. Team capability was another practical factor: the lack of mechatronics and electronics experts made purely mechanical solutions more feasible within the course timeframe. The Easy-Up gas-spring piston mechanism was chosen as the final concept for detailed design after careful consideration of the ADT analysis, decision matrix results, and Agile iteration cycles.

2.5 Our Design

The Easy-Up device uses a gas piston acting on a levered seat base to assist the user through the most joint-intensive portion of STS. As the user initiates standing, the piston applies an upward moment about the rear hinge of the seat, effectively “pushing” the pelvis of the user upwards while the legs remain engaged. This reduces peak loads in the hips and knees during the first third of the STS motion, where pain and failure to stand are most common.

For the base plate and lever components, 1060 aluminium alloy and other inexpensive, low-density materials are used to achieve lightweight construction. The device can be moved between chairs and repositioned by both carers and target uses as the geometry is compact and portable. Material components maintain under target weights and aim for maximum stiffness under worst-case loading.

A removable cushion module sits on top of the hinged seat base. This modularity allows the cushion to be supplied in different shapes, sizes, colours and materials to suit individual users and interior settings. Detachability also enables easy washing and replacement, extending the functional life of the device without replacing the underlying mechanism.

Gas - Spring Piston

The core piston element is an off-the-shelf long-travel office-chair gas spring rather than a proprietary actuator. A quick quasi-static sizing method was used to determine the required force (Appendix A). Note that the goal of Easy-Up is not to allow the user exert zero effort into lifting themselves into a standing position but rather to provide assistance and reduce the amount of effort exerted. As such, the mechanism intends to support a load range of 40 - 50 kg, giving vertical loads of approximately 392 - 490 N. The piston force was then estimated as

$$F_{piston} = \frac{W}{\sin(\theta)}$$

at the most critical seat angles, maximum incline and mid incline. This yielded forces between 1.8 - 3.8 kN. Therefore, we concluded that our target piston capacity is approximately 4.0 - 4.8 kN after an extra 20 - 30% margin was added to account for friction, self-weight of the seat, and manufacturing tolerances.

This range guided a catalogue search for a commercially available office chair gas spring with matching stroke length and force rating. Therefore, we decided on the Office Chair Gas Lift [9] piston, shown in Figure 2 below, which has matching stroke length and force rating.



Figure 2: Office Chair Gas Lift

Using a standard component reduces unit cost, allows users or maintenance staff to replace the piston with an identical part without returning the device to the manufacturer. It is also particularly beneficial for fleet management in nursing homes and hospitals, where minimising downtime is critical.

Living Hinge and Height Adjustment Mechanism

The living hinge is a singular, unibody component that allows for bending in the seat to accommodate the user's natural position throughout both sitting and standing motions. The design resembles a mesh-like structure that facilitates further bending on top of the materials' elastic capacity. A sliding plate mechanism, hidden under the seat, can be extended and held in place through friction applied by a user operated screw, and serves to limit the total bending area of the hinge component. This essentially "adds" more length to the rigid sections of the seat, increasing the overall height of the device when fully extended, thus enabling a high degree of user customisation, which can be observed in Figure 3 below.

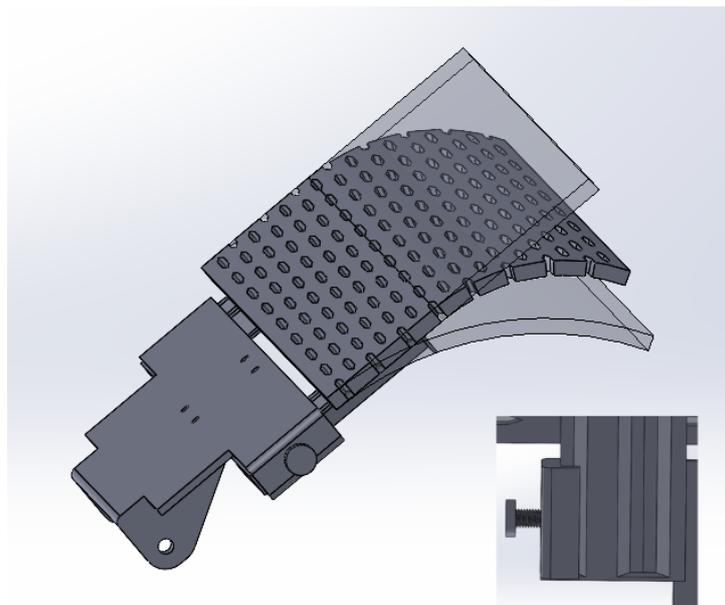


Figure 3: CAD model of hinge and plate mechanism.

3 Technical design and analysis: Material analysis

EasyUp’s materials were selected to ensure that the user is safely supported when the assist is in use and that our mechanisms perform their intended function effectively. Its main considerations are stiffness, yield strength and mass, however, for the relevant component we have also taken into consideration its elasticity.

3.1 Base plate

The base plate is the plate on which all the mechanisms are mounted. The constraints and objectives for the material selection of the base plate are summarised in Table 3 below.

Table 3: Function, constraints, and Objectives for Base Plate

Function	Support the load of the user and mechanism without plastic bending or failure.
Constraints	The mass of the base plate must be less than 3kg . Must support 120kg with a maximum of 0.02mm of elastic deformation.
Objective	Maximise stiffness so that the plate is more resistant to bending, and minimise mass.
Free Variables	Material has no constraints.

3.1.1 Justification of Constraints

As most other sit to stand assists that use a piston or another similar mechanism to support the user have a mass of around 5 - 6kg, it is believed to be appropriate to set the maximum mass of the base plate alone to be 3kg. This takes into consideration that our chosen piston is approximately 0.9kg, leaving around 1 - 2kg for the other mechanisms in our design. However, as one of our goals is to have a lightweight sit to stand assist, mass will be minimised when considering the possible materials within the constraints.

As the mass of an average adult male above 60 years is around 92kg [10], to ensure the safety of our user we have decided that the base plate must support at least 120kg of load. The maximum deflection will be 0.02mm. This is because some deflection is not an issue and can be preferable on certain surfaces such as cushions as they are soft and malleable, however, this deflection should not be noticeable by the user. On top of these constraints, stiffness will be maximised to ensure that plastic bending or failure does not occur.

3.1.2 Translation of Constraints to Material Properties

It is necessary to find the maximum density that ensures the base plate remains under the 3kg constraint. Using the measuring tools given in the SolidWorks CAD software, the volume of the base plate can be determined to be:

$$V = 515143.58 \text{ mm}^3$$

Then the maximum density can be determined using the formula:

$$\rho = \frac{m}{v}$$

Where ρ is density, m is our maximum mass and V is the base plate volume. As such the maximum density can be calculated to be:

$$\rho_{max} = 5823.62 \text{ kg/m}^3$$

The minimum Young’s Modulus that remains within our constraints of a 120kg load and 0.02mm deflection must also be found. The cross sectional area that will be used is the contact area between the support framing and the base plate. This includes the two frame rails and piston bottom mount. After

using the SolidWorks measuring tool again and summing the three contact areas, the total contact area can be found.

$$A_{total} = 21348.26 \text{ mm}^2$$

If the load mass is set to be 120kg, the load force can be calculated as

$$\begin{aligned} F &= ma \\ &= 120 \times 9.81 \\ &= 1177.2\text{N} \end{aligned}$$

Note that this force has been assumed to be applied directly downwards and evenly distributed across the contact areas between the base plate and support framing to simplify calculations. In reality, the force will be angled in the direction of the piston, the angle of which will be changing as the lift is in use, greatly reducing the vertical downwards force and therefore will accommodate for our evenly distributed assumption.

Bending force is not a great concern with the base plate as it will always be placed on another surface such as a chair and therefore, fully supported on the underside. As such the stress in the base plate can be calculated as

$$\begin{aligned} \sigma &= \frac{F}{A} \\ &= \frac{1177.2}{21.34826 \times 10^{-3}} \\ &= 55142.66736 \text{ Pa} \end{aligned}$$

The strain can then be calculated using the following formula:

$$\epsilon = \frac{\Delta L}{L_0}$$

With a constraint of 0.02mm and the thickness of the base plate being 5mm the strain is found to be

$$\epsilon = 4 \times 10^{-3}$$

With both the required stress and strain the minimum Young's Modulus can be found to be:

$$\begin{aligned} E_{min} &= \frac{\sigma}{\epsilon} \\ &= 0.0138 \text{ GPa} \end{aligned}$$

The full calculations can be found in Appendix B.

Now that the maximum density and minimum Young's Modulus have been derived from the constraints and geometry of the base plate, they can be plotted on a Young's Modulus to density Ashby Map from Material Selection In Mechanical Design [11] as shown in Figure 4. The maximum density has been rounded down to 5800 kg/m³ and the minimum Young's Modulus has been rounded up to 0.014GPa.

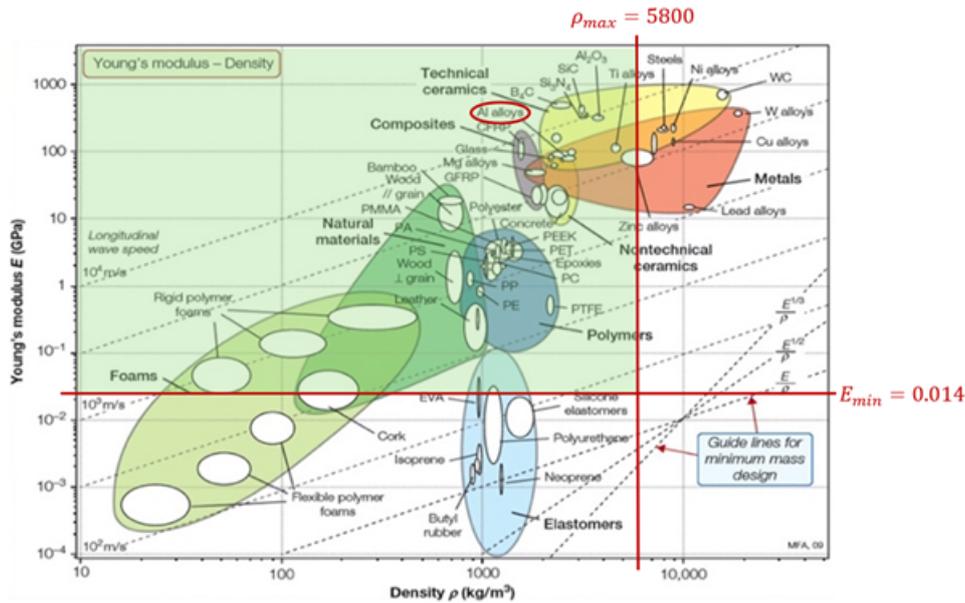


Figure 4: Base Plate Ashby Map Indicating Possible Materials

3.1.3 Screening

All the materials in the green quarter are within our constraints and possibilities for our base plate material. However, as stiffness must also be maximised to prevent the base plate from moving or bending much under the weight of the user on softer surfaces, all the elastomers and foams will be removed from our possible materials. Woods are a good option as they are light, however, they have installation concerns as screw inserts would be required to attach the supporting frames to the base plate. Polymers are also a consideration but are generally too expensive and have lower stiffness according to MIT’s Material Cost Type [12]. As such, our team has elected for Aluminium alloys. Aluminium alloys have high stiffness with a Young’s Modulus of around 70GPa, making it very unlikely to have any bending concerns when undergoing the expected load. Our team has more specifically chosen **1060 Aluminium Alloy** which has a Young’s Modulus of 69GPa and results in the base plate having a total mass of 1.39kg which is significantly lower than our target of 3kg.

3.2 Seat hinge

The seat hinge is the flexible component of the seat which allows for Easy-Up to have an adjustable seat height. This means that the hinge material must be flexible but must be able to support load without failing. The constraints and objectives for the seat hinge are recorded in Table 4 below.

Table 4: Functions, Constraints, Objectives and Free Variables for Seat Hinge

Function	<ul style="list-style-type: none"> Elastic hinge that bends to allow for seat adjustability without composite sections failing under user load.
Constraints	<ul style="list-style-type: none"> Hinge cannot fail, meaning stress must remain less than failure strength ($\sigma < \sigma_f$) throughout the hinge.
Objective	<ul style="list-style-type: none"> Maximise elastic flexure so the hinge can flex even at high height settings.
Free Variables	<ul style="list-style-type: none"> Material has no constraints.

3.2.1 Justification of Constraints and Objectives

It is crucial that the hinge does not fail under the load of the user. This means that the stress the hinge undergoes cannot exceed the failure strength of the chosen material. The elastic flexure of the material governs the extent to which the material can deflect within its elastic range. This property needs to be maximised due to the height adjustment feature. When the height adjustment is at the highest setting, the living hinge will have 140mm of length to bend which is approximately two thirds of the original length. As such, maximising elastic flexure ensures that the hinge flexes enough to the correct radius at the highest setting.

3.2.2 Material Index

The material index will be determined based on our constraints and objectives. The formula for strain due to bending is

$$\epsilon = \frac{t}{2R}$$

Where t is the original thickness of the hinge, R is the radius of curvature which the neutral axis makes during bending. Note that the distance between the neutral axis and surface, where stress is highest, is $\frac{t}{2}$. Thus using the Young's Modulus formula, stress can be derived as

$$\sigma = E \left[\frac{t}{2R} \right].$$

As stress must remain under failure stress, we see the following:

$$\sigma_f > E \left[\frac{t}{2R} \right]$$

After rearranging to make the curvature radius our subject we see that:

$$R > \frac{t}{2} \left[\frac{E}{\sigma_f} \right]$$

Since the radius, R , must be minimised to increase maximum curvature, the term $\frac{E}{\sigma_f}$ must be minimised. This is equivalent to the term $\frac{\sigma_f}{E}$ being maximised. As such the material index for the seat hinge has been derived.

$$M = \frac{\sigma_f}{E}.$$

The full derivation can be found in Appendix C.

From here, the material index can be plotted on a Young's Modulus to Strength Ashby Map as seen in Figure 5 below. The tie line for the material index has been moved down and to the right to minimise Young's Modulus and maximise strength. This means that the materials will have low stiffness, allowing them to bend and high failure strength, so they won't fail with large deformations. The material index plotted is 2×10^{-2} .

