University of Southern Queensland

Faculty of Engineering and Surveying

The Design of a System for the Removal of Underground Air Conditioning Units

A dissertation submitted by

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ABSTRACT

A design methodology for a system to remove and replace industrial air conditioning units in the underground coal mining environment is presented. Traditionally this task was completed with a method that had the potential to place coal mine workers at an unacceptable level of risk, which necessitated the design and implementation of a new method.

A number of concepts where evaluated which ultimately led to the decision to proceed with the detailed design of an interchangeable attachment used on an underground certified skid steer loader. The relevant Australian Standards were identified and used to guide the author toward appropriate design choices. Structural steel of differing grades was identified as most appropriate to use for the fabrication of system, with other standards also used to guide design choices such as the necessary welding techniques, the geometry with which the attachment point subsystem was designed, allowable stresses and recommended maximum deflection of the system.

Two load cases were identified that the system would be subjected to. The most severe load case was used to determine the required geometry and materials with which the primary support members would be fabricated. These members were designed as non-prismatic beams of constant strength due the necessity of mass reduction as the stability of the skid steer loader was identified as a significant design constraint. A rigorous mathematical description of the behaviour of these members is given, which is successfully verified by a finite element analysis.

Several finite element analyses were also conducted to ensure the von Mises stress and the maximum deflection defined in the relevant standards were not exceeded. The load factor and allowable stress factors given in the Australian Standards were used to determine a theoretical factor of safety, which was confirmed by the finite element analysis to be approximately 2.4 which agrees with published design papers and textbooks for this type of mechanical system.

A number of external engineering consultants were engaged for the fabrication of the system. The decision to proceed with one of these vendors was granted, with which the author is currently collaborating. The fabrication and testing that forms part of the certification is part of the ongoing further work that is required to have the system implemented on site.

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ENG4111 & ENG4112 Research Project

Limitations of Use

CERTIFICATION

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

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Acronyms

FEA Finite Element Analysis. x, 4, 22, 24, 34, 51, 57, 61, 62, 63, 65, 68, 69, 70, 75, 80, 91, 93, 95

FMEA Failure Modes and Effects Analysis. viii, ix, xii, 18, 19, 57, 78, 79, 80, 99, 121

FoS Factor of Safety. 11, 12, 13, 19, 20, 21, 26, 49, 50, 94, 95, 96

GMAW Gas Metal Arc Welding. 67

LHD Load, Haul, Dump. ix, 7, 8, 37, 38, 41, 45, 85, 86

LODMAT Lowest One Day Mean Ambient Temperature. 76

NDT Non-Destructive Testing. 80, 99

OEM Original Equipment Manufacturer. 30

POCV Pilot Operated Check Valve. 8

PPE Personal Protective Equipment. 16

PRP 'Product Realization Process'. ix, 27, 28

PTO Power Take Off. 8

SFD Shear Force Diagram. 23

SHS square hollow section. xi, 65, 69, 70, 72, 74, 75, 76, 77, 89, 90, 91, 92, 93, 95

VVVF Variable Voltage Variable Frequency. 2

WHS Work Health and Safety. 1, 2, 3

1 INTRODUCTION

The working environment in an underground coal mine is unlike any other workplace and requires the use of special machinery, equipment and procedures in order to perform the required work without exposing the coal mine workers to an unacceptable level of risk.

Many tasks that are completed regularly as part of the normal operation of a coal mine involve heavy and complex pieces of equipment, and require the use of purpose built machines and attachments to allow for the required manoeuvrability in the often enclosed areas in the underground workings. However, there are many tasks that require the use of manual handling and the carefully coordinated efforts of the underground workers, which exposes those personnel to the many hazards associated with human/machinery interaction.

One such task which requires the use of manual handling, and has the potential for exposing the workers to injury, involves the installation and removal of an industrial air-conditioning (AC) unit mounted to the top of a 1.5 MVA underground substation conveyor belt starter. This is a unique task as it is not one that is performed often, however it has been highlighted that in order to comply with the NSW Work Health and Safety (WHS) legislation, a new method for the completion of this task needs to be designed and implemented.

1.1 The Problem

There are a number of different coal extraction methods used in underground coal mines such as longwall mining and pillar extraction techniques. The one thing in common however is the need for a conveyor belt network to transport the coal from the point of extraction (the coal face) to the surface where it can be processed and ultimately sold to either the domestic or international markets.

These conveyor systems often consist of a number of individual conveyor belts which operate in a series type arrangement where one belt will feed onto the next at a transfer point until the coal has reached the surface. Each conveyor belt is driven by a number of electric motors (two drive motors per conveyor belt is quite common), each coupled to a drive pulley that is in contact with the belt which uses friction between the belt and pulley as the driving force. The belts often have to be started from a stationary position which requires the need for an electric drive system capable of efficiently operating the drive motors over a wide speed range. This is achieved with the use of a variable voltage, variable frequency (VVVF) drive unit housed inside the belt starter substation.

Each conveyor has an individual belt starter substation placed in an underground location close to the belt drive head, from which large electrical cables are run out to the drive motors. During their operation, it is normal for the temperature of the VVVF drive units to increase. To maintain a safe operating temperature, AC units are used which on some belt starters are mounted to the top. Figure 1.1 shows a typical underground belt starter installation with arrows indicating the position of the roof mounted AC units (a general arrangement of a typical AC unit is shown in Appendix B).

(a) Underground belt starter installation longitudinal view.

(b) Underground belt starter installation alternate view.

Figure 1.1: Underground belt starter installation showing the roof mounted AC units (Reid, N 2023, pers. comm., 4 March).

The difficulty in installing and removing the AC units is obvious from the above figure. The physical dimensions of the cut-throughs (c/t) in which the belt starters are located are approximately 5.5 m wide and 3 m high. The width of the belt starter is approximately 3 m, which in a typical belt starter installation, leaves 1.5 m on one side, and 1 m on the other side. This leaves little room for larger machinery to be able to assist in the replacement of the AC units. This has lead to a method in the past which involves the use of manual handling and lifting and slinging techniques that is believed to be in non-compliance the NSW WHS legislation by placing the coal mine workers that perform this task at an unacceptable level of risk.

Any piece of equipment in use on a mine or petroleum site in NSW must comply with Schedule

2 of the *Work Health and Safety (Mines and Petroleum Sites) Regulations (2022)* which details the requirement that the overall life cycle of that equipment be taken into account when determining how the risks to personnel will be controlled. As the replacement of the underground AC units form part of the life cycle of the substations on which they are situated, it has been identified that the risks associated with the installation and removal of these units may not currently be adequately controlled with the existing method of carrying out the work. Therefore a new method of replacing the AC units which minimises the level of manual handling and hence the risk to the coal mine workers is required.

1.2 Project Aim and Objectives

The primary aim of this project is to design and test a system for the safe installation and removal of the industrial AC units mounted to the top of an underground conveyor belt starter substation which complies with all relevant Australian Standards and WHS legislation.

The dimensions of the c/t in which the AC units and the substations on which they are mounted vary in size and grade, and often have other items of electrical infrastructure in close proximity. This increases the complexity of the maintenance requirements and means that a design or system that will satisfy the needs of the project in one location might not be suitable in other locations.

The objectives in order to successfully complete the project are as follows:

1. Determine the requirements and scope of the project by identifying the relevant equipment details including the locations where the AC units are required to be installed and removed, and the physical dimensions and weight of the units. Other constraints such as ground conditions and other items causing access issues will also be identified and evaluated. The restrictions in relation to access to the underground workings will also be evaluated which will aim to ensure compliance with any legislation and on site standards.
2. Review any existing underground material handling processes and the limitations of their use.
3. Identify and evaluate the design limitations of machinery used in underground coal mines for material handling purposes.

4. Develop a design that best meets the requirements and constraints of the project. This will include accurately modelling all reasonably expected load cases, performing a Finite Element Analysis (FEA) on the chosen design and producing the necessary engineering drawings to allow a prototype to be fabricated.
5. Engage with, and communicate with suppliers throughout the design and manufacturing process to ensure the equipment is fabricated to meet and exceed all relevant Australian Standards and legislative requirements.
6. Develop a life cycle management plan for the design.
7. Conduct on site testing of the design.

1.3 Research Questions

In addition to the above mentioned objectives, a number of research questions will aimed to be answered as part of the project and are listed below:

1. What are the appropriate design options for the problem discussed in Section 1.1?
2. What available materials are best suited for the fabrication of the system?
3. What is the recommended maximum displacement and maximum permissible stress values for the components of the design for the intended application?
4. What factor of safety is applicable for the system?

The research questions within the project are intended to guide the author toward appropriate design choices which will aid in the realisation of an end product that is safe and reliable which meets all relevant Australian Standards and legislation. Theoretical techniques that provide values of stresses and displacements similar to those produced with FEA software will be considered and evaluated for their effectiveness and ability to provide model verification which is thought to be a critical exercise to determine the model accuracy. These techniques will also be able to to guide the author toward appropriate engineering choices throughout the design process in an effort to reduce material usage and decrease the overall cost of the system.

1.4 Dissertation Structure

This section will briefly explain the layout and structure of the dissertation with the intention of giving the reader a brief understanding of the information included in each section. The structure of the dissertation is as follows:

- **Literature Review**
  - A review of the available literature is presented with the aim of providing the information required to answer the project research questions. Information discussed will include details relating to equipment handling in the underground environment, material selection, legislation, stress analysis and theoretical techniques required for model verification.
- **Methodology**
  - The methods used to carry out the design project are discussed in this section. A discussion of the recognition of need, conceptualisation, feasibility assessment, preliminary and detailed design, production and testing is included which details the engineering design process that will be followed during the project. The high level design constraints will be discussed in this section. The tools used to model the design are also briefly discussed.
- **Design Concepts**
  - In this section, a number of potentially suitable design options are presented detailing the advantages and disadvantages of each. A parallel discussion of the feasibility of each concept will be presented, with the aim of justifying the choice to proceed with the preferred design.
- **Preliminary Design**
  - The design concept that best meets the higher level constraints will be further discussed in this section, with the aim of introducing the lower level design requirements that guide the decision making process. Load cases will be presented that will used in the finite element analysis as part of the detailed design phase.

- **Detailed Design**
  - The finite element analysis and results will be presented in this section. The results will be compared with theoretical results for the purposes of determining the accuracy and validity of the model used. A failure modes and effect analysis will also be conducted and discussed.
- **Fabrication**
  - The manufacturing processes employed for the fabrication of the system will be discussed in this section with the aim of presenting the methods used that comply with the relevant Australian Standards and legislation.
- **Discussion and Conclusion**
  - This section is devoted to discussing the level to which the project has met the objectives as intended. The results of all analyses and testing will be discussed with a comparison against the expected outcomes. The problems encountered throughout the project and methods used to overcome them will also be presented.
- **Further Work**
  - Any work that is yet to be completed, or improvements that are possible will be presented in this section.

2 LITERATURE REVIEW

This section will aim to identify and evaluate the existing systems currently in use for the purpose of transporting and handling material in an underground environment. The relevant standards and legislative requirements with regard to the design and manufacture of a new system or piece of equipment will also be reviewed.

As it is anticipated that a new type of mechanical piece of equipment will need to be designed and fabricated to achieve the goals of the project, the finite element method, relevant Australian Standards and theoretical techniques will also be explored to demonstrate the understanding of the theoretical and legislative requirements with regard to stress analysis in machine component design.

2.1 Existing Methods for Material Handling

The primary method in which materials are transported in an underground mine is with the use of a machine called a Load, Haul, Dump (LHD) loader. LHDs are used in a number of applications with the use of different attachments which allow them to transport raw materials and consumables such as roof bolts and chemicals, perform roadworks and provide hydraulic power to other specialised attachments. A typical LHD and examples of attachments are shown in Figures 2.1 and 2.2 respectively.

Figure 2.1: A typical LHD used in an underground mine for the transport of materials (shown with bucket attachment) (Impact Mining Equipment 2023).

(a) Typical bucket attachment for use on an LHD.

(b) Typical LHD man-basket for reaching higher areas in an underground coal mine.

Figure 2.2: Examples of attachments available for use with equipment handling machinery in underground coal mines (Impact Mining Equipment 2023).

The attachments are designed to be quickly interchanged with the use of a quick disconnect system. This is achieved with the use of the horns on the front of the LHD that provide the main support for the attachment, and a locking plate operated by a hydraulic cylinder locked in place by a pilot operated check valve (POCV). Various hydraulic attachments are also available which are powered with the use of Power Take Off (PTO) ports on the front of the LHD.

The skid steer loader is another example of a specialised piece of machinery used in an underground coal mine. They are often used for the transport of loose material such as coal and rock with the use of a bucket attachment (Figure 2.3) however they can also perform other unique tasks such as the handling of other more critical components with the use of a fork tyne attachment. The method with which the attachments are interchanged are similar to that of an LHD, however rather than a hydraulic cylinder locking the attachment in place, the locking effect is achieved with the use of manually actuated over-centre locking pins.

Figure 2.3: Typical skid steer loader used in an underground coal mine (shown with hydraulic bucket attachment) (Hunter Plant Hire 2023).

Many skid steer loader attachments are commercially available as described by Himac (2023) and Clark Equipment (2017) which demonstrates the feasibility of designing a purpose built attachment for almost any use deemed necessary. An exhaustive review however has been unable to identify any existing systems or machinery attachments currently in use that meet the requirements and constraints of the problem. The diesel equipment operating constraints shown in Table 1 demonstrate however that the machines available would be adequate if the design and manufacture of an attachment is identified as the preferred method moving forward with the project.

Table 1: Diesel Equipment Operating Constraints (formulated from Hunter Plant Hire (2018a) and Sandvik Mining (2016)).

Machine	Lifting Capacity (kg)	Maximum Grade, long.	Maximum Grade, trans.
LS170	7000	1:4 (14°)	1:8 (7.13°)
LS190	10000	1:4 (14°)	1:8 (7.13°)
LS190S	12000	1:4 (14°)	1:8 (7.13°)
Skid Steer	748	1:8 (7.13°)	1:8 (7.13°)

As new areas are exposed as part of the normal mining operations, roof support must be installed to ensure the integrity of the over laying strata is maintained. This support typically consists of steel roof bolts that are inserted into pre-drilled holes in the roof, and anchored into place with a chemical resin. These roof bolts typically have a threaded portion on the end, which is primarily used for clamping and securing the immediate strata material in the roof to allow it to support itself (Mark, Molinda & Dolinar 2001), however they can also be utilised for the lifting and manoeuvring of larger components with specialised lifting equipment due to the high tensile strengths of the bolts (Jennmar 2023) and the ability of the resin to withstand high pulling forces (Mark et al. n.d.). An image of these roof bolts is shown in Figure 2.4 with an example in Figure 2.5 showing a monorail system which supports a longwall services system that is supported by several roof bolts along its length.

Figure 2.4: Roof support in an underground coal mine used for strata support, also can be used for lifting pieces of equipment (Mining Weekly 2023).

Figure 2.5: A longwall monorail system supported by underground roof bolts (Macquarie Manufacturing 2023).

2.2 Standards

There are numerous Australian Standards which must be referenced when designing a new component to ensure it is manufactured to be as safe and compliant as any member of the community would reasonably expect. It would be unnecessary to discuss in great detail each of the standards believed to be applicable with regard to the problem and potential solutions, however a short review of the standards that may apply based on initial assumptions of the problem has produced the list below that may prove useful in the design phase of the project:

- AS 3990:1993 - Steelwork
- AS 4100:2020 - Steel Structures
- AS/NZS 1554.1:2014 - Structural Steel Welding
- AS 2312.1:2014 - Guide to the protection of structural steel against atmospheric corrosion by use of protective coatings
- AS/NZS 3750 - Paints for Steel Structures
- AS ISO 24410:2021 - Earth-Moving machinery – Coupling of Attachments to Skid Steer Loaders
- AS 1171-1998 - Magnetic Particle Testing

Of the above mentioned standards, AS 3990:1993 and AS 4100:2020 are anticipated to be referenced heavily throughout the design phase and are discussed in further detail below.