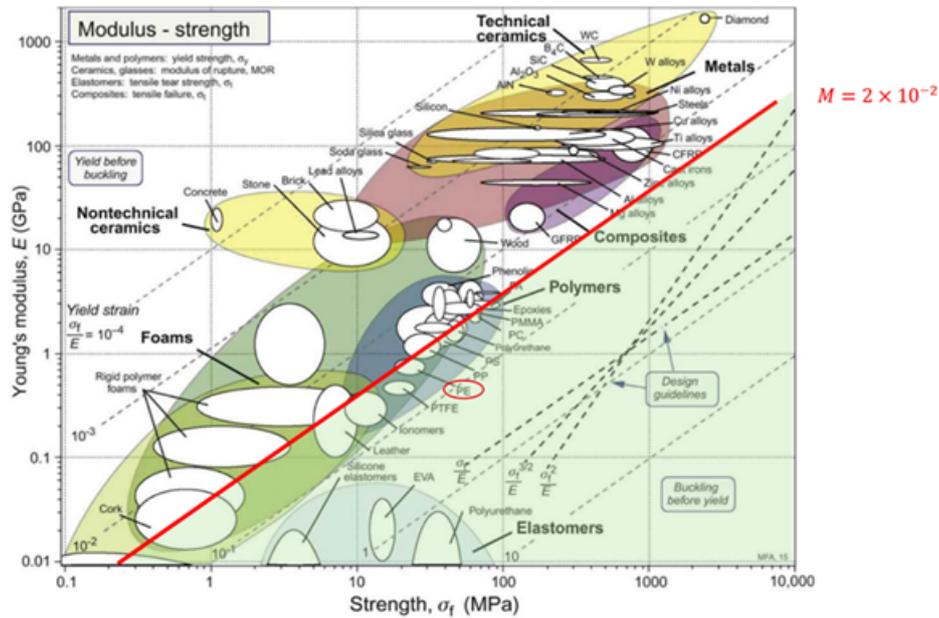


Figure 5: Seat Hinge Ashby Map Indicating Possible Materials

3.2.3 Screening

The green section of the chart indicates our possible material candidates for the seat hinge. Table 5 has ranked the considered materials for the seat hinge based on their material indexes and will be referred to for our final decision. Elastomers have the highest material index however, due to them having a comparably low Young's Modulus, they are generally considered not stiff enough to support a considerable load. PTFE is also an excellent choice but is more expensive than options such as polyethylene or polypropylene. As such, our team has decided on **polyethylene** as it has a higher material index than polypropylene and is frequently used for living hinges.

Table 5: Seat Hinge Material Ranking Based On Material Index

Rank	Materials	$M (\times 10^{-3})$
1	Elastomers	100 - 1000
2	PTFE	35
3	Polyethylene	32
4	Polypropylene	30
5	Nylon	30

3.3 Seat

The non-bendable part of the seat should be able to handle the load of a user under the intended usage without being subjected to elastic deformation. The constraints and objective are summarised in Table 6.

Table 6: Function, constraints, objectives, and free variables of the seat

Function	<ul style="list-style-type: none"> • <i>Support the load of the mechanism along with the user</i>
Constraints	<ul style="list-style-type: none"> • <i>Dimensions are 450mm by 425mm by 8mm</i>
Objective	<ul style="list-style-type: none"> • <i>Minimise mass</i> • <i>Maximise stiffness</i>
Free variables	<ul style="list-style-type: none"> • <i>Material</i>

It is important that the non-bendable part of the seat is stiff so that the seat retains its shape to avoid ergonomic issues and maintain stability when the user is seated. Furthermore, it has to be lightweight to make it portable for an elderly person and conform to restrictive guidelines such as baggage allowance when bringing it on the plane.

3.3.1 Material index

In order to maximise the stiffness and minimise the mass, the following formula is derived with its full derivation in Appendix D

$$m = \left(\frac{12S^*}{C_1b} \right)^{1/3} (bL^2) \left(\frac{\rho}{E^{1/3}} \right).$$

The formula above allows it to be segmented in terms of material properties and defined constraints. In this case, the material properties component in the equation has an index as follows

$$\frac{\rho}{E^{1/3}}.$$

However, since this maximisation of the stiffness is sought after, it needs to be inverted so that it becomes

$$\frac{E^{1/3}}{\rho}.$$

This material index will allow the tie line to be plotted on the Young's modulus to density Ashby chart as seen in Figure 6. Therefore, by moving the tie line at our discretion towards the left-hand corner, a few suitable materials remain for selection.

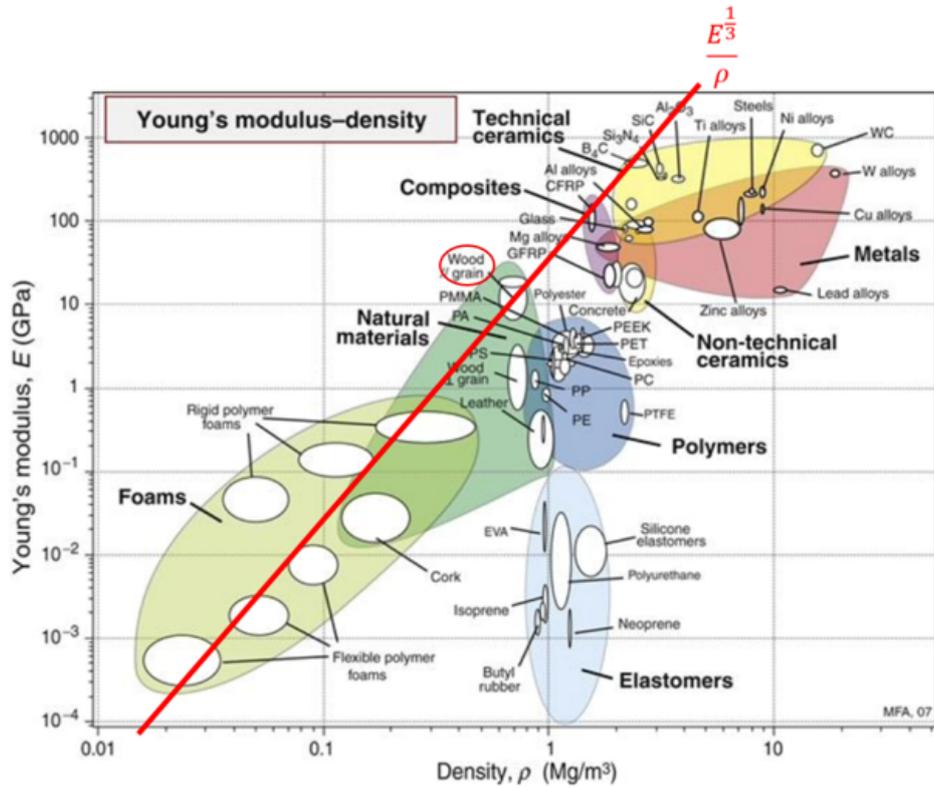


Figure 6: Ashby chart for seat

3.3.2 Material ranking

The Ashby chart allows us to select a few notable materials and rank them. In the following scenario, wood, aluminium, steel, and carbon fiber (CFRP) are selected during this ranking process. As seen in Table 7, CFRP has the highest index number, which correlates to the best performance materials in terms of providing maximum stiffness and minimising mass. Following that, wood has a score just slightly under CFRP. On the other end of the spectrum, steel and aluminium has the lowest score due to its high stiffness but significantly more heavy.

Table 7: Material Ranking Based on Performance Index

Rank	Materials	Performance Index, $E^{1/3}/\text{Density}$
1	CFRP	$150/1600 = 0.003321$
2	Wood	$16/800 = 0.00315$
3	Magnesium alloys	$47/2000 = 0.00184$
4	Aluminium alloys	$82/2900 = 0.00150$
5	Steel	$200/7800 = 0.000750$

In the end, wood was selected due to its good compromise between stiffness and price. CFRP is significantly more expensive, although it provides high stiffness. Aluminium, magnesium and steel were negated as the heavier mass outweighs any benefit of the higher stiffness compared to wood. Furthermore, wood has a more pleasing element in terms of aesthetics since it is easier to blend in with other furniture.

3.3.3 Wood selection

Subcategories of wood can be further determined using the same tie line shown in Figure 6 and plotted on the Ashby map for natural materials, as seen in Figure 7.

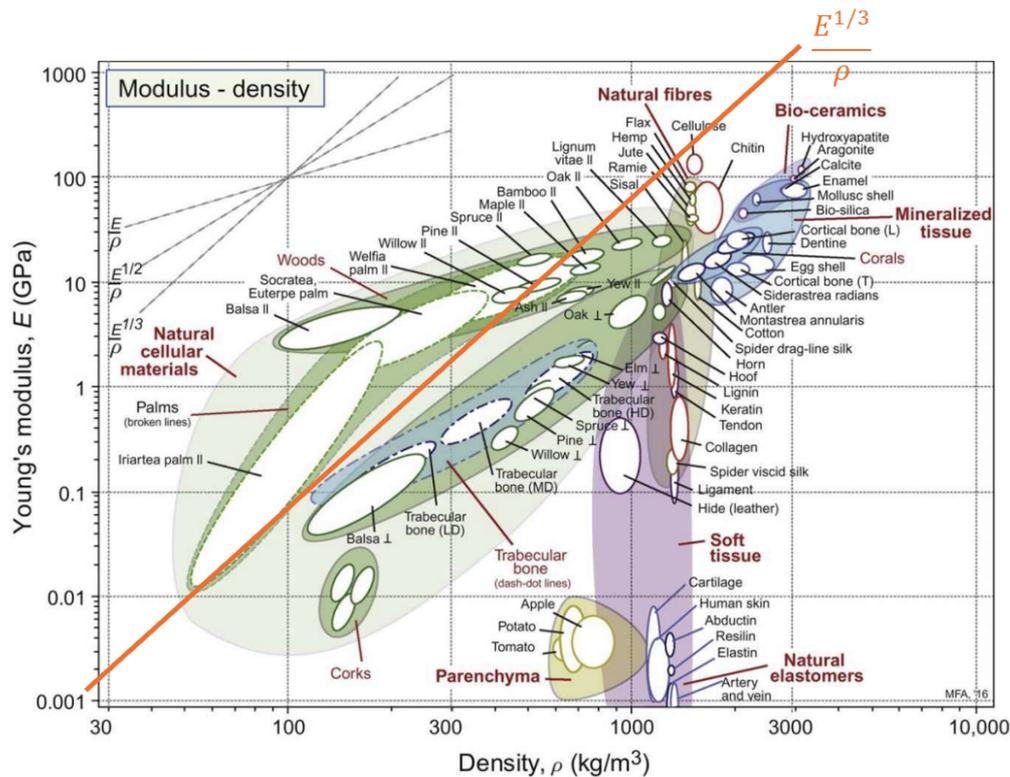


Figure 7: Ashby density-Young's modulus chart for natural materials

The line is translated in the same way since maximising stiffness and minimising mass is still the objective. However, material selection is narrowed down to natural materials, and wood is the desired material in this case. Therefore, the green area labelled wood in Figure 7 is the region of interest, and those located to the left of the tie line will be the specific wood choices. In this instance, Spruce, Pine, Willow Welfia palm and Balsa will be selected. It is also important to note that most possible wood candidates in Figure 7 are loaded in the parallel direction as seen in Figure 8. The Young's modulus in this direction is higher because the cell walls are axially compressed, whereas they bend and act like a foam when loaded perpendicular to the grain [13].

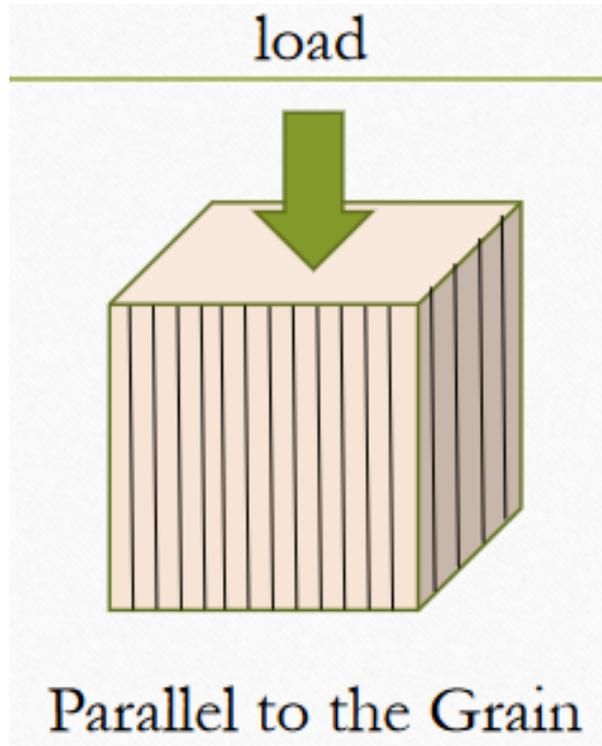


Figure 8: Load parallel to grain

Taken this into consideration, the Table 8 below summarises the density and Young’s modulus of the selected wood.

Table 8: Properties of Selected Wood Types

Wood	Density (kg/m ³)	Young’s modulus to grain (GPa)	Material Index E/ρ
Spruce	400-705	9.6-10.8	0.015-0.024
Pine	352-849	16.3	0.019-0.046
Palm	400-1000	11.41	0.011-0.029
Balsa	100-300	4	0.013-0.04

The material index value in Table 8 represents how well each wood meets the objectives, and the higher value is more desirable in this case. Evidently, pine wood has the highest numbers at both ends of the spectrum as seen in Table 8. Therefore, pine wood will be chosen for the seat of the sit-to-stand device. Pine wood is also proven to be versatile and is widely used for furniture products [14]. Moreover, it is quite affordable, coming at 7.75 per linear meter [15]. Thereby, circumvent the unnecessarily high material cost to achieve the goal of providing an affordable product.

4 Technical design and analysis: Fasteners selection

Selecting the correct fasteners ensure that the sit-to-stand device is safe under operation. It further enhances the longevity of the device. The fasteners selection and analysis include bearings, screws and bolt and nut during the installation of the device.

4.1 Bearing selection

The bearings connected to the shaft of the sit-to-stand device piston must facilitate a smooth pivotal motion. These bearings must also sustain radial load applied by the users when the device is in use, and must be readily accessible and inexpensive. The ideal bearing for such a situation is a rolling-contact bearing, which includes ball bearings and roller bearings.

4.1.1 Roller bearing vs ball bearing

Roller bearings and ball bearings, shown below in Figure 9, use the same principle to provide smooth rotation of the part. The differences between the ball and the rolling bearing are due to their point of contact with the inner ring raceways [16]. A rolling bearing is able to handle a greater load at a cost of slower and less smooth motion due to friction from the surface area that comes into contact with the ring raceways [16]. In contrast, ball bearings are more efficient by providing a smooth motion due to less friction, but comes at a cost of having lower radial load [17]. However, due to the low load requirements and simplicity of the sit-to-stand mechanism, it is more ideal to go with the ball bearing, specifically the deep grooved ball bearing, due to its wide availability. Although roller bearings can handle greater loads, there is also a limited selection based on the device dimensions.

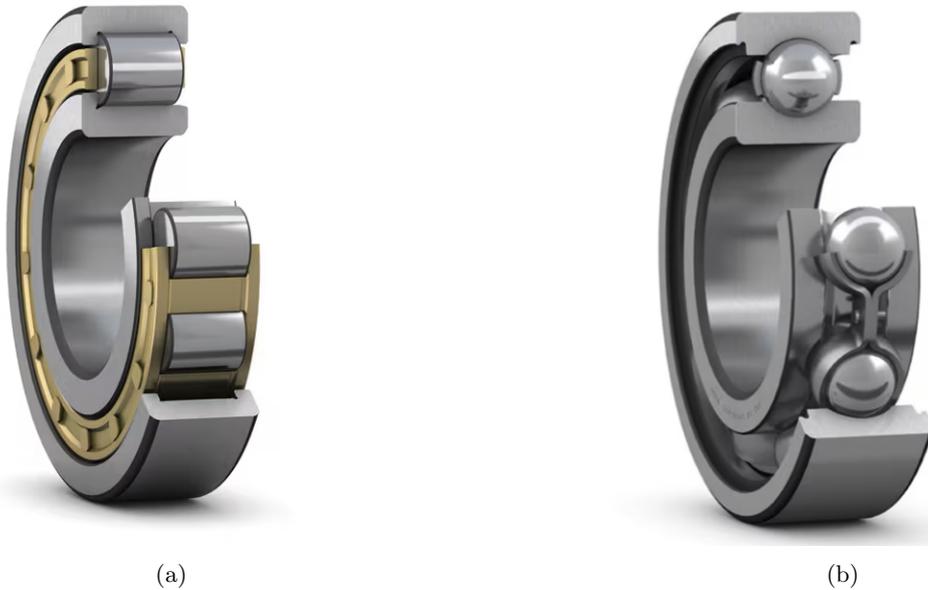


Figure 9: (a) roller bearing and (b) ball bearing

4.1.2 Expected life

Further classification of the potential deep grooved ball bearing model requires calculation of the expected and recommended life of the bearing. The expected life can be calculated given the following equation

$$L = \frac{60nL_n}{10^6} \text{ million revolutions.} \quad (1)$$

The formula will require the desirable conditions, such as the rpm denoted as n and the desirable life span of the bearing under operation, denoted as L_n in hours. Based on the desirable condition of the device, which is assigned under the worst-case scenario, it will need to withstand 58240 hours, which equates to 112 hours per week for 10 years of operation at a rate of 2 rpm. The chosen rpm is converted from degrees per second, in which case the piston will need to move the seat to a 45-degree angle in 4 seconds, which is roughly 2 rpm. Substituting these values into the equation 1 gives 6.99 million revolutions.

$$L = \frac{60(2)(58240)}{10^6}.$$

$$= 6.99 \text{ million revolutions}$$

4.1.3 Bearing selection and Recommended life

Next, a selection of bearings based on the shaft diameter, which is the bore size of the bearing, and also the outer diameter of the bearing, based on geometric constraints. The bearing suitable for this mechanism is provided by SKF, where the outer diameter should not be larger than 40 mm, and a bore size of 20 mm. Therefore, any bearings from SKF's bearings catalogue, as seen in Figure 10, should fit the mechanism. In this case, bearing 61804-2RS1 will be selected to evaluate if it will meet the desirable condition by calculating its recommended life.

Designation	Bore diameter	Outside diameter	Width	Basic dynamic load rating	Limiting speed
	[mm]	[mm]	[mm]	f_t [kN]	[r/min]
W 61704-2ZS	20	27	4	0.585	26 000
W 61704	20	27	4	0.585	32 000
W 61804-2RS1	20	32	7	3.12	13 000
W 61804-2RS1/VT378	20	32	7	3.12	13 000
W 61804-2Z	20	32	7	3.12	24 000
W 61804-2RZ	20	32	7	3.12	24 000
61804-2RS1/W64	20	32	7	4.03	9 200
61804-2RS1	20	32	7	4.03	13 000
61804-2RZ	20	32	7	4.03	22 000
61804	20	32	7	4.03	28 000
W 61904-2RS1	20	37	9	5.53	12 000
W 61904	20	37	9	5.53	26 000
61904-2RS1	20	37	9	6.37	12 000
61904-2RZ	20	37	9	6.37	20 000
61904	20	37	9	6.37	26 000
61904/C3	20	37	9	6.37	26 000

Figure 10: List of possible bearings subjected to geometric constraints [18]

The values seen in Figure 10 should also aid in calculating the recommended life of the bearing, specifically the dynamic bearing, which is denoted as C in the recommended life equation below

$$C = P \left(\frac{L}{L_{10}} \right)^{\frac{1}{k}}. \quad (2)$$

From equation 2, the basic durability L_{10} is 1 (million), and the calculated equivalent load value denoted as P found in Appendix E with k has a value of 3 for a deep-grooved ball bearing. By rearranging and substituting and substitute these values into equation 2 gives a recommended bearing life of 40.2 million revolutions.

$$L = L_{10} \left(\frac{C}{P} \right)^3.$$

$$= (1) \left(\frac{4030}{1412.64} \right)^3$$

$$\approx 23.2 \text{ million revolutions}$$

4.1.4 Selected bearing

Conclusively, as the value of the selected recommended life of the bearing is significantly greater than the expected life of the bearing, the bearing 61804-2RS provided by SKF would safely withstand the desirable condition. Furthermore, the bearing has the necessary geometric properties to be fitted into the shaft and has enough clearance around the surrounding area. The bearing is also quite affordable, coming at just \$5, which makes it quite feasible in terms of repairs [16].

4.2 Fastener selection for the cushion base

This segment of the report examines the fasteners used in the cushion base, with a focus on the bolted joints that transfer load between the cushion, the lifting mechanism and the stabilising elements. In this device, the base is the main structural interface between the user and the ground; any weakness here would directly compromise safety, particularly for elderly users who may rely on the device for support. The analysis therefore concentrates on three key bolted connections: the joint between the base plate and the piston bracket, the side joints between the base plate and the main lifting links, and the joints that secure the stabiliser blocks to the cushion base. For each of these connections, the fastener arrangement, geometry and loading are described, then the five standard failure modes for bolted joints are evaluated and compared with the expected service loads. The aim of this section is to show that the chosen bolt sizes and layouts provide sufficient strength and robustness for the intended operating conditions.

4.2.1 Assumptions

The maximum intended user mass is taken as

$$m = 110 \text{ kg},$$

corresponding to a static weight

$$W = mg = 110 \times 9.81 \approx 1080 \text{ N}.$$

Because real sit–stand movements are not purely static, a dynamic amplification factor of

$$k_{\text{dyn}} = 1.5$$

is applied to account for users dropping into the seat or pushing off more rapidly than a quasi–static load. An additional factor

$$k_{\text{unc}} = 1.25$$

covers modelling uncertainties and uneven load sharing between bolts. Together, these factors give a conservative design vertical reaction at the cushion

$$\begin{aligned} F_{\text{seat}} &= W k_{\text{dyn}} k_{\text{unc}} \approx 1080 \times 1.5 \times 1.25 \\ &\approx 2.0 \text{ kN}. \end{aligned}$$

The mechanism is assumed to be symmetric, so this reaction is shared equally between the left and right sides of the device. Each side therefore carries

$$F_{\text{side}} = \frac{F_{\text{seat}}}{2} \approx 1.0 \text{ kg}$$

through the base–to–mechanism side joint. The piston joint must transmit essentially the full F_{seat} between the base and the actuator, while the stabiliser joints primarily see lateral loads when the user leans; a conservative lateral design load of

$$F_{\text{stab}} = 0.3 \text{ kN}$$

per stabiliser is taken for the analysis.

The structural fasteners are metric M8 bolts of property class 8.8, with a nominal yield strength

$$S_{y,b} = 640 \text{ MPa}.$$

The smaller stabiliser joints use M3 machine screws of similar grade. The base plate and side links are manufactured from an aluminium alloy with yield strength

$$S_{y,Al} = 250 \text{ MPa},$$

representative of a 6000–series structural aluminium.

A design factor of safety of

$$n_d = 1.5$$

is applied to both materials, giving allowable normal stresses

$$\sigma_{b,\text{allow}} = \frac{S_{y,b}}{n_d} = \frac{640}{1.5} \approx 427 \text{ MPa},$$

$$\sigma_{Al,\text{allow}} = \frac{S_{y,Al}}{n_d} = \frac{250}{1.5} \approx 167 \text{ MPa},$$

and corresponding allowable shear stresses (von Mises)

$$\tau_{b,\text{allow}} \approx 0.577 \sigma_{b,\text{allow}} \approx 0.577 \times 427 \approx 246 \text{ MPa},$$

$$\tau_{Al,\text{allow}} \approx 0.577 \sigma_{Al,\text{allow}} \approx 0.577 \times 167 \approx 96 \text{ MPa}.$$

All joints are idealised as single-shear connections, with fasteners primarily loaded in shear rather than tension. The bolts are treated as pins that do not carry bending moment, and friction from preload is neglected when calculating capacities, giving conservative estimates of the required failure loads. Threads are positioned so that the smooth shank, not the threaded portion, lies in the shear plane of the main structural joints [17] [19].

4.2.2 Methods

The analysis follows the DESN2000 fastener workshop approach [19]. Rather than presenting a full free-body diagram, the load path is described verbally. The cushion load from the user is first transferred into the cushion base, then into the piston bracket at the centre and the two mechanism links on either side, before finally being carried to the ground through the rest of the frame. In this description, three fastener groups are of interest: the base-piston joint, the base-to-mechanism side joints and the stabiliser joints at the rear of the base.

Conceptually, the cushion is subjected to a downward force equal to the design user load. This is resisted by an upward reaction at the piston joint and two upward reactions at the side joints. Because the mechanism is symmetric, the side reactions are taken as equal and each side carries half of the remaining load after the piston reaction. The stabiliser joints are not part of this primary vertical load path, but they experience horizontal forces when the user leans to one side or when the device sits on slightly uneven flooring, so a separate lateral design load is assigned to them.

For each joint, the geometry of the connection is obtained from the CAD model: plate thicknesses, bolt diameters, distances from bolt centre to the nearest free edge and the net width of material behind the bolt row. From these dimensions, four key areas are defined: the total bolt shear area A_{shear} , the bearing area between the bolt and plate A_{bearing} , the edge shear-out area A_{edge} and the net cross-section in tension A_{net} .

The same basic formula is used for every failure mode:

$$F_{\text{req}} = A \sigma_{\text{allow}} \quad \text{or} \quad F_{\text{req}} = A \tau_{\text{allow}},$$

where F_{req} is the force required to reach the allowable stress on that particular failure path. Because $1 \text{ MPa} \times 1 \text{ mm}^2 = 1 \text{ N}$, the areas can be kept in mm^2 and the stresses in MPa to yield forces directly in newtons.

The five standard failure modes checked for each joint are: (1) bolt or screw shear, (2) bolt or screw bearing, (3) bearing of the aluminium plate, (4) edge shear-out of the aluminium and (5) net-section tensile failure of the aluminium member. The numerical working for each joint is shown in the following subsections, and the complete sets of required forces are summarised in tables.

4.2.3 Fastener geometry and materials

The base-piston joint connects the 5 mm thick aluminium base plate to a 15 mm thick piston mounting block using four M8 flat-head bolts arranged approximately in a rectangle around the piston axis. Two bolts sit on each side of the piston, providing both vertical load transfer and rotational stability. Bolt centres are set so that the distance from each centre to the nearest free edge of the base plate, in the direction of shear, is about 18 mm. Washers are used under the nuts to spread local bearing pressure on

the aluminium and to reduce surface damage.

Each base-to-mechanism side joint uses two M8 bolts to secure a 20 mm thick side link to the 5 mm base plate. The bolts are aligned vertically along the load path and share the reaction taken by that side of the mechanism. The same nominal edge distance of approximately 18 mm is maintained, and the effective net width of base material behind the bolt pair is roughly 34 mm. The bolts are standard M8×25 mm and M8×12.5 mm items, matching the bill of materials.

The stabiliser joints use two M3×20 mm machine screws per stabiliser block. The screws clamp a 10 mm thick stabiliser feature to a 5 mm aluminium plate. Their centres are set around 9 mm from the nearest free edge, and the net width of material behind the pair is about 19 mm. Although smaller, these screws only need to resist lateral forces associated with tipping and rocking, not the full user weight.