2.2.1 AS 3990:1993 - Steelwork

AS 3990:1993 is the standard that is applicable to the design and fabrication of components made of steel using the working stress method. The scope of the standard states that it 'applies to the design, fabrication, erection, repair and alteration of steelwork associated with boilers and pressure vessels, lifts, cranes, mining equipment, gas and liquid petroleum piping systems, bulk handling equipment and the like, in accordance with the working stress design method' (Standards Australia 2016a, p. 6).

A number of methods are provided in AS 3990-1993 which describe the ways in which a component or structure must be designed. The most applicable of these methods to this project is called the 'Fully rigid design method' (Standards Australia 2016a, p. 14) which states that the connections of the supporting members of a component will have negligible rotation angles due to the level of rigidity of those members. It can be left to the design engineer to determine the most appropriate Factor of Safety (FoS), however the standard offers a quantitative means within the fully rigid design method which states that the capacity of the structure to withstand the expected load cases shall not be less than 1/0.60 times the magnitude of those load combinations.

The load combinations for use in determining the maximum permissible stress are provided in Section 3.3 of AS 3990-1993, which for the case of this project will include both the dead load and the live load associated with the plant or structures intended use described in Equation 1 (Standards Australia 2016a, p.15):

$$X_D + X_L \tag{1}$$

where X_D is the load due to the weight of the permanent fixtures of the plant (the dead load), and X_L is the load which the component is subjected to as a result of its intended use (the live load) (Standards Australia 2016a, pp. 7-8).

When the load capacity factor described above is applied to these loads, Equation 2 is obtained that describes the load, P, that the structure must be able to withstand without the maximum permissible stress being exceeded.

$$P = \frac{1}{0.60}\left(X_D + X_L\right) \tag{2}$$

It is intended that Equation 2 will be used moving forward in the design phase of the project to determine the load that the proposed system must be able to withstand.

The maximum permissible stress, σ_b, for round, square and rectangular bars subjected to moments about the axis which provides least resistance to bending is described in the above standard and is to be taken as (Standards Australia 2016a, eq. 5.2(1)):

$$\sigma_b = 0.75\sigma_Y \tag{3}$$

and for all other beams (Standards Australia 2016a, eq. 5.2(2)):

$$\sigma_b = 0.66\sigma_Y \tag{4}$$

where σ_Y is the yield stress of the chosen material as described by AS 4100:2020 (Standards Australia 2020, tab. 2.1).

Inherent in the application of Equations 2, 3 and 4 is the production of a reasonable FoS, as the multiplication factor of $1/0.60$, and the reduction factors of 0.75 and 0.66 will ensure that the maximum permissible stress is not exceeded under the given load conditions. Further discussion on the choice of a reasonable FoS is discussed in Section 2.5.

The standard also states that methods must be employed to provide adequate protection against corrosion, the details of which can be found in AS 2312.

2.2.2 AS 4100:2020 - Steel Structures

AS 4100:2020 states that the yield and tensile stresses used in the design of steel structures shall not exceed the values stated for the chosen material grade (Standards Australia 2020, tab. 2.1). This provides the designer with a comprehensive list of allowable yield and tensile stresses depending on the material grade to which a component must be designed to withstand. The standard also specifies the type of steels allowable in the design of steel structures and the applicable standard to which each material type applies. Examples of such standards which may also be referenced as part of the design are as follows:

- AS/NZS 1163 - Cold-formed structural steel hollow sections
- AS/NZS 1594 - Hot-rolled steel flat products
- AS/NZS 3679.1:2016 - Structural steel - Hot-rolled bars and sections
- AS/NZS 3678:2016 - Structural steel - Hot-rolled plates, floor plates and slabs
- AS/NZS 1554 - Structural Steel Welding

The standard also describes the mechanical properties which must be used for design purposes which is summarised in Table 2:

Table 2: Mechanical properties for use in design calculations for structural steel (Standards Australia 2020, p. 29).

Property	Value
Modulus of Elasticity (E)	200 x 10^3 MPa
Shear Modulus of Elasticity (G)	80 x 10^3 MPa
Poisson's Ratio (ν)	0.25
Coefficient of Thermal Expansion (α_T)	11.7 x 10^{-6} / °C

2.2.3 Deflection Limits

The standards previously discussed also offer recommendations on the maximum allowable deflection limits for a number of situations. AS 3990:1993 states that if a specific application standard exists, then the deflection limits given in that standard are to be used and the application of the limit chosen is the responsibility of the design engineer (Standards Australia 2016a, p. 16). AS 4100:2020 however, offers suggested deflection limits for span and cantilever type beams. The equations used to determine the allowable deflection for a span and cantilever type beam are given in Equations 5 and 6 respectively (Standards Australia 2020, p. 185):

$$\frac{\Delta}{l} \leq \frac{1}{250} \tag{5}$$

$$\frac{\Delta}{l} \leq \frac{1}{125} \tag{6}$$

where:

- $\Delta =$ the deflection limit
- $l =$ the length of the span or cantilever

At this stage in the process, it is unclear in which direction the design will progress. From the authors own personal experience, there are many purpose built components used in the mining industry constructed from steel which is the reason for the above discussion. As the design process progresses and it becomes more clear which type of system or component will be chosen for the preferred design, the standards which are applicable will be discussed in further detail to ensure compliance is achieved.

2.3 Material Selection

It can be argued that the most important factors in the selection of a material are availability, cost, required material properties and the available manufacturing processes. The information on availability and cost of materials can often be easily gained, however a much deeper understanding of the functional requirements of the system need to be understood prior to the selection of a material which is to make up a component to perform an intended system function. The service characteristics of a system such as expected loads, operating environment and reliability for example, determine the functional requirements and form an integral part of the material selection process (Juvinall & Marshek 2012, pp. 116-118).

When the functional requirements of the system are understood, a professional engineer must exercise caution in deciding whether the tabled data for a particular material is appropriate to use when determining a materials suitability. A thorough research effort would determine if the tabled values of a material under scrutiny are those derived from single tests, average values over a larger number of tests or even if the material is tested under conditions similar to those which the proposed component is expected to experience (Juvinall & Marshek 2012, p. 100).

It is for this reason that all relevant Australian Standards must be adhered to when designating a specific material grade for the manufacture of any system with respect to the given problem. The Australian Standards which are anticipated to be applicable to the proposed system have been discussed previously and describe the relevant properties that the chosen materials must exhibit. It is expected that the most important materials properties with respect to the project are the yield and ultimate strengths of a material, as it is these properties that ultimately determine the allowable loads that a component or structure is able to withstand.

Given that the majority of the equipment used in the underground environment is fabricated using steel as identified in section 2.1, it would be beneficial to determine the types and grades often used to fabricate these components. The standards identified in the previous section not only specify the properties that the material must exhibit, but also the types of available and regularly produced shapes and sections that are commonly used to manufacture system components from structural steel. Each of these standards state the specifications and requirements of the production and supply of various grades which are available to be produced in a number of shapes and sections for different applications.

The standards designate the necessary properties that the steel grade must exhibit, which gives a design engineer the appropriate information with regard to material choice once the functional requirements of the components are understood.

2.4 Legislation

The Work Health and Safety Regulation (2017, p. 66) states that a duty holder must identify any hazards that could risk the health and safety of persons and eliminate those hazards if it is possible to do so. In the event that the hazards cannot be completely eliminated, the hierarchy of controls (represented graphically in Figure 2.6) shall be followed by either substituting, isolating or establishing engineering methods to control the hazards. Administrative controls and personal protective equipment (PPE), or a combination of any of the controls in the hierarchy may then be used to minimise the risks if they still remain.

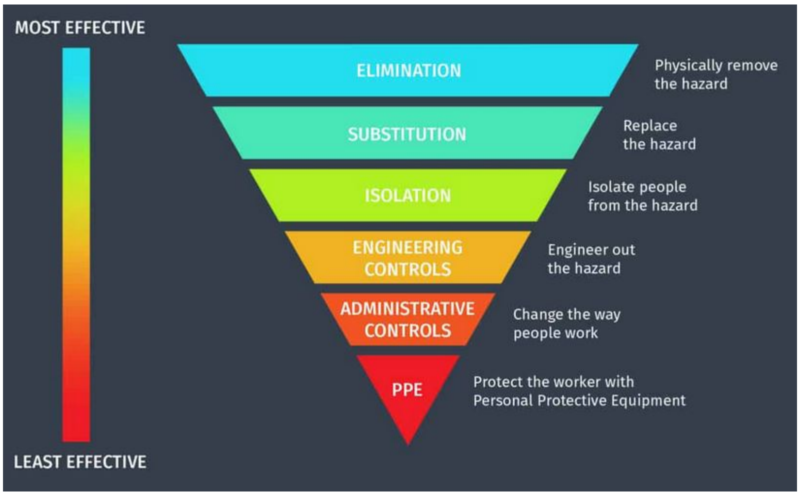


Figure 2.6: Graphical representation of the Hierarchy of Controls (Tap into Safety 2019).

When determining the most appropriate controls to mitigate those hazards that have been identified, individuals who hold a duty under *The Work Health and Safety Act* (2011, p. 22) need to consider the following:

- the likelihood of the hazard occurring
- the reasonable consequence of the hazard
- what the duty holder knows, or in their position is required to know about the hazard and methods to minimise the hazard
- the availability of the control methods
- whether the financial cost associated with the method of control is proportionate to the risk

Following the control hierarchy with regard to the replacement of the underground AC units, elimination of the hazards would be the most effective way of reducing risk. To achieve this, the AC units would need to remain in place atop the transformers and have the required maintenance carried out in situ. This may be possible for certain routine tasks, however there are tasks that require the units to be removed for the work to be carried out effectively, therefore complete elimination of the hazard is not possible.

Substitution is the next control method in the hierarchy. This may be achieved by replacing the top mounted AC units with units which are mounted at ground level that have ducts that are run remotely to achieve the same cooling result. This method is definitely possible as these types of units are available, however considering the overall cost to retrofit new equipment to the existing transformers, it is considered to be disproportionate to the risk as there are other means with which the risk can be brought down to an acceptable level.

Isolation as a control method is not considered feasible with the type of problem discussed. It is therefore considered necessary to design and implement an engineering control as a method for reducing the risks associated to a level considered to be acceptable. This type of control comes with many challenges, as the designer of plant has several duties under the legislation which must be adhered to and thoroughly referenced throughout the design process to ensure compliance.

The Work Health and Safety Regulation (2017, ch. 5) describes the many duties of persons who design plant including the obligation to provide appropriate information to the manufacturer of the plant including information in relation to guarding, operational and emergency stop controls and warning devices. In regard to the problem detailed in section 1, clause 219 of the legislation discusses the following regarding plant that is designed to lift or suspend loads:

- The person who manages the plant must ensure it designed specifically to support the load
- The loads being lifted are within the safe working load of the attachment being used
- The load must remain under control throughout the duration of the lift

All of the above items must be considered by persons who design plant for use in any industry in New South Wales. It is up to the designer to determine how the above items are achieved, however as long as the relevant Australian Standards are adhered to and that process is clearly documented, a safe and compliant design is achievable.

2.5 Safety in Design

There is a level of safety and quality which is expected by consumers of any type of product in Australia which a design engineer must accept. Smith (2002, p.72) describes three different types of defects with which design engineers must be familiar. Those are manufacturing, design and (arguably less obvious) marketing defects. Most applicable to this project of course is ensuring that the proposed system is free from defects which are created as part of the design. A design defect can be defined as a situation in which there has been a failure of the system to perform a function as part of its intended use in a manner which can be reasonably expected (Smith 2002, p. 72).

There are many techniques which can be used in order to ensure that all reasonably foreseeable hazardous situations are accounted for, the most applicable of which includes a technique called a Failure Modes and Effects Analysis (FMEA).

A FMEA is a systematic process for the component level analysis of a design with the aim of identifying all possible ways in which failures can occur and how those failures affect the system as a whole. The technique provides a design engineer with a methodical process to identify the relationships between system components with the aim of the implementation of methods for the mitigation, or prevention of the effects of the failures that have been identified (Bowles 2002, p. 50).

The process, more specifically involves the following as described by Bowles (2002, p. 53):

1. Identification of the failure modes of each item in the system
2. Determination of the component and system level effect of the failures
3. Failure mode probability ranking
4. Recognition of how the failure is identified
5. Proposition of any design changes to mitigate the effects

The above information can be captured in worksheets and more graphically as in Figure 2.7 to help with the tracking of the failure modes and effects.

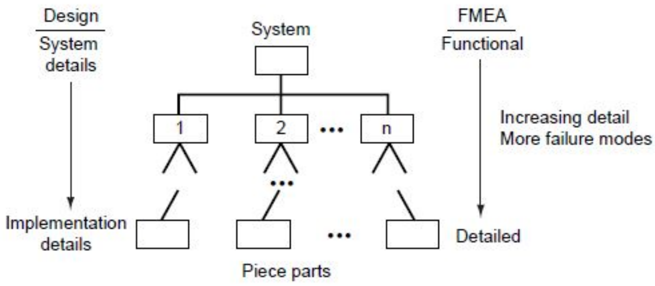


Figure 2.7: Typical FMEA flow cart (Bowles 2002, p. 53).

The method described above will be used throughout the design process to ensure a system is proposed that meets and exceeds all safety expectations of the intended users.

An important part of the design process is the choice of a factor of safety, and is described by Beer et al. (2015, p. 33) to be one of the most important activities to be undertaken by a design engineer. They describe many factors to consider when choosing an appropriate factor of safety which include the level to which the properties of a component are known to vary, expected loading cycles, types of failure, confidence in analyses and the likelihood of total system failure given the failure of a single component. Juvinall and Marshek (2012, p. 274-276) agree with this reasoning and go on to describe appropriate numerical values for a FoS that are dependent on certain situations. A FoS of 1.5 to 2 is considered reasonable for materials with properties that are known with a high degree of confidence, and when the component is subjected to well understood loads in a normal environment.

Two equations to determine the FoS are often used in engineering applications as described below (Juvinall & Marshek 2012, p. 273):

$$FoS = \frac{significant\ strength\ of\ the\ material}{significant\ stress\ from\ expected\ loads} \tag{7}$$

$$FoS = \frac{design\ overload}{normally\ expected\ load} \tag{8}$$

Equation 8 is often preferred due to situations where failure can occur well before the yield strength of a material is reached. An example of such a situation in which Equation 7 may not be suitable is when buckling is expected, and as such the critical buckling load of a member is lower than that which would cause yielding under normal situations.

Given the necessity to use materials that are of a known quality as prescribed by the relevant Australian Standards, and loading conditions that will be well understood, a FoS of 1.5 to 2 using Equation 8 will be considered reasonable for this project. This may be subject to change at a later stage during the process however as the design is further developed and understood.

2.6 Stress Analysis

Section 2.2 defined an equation (Equation 2) that describes the load that a system must be able to withstand without exceeding the maximum permissible stress of the chosen material. Before this equation can be used, a thorough understanding of the types of stresses that a material will be subjected to must be achieved.