4.2.4 Results: base-piston bolted joint

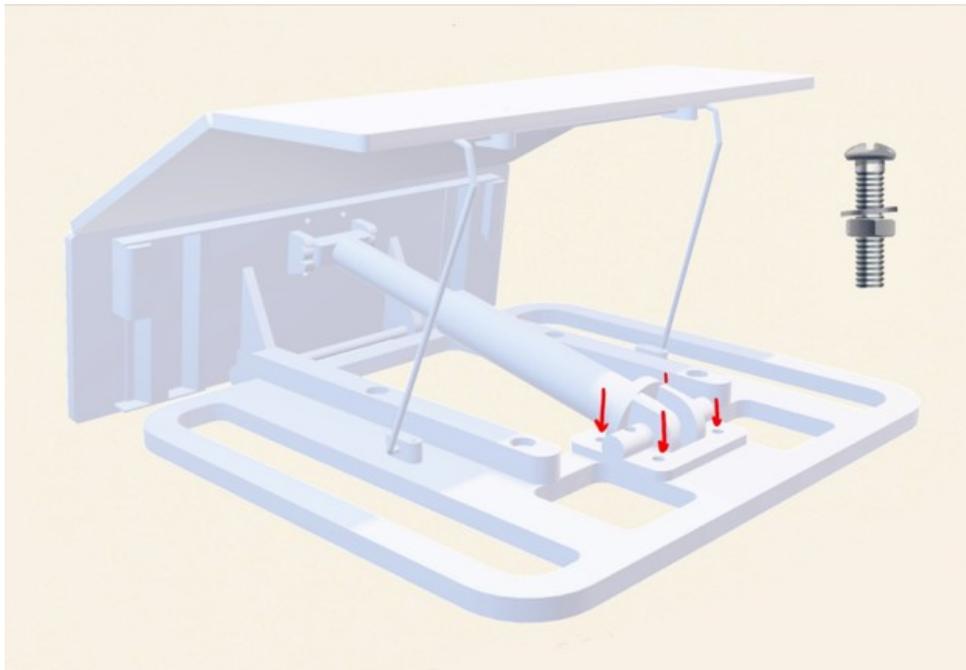


Figure 11: Shows the base-piston bolted joints location

For the base - piston connection as shown in Figure ??, the design shear load is

$$F_{\text{design}} = F_{\text{seat}} = 2.0 \text{ kN}.$$

The four M8 bolts are in single shear, so the total bolt shear area is

$$\begin{aligned} A_{\text{shear}} &= 4 \times \frac{\pi d^2}{4} \\ &= 4 \times \frac{\pi(8)^2}{4} \\ &= 4 \times 50.3 \approx 201 \text{ mm}^2. \end{aligned}$$

Using the bolt shear allowable, the force required to cause bolt shear failure is

$$\begin{aligned} F_{\text{req, bolt shear}} &= A_{\text{shear}} \tau_{\text{b,allow}} \\ &\approx 201 \times 246 \\ &\approx 49500 \text{ N} = 49.5 \text{ kN}. \end{aligned}$$

The projected bearing area in the base plate is

$$\begin{aligned} A_{\text{bearing}} &= 4 \times d \times t_{\text{base}} \\ &= 4 \times 8 \times 5 = 160 \text{ mm}^2, \end{aligned}$$

so the required force for aluminium bearing is

$$\begin{aligned} F_{\text{req, Al bearing}} &= A_{\text{bearing}} \sigma_{\text{Al,allow}} \\ &= 160 \times 167 \\ &\approx 26700\text{N} = 26.7 \text{ kN}. \end{aligned}$$

The strip of base material between the bolt line and the free edge provides an edge shear-out area

$$\begin{aligned} A_{\text{edge}} &= 8 \times a \times t_{\text{base}} = 8 \times 14 \times 5 \\ &= 560 \text{ mm}^2, \end{aligned}$$

so the force required to cause shear-out is

$$\begin{aligned} F_{\text{req, edge shear}} &= A_{\text{edge}} \tau_{\text{Al,allow}} \\ &= 560 \times 96 \approx 53900 \text{ N} = 53.9\text{kN}. \end{aligned}$$

The net tension area behind the bolt row is at least

$$A_{\text{net}} \approx 170\text{mm}^2,$$

giving

$$\begin{aligned} F_{\text{req, net tension}} &= A_{\text{net}} \sigma_{\text{Al,allow}} \\ &\approx 170 \times 167 \approx 28300\text{N} = 28.3\text{kN}. \end{aligned}$$

Carrying out the same process for bolt bearing in the steel gives the values listed in Table 9.

Table 9: Required force for each failure mode (base-piston joint)

Failure mode	Required force / kN
Screw shear failure	49.5
Screw bearing failure	68.3
Bearing failure of base plate	26.7
Edge shear-out of base plate	53.9
Net-section tensile failure	≥ 28.3

The lowest value corresponds to bearing failure of the aluminium plate around the bolt holes, at approximately 26.7 kN. Figure 12 graphs these values against the 2.0 kN, design load.

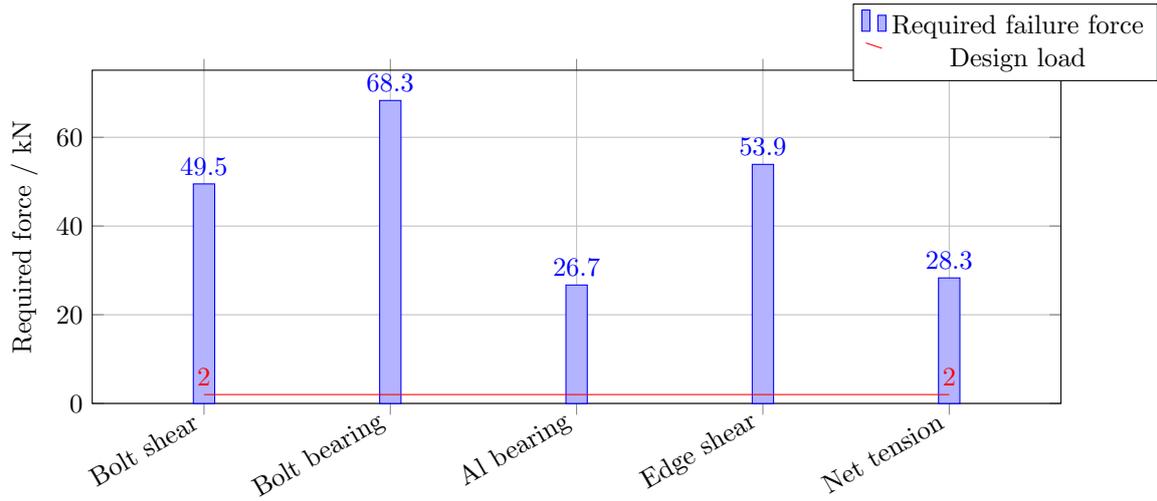


Figure 12: Failure mode capacities and design load for the base-piston joint.

4.2.5 Results: base-to-mechanism side joints

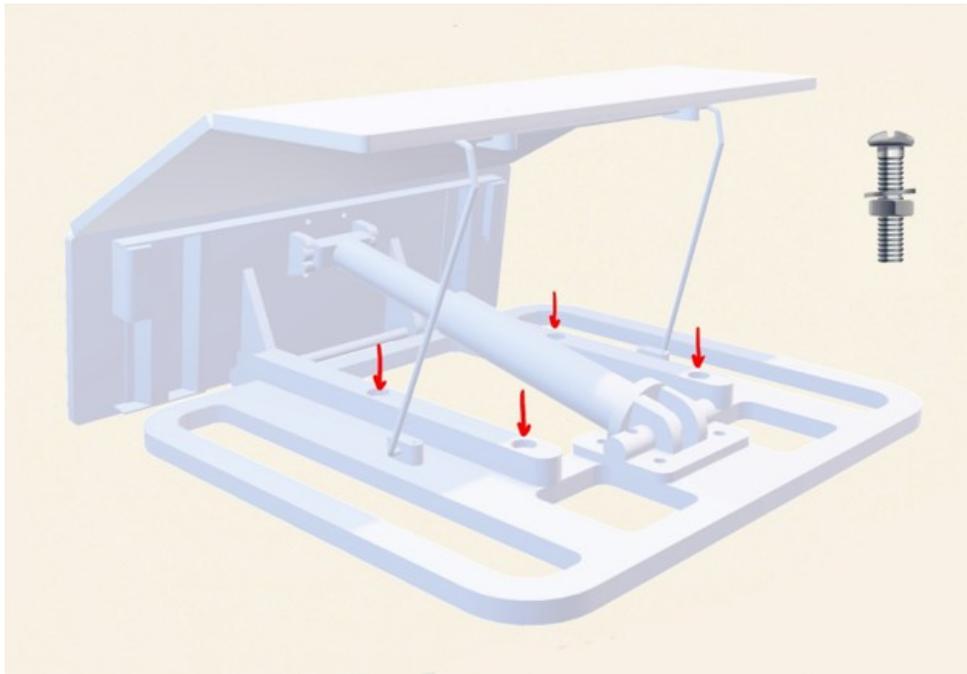


Figure 13: Base-piston bolted joints location

Each base - to - mechanism side joint as shown in Figure 13 carries

$$F_{\text{design}} = F_{\text{side}} = 1.0 \text{ kN.}$$

With two M8 bolts in single shear, the total bolt shear area on one side is

$$A_{\text{shear}} = 2 \times \frac{\pi d^2}{4} = 2 \times 50.3 \approx 100.5 \text{ mm}^2,$$

giving a required force for bolt shear failure of

$$\begin{aligned} F_{\text{req, bolt shear}} &= A_{\text{shear}} \tau_{b,\text{allow}} \\ &\approx 100.5 \times 246 \approx 24\,700 \text{ N} = 24.7 \text{ kN.} \end{aligned}$$

The bearing area in the base plate is

$$A_{\text{bearing}} = 2 \times d \times t_{\text{base}} = 2 \times 8 \times 5 = 80 \text{ mm}^2,$$

so the aluminium bearing capacity is

$$\begin{aligned} F_{\text{req, Al bearing}} &= A_{\text{bearing}} \sigma_{\text{Al,allow}} \\ &= 80 \times 167 \approx 13\,300 \text{ N} = 13.3 \text{ kN}. \end{aligned}$$

The edge strip area and net tension area are

$$\begin{aligned} A_{\text{edge}} &= 4 \times a \times t_{\text{base}} = 4 \times 14 \times 5 = 280 \text{ mm}^2, \\ A_{\text{net}} &= b_{\text{net}} \times t_{\text{base}} = 34 \times 5 = 170 \text{ mm}^2, \end{aligned}$$

which lead to required forces of

$$\begin{aligned} F_{\text{req, edge shear}} &= A_{\text{edge}} \tau_{\text{Al,allow}} = 280 \times 96 \approx 26\,900 \text{ N} = 26.9 \text{ kN}, \\ F_{\text{req, net tension}} &= A_{\text{net}} \sigma_{\text{Al,allow}} \approx 170 \times 167 = 28\,300 \text{ N} = 28.3 \text{ kN}. \end{aligned}$$

The complete set of values is shown in Table 10.

Table 10: Required force for each failure mode (base-to-mechanism side joint, one side)

Failure mode	Required force / kN
Screw shear failure	24.7
Screw bearing failure	34.1
Bearing failure of base plate	13.3
Edge shear-out of base plate	26.9
Net-section tensile failure	28.3

The smallest value is again the aluminium bearing capacity. Comparing 13.3 kN with the 1.0 kN design load gives. Figure 14 graphs these values against the 1.0 kN design load.

$$\text{FoS}_{\text{side}} = \frac{13.3}{1.0} \approx 13.$$

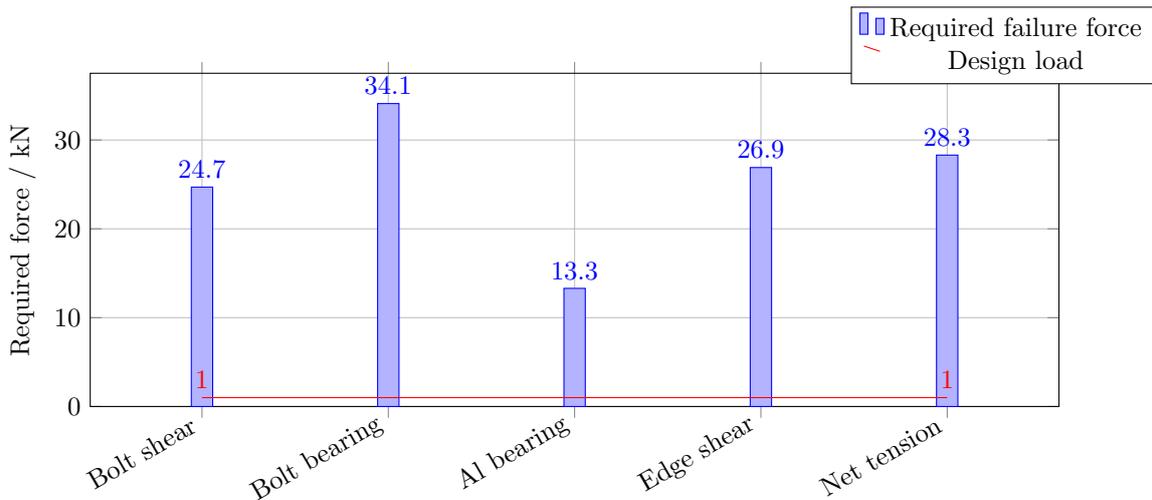


Figure 14: Failure mode capacities and design load for one base-to-mechanism side joint.

4.2.6 Results: stabiliser joints

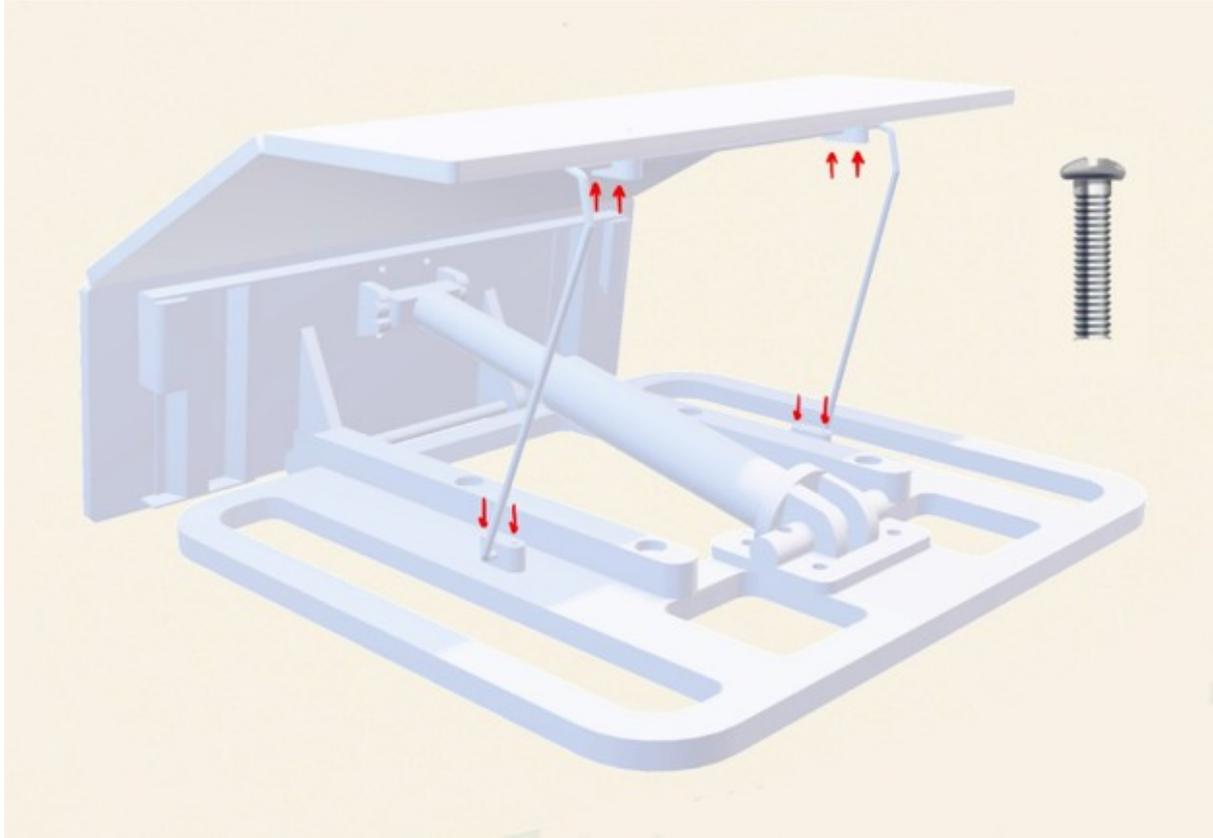


Figure 15: Shows the base-to-mechanism side joints location

Each stabiliser joint as shown in Figure 15 is checked against a lateral design load of

$$F_{\text{design}} = F_{\text{stab}} = 0.3 \text{ kN}.$$

With two M3 screws in single shear, the total shear area is

$$\begin{aligned} A_{\text{shear}} &= 2 \times \frac{\pi d_3^2}{4} = 2 \times \frac{\pi(3)^2}{4} \\ &\approx 14.1 \text{ mm}^2, \end{aligned}$$

and the corresponding required force for screw shear is

$$\begin{aligned} F_{\text{req, screw shear}} &= A_{\text{shear}} \tau_{\text{b,allow}} \\ &\approx 14.1 \times 246 \approx 3500 \text{ N} = 3.5 \text{ kN}. \end{aligned}$$

The aluminium bearing area at the holes is

$$A_{\text{bearing}} = 2 \times d_3 \times t_3 = 2 \times 3 \times 5 = 30 \text{ mm}^2,$$

giving an aluminium bearing capacity of

$$\begin{aligned} F_{\text{req, Al bearing}} &= A_{\text{bearing}} \sigma_{\text{Al,allow}} \\ &= 30 \times 167 \approx 5000 \text{ N} = 5.0 \text{ kN}. \end{aligned}$$

For the edge strip and net section we have

$$\begin{aligned} A_{\text{edge}} &= 4 \times a_3 \times t_3 = 4 \times 7.5 \times 5 = 150 \text{ mm}^2, \\ A_{\text{net}} &= b_{\text{net},3} \times t_3 = 19 \times 5 = 95 \text{ mm}^2, \end{aligned}$$

leading to required forces of

$$F_{\text{req, edge shear}} = A_{\text{edge}} \tau_{\text{Al,allow}} = 150 \times 96 \approx 14\,400 \text{ N} = 14.4 \text{ kN},$$

$$F_{\text{req, net tension}} = A_{\text{net}} \sigma_{\text{Al,allow}} = 95 \times 167 \approx 15\,800 \text{ N} = 15.8 \text{ kN}.$$

The complete set of required forces is listed in Table 11 and shown in Figure 16.

Table 11: Required force for each failure mode (stabiliser joint)

Failure mode	Required force / kN
Screw shear failure	3.5
Screw bearing failure	12.8
Bearing failure of base plate	5.0
Edge shear-out of base plate	14.4
Net-section tensile failure	15.8

Here the limiting mode is again aluminium bearing, at 5.0 kN. Compared with the 0.3 kN lateral design load, this gives

$$\text{FoS}_{\text{stab}} = \frac{5.0}{0.3} \approx 17.$$

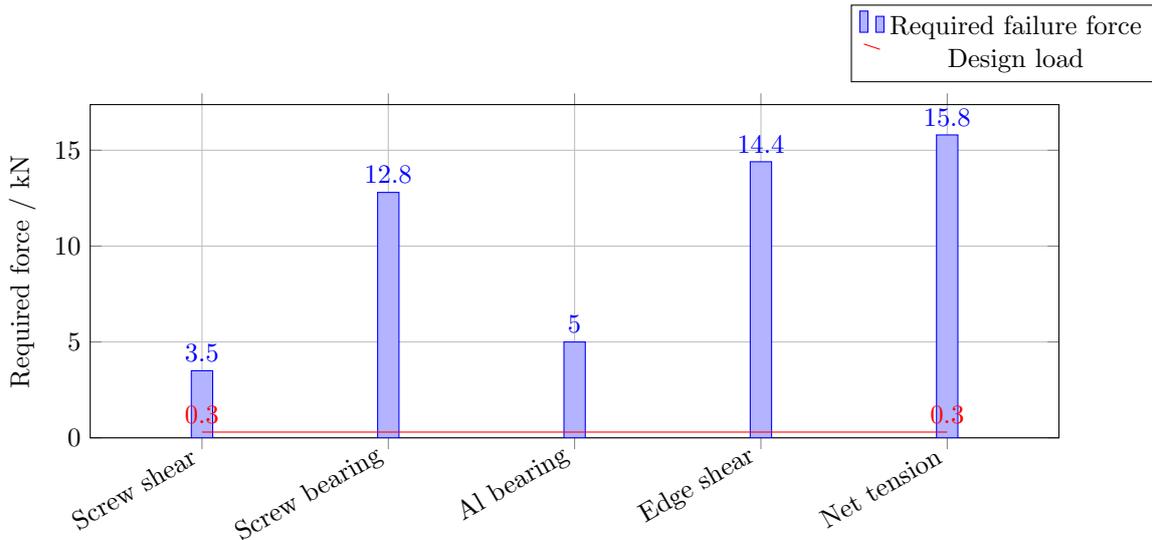


Figure 16: Failure mode capacities and design load for one stabiliser joint.

4.2.7 Discussion and design justification

Across all three bolted connections in the cushion base, a consistent pattern emerges. The fasteners themselves, made from property class 8.8 steel, have much higher capacities than the surrounding aluminium plates. In every joint, the governing failure mode is bearing deformation of the aluminium around the bolt or screw holes, not shearing of the fasteners. This is a typical outcome in bolted aluminium structures and is often preferred, as it means the less brittle aluminium would begin to yield locally before any sudden bolt fracture occurs, giving some warning and redistributing load to neighbouring fasteners [17].

The calculated safety factors for the controlling aluminium bearing mode are approximately 13 for both the base-piston joint and the base-to-mechanism side joints, and around 17 for the stabiliser joints.

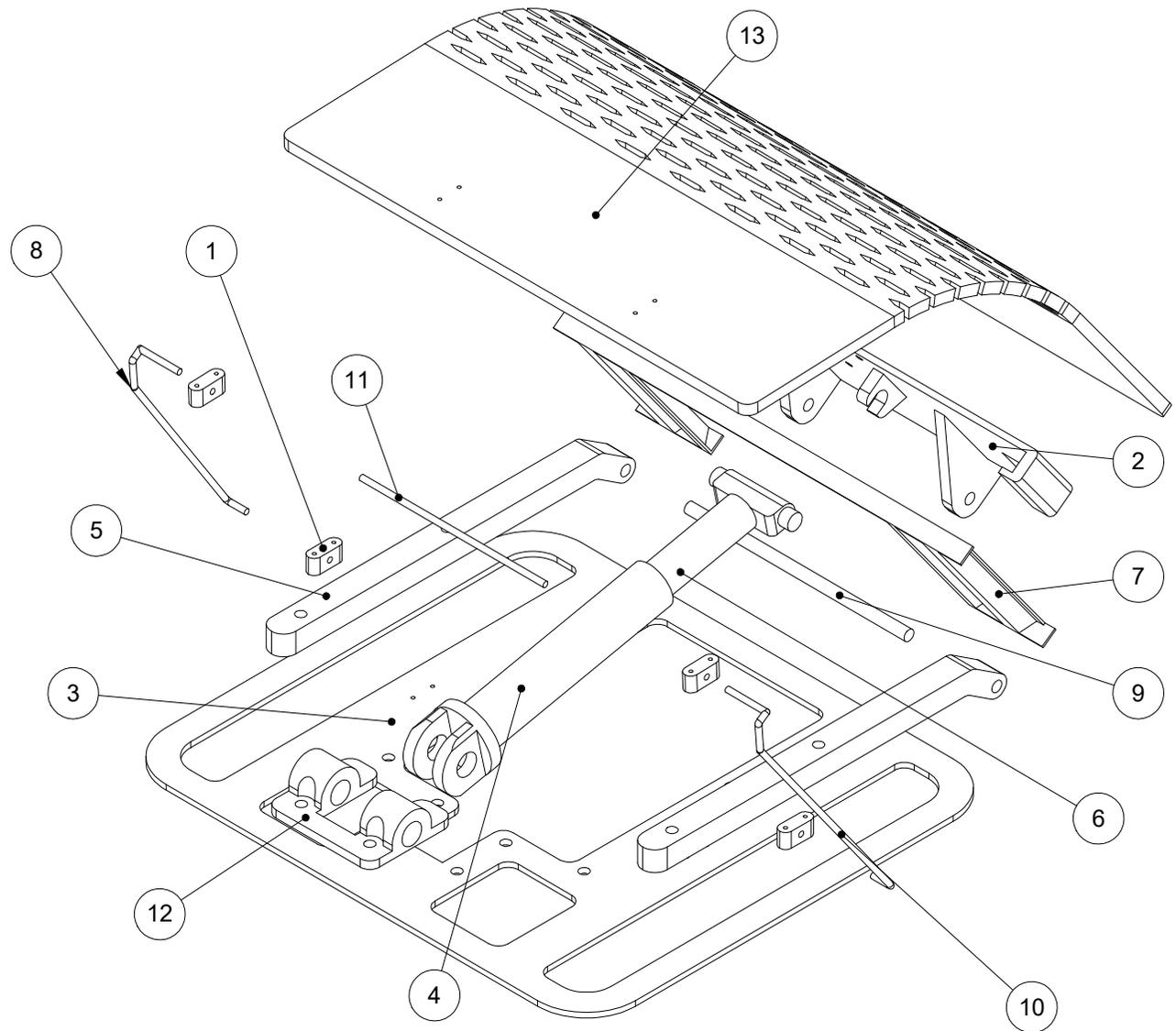
These values are considerably higher than typical minimum factors for static mechanical design and reflect a deliberate decision to build in generous margins, given the uncertainties in user behaviour and the vulnerability of the target user group. The results also show that increasing the bolt grade would not materially improve the safety of the design, because the aluminium members, not the bolts, are the limiting components. Any future optimisation aimed at reducing mass or material usage would therefore need to adjust plate thicknesses, edge distances and bolt spacing rather than bolt strength.

From a practical perspective, the use of standard metric bolts and screws keeps manufacturing and maintenance simple. Bolted joints allow the device to be assembled, inspected and disassembled for transport or part replacement, which is advantageous for a product intended for daily use in domestic environments. Compared with welding or permanent riveting, they also avoid heat-affected zones and allow damaged parts to be replaced without scrapping the entire base assembly [20]. Overall, the calculations and failure-mode plots demonstrate that the selected fastener arrangements in the cushion base comfortably meet the design loads with substantial safety margins and provide a sound engineering basis for the device's structural performance.

5 CAD Model and Mechanical Part Drawings



PART NO.	DRAWING NO.	PART NAME	DESCRIPTION	QTY.
1	9	SUPPORT HOUSING	-	4
2	7	SEAT BASE MOUNT	-	1
3	1	BASE PLATE	-	1
4	-	GAS PISTON 1	OFF THE SHELF SINGLE UNIT WITH GP2	1
5	2	FRAME RAIL	-	2
6	-	GAS PISTON 2	OFF THE SHELF SINGLE UNIT WITH GP1	1
7	8	SLIDING PLATE	-	1
8	4	REAR SUPPORT LEFT	-	1
9	-	PIVOT	OFF THE SHELF	1
10	5	REAR SUPPORT RGT	-	1
11	-	LINKING ROD	OFF THE SHELF	1
12	3	PISTON BOTTOM MOUNT	-	1
13	6	SEAT BASE	-	1



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SURFACE FINISH UNLESS NOTED OTHERWISE

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TITLE

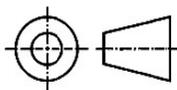
EASYUP ASSEMBLY DRAWING

DRAWN BY

MUNEEL NAQVI (Z5480559)