Only components that are uni-axially loaded will have predictable failure points when compared to tensile tests of specimens of the same material due to the component and the test specimen being subjected to the same stress state (Beer et al. 2015, p. 507). Most machinery components are not subjected to uni-axial stress however, which requires the use of a failure theory appropriate for the material under scrutiny. The most common failure theory in use for ductile materials (of which structural steels are considered (Callister & Rethwisch 2012, p. 544)) is termed 'The Maximum-Distortion Energy Criterion', or the 'von Mises Criterion' which states that a component will generally resist failure if the maximum amount of energy required to distort the material per unit volume remains under the value required to cause it to yield in a specimen of the same material (Beer et al. 2015, p. 508; Juvinall & Marshek 2012, p. 267). The equation describing this criterion based on the three principle stresses is as follows:

$$\sigma_e = \sqrt{\frac{(\sigma_2 - \sigma_1)^2 + (\sigma_3 - \sigma_1)^2 + (\sigma_3 - \sigma_2)^2}{2}} \tag{9}$$

The equivalent stress in Equation 9, σ_e, is defined as the magnitude of uni-axial stress that would cause the same probability of failure based on the three principle stresses (Juvinall & Marshek 2012, p. 267). As long as the equivalent stress in a component when loaded, remains under the known value of yield strength of the material with which it is made, the component will be considered to withstand that loading. This inequality (including the FoS) that must be used to ensure a design will not fail is described below in Equation 10.

$$\sigma_e \leq \frac{\sigma_Y}{FoS} \tag{10}$$

A short review of available design literature supports the above discussion. Saldana-Robles et al. (2020) for example, propose a number of design techniques that use von Mises stress and an appropriate safety factor as output variables that guide their design of an agricultural back-hoe in an effort to reduce mass and cost.

When designing a machinery or system component, an engineer must be satisfied that the item will resist the reasonably expected loads and not deform excessively or yield to the point of failure as previously discussed. Ideally, an analytical solution that describes the stress and strain at every point on a component would be preferred, however due to the geometries with which many machinery components are designed, these analytical solutions are not usually possible. The finite element method is a technique used in engineering that can be used to predict the approximate values of stress and strain at critical areas by discretising the component into small elements, where the solution is only obtained at the junction, or nodes, of each interconnecting element (Logan 2017, p.2).

The forces and displacements in a component that has been discretised are related by a global stiffness matrix, K, as shown in Equation 11 (Logan 2017, eq. 1.2.4):

$$\{F\} = [K]\{d\} \tag{11}$$

or in expanded form in Equation 12 (Logan 2017, eq. 1.2.5):

$$\begin{Bmatrix} F_{1x} \\ F_{1y} \\ \dots \\ F_{nz} \end{Bmatrix} = \begin{bmatrix} K_{11} & K_{12} & \dots & K_{1n} \\ K_{21} & K_{22} & \dots & K_{2n} \\ \vdots & & & \\ K_{n1} & K_{n2} & \dots & K_{nn} \end{bmatrix} \begin{Bmatrix} u_1 \\ v_1 \\ \dots \\ w_n \end{Bmatrix} \tag{12}$$

The above notation of the force matrix describes the force experienced at the node and in the direction in which it is applied. For example, F_{1x} describes the force in the x direction at node 1. Similarly, the displacement matrix describes the direction in which each node is displaced, whilst the global stiffness matrix, $[K]$, represents the coefficients that relate the force and displacement matrices.

The above equations yield a set of simultaneous algebraic equations that are able to be solved in a relatively short amount of time with the use of modern computers, with a number of outputs of interest to engineers including principle and von Mises stress, displacements and safety factors.

Logan (2017, p. 9) argues that it is up to the engineering professional to decide on how many elements are required and which type best suit the problem depending on the accuracy required, and the time available to execute the computation. It is generally accepted that higher numbers of elements are required in areas that are believed to have more rapidly changing properties such as stress and displacement (as shown by Alegre and Tremblay (2022, p. 5) in their four point bending test of C-channels), and also that higher numbers of elements produce results that are closer to an exact solution of the given problem (Logan 2017, p. 408).

An FEA study will be conducted on a proposed design using PTC Creo with the adoption of the meshing methodology described, by first establishing an automatically generated mesh, then with manual refinement in areas believed to exhibit higher stress and strain rates for greater accuracy in these areas.

2.7 Theoretical Techniques

Before any FEA model can be verified, the theoretical nature with which a component is expected to experience stress and displacement as a result of the applied loads should be well understood. Although it is not yet understood how the system will be designed, the author has been exposed to the necessary information throughout his undergraduate program with regard to the theoretical behaviour of materials, and it is thought to be beneficial to identify and review some theoretical techniques that may provide some insight on how to best predict the behaviour of the design. As the design will by necessity involve some type of support for the AC units, this section will briefly discuss the techniques thought to be useful in understanding the response of the system for model verification purposes.

A prismatic beam is a member of constant cross-section designed to support loads that are normally applied perpendicular to its longitudinal axis (Beer et al. 2015, p. 346). Depending on the loading application, the member will be subjected to shear forces and bending moments of varying magnitudes which can be predicted with the use of shear force and bending moment diagrams (SFD and BMD respectively).

A useful relationship exists between the applied load (w), the shear force (V) and bending moment (M) and can be used to derive the SFD and BMD. The equations that relate these variables are given in Equations 13 and 14 (Beer et al. 2015, pp. 360 - 361).

$$-w = \frac{dV}{dx} \tag{13}$$

$$V = \frac{dM}{dx} \tag{14}$$

Upon inspection of Equation 14, it can be seen that at a point where the magnitude of the shear stress is zero, a point of maxima or minima with respect to the bending moment exists which is useful in determining the location along the beam at which the bending moment is greatest.

Beer et al. (2015, p. 347) argue that the maximum normal stress in a beam designed for strength is of most importance, which can be obtained from the elastic flexural formula given below (Beer et al. 2015, p. 245) once the maximum bending moment has been defined using Equation 14:

$$\sigma_x = -\frac{My}{I} \tag{15}$$

where σ_x is the normal stress at any vertical distance, y, from the neutral axis, M is the magnitude of the moment couple applied to the beam and I is the second moment of area for the cross-sectional shape of the beam. It is important to note that Equation 15 is only applicable in the elastic range prior to the material yielding, which is appropriate if failure is to be avoided.

Beams are often designed to be optimised for deflection rather than stress however, as the largest displacement often occurs when the maximum permissible stresses in the beam are well below the yield point of the material (Juvinal & Marshek 2012, p. 206). Equations for the slope and deflection of prismatic beams under simple loading conditions are given in Appendix D in Beer et al. (2015, p. A29) and also by Juvinal & Marshek (2012, p. 850) which may prove to be useful in calculating the theoretical amount of deflection for comparison against a model produced as part of an FEA study.

Upon inspection of the BMD shown in Figure 2.8, it is quite clear that the maximum bending moment occurs at the point of the beam at which the concentrated load is applied. A prismatic beam subjected to a bending moment will normally be designed to withstand this magnitude, and hence the geometry of the beam along the entirety of its length is dictated by the largest bending moment. The bending moment at the supports is zero, therefore it can be concluded that a prismatic beam designed for the loading condition described may be significantly over designed in areas far from the point of load application, leading to a considerable waste of material (Beer et al. 2015, p. 396).

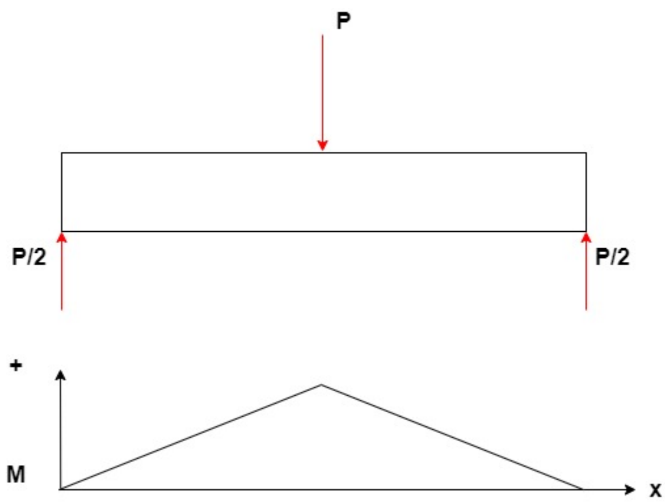


Figure 2.8: Typical BMD of a prismatic beam with a concentrated load, simply supported at both ends.

A method used to overcome this problem is to design a 'beam of constant strength' as described by Beer et al. (2015, p. 396). A member which has been designed so that it withstands the bending moment as a function of its length leads to a beam which has a varying cross section, and is called a non-prismatic beam (Vo et al. 2021, p. 1). Beams designed in this manner are optimised so that every point along the length of the beam satisfies Equation 15, rather than choosing a single geometry capable of withstanding the maximum stress. Since the allowable stress (σ_{all}) is the same at all locations and in all directions of a homogeneous, isotropic material, Equation 15 can be rearranged to solve for the section modulus (S) of a material which is defined as I/c, where c is the farthest location from the neutral axis. This results in the derivation of Equation 16 (Beer et al. 2015, p. 396).

$$S = \frac{|M|}{\sigma_{all}} \tag{16}$$

A member designed to optimise the section modulus leads to a varying cross-section as a function of its length. Vilar et al. (2021, p. 1) argue that beams of this type are becoming more popular in industrial applications due to a number of reasons including cost savings and the ability to produce a stronger member with a lower mass. The stress fields in these types of beams are more complex however, with the formulation of the general stress case being the subject of current research.

It is believed that the above theoretical techniques for the design of prismatic and non-prismatic beams will be useful in both the design and verification of the behaviour of a system for the solution to the problem outlined in Section 1.

2.8 Conclusion

The preceding discussion has identified the relevant literature that will enable the author to produce an adequate solution to the problem described in Section 1. The literature review has not been able to identify any current system or machinery attachment that will be able to perform the function of removing and replacing the AC units with a risk that is as low as reasonably practicable as is required by the relevant legislation.

The research questions introduced in Section 1.3 have been answered to the level thought to be acceptable for moving forward with the project.

The design options available have been explored to enable appropriate conceptualisation to begin. The materials available for the production of those concepts have been identified, with a review of the supporting literature associated with the maximum deflection and maximum permissible stress limits presented also. The methods identified in the literature review will be used in the design phase of the project where applicable to ensure a design is produced that meets and exceeds the relevant Australian Standards. Those techniques will also be used to determine the geometries required, both theoretically and through model verification, to produce a design with an acceptable deflection limit, allowable material stresses and an appropriate FoS that is applicable for the intended use of the system as designed.

3 METHODOLOGY

There are many requirements that need to be satisfied if a good design is to be realised. Arguably the most important requirement is that of the intended function of the system which takes into account many factors including physical size and weight, ease of use, maintainability, operating environment and cost. Other requirements include factors relating to materials such as strength and fatigue requirements, and manufacturing requirements including shape and size, fabrication, machinability and quality (ed. Davis 1998, p. 122). Of course it is the materials and manufacturing requirements that allow the intended function to be realised, so it can be thought of as an iterative process with which the need of the design is always kept in mind when making decisions that relate to the component design as summarised in Figure 3.1.

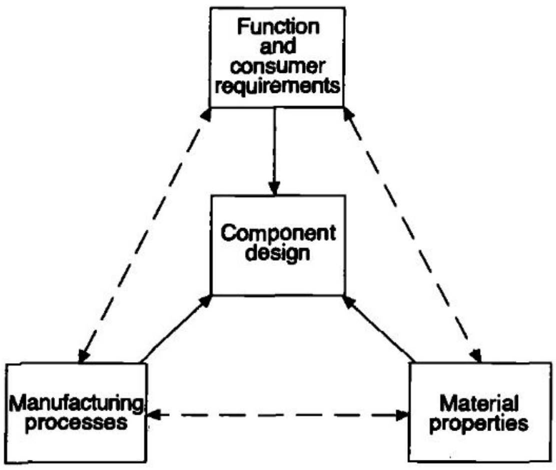


Figure 3.1: The design framework and inter-dependability with important factors for consideration in the design process (ed. Davis 1998, p. 122).

Engineering design is part of a much larger framework termed the 'Product Realization Process' (PRP) (ed. Davis 1998, p. 122) (Figure 3.2) which considers the whole undertaking from the initial consultation with customers to the disposal of the system after the service life has been reached. It is not the intent of this project to take into account the marketing or sales portion of the PRP, as the focus will be on the process with which the design will be realised. However it is important to realise that the engineering design process is just part of a much larger framework that has the ability to effect the decisions made by design engineers throughout the process of designing a new component or system.

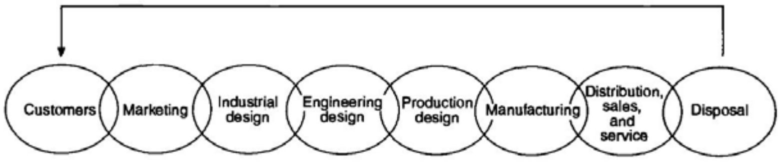


Figure 3.2: The entirety of the 'Product Realization Process' (ed. Davis 1998, p. 122).

It is important to note that the proposed design is expected to be manufactured by an outsourced engineering consultancy experienced in the design and fabrication of small scale mining support equipment. This is not uncommon in the industry as the project engineers on a mine site are often involved in many projects simultaneously. It is their role to define the project requirements and liaise with experienced design engineers so that the design can be realised adhering to all relevant standards and legislation. AS 3990-1993 specifically states 'The design of a structure or the part of a structure to which this standard is applied shall be the responsibility of an engineer experienced in the design of such structures.' (Standards Australia 2016a, sec. 1.5.1). It is therefore expected that the author will conduct the complete design process individually from the conceptualisation through to the completion of the detailed design phase (discussed below) as an academic exercise, then once the preferred system has been agreed upon by the relevant workplace management, a third party will be engaged to conduct their own analysis and fabrication of the agreed design.

During the discussions between the author and the experienced design engineer used to perform the parallel design analysis, the lower level requirements will need to be agreed upon. Ertas and Jones (1996, p. 17) argue that it is a critical step in this phase of the project that design requirements be well thought out to prevent a number of difficulties including cost blow-outs, negotiation issues with suppliers and in the extreme case, abandonment of the project due to the inability to redefine the requirements after contracts between the end user and the supplier have been agreed upon.

The following sections aim to describe the proposed method with which the engineering design process will be carried out by the author. It will briefly describe the basic functions of each stage of the design process that will be used to complete the project.

3.1 The Engineering Design Process

In order to complete the project successfully, a sound methodology will be required which begins with the thorough understanding of the design process. The Accreditation Board for Engineering and Technology (cited in Ertas & Jones 1996, p. 2) describe design as '. . . the process of devising a system, component, or process to meet desired needs'. The major steps involved in the engineering design process are described in Figure 3.3.

A similar process with regard to the design of machine components is described by Juvinall and Marshek (2012, p. 19) which gives the author confidence that with the use of the proposed methodology, a design of appropriate quality will be produced to meet the required needs.