DATE

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N/A

MATL

N/A

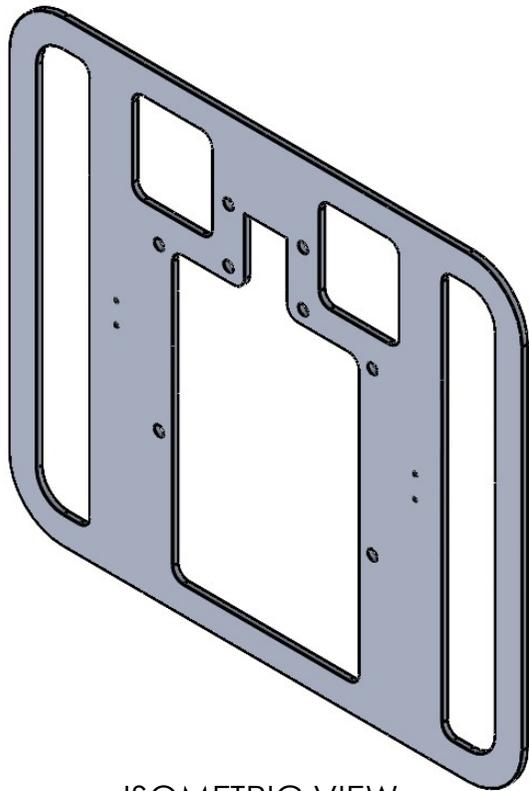
SCALE

1:5

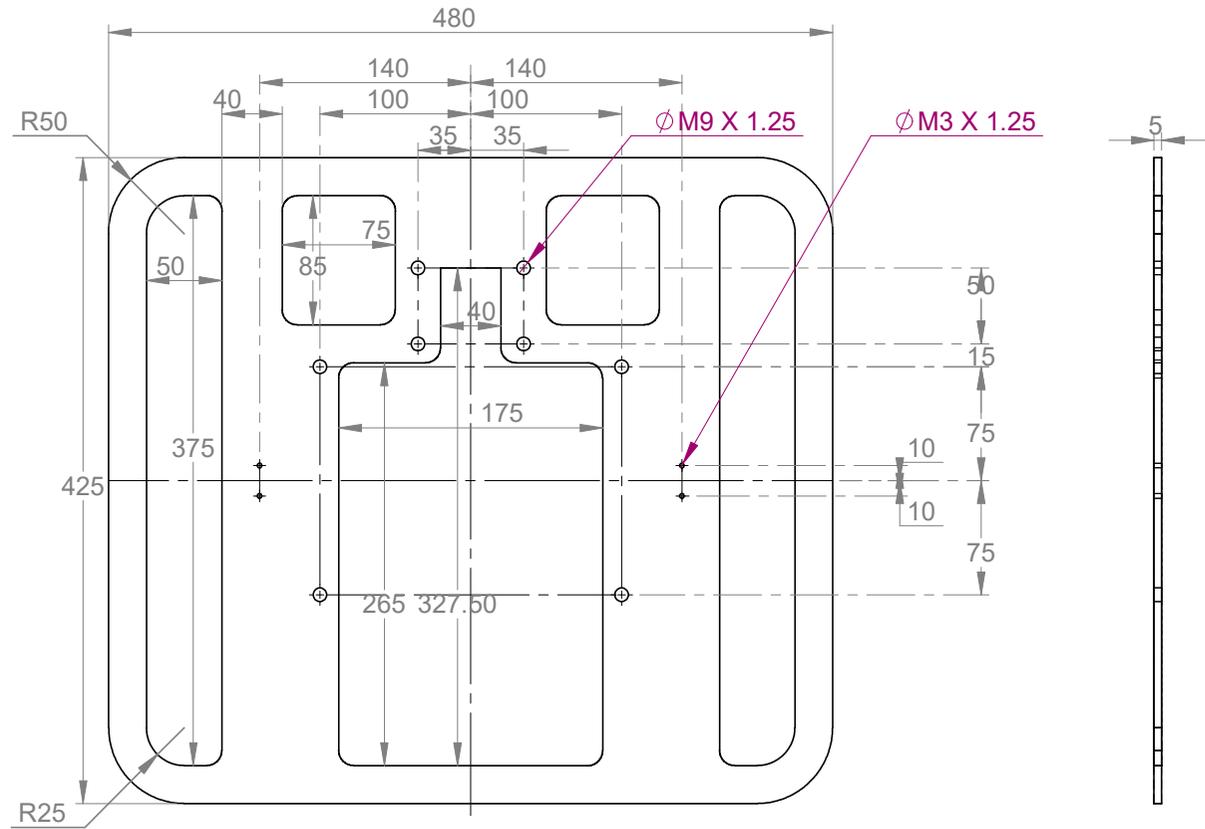
DO NOT SCALE

DIMENSION IN MILLIMETRES

A4



ISOMETRIC VIEW



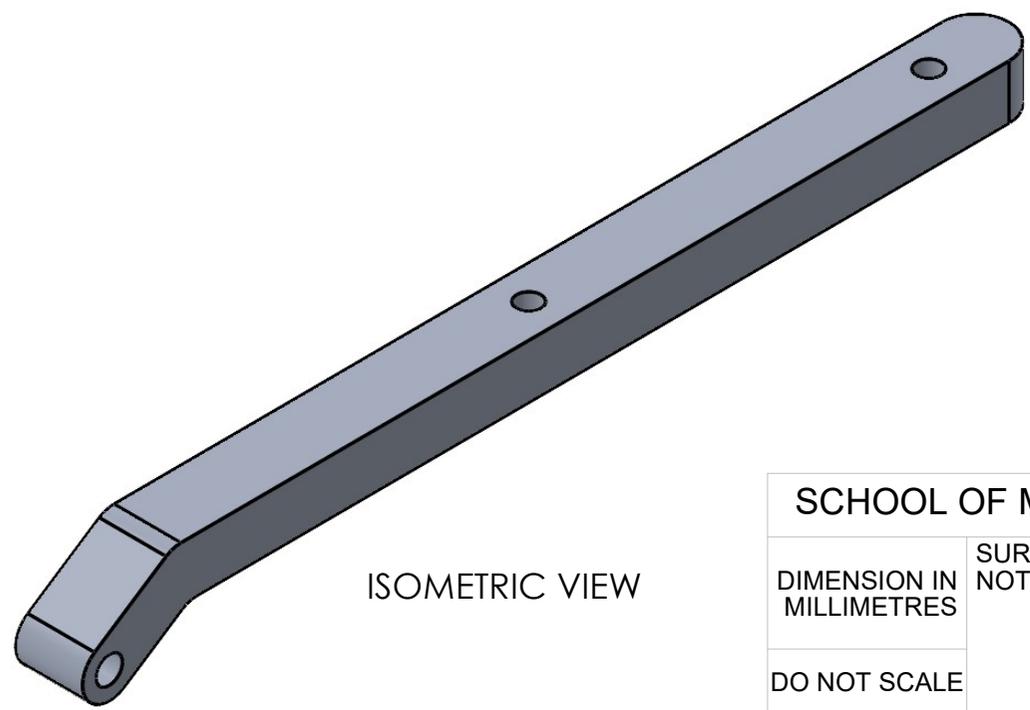
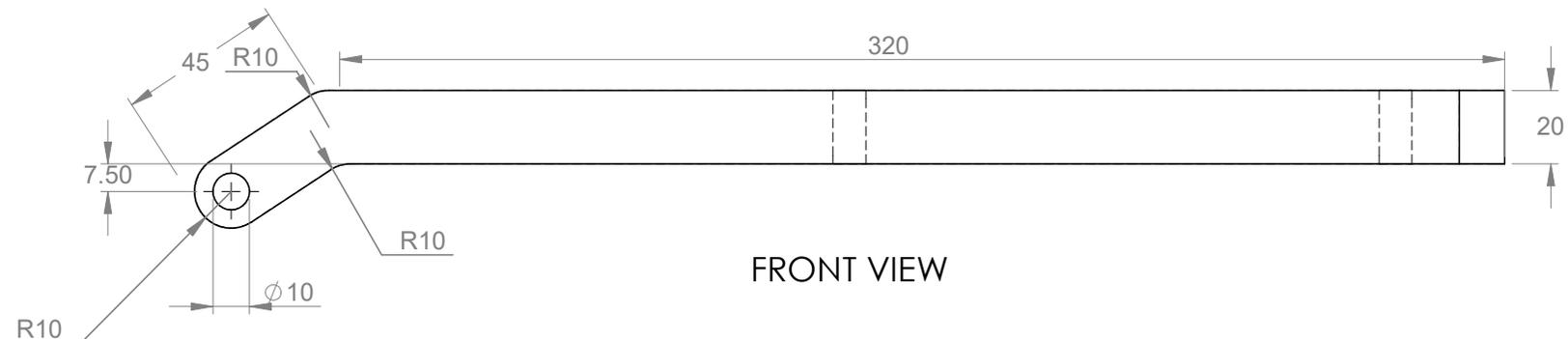
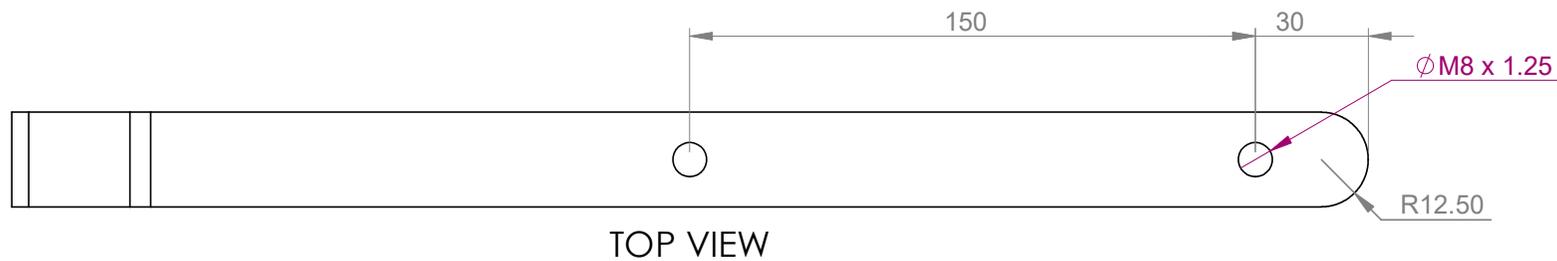
FRONT VIEW

RIGHT SIDE VIEW

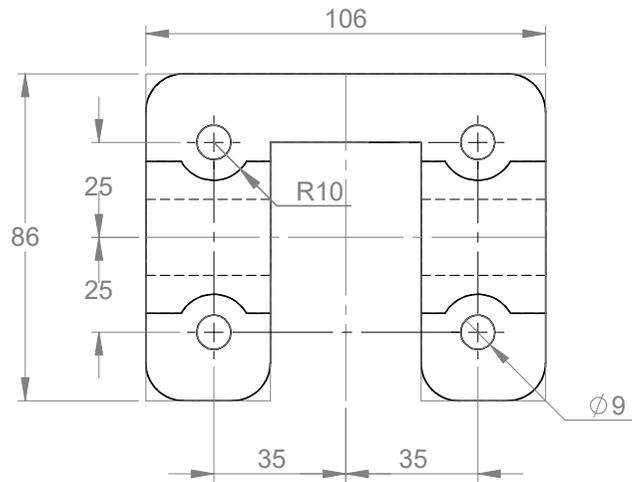
SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

DIMENSION IN MILLIMETRES	SURFACE FINISH UNLESS NOTED OTHERWISE ✓	DRAWN BY MUNEEL NAQVI (Z5480559)		TITLE BASE PLATE	
		CHECKED BY WILSON ZHAO (Z5591831)		DRAWING NUMBER 1	
DO NOT SCALE	TOLERANCE UNLESS NOTED OTHERWISE ±0.05	APPROVED BY WILSON ZHAO (Z5591831)		FIRST RELEASE DATE 29/11/2025	
		QTY 1	MATL ALLOY ALUMINIUM	SCALE 1:5	REV 1

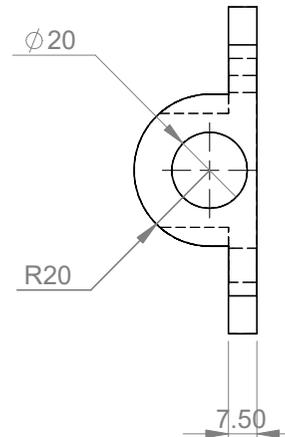
A4



SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW							
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		MUNEEL NAQVI (Z5480559)		FRAME RAIL			
DO NOT SCALE	✓	CHECKED BY		DRAWING NUMBER			
		WILSON ZHAO (Z5591831)		2			
	TOLERANCE UNLESS NOTED OTHERWISE	APPROVED BY		FIRST RELEASE DATE			
		WILSON ZHAO (Z5591831)		29/11/2025			
	±0.05	QTY	MATL	SCALE	REV	DATE	A4
		2	ALLOY ALUMINIUM	1:2	1	29/11/2025	

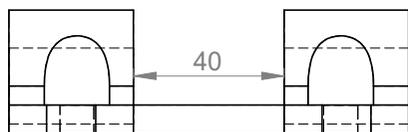


FRONT VIEW

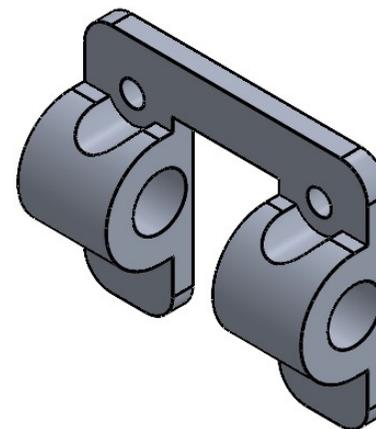


RIGHT SIDE VIEW

FILLETS R10



BOTTOM VIEW

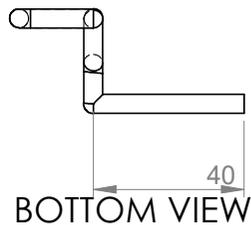
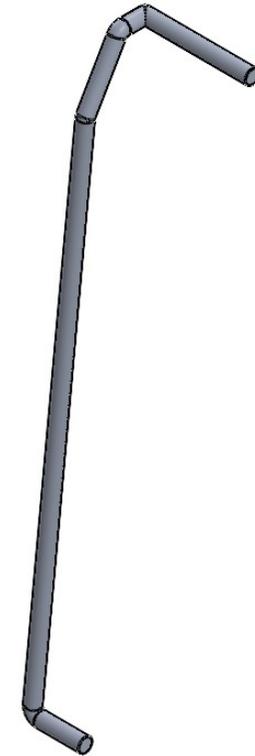
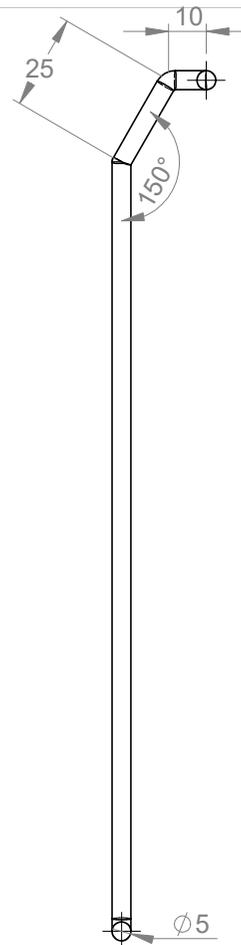
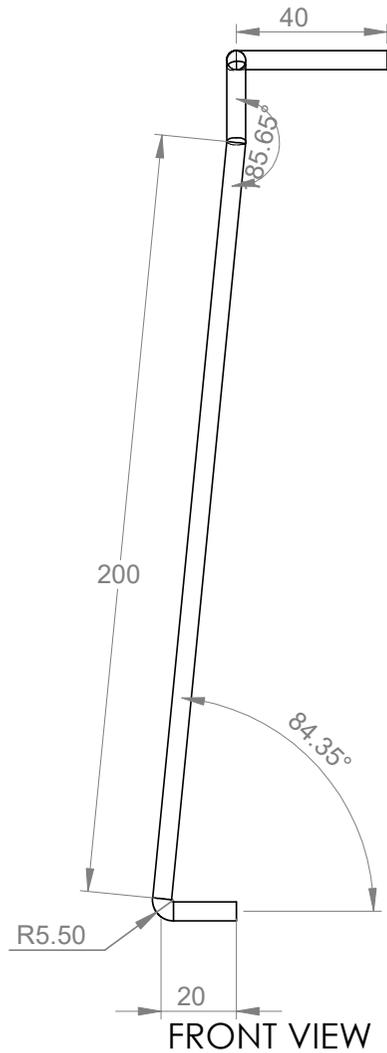