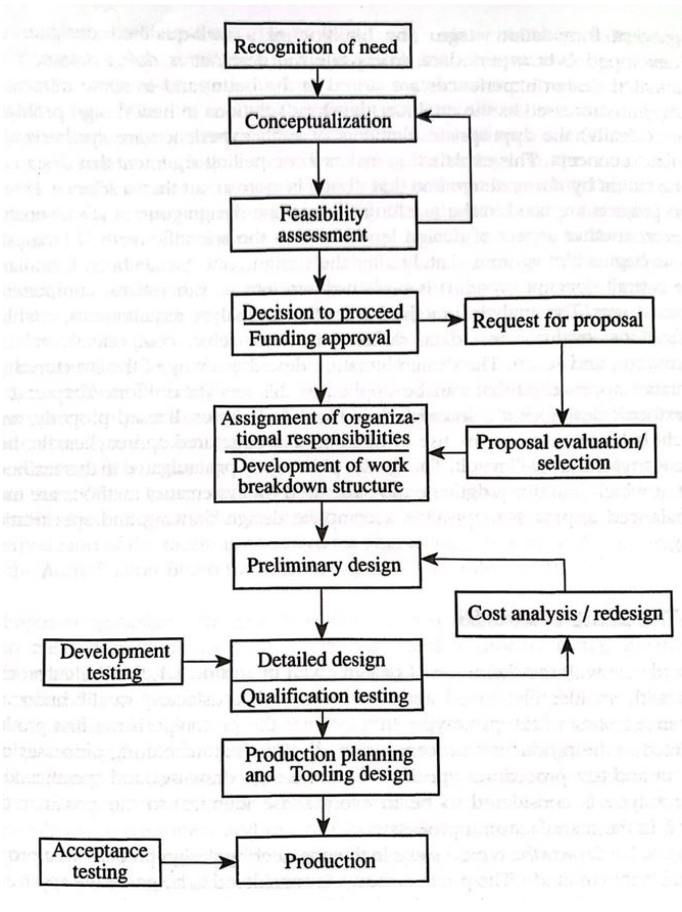


Figure 3.3: The typical process used in engineering design (Ertas & Jones 1996, p. 4).

3.1.1 Recognition of Need

The first step in this process requires that the need for the project be thoroughly recognised and understood. This is broadly covered by the aim of the project being the need for a system for the safe installation and removal of the AC units previously discussed.

Section 1 identified some initial constraints relative to the problem, however further information was needed to gain a complete and thorough understanding of what was required. The AC units mass is designated as 320 kg and was ascertained by the information plaque situated on the unit itself (Figure 3.4).

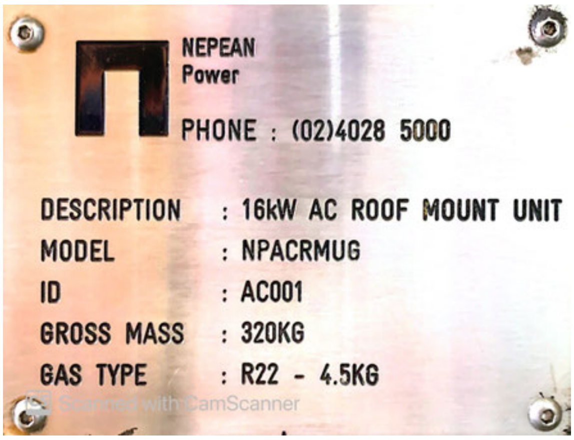


Figure 3.4: AC unit mass shown on plaque .

The author made contact with the OEM in an effort to locate any information or literature which indicated the location of the Centre of Gravity (CoG) of the AC units. The OEM was unable to provide this information, so a manual method was used which involved placing a length of pipe under the unit and determining the location at which it was balanced. Figure 3.5 shows the experiment being carried out.

(a) Photo showing the method for obtaining the CoG on the longitudinal axis of the AC unit.

(b) Photo showing the method for obtaining the CoG on the lateral axis of the AC unit.

Figure 3.5: Photos indicating the method for which the Centre of Gravity (CoG) was determined

Figure 3.5a shows that the longitudinal CoG is approximately 50 mm away from the geometric axis. Given the conditions under which the CoG was determined (imperfect workshop floor), the author was satisfied that any calculations used for the purpose of a force analysis would be appropriate using the longitudinal geometric centre of the AC unit. However for the lateral axis, the CoG was measured to be 635 mm from the right edge, 165 mm from the geometric centre (Figure 3.5b).

The lateral CoG also seems to coincide with lifting points as shown in Figure 3.5b, however not all of the AC units have lifting points in this location. Upon further investigation, certification for these lifting points was unable to be verified and therefore it was decided that the final design used in the removal the units would aim to disregard the lifting points as a potential method.

The physical dimensions of the AC units was also established via direct measurement and are summarised as 1600 mm x 1000 mm x 1000m. A general arrangement drawing of the AC units is provided in Appendix B.

3.1.2 Conceptualisation

Conceptualisation is the next step in the process once all of the relevant constraints have been identified. The author will be able to draw on a number of years experience working in the underground coal industry as a mechanical trades person to devise several concepts that will be suitable for forwarding on to the feasibility stage. This experience is anticipated to prove beneficial to the design process with the recognition and understanding of what is deemed to be acceptable in terms of materials and processes currently used in the industry. Careful reflection during this stage will be important however, as the current knowledge of what is normal and accepted in the industry may suppress new ideas and processes that have the potential to improve the final design.

3D modelling software is useful as a communication tool during this stage of the design process, as the ideas can be visualised and communicated to all relevant stakeholders in order to gain important feedback. The author has had a significant level of exposure to PTC Creo during the undergraduate program which has lead to the successful completion of a number of small scale design projects. PTC Creo is therefore intended to be used in the conceptualisation phase to provide the 3D modelling capabilities required for the visualisation and communication of the design concepts.

The conceptualisation phase is often described as having to occur prior to the feasibility assessment, however for smaller scale projects they can occur simultaneously which is believed to provide more achievable concepts to be pursued further throughout the design process (Ertas & Jones 1996, p. 11). Using this methodology, a number of ideas will be conceptualised keeping in mind a number of higher level design requirements discussed in the following section.

3.1.3 Feasibility Assessment

Identifying the design requirements is arguably one of the most difficult, yet important tasks of the entire design process (Ertas & Jones 1996, pp. 12-13). A significant amount of time and effort has been spent identifying the needs of the project as a whole, however for a successful completion, specific design requirements need to be established in enough detail to ensure the needs of the project can be met by the proposed design.

Ertas and Jones (1996, p.12) argue that the design requirements must be completed after a concept has been chosen, and before the detailed design process takes place. This may be the case for lower level requirements, however it is the opinion of the author that it may be beneficial for the establishment of higher level design requirements prior to the conceptualisation stage. It is believed that this will reduce the time and effort pursuing designs that although they may work on a theoretical basis, may have little chance of being further refined in the detailed design process due to preferred methods for carrying out the work. The higher level requirements have been established by reviewing the projects basic needs and are summarised below:

- The locations of the belt starter substations vary in size, grade and orientation, therefore the design must be suitable for all existing belt starter substation locations and orientations.
- The design must limit the amount of manual handling involved to reduce risk to personnel to a level as low as reasonably practicable.
- The use of lifting and slinging devices must be minimised to reduce the risks associated with load shifting.
- The use of common and available material types and grades will be the preferred choice for the fabrication of the design.
- The use and operation of the design or system must be appropriate for the skills and competencies of the existing personnel.
- The system will be designed to adhere to all relevant Australian Standards and legislation.

In the case of this project and the needs previously discussed, there are many methods with which the problem can be solved. Therefore it is important that these initial requirements be used as a guide in the conceptualisation phase in an effort to reduce time spent exploring unsuitable designs. After a number suitable concepts have been chosen, it is at this stage that lower level design requirements can be identified which the chosen design must adhere to.

3.1.4 Preliminary Design

The preliminary design phase of the project will aim to further refine the concept that best suits the higher level requirements defined in the previous section. In the case that more than one concept may be suitable, this phase will assist in determining which concept will be best to choose for the final design.

The activities involved during this process according to Ertas and Jones (1996, p. 17) include defining the lower level system and component design requirements. This process is anticipated to include a thorough force analysis on the proposed design to ensure that it is suitable. In the case of the design of an attachment for an existing piece of machinery for example, a static force analysis will be required to ensure the appropriate stability requirements can be met prior to the detailed design phase. If the force analyses alone do not provide enough information that give confidence for the proposed design to move forward, it is in this phase that testing may be carried out to support the analyses conducted.

It is intended that an FEA be conducted during the detailed design phase of the chosen concept (discussed in the following section), however it is argued that such an analysis will be of little benefit unless the loads to which the system is subjected, are accurately modelled (Juvinall & Marshek 2012, p. 45). Hence it is intended during the preliminary design phase that the force analysis be thoroughly studied and conducted with the recognised engineering practice and rigour that the author has been exposed to throughout the undergraduate program, to ensure the system that is fabricated will perform the intended function whilst being able to withstand the expected operating environments to which it will be subjected.

If successful in the creation of a concept that is suitable and the lower level design requirements have been established, it is at this stage that the detailed phase of the design will begin, as discussed in the following section.

3.1.5 Detailed Design

The next phase of the project in which all of the necessary details are established to allow the system to be manufactured is called the detailed design phase. It is intended that this is the phase in which the chosen concept is validated through the completion of a finite element analysis using PTC Creo to ensure the design is manufactured to withstand the expected load cases, taking into account the appropriate load factors required by the relevant standards. The FEA model will be verified using the theoretical methods described in Section 2 where applicable to ensure the system responses and behaviours are well understood.

The detailed design phase will also aim to verify that the maximum deflection and maximum permissible stress values identified in Section 2.2 are not exceeded by reviewing the information gained through the FEA study.

The project is intended to reduce the risks associated with the replacement of the underground AC units, and therefore has real world consequences with regard to the design of the system to perform the intended function. It is for this reason, and for the purposes of responsibility under the Work Health and Safety Regulation (2017, p.150) that a separate design analysis will be conducted by an experienced design engineer as previously discussed. It is intended that the system designed by the author will be discussed with the design engineer for the relevant legislative reasons. If the proposed method needs to be altered in any way to achieve the required standard for any reason such as to improve the load carrying capacity, ability to withstand stress or ease with which it can be fabricated, this feedback will be acknowledged and reported appropriately.

Engineering drawings will be produced at this stage of the project and will include general assembly drawings and detail drawings that will provide the means with which the system can be communicated for fabrication. All drawings will be produced to adhere to AS 1100 and will be created from the 3D models using PTC Creo.

3.1.6 Production

As previously discussed, the fabrication of the system is intended to be outsourced as the facilities of external engineering and manufacturing firms are superior to those that the author has access to. During this stage, communication between the author and the company tasked with the fabrication will be maintained to ensure an on-time delivery can be met, to which both parties have agreed.

3.1.7 Testing

Once the system has been fabricated and delivered, on-site testing will be completed after the appropriate risk management procedures have been implemented. The relevant Australian Standards with regard to testing will be identified and adhered to once the preferred design choice has been made.

3.1.8 Ethics

There are no anticipated ethical issues with the proposed design methodology. Any concerns relating to the introduction of the new piece of plant will be captured and controlled as part of the change management system on-site.

3.2 Conclusion

This section has provided a detailed description of the methodology with which the design will be carried out using all of the relevant information identified in the literature review.

4 DESIGN CONCEPTS

This section describes a number of potentially suitable design concepts for the installation and removal of the AC units.

4.1 LHD Attachment

One common method for material handling in the underground environment is with the use of an LHD as described in section 2. It was therefore a natural choice to explore the options for the replacements of the AC units using one of these machines. A number of concepts were devised which centred around the idea of a simple support platform with the ability to be easily attached using the quick disconnect system of an LHD (Figure 4.1). The advantages of this system are the relatively high load carrying capacity of the LHDs of up to 12 tonne depending on the model. This capacity is not needed of course for supporting the AC units, however it indicates the machines may be capable of supporting the units at a longer distance if required, whilst maintaining the stability required on a range of grades.

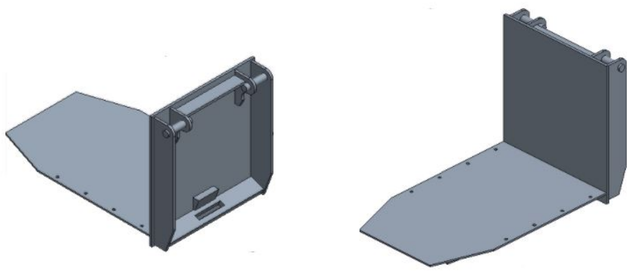


Figure 4.1: LHD attachment concept with simple plate for supporting the AC units.

Figure 4.2 below depicts the approximate positioning of the LHD with the attachment concept adjacent to the belt starter when replacing the AC units. The operator would be able to position the machine adjacent the belt starter, raise the attachment to the required height to allow the units to be manoeuvred onto the attachment, lowered to the ground and secured for transport.

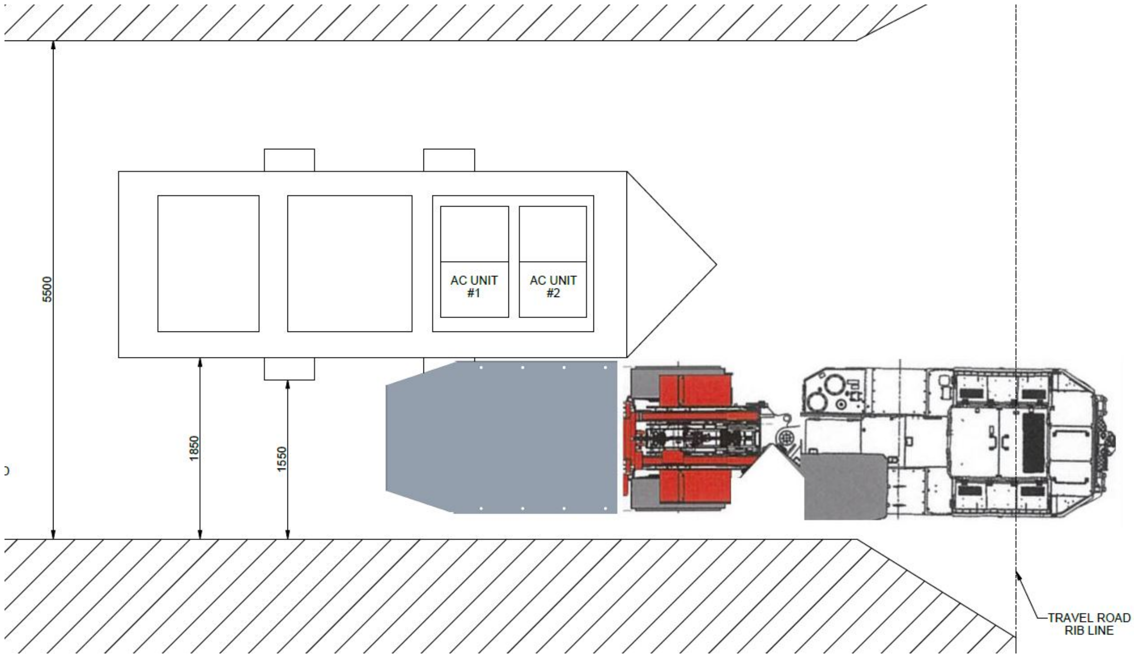


Figure 4.2: Image depicting the approximate positioning of the LHD with attachment concept used for the replacement of the AC units (LHD image from Sandvik Mining 2016).

The disadvantages with the use of an LHD are the issues with access to the belt starters due to the physical size of the machines. As previously mentioned, some of the belt starters are oriented differently, in particular, in some locations the AC units would be at the farthest end of the cut-through from where the LHD has access. This would mean that an exceedingly long plate type attachment similar to the one depicted in Figure 4.1 would be required in order for the replacement of the AC units to be feasible.

4.2 Skid Steer Loader Attachment

The second concept involves the use of a purpose built attachment for use with an underground certified skid steer loader similar to the previous concept. As discussed in the literature review, these loaders are capable of operating over a wide range of grades and they are compact with good manoeuvrability. Other advantages include the availability of personnel with the competency to operate a skid steer loader, meaning little to no further training will be required apart from the familiarisation with the safe working procedure for the new attachment. A number of early concept models are shown in Figures 4.3, 4.4 and 4.5.

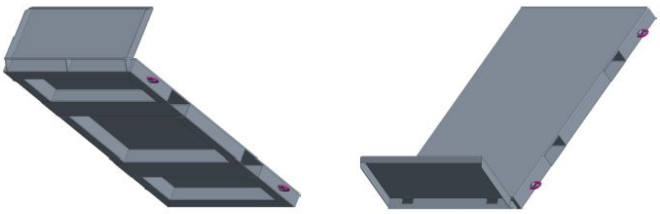


Figure 4.3: Skid steer loader attachment concept with tyne slots and rud lugs for use as tie-down points.

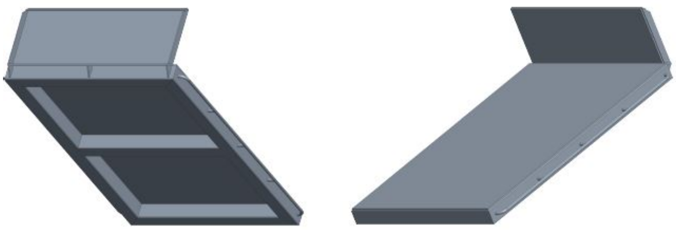


Figure 4.4: Skid steer loader attachment concept with tie-down rails.

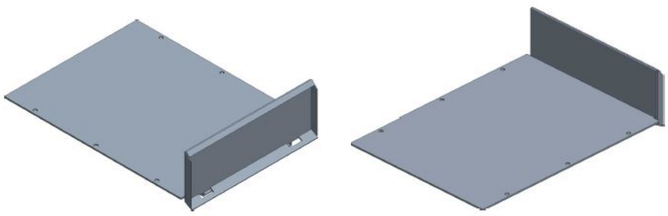


Figure 4.5: Skid steer loader attachment concept with lighter weight construction and holes for tie-down points.

The skid steer loader available for use has been measured to be 1500 mm in width, which would allow for the machine to be manoeuvred up next to the belt starter as the typical width adjacent to the rib with the wheels removed from the substation skid base is 1850 mm as shown in Figure 4.6 below.

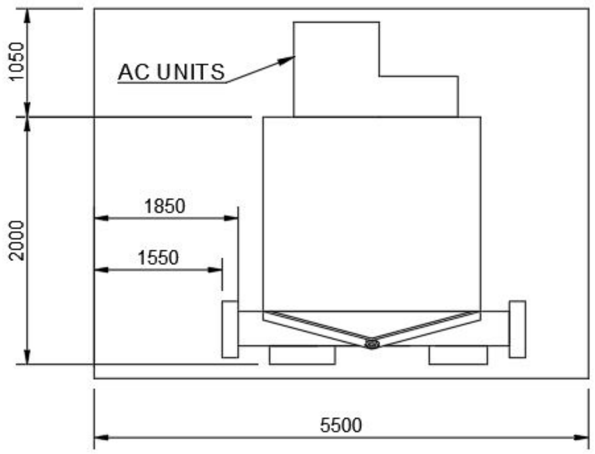


Figure 4.6: Belt starter substation typical arrangement (front view).

It is believed that this concept is able to meet all of the higher level design requirements described in the previous section as discussed below:

- it will allow the replacement of the AC units at all underground belt starter locations;
- it has the potential to virtually eliminate the requirement for manual handling;
- the AC units will be supported from their base, eliminating the need for lifting and slinging equipment;
- it is able to be manufactured from common grades of available materials;
- several operators have the skill and competency for the operation of a skid steer loader in the underground environment;
- and the design can be made to adhere to all relevant Australian Standards and legislation.

One significant problem that will need to be overcome with this concept is the issue of stability. Skid steer loaders are rated according to their tipping load and their lift capacity as described in AS ISO 14397.1:2021 (Standards Australia 2021c, p. 2) which states that the tipping load is the force exerted by a mass acting through the centroid of the loader bucket that causes the wheels farthest from the pivot point to leave the ground. The rated lifting capacity is defined as the tipping load multiplied by a stability factor of 0.5 for wheel loaders. Given the geometry of the AC units and the experiment previously carried out (Figure 3.5), the point at which the centre of gravity is acting will be further out from the front wheels of the skid steer loader, meaning the rated capacity will be affected.

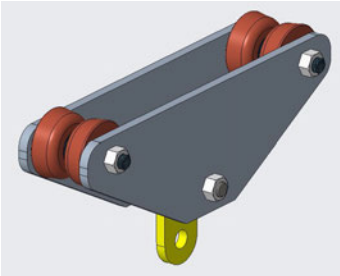
Given the rated lifting capacity of the available loader is 748 kg and the AC units are 320 kg, this method is thought to be feasible, however a design of this type may have to be optimised to reduce its mass to ensure the stability of the skid steer loader is maintained.

The attachment concept shown above would be used in a similar manner to the LHD that is depicted in Figure 4.2 by manoeuvring adjacent to the belt starter and using the loader hydraulic controls, will raise the attachment to allow the AC units to be manoeuvred onto the implement to be lowered to the ground.

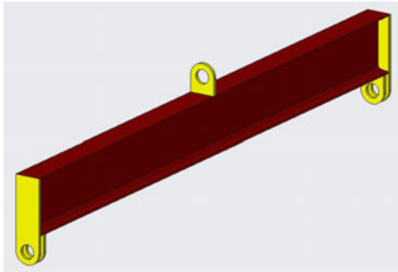
4.3 Monorail Concept

Another method thought to be suitable was the use of a roof mounted monorail system that was used in conjunction with a lifting jig. The system would require the installation of a monorail beam secured to the roof using standard roof bolts depicted Figures 2.4 and 2.5 (Section 2). A simple monorail trolley and jig concept is depicted in Figures 4.7a and 4.7b respectively with a possible configuration depicted in Figure 4.8. The following image (Figure 4.9) shows the potential removal path that would allow the units to be removed from above the belt starter. Using this configuration, the AC units would be carefully traversed along the monorail, then lowered and placed directly into an LHD implement used for transporting the units out of the mine.

After the first AC unit is removed, the next unit could be carefully manoeuvred under the monorail beam to be lifted and removed.



(a) Simple monorail trolley concept.



(b) Lifting jig concept used in conjunction with monorail trolley.

Figure 4.7: Monorail trolley and lifting jig concept used for the removal of the AC units.

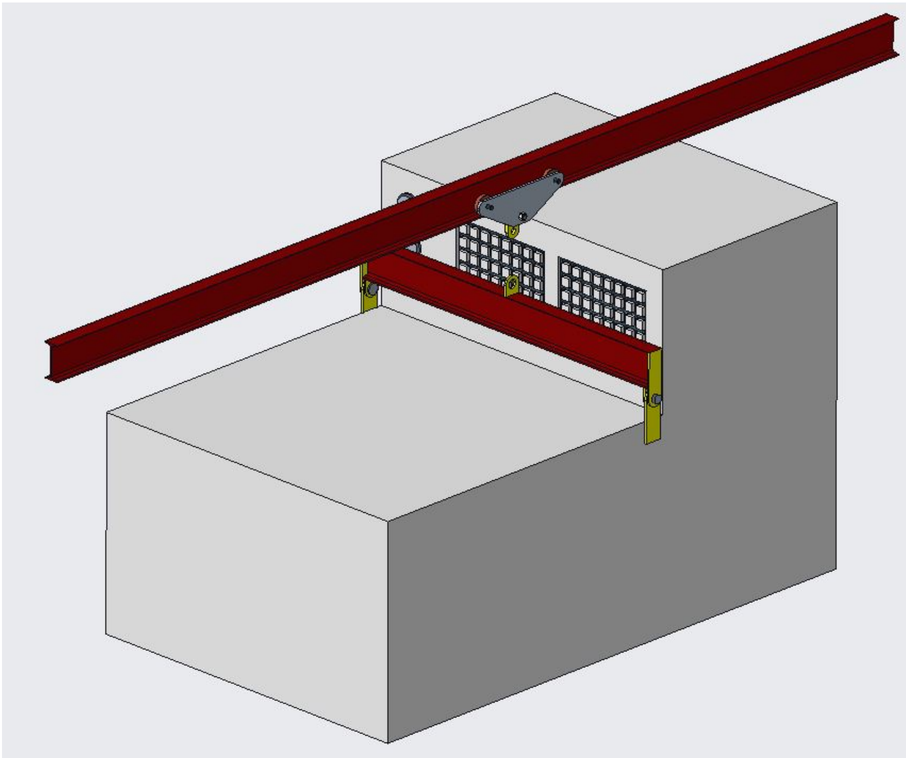


Figure 4.8: Possible configuration of monorail beam and trolley for AC unit removal.

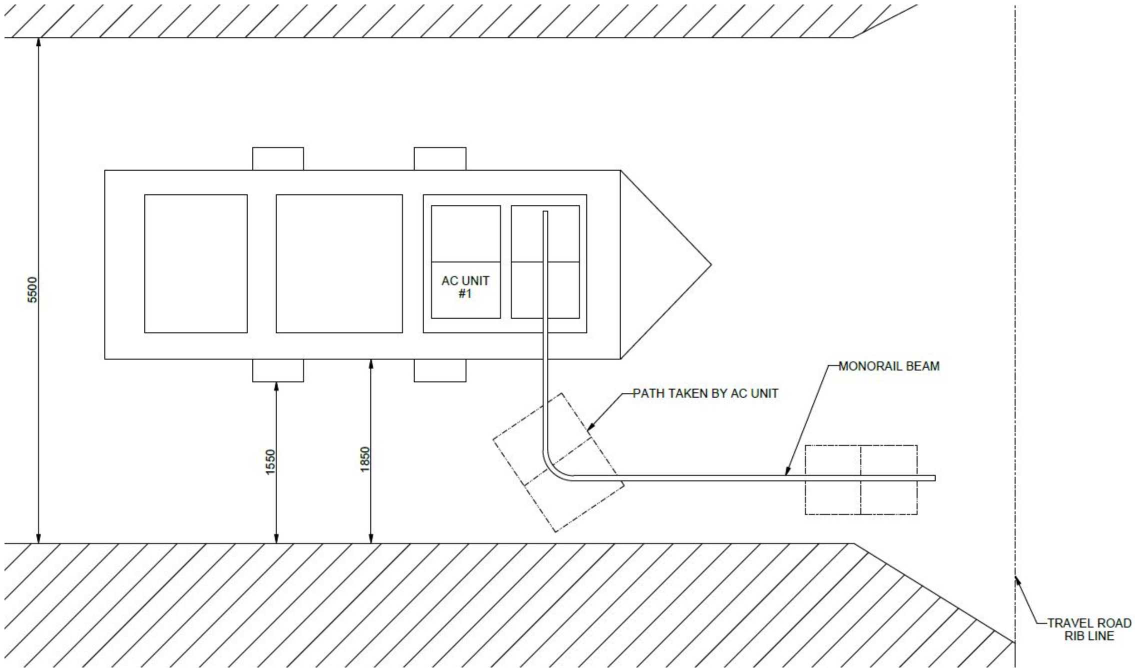


Figure 4.9: Top view of belt starter showing method with which the AC units could be removed with an overhead monorail system.

Although this concept may have worked for some belt starters, issues with this concept arose quickly with the realisation that there is little height above some other belt starters, thereby violating the higher level constraint defined earlier which requires the design to be suitable in all locations. This is easily visualised with reference to Figure 4.10 below and Figure 1.1 in Section 1 which shows how close the AC units can be to the roof of the underground workings. Adding to the height constraint is the necessity to use some form of mechanical lifting device such as a chain or lever hoist between the monorail trolley and lifting jig designated lifting lugs (shown in yellow in the above images). This would be required to enable the units to be lifted slightly for manoeuvring away from the belt starter, and then to be lowered to the ground or in an implement ready for transport.

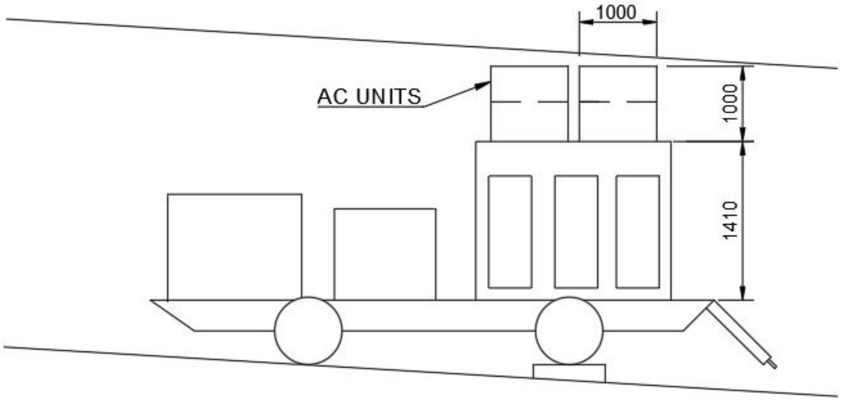


Figure 4.10: Belt starter substation typical arrangement (side view).

It would be possible to design a lifting jig that supported the units from the underside to remove the requirement for the certification of the lifting points, however it is thought that this would be difficult to achieve given the height constraints and the hazards presented when attempting to lift and place lifting equipment underneath the units. The reluctance to certify and use lifting points is derived from the hierarchy of controls shown in Figure 2.6. The act of lifting and load shifting itself presents additional hazards, and the preferred method with which hazards are controlled is through elimination.

Although it is unlikely that this concept would further be pursued, it has been necessary to explore this option to ensure the best possible decision could be made that would satisfy the given problem constraints.

4.4 Conclusion

To assist in the quantification of the level to which the concepts meet the high level design requirements proposed in Section 3.1.3, Table 3 has been created and is shown below.

Table 3: Quantification of adherence to proposed design constraints.

Constraint	LHD	Skid Steer	Monorail
All locations		X	
Manual Handling	X	X	X
Lifting/Slinging	X	X	
Materials	X	X	X
Skills	X	X	X
Standards	X	X	X

The above table demonstrates that the skid steer loader attachment concept is the method with which the problem is able to be solved whilst meeting all of the described design constraints. This design concept will be further discussed in the next section.

5 PRELIMINARY DESIGN

As discussed in the previous section, the concept which best meets the high level design requirements is the skid steer loader attachment. Of the three early design concepts created, the one depicted in Figure 4.5 is the concept that will be pursued further due to its lighter weight construction for reasons discussed below. The design constraints and requirements that apply to this design concept are further defined in this section.

5.1 Stability

It is believed that the biggest difficulty to overcome with this concept is ensuring the rated capacity is not exceeded with regard to the loaders stability. With reference to AS ISO 14397.1:2021 (Standards Australia 2021c, p.2) the available skid steer loader has a tipping capacity of 1496 kg, and a rated capacity of 748 kg (Hunter Plant Hire 2018a). The above standard describes the rated capacity acting through the centroid of the standard bucket. In this case, the bucket is assumed to have a cross section with the shape of a triangle with a length of 637 mm, giving a centroid of approximately 212 mm from the rear of the bucket. Measurement of the centroid taken from Hunter Plant Hire (2018b) which has been confirmed through physical measurement by the author, is calculated to be approximately 850 mm from the front axle (Figure 5.1).