ISOMETRIC VIEW

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

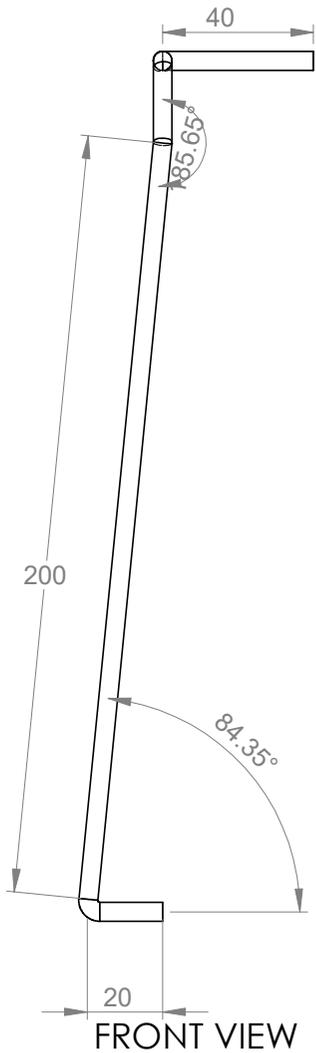
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		CHECKED BY WILSON ZHAO (Z5591831)		DRAWING NUMBER 3		
DO NOT SCALE		APPROVED BY WILSON ZHAO (Z5591831)		FIRST RELEASE DATE 29/11/2025		
	TOLERANCE UNLESS NOTED OTHERWISE ± 0.05	QTY	MATL	SCALE	REV	DATE
		1	ALLOY ALUMINIUM	1:2	1	29/11/2025

A4

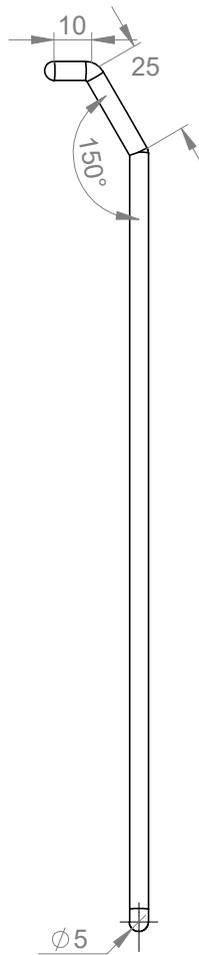


SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

DIMENSION IN MILLIMETRES	SURFACE FINISH UNLESS NOTED OTHERWISE	DRAWN BY		TITLE			
		MUNEEL NAQVI (Z5480559)		REAR SUPPORT LEFT			
DO NOT SCALE		CHECKED BY		DRAWING NUMBER			
		WILSON ZHAO (Z5591831)		4			
	TOLERANCE UNLESS NOTED OTHERWISE	APPROVED BY		FIRST RELEASE DATE			
		WILSON ZHAO (Z5591831)		29/11/2025			
	±0.05	QTY	MATL	SCALE	REV	DATE	A4
		1	ALLOY ALUMINIUM	1:2	1	29/11/2025	



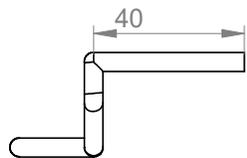
FRONT VIEW



RIGHT SIDE VIEW



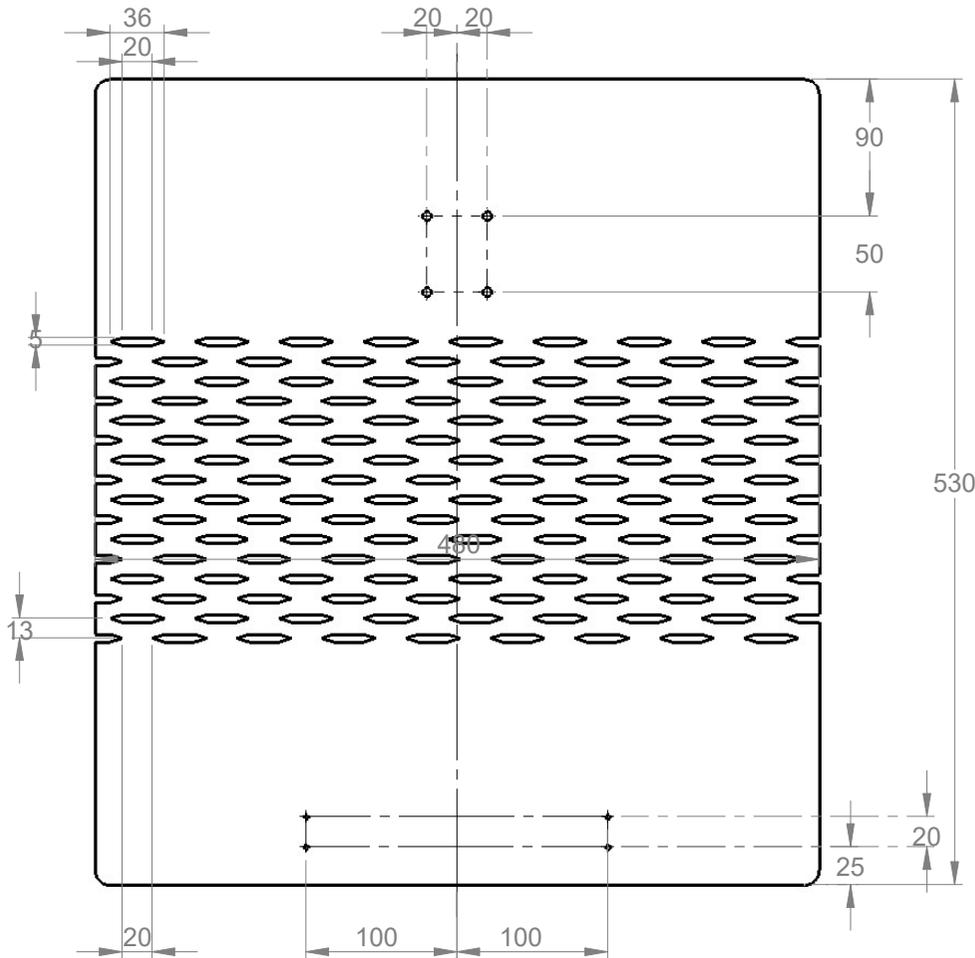
ISOMETRIC VIEW



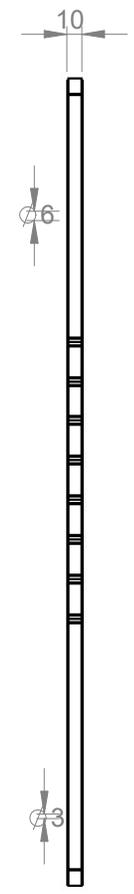
BOTTOM VIEW

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

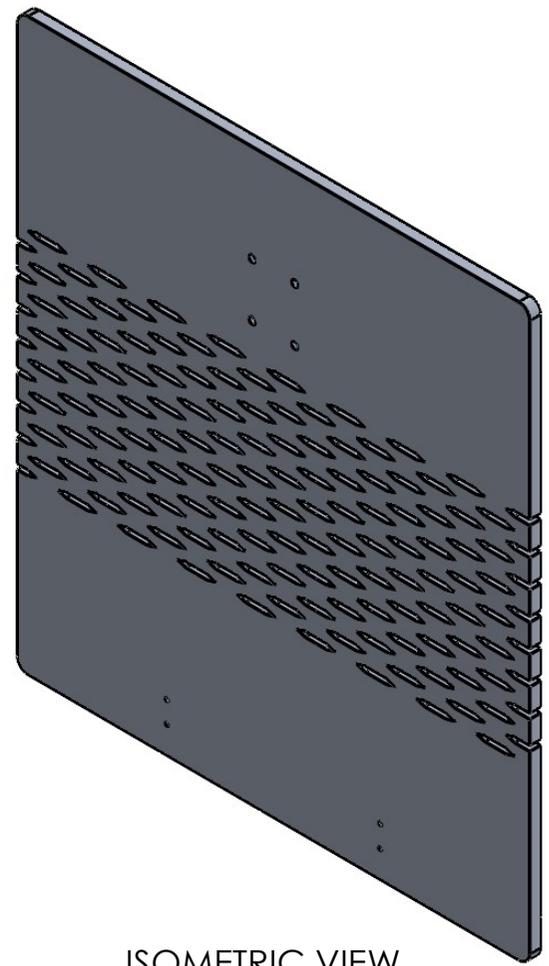
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		MUNEEL NAQVI (Z5480559)		REAR SUPPORT RIGHT			
DO NOT SCALE		CHECKED BY		DRAWING NUMBER			
		WILSON ZHAO (Z5591831)		5			
	TOLERANCE UNLESS NOTED OTHERWISE ± 0.05	APPROVED BY		FIRST RELEASE DATE			
		WILSON ZHAO (Z5591831)		29/11/2025			
		QTY	MATL	SCALE	REV	DATE	A4
		1	ALLOY ALUMINIUM	1:2	1	29/11/2025	



FRONT VIEW



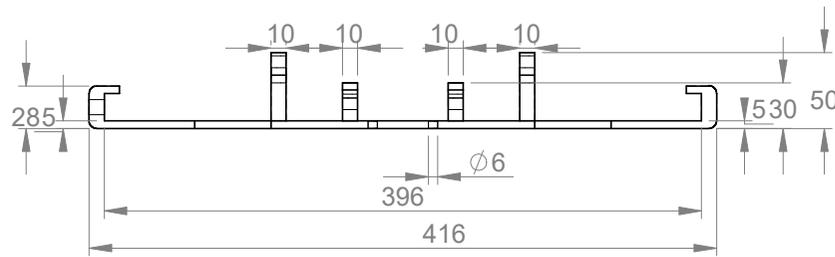
RIGHT SIDE VIEW
FILLET R10



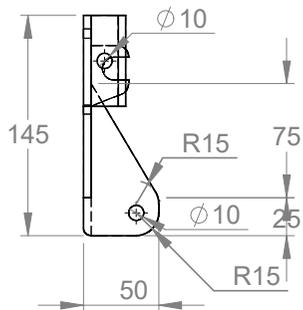
ISOMETRIC VIEW

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

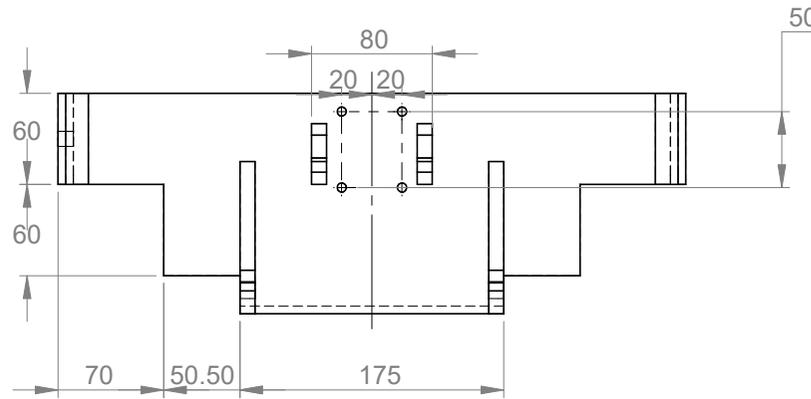
DIMENSION IN MILLIMETRES	SURFACE FINISH UNLESS NOTED OTHERWISE	DRAWN BY		TITLE			
		MUNEEL NAQVI (Z5480559)		SEAT BASE			
DO NOT SCALE		CHECKED BY		DRAWING NUMBER			
		WILSON ZHAO (Z5591831)		7			
	TOLERANCE UNLESS NOTED OTHERWISE	APPROVED BY		FIRST RELEASE DATE			
		WILSON ZHAO (Z5591831)		29/11/2025			
	±0.05	QTY	MATL	SCALE	REV	DATE	A4
		1	PE / PINE COMPOSITE	1:5	1	29/11/2025	



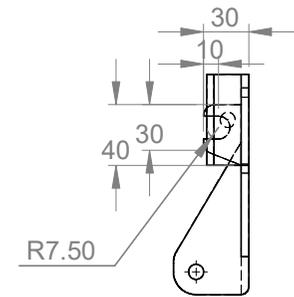
TOP VIEW



LEFT SIDE VIEW

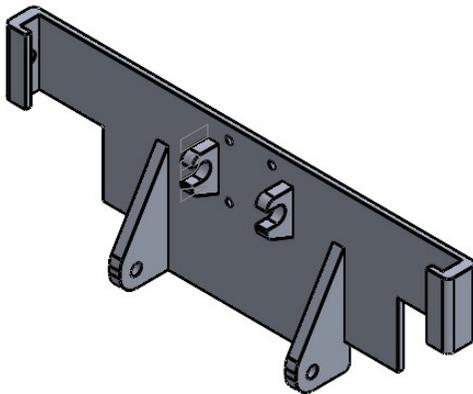


FRONT VIEW



RIGHT SIDE VIEW

FILLETS R5

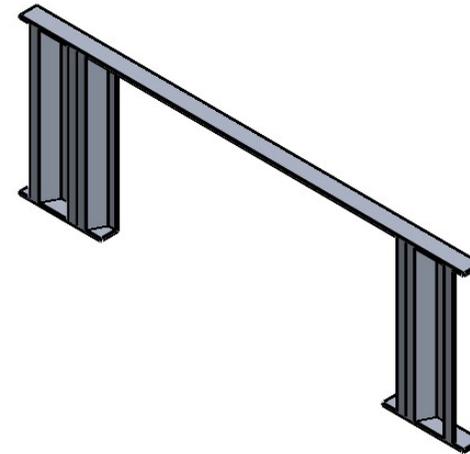
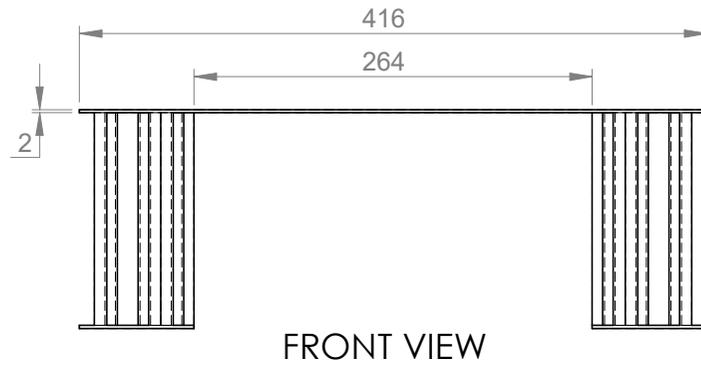