It is important that the operational stability of the machine be maintained with the attachment to be manufactured. In order to achieve this condition, a moment caused by the attachment and AC unit must be taken about the tipping line (front axle) that must not exceed the moment caused by the rated capacity acting about the same point. The geometry of the AC unit and the attachment point on the loader dictates that the approximate CoG will be acting at a distance of 1.1 m from the tipping line (Figure 5.2b).

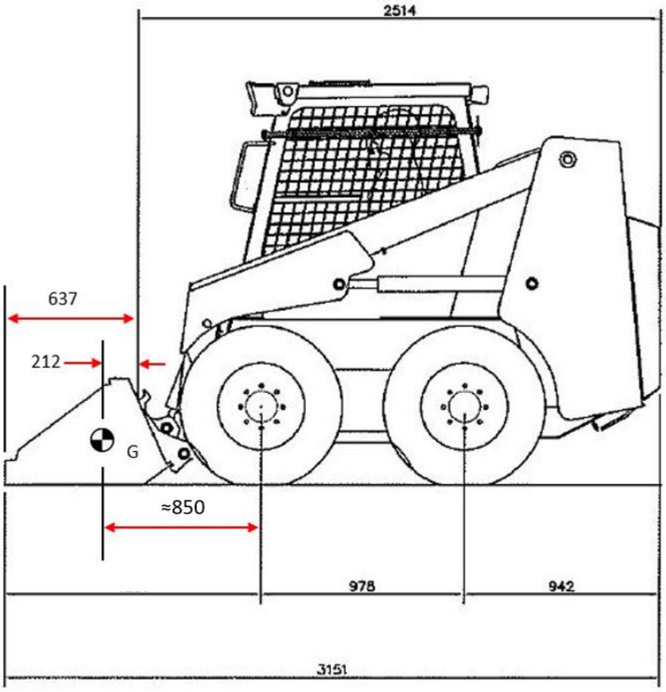
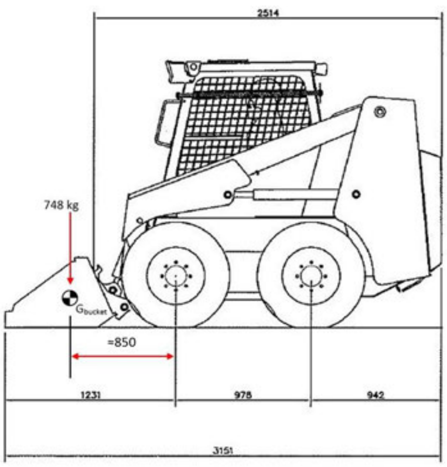
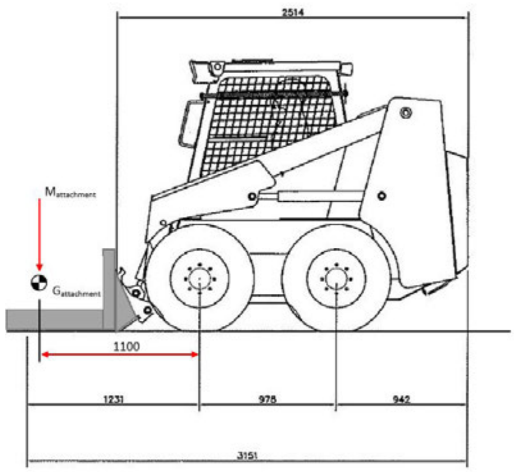


Figure 5.1: Skid steer loader bucket CoG (image adapted from Hunter Plant Hire (2018b)).



(a) Image depicting the location at which the rated load acts through the centroid of the bucket.



(b) Image depicting the location at which the attachment load acts through the centroid.

Figure 5.2: Moment diagrams indicating the points at which the different loads act with respect to the front axle

Therefore:

$$748(0.850) = M_{attachment}(1.1) \tag{17}$$

$$M_{attachment} = \frac{748(0.850)}{1.1}$$
$$= 578kg$$

With the weight of the AC unit being 320 kg, this allows an attachment mass of 258 kg which is deemed to be achievable. This analysis provides the detail necessary to define the first lower level design constraint, which is:

- the mass of the attachment must be minimised in an effort to improve the stability of the skid steer loader.

5.2 Attachment Coupling

The second lower level design constraint comes from the requirement that the attachment must be securely coupled to the skid steer loader. AS ISO 24410:2021 (Standards Australia 2021b, pp. 6-7) describes the geometry to which the attachment system must be designed and fabricated (shown in Figures 5.3 and 5.4). Hence, the second design constraint is as follows:

- The coupling point of the attachment must be designed to meet the geometry requirements of AS ISO 24410:2021 - Earth-moving machinery - Coupling of attachments to skid steer loaders.

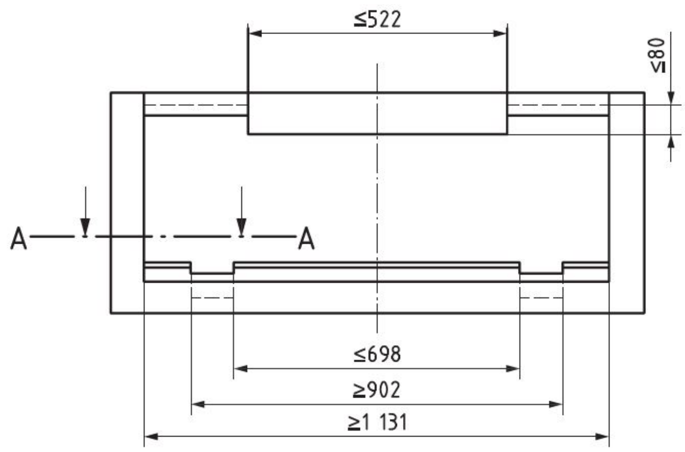


Figure 5.3: Front view of skid steer loader attachment geometry (Standards Australia 2021b, p. 6).

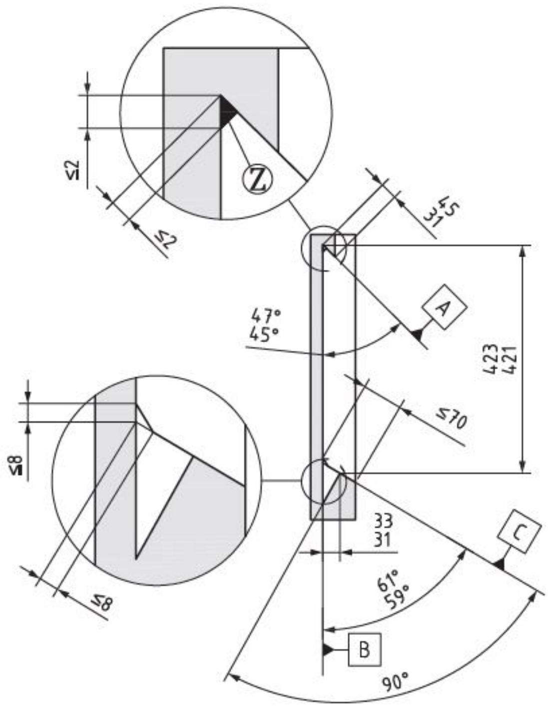


Figure 5.4: Sectional view of skid steer loader attachment geometry (Standards Australia 2021b, p. 7).

5.3 Functional Requirements

The establishment of the necessary functional requirements formed part of the research component of the project which involved determining the maximum deflection, maximum permissible stress and an appropriate FoS for the system. This section will provide a discussion on the determination of these requirements based on the information provided in Section 2.

5.3.1 Maximum Deflection

The literature review identified the relevant Australian standard (AS 4100:2020) that provides the design engineer with the necessary detail to determine the acceptable amount of vertical deflection for a beam under a number of circumstances. The attachment can be considered to be a support structure behaving as a cantliver system, and therefore, Equation 6 can be utilised to determine the recommended maximum deflection that would be applicable to the design.

Considering the primary support member to be approximately 1600 mm in length (l), the recommended maximum deflection can be tentatively given as:

$$\Delta = \frac{1}{125}(1600)$$
$$= 12.8mm$$

This value can be used as an absolute maximum as the length of the support is subject to change as the design is iterated, however until the geometry is known, the design constraint will be decided by Equation 6.

5.3.2 Maximum Permissible Stress

Equation 3 provides the detail necessary to define the maximum permissible stress of any member of the system bent about the axis of minimum strength. The standard from which this equation was taken (AS 3990:1993) also defines the maximum permissible stress for other members which are not covered by the same clause and therefore the maximum permissible stress for those members are given by Equation 4.

5.3.3 Factor of Safety

It has been stated previously that the factor of safety for the design is established due to the application of the load factor given in Equation 2 and the reduction factor applied to the maximum permissible stress in Equation 3 for beams subjected to bending moments about the axis of minimum strength, and Equation 4 for all other beams.

5.3.4 Functional Design Requirement Summary

The lower level preliminary design constraints based on the functional requirements can be summarised as follows:

- The maximum vertical deflection of the support structure of the system is to be established from Equation 6
- The maximum permissible stress of any component in the system must adhere to AS3990:1993, Section 5
- The FoS of the system is to be determined from the application of the load factor in Equation 2 and the maximum permissible stress as defined in Section 5 of AS 3990:1993

5.4 Load Cases

Before the FEA study can be conducted, a force analysis is required to determine the load cases that the attachment will be subjected to. There are two load cases that have been identified which are described below.

5.4.1 Load Case 1

The first load case describes the situation in which the attachment is used as designed by supporting the load of the AC unit and also includes the self weight of the attachment. Using the geometry of the coupling point shown in Figure 5.4, the free body diagram (FBD) in Figure 5.5 is produced where:

- W_{AC} = the weight of the AC unit
- W_{SW} = the self weight of the attachment
- F_{A_x} = the force transmitted by the top of the skid steer coupling system to the attachment in the x direction
- F_{A_y} = the force transmitted by the top of the skid steer coupling system to the attachment in the y direction
- F_{B_x} = the force transmitted by the bottom of the skid steer coupling system to the attachment (only the x component is considered)
- a = the moment arm to the point at which the weight of the AC unit acts
- b = the moment arm to the point at which F_A acts
- c = the moment arm to the point at which the self weight of the attachment acts

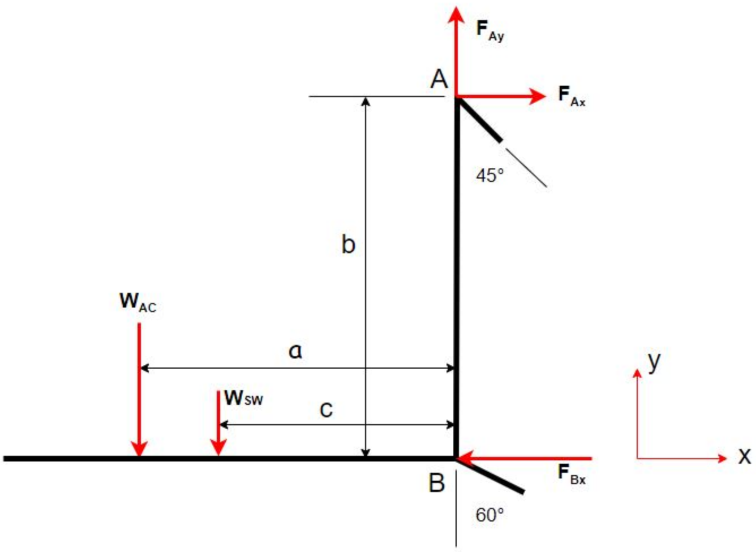


Figure 5.5: Free body diagram of the attachment used to determine the forces exerted on the coupling attachment point for load case 1.

Taking a moment about point B, the equilibrium conditions that satisfy Equation 18 gives the magnitudes of the unknown forces, F_{A_x}, F_{A_y} and F_{B_x}, as shown in Equations 19, 20 and 21 respectively.

$$\circlearrowleft + \sum M_B = 0 \tag{18}$$

$$F_{A_x} = \left(\frac{aW_{AC} + cW_{SW}}{b} \right) \tag{19}$$

$$F_{A_y} = W_{AC} + W_{SW} \tag{20}$$

$$F_{B_x} = \left(\frac{aW_{AC} + cW_{SW}}{b} \right) \tag{21}$$

5.4.2 Load Case 2

The second load case arises from the ability of the skid steer loader to potentially lift the front wheels by tilting the attachment forward and into the ground. This would not be uncommon in the underground environment and therefore the attachment must be designed to withstand this action. The FBD for this load case is shown in Figure 5.6 where:

- F_C = the force transmitted by the ground to the tip of the attachment
- d = the moment arm to the point at which the force, F_C, from the ground acts

Also note that the direction of the forces are assumed to have changed due to the force, F_C, pushing up on the attachment. F_A is assumed to now only be acting in the negative x direction, whilst F_{B_x} and F_{B_y} are now considered due to the geometry shown in Figure 5.4. This is in agreement with the working force application description given by Standards Australia (2021b, pp. 8-9).

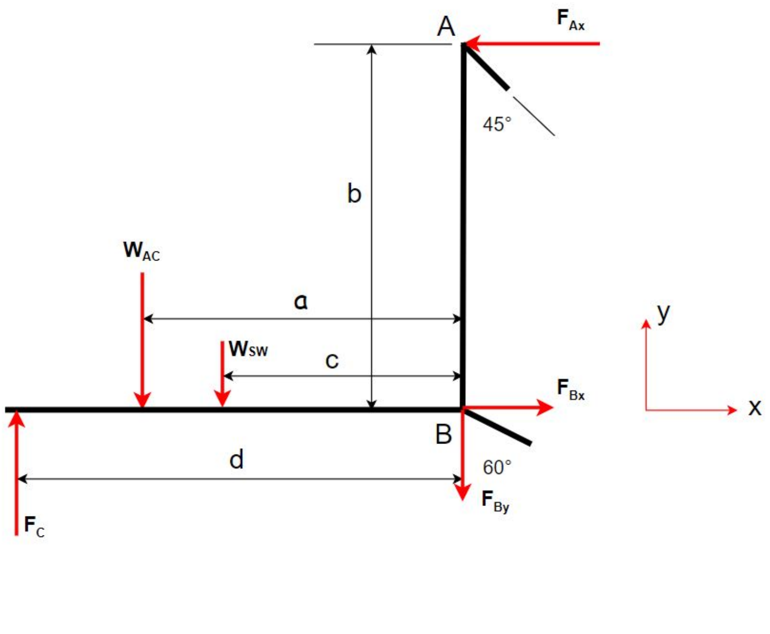


Figure 5.6: Free body diagram of the attachment used to determine the forces exerted on the coupling attachment point for load case 2.

Again, taking a moment about point B and using Equation 18, the magnitudes for F_{A_x}, F_{B_x} and F_{B_y} for the second load case are given in Equations 22, 23 and 24 respectively:

$$F_{A_x} = \left(\frac{dF_C - aW_{AC} - cW_{SW}}{b} \right) \tag{22}$$

$$F_{B_x} = \left(\frac{dF_C - aW_{AC} - cW_{SW}}{b} \right) \tag{23}$$

$$F_{B_y} = F_c - W_{AC} - W_{SW} \tag{24}$$

The magnitude of F_C (Equation 25) has been determined by taking a moment about the rear wheels at the point when the front wheels no longer contact the ground due to the tilting action of the skid steer loader. The nomenclature from Figure 5.6 has been kept for simplicity, with three additional terms introduced to determine the force, F_C. Referencing Figure 5.7:

$$F_C = \frac{(a+e)W_{AC} + (c+e)W_{SW} + fW_{SS}}{d+e} \tag{25}$$

Where:

- W_{SS} = the weight of the skid steer loader
- F_R = the reaction force from the ground on the rear wheels
- e = the distance between the rear wheels and the vertical face of the attachment
- f = the distance from the rear wheels to the skid steer CoG

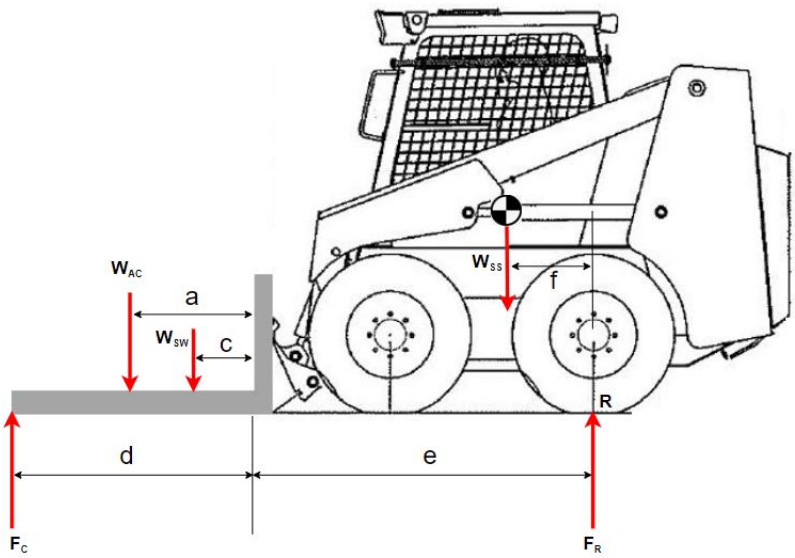


Figure 5.7: Free body diagram of the attachment used to determine the F_C in load case 2 (image adapted from Hunter Plant Hire (2018b)).

5.5 Load Case Summary

A number of values are yet to be determined due to the dependence on the mass and geometry of the attachment to be modelled. Table 4 summarises the variables of each load case where they are already known, with the asterisks next to the other variables denoting approximate values as they are subject to change depending on the model.

Table 4: Physical values of terms used to determine forces in previously discussed load cases.

Variable	Load Case 1	Load Case 2
a	0.635m	0.635m
b	0.422m	0.422m
c	0.7m*	0.7m*
d	—	1.6m*
e	—	1.58m*
f	—	0.342m
W_{AC}	3140N	3140N
W_{SW}	2543N*	2543N*
W_{SS}	29430N	29430N
F_{A_x}	8942N*	18262N*
F_{A_y}	5682N*	—
F_{B_x}	8942N*	18262N*
F_{B_y}	—	1493N*
F_C	—	7110N*

The description of the known values in Table 4 are given below:

- a - the distance from the rear face of the AC unit to the CoG (discussed in Section 3.1.1).
- b - the vertical distance stipulated by the geometry in Figure 5.4.
- f - The distance from the rear wheels to the skid steer CoG. This was determined from the known weight distribution between the front and rear wheels (65% rear, 35% front) given by Hunter Plant Hire (2018c)
- W_{AC} - the weight in Newtons of the AC units
- W_{SS} - The weight in Newtons of the skid steer loader

6 DETAILED DESIGN

The preliminary design has defined the lower level design constraints that will now enable the detailed design features to be realised. This section will discuss how these features will meet the requirements by giving a theoretical analysis where applicable that will aim to provide the necessary verification of the FEA model to follow.

This section will conclude by providing the details of the FMEA that is required to identify the failure modes of the design, with the details of the design features which mitigate those failure modes.

6.1 Primary Support Member

The design constraint in Section 5.1 requires the attachment mass to be minimised as much as practicable. One method of ensuring this is achieved is to design the primary support members to be non-prismatic in geometry, as opposed to having an area of constant cross-section. To achieve this, the geometry of the beam must be designed so that every point along the beam is optimised for the maximum allowable stress.

Due to the material availability and the adherence to the relevant Australian Standards, the primary support members will be manufactured from 10 mm AS/NZS 3678-350 plate which has a minimum yield strength of 360 MPa (Standards Australia 2016b, p. 21). This grade has been chosen for ease of manufacturing and availability. The height of the section will be determined via the optimisation process detailed below.

The section modulus of a member with a rectangular cross-section is described by the following equation (Beer at al. 2015, p. 348):

$$S = \frac{1}{6}bh^2 \tag{26}$$

where:

- b = the width of the cross-section
- h = the height of the cross-section

Substituting Equation 26 into Equation 16 and solving for the height h, of the cross-section gives:

$$h = \sqrt{\frac{6|M|}{b\sigma_{all}}} \tag{27}$$

To determine the value of h, the magnitude of the bending moment ($|M|$) along the length of the beam needs to be determined. Modelling the AC unit as a point load acting through its CoG and determining the magnitudes of the shear force and bending moments from a FBD, it can be shown that the optimum height of the cross-section that gives a constant strength is proportional to the square root of the distance from the support (Equation 28), giving the shape of the member shown in Figure 6.1:

$$h = \sqrt{\frac{6(Px)}{nb\sigma_{all}}} \tag{28}$$

where σ_{all} is defined by Equation 4 to be 237.6 MPa for the chosen material grade (multiplied by a reduction of 0.66), and the force, P (F_C from Figure 5.6), is multiplied by a factor of $1/0.60$ as is required by Equation 2 in AS 3990:1993, then divided by the number, n, of support members used (four members in this case), assuming each member shares the load equally.

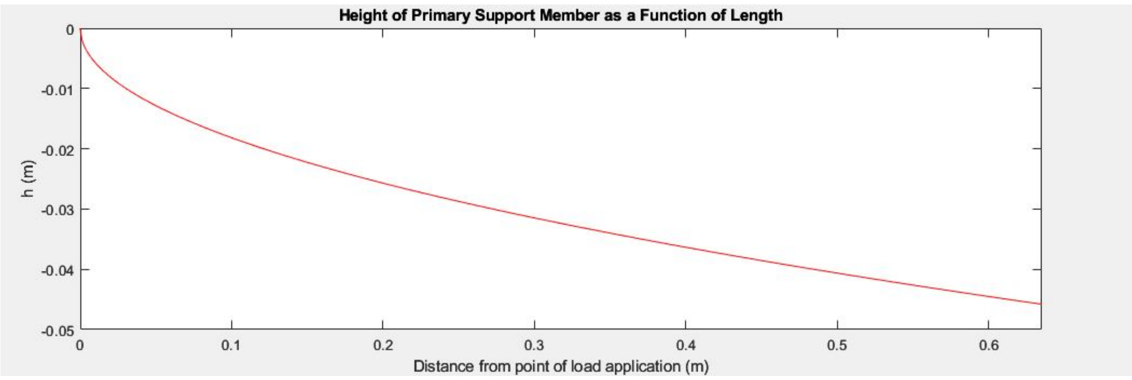


Figure 6.1: Cross-section of the non-prismatic primary support member used to support the AC unit modelled as a point load.

Although a support of this cross-section gives optimum results in terms of constant strength for the point load in Load Case 1, Figure 6.1 shows that the support abruptly ends where the point load is applied, in this case, the location at which the CoG is acting (a distance of 0.635 m from the support). This would no doubt result in issues of instability due to the lack of support further out along the length of the attachment. It is for this reason that the analysis will continue, modelling the AC unit as a linearly varying distributed load, using the same approach as above where the height, h, will be determined to give a cross-sectional area of constant strength.

Using the weight of the AC unit of 3140 N and again multiplying the load by a factor of 1/0.60, a linearly varying distributed load, w, can be modelled as a function of the length from the free end of the attachment as described by Equation 29:

$$w = \frac{2W_{AC}}{n}\frac{x}{L} \tag{29}$$

Modelling the load in this manner ensures the total load equals the weight of the AC unit (taking into account the load factor). With a distance, L, of 1.6 m, this also dictates that the centroid of the linearly varying distributed load is acting at a distance of approximately 533 mm from the supported end, which is considered to be sufficiently close to the actual case. Using Equations 13 and 14, the shear force and bending moments respectively, can be described by the following equations, again using four primary support members equally sharing the load. The shear force and bending moment diagrams are shown in Figure 6.2.

$$V = -\frac{W_{AC}x^2}{nL} \tag{30}$$

$$M = -\frac{W_{AC}x^3}{3nL} \tag{31}$$

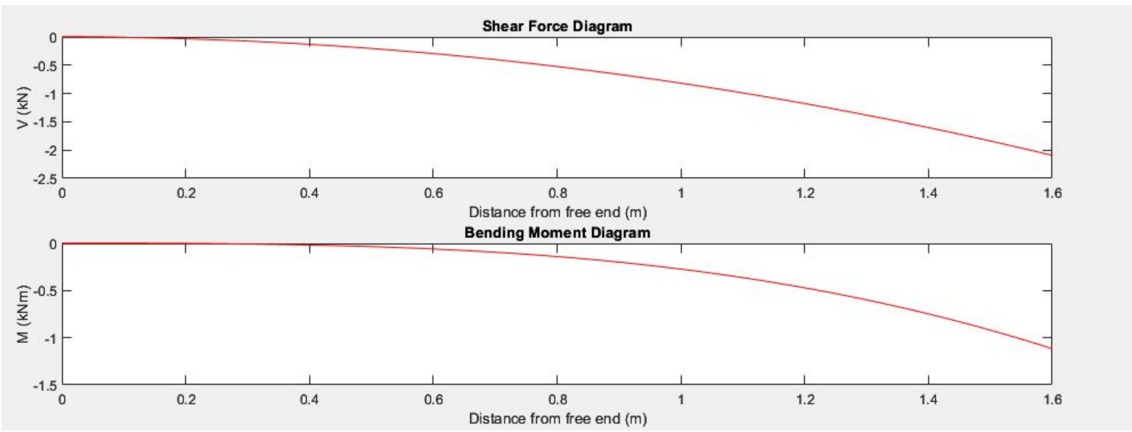


Figure 6.2: Shear force and bending moment diagrams of the AC unit modelled as a linearly varying distributed load.

Assuming four members are used as the primary supports, Equations 30 and 31 describe the magnitudes of the maximum shear force and bending moments which for this load case are 2.09 kN and 1.12 kNm respectively. Substituting Equation 31, the magnitude of the bending moment as a function of length from the free end, into Equation 27, the non-prismatic shape of the primary support member can be described by Equation 32 which is shown in Figure 6.3 below:

$$h = \sqrt{\frac{2W_{AC}}{nLb\sigma_{all}}} x^{3/2} \tag{32}$$

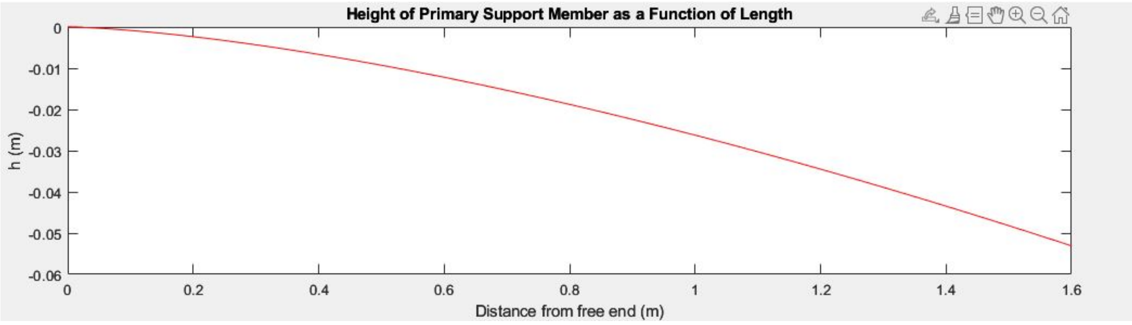


Figure 6.3: Shape of the primary support member based on the linearly varying distributed load.

Equation 32 and Figure 6.3 describe the shape of the primary support member designed to be of constant strength for the linearly varying load case. Section 5.4.2 however describes the load case in which the attachment is tilted forward into the ground, giving a reaction force of approximately 7.11 kN acting upwards on the end of the attachment. The primary support needs to be designed to resist the largest bending moment, and therefore the above analysis needs to be repeated to determine the magnitude of the bending moment as a result of the second load case.

An elementary force analysis of this load case produces an equation for the height of the member as a function of its length that is identical to Equation 28, however the sign of both the shear force and bending moments in this case are positive. Using a point load of 7.11 kN with a multiplication factor of 1/0.60, and assuming the four members share the load equally, the shear force is constant across the length of the beam with a magnitude of 2.96 kN, whilst the bending moment varies linearly from zero at the free end to a maximum of 4.74 kNm at the support. The shear force and bending moment diagrams, and the height of the member as a function of length is shown in Figure 6.4.

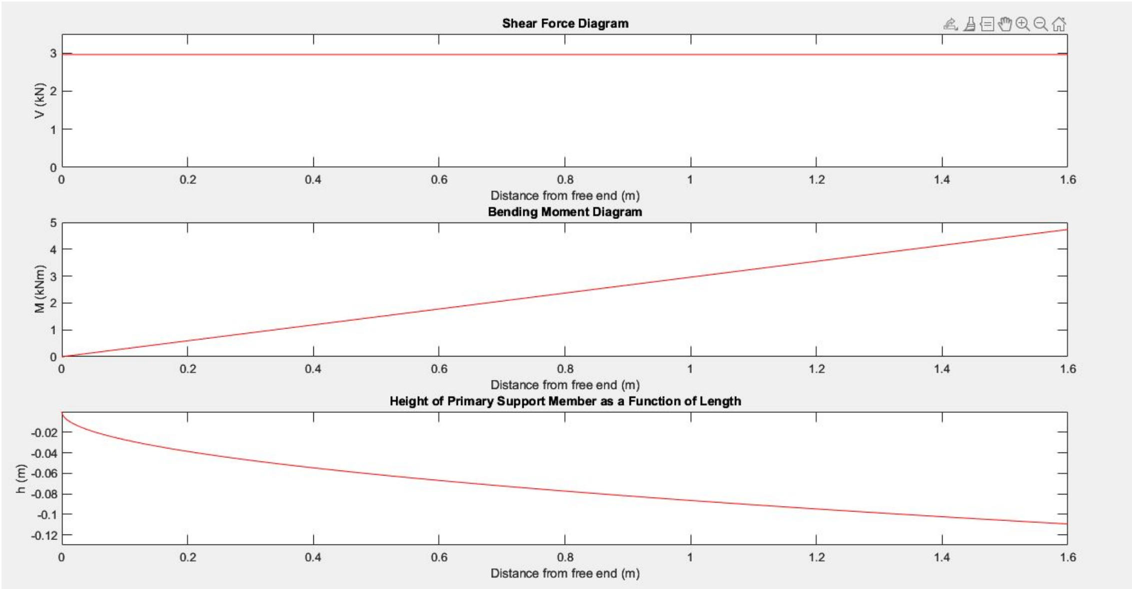


Figure 6.4: Shear force and bending moment diagrams, and the shape of the primary support member based on the application of the point load in load case 2.

It can be seen that Load Case 2 produces a greater bending moment on the primary support members, and therefore they must be designed to resist the associated stresses and deflections. A preliminary FEA study was conducted on the primary support members to verify that the beams showed values of constant stress levels along the entirety of their lengths. Figure 6.5 shows that this is the case with the exception of the location in close proximity to the application of the load and the restraint point which can be ignored.