ISOMETRIC VIEW

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

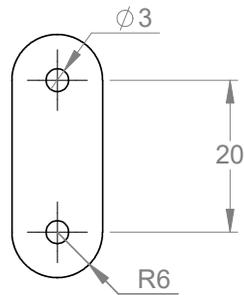
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		CHECKED BY WILSON ZHAO (Z5591831)		DRAWING NUMBER 6	
DO NOT SCALE	TOLERANCE UNLESS NOTED OTHERWISE ±0.05	APPROVED BY WILSON ZHAO (Z5591831)		FIRST RELEASE DATE 29/11/2025	
		QTY 1	MATL ALLOY ALUMINIUM	SCALE 1:5	REV 1

A4

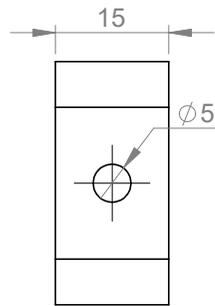


SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

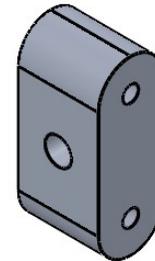
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DO NOT SCALE		CHECKED BY		DRAWING NUMBER			
		WILSON ZHAO (Z5591831)		8			
	TOLERANCE UNLESS NOTED OTHERWISE ± 0.05	APPROVED BY		FIRST RELEASE DATE			
		WILSON ZHAO (Z5591831)		29/11/2025			
		QTY	MATL	SCALE	REV	DATE	
		1	ALLOY ALUMINIUM	1:5	1	29/11/2025	A4



FRONT VIEW



RIGHT SIDE VIEW



ISOMETRIC VIEW

SCHOOL OF MECHANICAL AND MANUFACTURING ENGINEERING - UNSW

DIMENSION IN MILLIMETRES	SURFACE FINISH UNLESS NOTED OTHERWISE 	DRAWN BY MUNEEL NAQVI (Z5480559)		TITLE SUPPORT HOUSING	
DO NOT SCALE		CHECKED BY WILSON ZHAO (Z5591831)		DRAWING NUMBER 9	
	TOLERANCE UNLESS NOTED OTHERWISE ± 0.05	APPROVED BY WILSON ZHAO (Z5591831)		FIRST RELEASE DATE 29/11/2025	
		QTY 4	MATL ALLOY ALUMINIUM	SCALE 1:1	REV 1

A4

6 Conclusion

Overall, EasyUp is a sit-to-stand device developed with the primary goal of supporting elderly users; reducing susceptibility to injury, improving day-to-day safety and bolstering their independence, just to name a few. EasyUp leverages common production materials, simple mechanisms, and user friendly design principles to establish itself as a price effective, simple to use, and easy to repair solution.

Conceptualisation of the product consisted of equal parts market research and technical design methods under the curriculum. Field observations consisted of analysing customer reviews under competing products, and conducting interviews with mobility device distributors. In terms of design methods, our team isolated and developed several functional requirements with function design methods, which were then employed in axiomatic design theory independence design matrices to weigh our 4 concepts against each other, before finally settling on the gas-spring mechanism.

Technical analysis involved material selection and fastener analysis. Due to the sheer amount of parts in the final design, and their varied functions, the team resolved to separately perform Ashby graph analysis for the four major components.

The base plate favoured rigid but lightweight materials, for which we chose 1060 aluminum alloy for its high Young's Modulus to density ratio. The seat shared these same requirements as the base, with the additional, secondary constraint to serve as a comfortable sitting surface, for which the team settled on pine wood, owing to its high rigidity, low mass, and widespread application to similar functions.

The living hinge component followed a completely different set of constraints, requiring high yield stress above all else. Polyethylene (PE) was ultimately deemed suitable for its low modulus of elasticity, and ease of manufacturing into a complex mesh structure via 3D printing or laser cutting.

The Easy-Up cushion base and bearing assemblies comfortably meet the structural and functional requirements of the sit-to-stand device, according to the fastener selection and analysis. Since the aluminium bearing in the base plate consistently governs rather than bolt or screw failure, all critical joints achieve safety factors well above typical design. This also suggests that increasing bolt strength would not significantly increase safety. Future weight or cost optimisation would instead concentrate on bolt spacing, edge distances, and plate thicknesses. Standard metric bolts, screws, and deep-groove ball bearings also make manufacturing, assembly, and maintenance easier by enabling component replacement without the need for specialised tools or heat-affected repairs. When put together, these options offer a reliable, affordable, and serviceable fastening method that promotes the sit-to-stand device's long-term, safe operation.

From a mechanical side, the piston and bearing systems have been chosen to increase the reliability. The team has analysed the bearing life by comparing the expected life cycles to the provided load capacities, to guarantee that bearings meet the life cycle standard. Additionally, gas piston has been considered for reasonable cost and can be maintained easily. Since the pistons are mass produced, leading to low unit price and easy to find a replacement.

In conclusion, the EasyUp sit-to-stand system is not only technically sound, but also practical, cost effective, and aligned with the needs of elderly. By combining robust structural design, choosing best material from the analysis, and manufacturable mechanisms, the final design was ready for future prototyping and user testing. With the final adjustment from stress-strain analysis and cycle-fatigue calculation, the device has the ability to improve safety, independence and quality of life for elderly. The EasyUp project not only achieves its initial engineering objectives but also contributes meaningfully to the challenge of designing accessible assistive technologies for elderly and people with limited mobility.

7 Acknowledgement

Our team wishes to acknowledge the support and assistance we received during the project. We acknowledge our workshop demonstrator, Catherine Hu, who supported us through her guidance on the specification of what needed to be included in our report. She also gave us guidance on the level of detail required in areas such as the Bill of Materials and manufacturability considerations. Her feedback from the presentation also gave our team ideas on where to improve our explanations in the report. We also acknowledge the Course admin staff and our lecturer Ang Liu for their consistent support and provision of material necessary during the development of this report.

We also wish to acknowledge the staff of the mobility aid store Mounties Care Health Concierge whom one of our team members conducted an interview with during the early stages of research. Their responses gave us insights on which customer needs should be prioritised and what our target audience considered when purchasing mobility aid devices. We were able to use this to develop our functional requirements which further aided the development of Easy-Up.

Additionally, we utilised the large language model ChatGPT in a few sections of our report. Notably, our Easy-Up logo was generated by ChatGPT, however, it was later photoshopped to centre the text with the image to suit our team's design preferences. ChatGPT was also used in the initial stages of our research to efficiently gather existing solutions, allowing our team to choose a design which suited our functional requirements and goals. Lastly, ChatGPT was used in certain areas of our technical analysis as clarification on certain analysis methods.

Overall, the quality of AI generated outputs was useful in collating information in a fast manner, however, it is not fully reliable. Due to LLMs not always understanding the full context of our design their responses are at times questionable and have incorrect sections. As such it was required for our team members to cross reference with textbooks and provided lecture slides to ensure their information is correct. None of ChatGPT's outputs were pasted directly into our report and only ever used to gather information and support idea generation.

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A Piston force requirement

Weight

$$W = mg, g \approx 9.81 \text{ m/s}^2$$

$$\therefore W_{40} = 40 \times 9.81 \approx 392 \text{ N}$$

$$\therefore W_{50} = 50 \times 9.81 \approx 490 \text{ N}$$

Therefore the range is 400 – 500 N, and the piston will need to be able to push 400 – 500 N vertically.

Max-incline

$$\theta = 12.47^\circ$$

12.47° based of the CAD model range.

$$F_y = F_{piston} \times \sin \theta$$

$$F_{piston} \times \sin \theta \approx W$$

$$F_{piston} \approx \frac{W}{\sin \theta}$$

For 40kg W_{40} weight:

$$F_{piston} \approx \frac{392}{\sin(12.47^\circ)}$$

$$F_{piston} \approx 1.8 \text{ kN}$$

For 50kg W_{50} weight:

$$F_{piston} \approx \frac{490}{\sin(12.47^\circ)}$$

$$F_{piston} \approx 2.3 \text{ kN}$$

Mid-incline

$$\theta = 7.36 \Rightarrow \sin(7.36^\circ) = 0.1281$$

7.36° based of the CAD model range.

For 40kg W_{40} weight:

$$F_{piston} \approx \frac{392}{0.1281}$$
$$\approx 3.06 \text{ kN}$$

For 50kg W_{50} weight:

$$F_{piston} \approx \frac{490}{0.1281}$$
$$\approx 3.83 \text{ kN}$$

Further 20% – 30% has been taken into account for friction, seat weight and tolerance. Therefore, the piston is roughly 4.0 – 4.8 kN.

B Base Plate Material Selection Calculations

Maximum Density Calculations

$$\begin{aligned}\rho &= \frac{m}{V} = \frac{3}{515143.58 \times 10^{-9}} \\ &= 5823.62 \text{ kg/m}^3 \\ &\approx 5800 \text{ kg/m}^3\end{aligned}$$

Minimum Young's Modulus Calculations

$$\begin{aligned}A &= \text{Sum of contact points between base plate and support frames} \\ &= 2 \times \text{frame rails} + \text{bottom mount} \\ &= 2 \times 7331.36 + 5885.54 \\ &= 21348.26 \text{ mm}^2\end{aligned}$$

$$\begin{aligned}F &= ma \\ &= 120 \times 9.81 \\ &= 1177.2 \text{ N}\end{aligned}$$

$$\begin{aligned}\sigma &= \frac{F}{A} = \frac{1177.2}{21.34828 \times 10^{-3}} \\ &= 55142.66736 \text{ Pa}\end{aligned}$$

$$\begin{aligned}\epsilon &= \frac{\Delta L}{L_0} = \frac{0.02 \text{ mm}}{5 \text{ mm}} \\ &= 4 \times 10^{-3}\end{aligned}$$

$$\begin{aligned}E &= \frac{\sigma}{\epsilon} = \frac{55142.66736}{4 \times 10^{-3}} \\ &= 13785666.84 \text{ Pa} \\ &= 0.0138 \text{ GPa}\end{aligned}$$

$$\begin{aligned}V &= 515143.58 \text{ mm}^3 \\ m &= 3 \text{ kg}\end{aligned}$$

C Seat hinge mass minimisation derivation

Surface strain (max strain):

$$\begin{aligned}\epsilon &= \frac{y}{R} \\ &= \frac{t/2}{R} \\ &= \frac{t}{2R}\end{aligned}$$

Max stress:

$$\begin{aligned}\tau &= E\epsilon \\ &= E\frac{t}{2R}\end{aligned}$$

cannot exceed failure stress σ_f :

$$\begin{aligned}\sigma_f &> E\frac{t}{2R} \\ R &> E\frac{t}{2\sigma_f} \\ R &> \frac{t}{2} \left[\frac{E}{\tau_f} \right]\end{aligned}$$

Minimise R so maximise $\frac{\tau_f}{E}$

$$M = \frac{\sigma_f}{E}$$

D Mass minimisation in relation to maximising stiffness

In order to minimize the weight, we have the mass equation below:

$$m = AL\rho = b^2L\rho. \quad (3)$$

The bending stiff S of the plate must be at least S^*

$$S = \frac{C_2EI}{L^3} \geq S^*, \quad (4)$$

where C_2 is a constant. The second moment of area, I , for a plate is

$$I = \frac{b^4}{12} = \frac{A^2}{12}. \quad (5)$$

For a given length, L, the stiffness S^* is adjusted by adjusting the size of the rectangular section. We can plug either equation of 5 into 4. Thereby, we can eliminate b or A by plugging it back into 3, which gives the equation

$$m = \left(\frac{12S^*}{C_1b} \right)^{1/3} (bL^2) \left(\frac{\rho}{E^{1/3}} \right).$$

E Equivalent Load

The equivalent load P is required to calculate the recommended load, which is a combination of the axial load and the radial load given as

$$P = (XF_r + YF_a)S. \quad (6)$$

The radial load F_r is 1177.2N since we would like to accommodate users up to 120 kg to use the device. Conversely, the mechanism does not have any axial load, the equivalent load formula would simply be

$$P = (XF_r)S. \quad (7)$$

In order to find the X value, we have to use the following equation

$$\frac{F_a}{C_0} \quad \text{and} \quad \frac{F_a}{VF_r}.$$

Since there is no axial load, both equation will be equal to zero. According to the radial load table in Figure E.1, the X value would just be 1.

Table 11–1 Equivalent Radial Load Factors for Ball Bearings

F_a/C_0	e	$F_a/(VF_r) \leq e$		$F_a/(VF_r) > e$	
		X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Figure E.1: Radial load table [17]

Therefore, substituting all the values with a safety factor of 1.2 into equation 7 gives the following

$$\begin{aligned}
 P &= (1.2)(1177.2) \\
 &= 1412.64 \text{ N.}
 \end{aligned}$$

F Bill of Material

PART NO.	DRW NO.	PART NAME	DESC.	QTY	MTRL	MASS PER PART (g)	SOURCE
1	9	SUPPORT HOUSING	-	4	ALLOY ALUMINIUM	13.09	-
2	7	SEAT BASE MOUNT	-	1	ALLOY ALUMINIUM	879.07	-
3	1	BASE PLATE	-	1	ALLOY ALUMINIUM	1390.89	-
4	-	GAS PISTON 1	OFF THE SHELF SINGLE UNIT WITH GP2	1	N/A	450	EXTERNAL SUPPLIER
5	2	FRAME RAIL	-	2	ALLOY ALUMINIUM	474.06	-
6	-	GAS PISTON 2	OFF THE SHELF SINGLE UNIT WITH GP1	1	N/A	450	EXTERNAL SUPPLIER
7	8	SLIDING PLATE	-	1	ALLOY ALUMINIUM	188.11	-
8	4	REAR SUPPORT LEFT	-	1	ALLOY ALUMINIUM	15.69	-
9	-	PIVOT	OFF THE SHELF	1	ALLOY ALUMINIUM	47.71	EXTERNAL SUPPLIER
10	5	REAR SUPPORT RIGHT	-	1	ALLOY ALUMINIUM	15.69	-
11	-	LINKING ROD	OFF THE SHELF	1	ALLOY ALUMINIUM	9.97	EXTERNAL SUPPLIER
12	3	PISTON	-	1	ALLOY ALUMINIUM	204.64	-
13	6	SEAT BASE	-	1	COMPOSITE PE/PINE WOOD	798.91	-