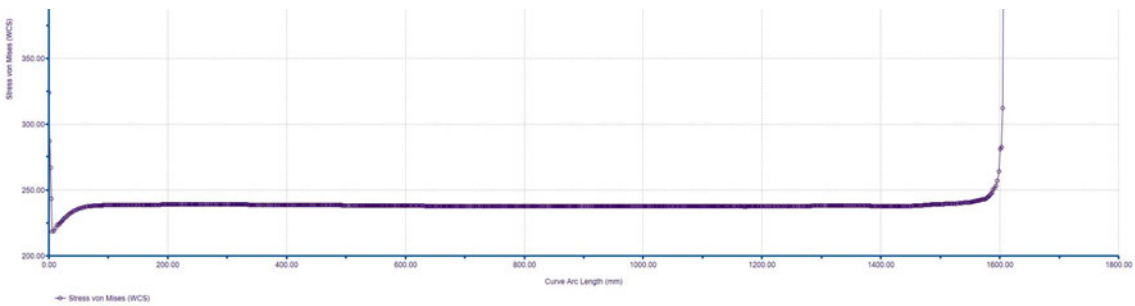


Figure 6.5: Plot of the expected constant stress magnitude along the lower edge of the primary support members.

The beams have been designed to resist the maximum permissible stress, however the expected deflection is not yet known. To calculate the deflection of the beam, the equation for the elastic curve (Equation 33) must be integrated twice to get the function which describes the vertical displacement at any location along the beam. However as the beams are non-prismatic, the second moment of area, I, must first be expressed as a function of length (Beer et al. 2015, pp. 603-604).

$$\frac{d^2y}{dx^2} = \frac{M(x)}{EI} \tag{33}$$

Equation 34 describes the vertical displacement of the primary support member as a function of its length after the double integration of Equation 33 (full derivation given in Appendix C). At a distance of $x = 0$ at the free end, $L = 1.6$ m, $b = 10$ mm, $P = 11.85$ kN, $n = 4$ members, $E = 200$ GPa, and $\sigma_{all} = 237.6$ MPa, the vertical displacement of the beam is calculated to be 37.06 mm. The preliminary FEA study was also used to verify the expected results which shows agreement with the above analysis with a maximum displacement of 37.38 mm, a difference of approximately 0.8% (Figures 6.6, 6.7 and 6.8).

$$y = \left[\frac{12\frac{P}{n}}{Eb\left(\frac{6P}{nb\sigma_{all}}\right)^{\frac{3}{2}}}\right]\left(\frac{4}{3}x^{3/2} - 2L^{1/2}x + \frac{2}{3}L^{3/2}\right) \tag{34}$$

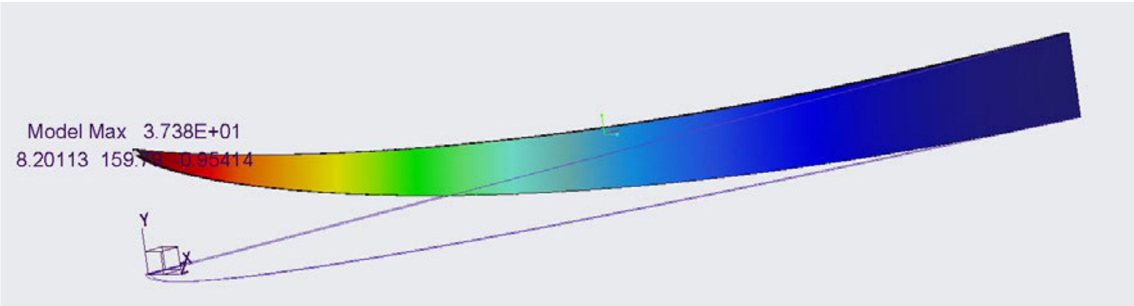


Figure 6.6: FEA study showing the simulated maximum displacement agrees with the results of the theoretical analysis.

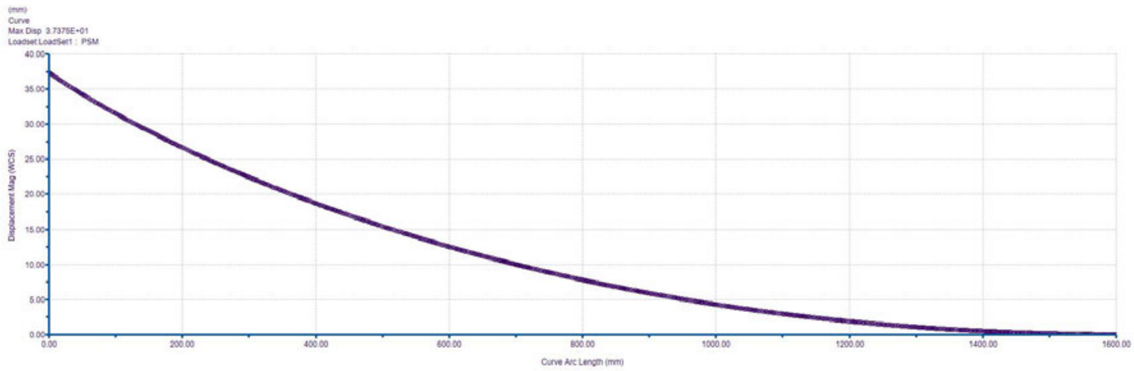


Figure 6.7: PTC Creo results showing the displacement of the primary support member.

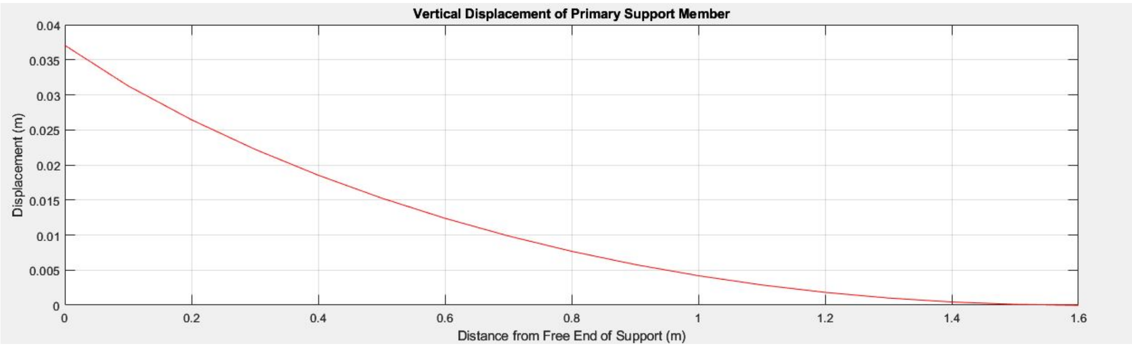


Figure 6.8: Plot of the theoretical analysis showing the displacement agrees with the simulated results.

The above analysis shows a full understanding of the behaviour of the primary support members based on the applied loading conditions. The theoretical analysis has successfully verified the results of the FEA simulation which will allow the completion of a subsequent FEA analysis on the system as a whole with the confidence that the resulting simulations are accurate.

Although the maximum displacement exceeds the recommended value of 12.8 mm given by Equation 6, it will be shown that with the addition of the main supporting plate, the maximum deflection of the system once fabricated will fall within the recommended limits.

6.2 Main Support Plate

The main support plate has been designed to provide a platform on which the AC units will be supported. It could be argued that the primary support members alone could resist the expected forces, however the main support plate performs a number of critical functions. The first function is to provide a stable platform which will allow the AC units to be safely manoeuvred from the top of the belt starters and onto the attachment. It is crucial that the system allows this manoeuvre to be made as safely as possible, and the flat, relatively low friction surface makes this possible.

The second function is to provide the means with which the AC units are secured during transport from the belt starters. This is achieved with a number of holes along the edge of the plate as can be seen in Figure 6.9. These holes allow tie down straps to be run over the top of the units, preventing unexpected movement during transport. Although these holes are not strictly designed to be used with fasteners such as bolts or pins, the location of the holes has been chosen to comply with Section 9.6.2 of AS 3990:1993 which gives the minimum distance permitted from the centre of a fastener to the edge of a plate (Standards Australia 2016a, p. 58).

Another function that is equally important is to provide both lateral and longitudinal stability to the primary support members which will aid in the prevention of localised buckling, and also as previously discussed, it will allow the deflection limits described by Equation 6 to be met.

The material with which this component will be fabricated is 10 mm AS/NZS 3678-350 plate which is the same as the primary support members. This will aid in the ease of fabrication and also provide the strength required to minimise the expected stresses and deflections.

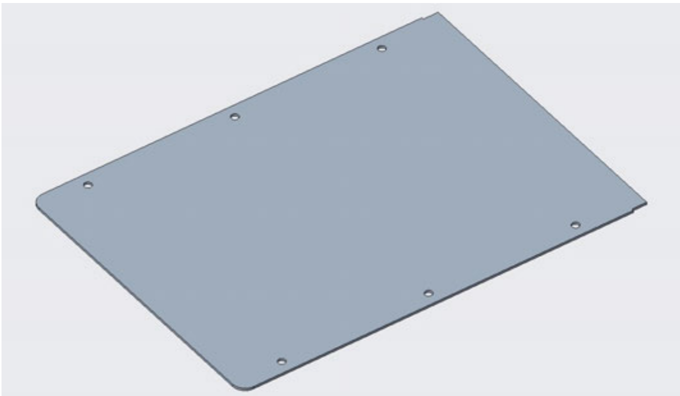


Figure 6.9: Isometric view of the main plate used to support the AC units.

6.3 Attachment System Design

In order for the system to function as intended, a subsystem needs to be designed to act as an interface between the main supporting structure and the skid steer loader. AS ISO 24410:2021 (Standards Australia 2021b) gives the geometry to which this system must comply which is shown in Figures 5.3 and 5.4 in Section 5.2. The required geometry can be realised with simple manufacturing processes applied to a number of flat steel plates, the properties of which can be chosen to withstand the expected forces that have been previously calculated with a static force analysis, which can then be verified through the FEA study.

Figures 6.10 and 6.11 show the system that has been designed. The subsystem consists of two backing plates which provide the means with which the lower locking mechanism of the skid steer loader engages. These are then attached to the lower square hollow section (SHS) member which provides the majority of the strength required to resist the expected forces transferred from the primary support members and the main support plate discussed in the previous sections. The lower SHS member is connected by vertical side plates which transfer those forces to the upper SHS member. The function of this member is to provide the means with which the upper section of the skid steer coupling mechanism engages and is achieved with the addition of a flat steel plate which forms the upper section of the geometry required by AS ISO 24410:2021.

Two additional side restraint plates and centre support tabs (shown in Figure 6.11) have been added to complete the required geometry. The side restraint plates are placed 1135 mm apart and perform the function of limiting lateral movement of the coupling mechanism within the attachment system. The centre support tabs have been placed 435 mm apart (placed on the underside of the top load plate), satisfying the dimensional requirements shown in Figure 5.3 and also provide additional strength for the top load plate.

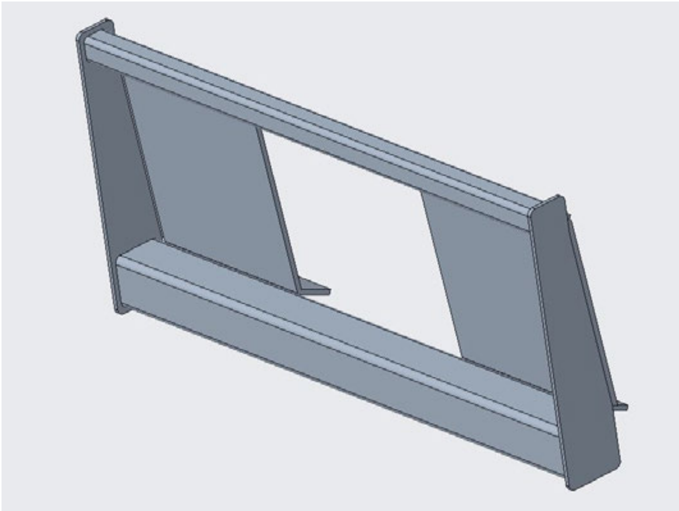


Figure 6.10: Front view of backing plate weldment.

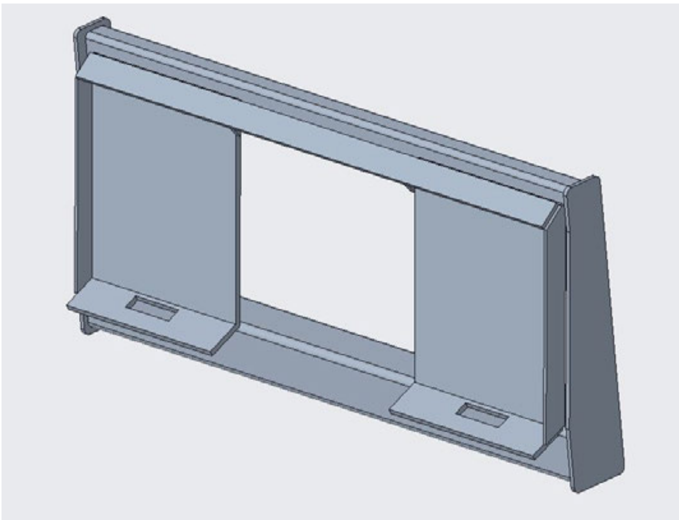


Figure 6.11: Rear view of backing plate weldment.

All of the connections between the components are achieved with welding techniques, the majority of which are fillet welds which conform to AS/NZS 1554.1:2014 (discussed further in Section 6.4).

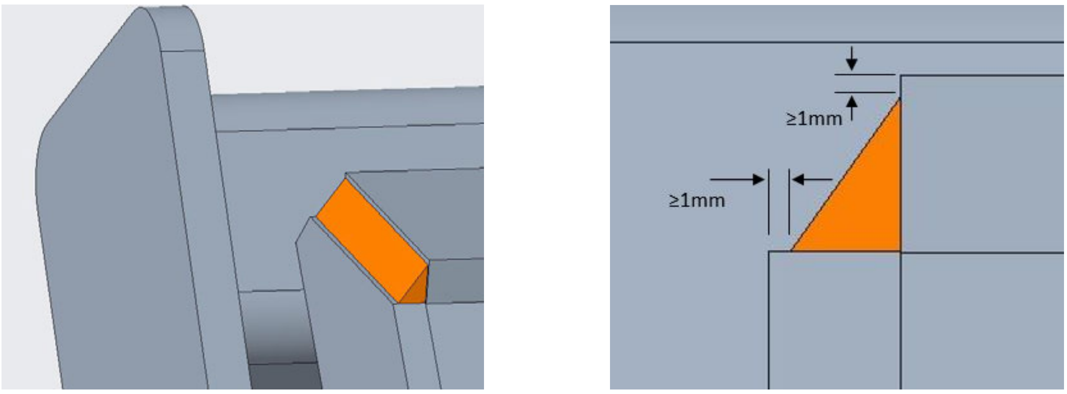
6.4 Welding

The method with which the individual components will be joined is by Gas Metal Arc Welding (GMAW). This method has been chosen for the ease with which it can be completed and also the strength that can be achieved when adhering to the recognised standards, which for the case of structural steel welding is AS/NZS 1554.1:2014 - Structural Steel Welding, Part 1:Welding of steel structures.

The majority of the assembly is welded using fillet welds. Section 3.3.5 of the above standard gives the minimum leg length that a fillet weld must be designed to based on the thickness of the parent metal being joined (Standards Australia 2014b, p. 13), however these sizes must carefully be chosen to ensure the maximum equivalent stress and maximum shear stress experienced by the welds does not exceed 57% and 33% (respectively) of the ultimate tensile strength of the electrode material which is prescribed by AS 3990:1993 (Standards Australia 2016a, p. 60).

With reference to table 4.6.1(A) in AS/NZS 1554.1:2014, all of the steel types used in the assembly are classified as steel type 4 with the exception of the top load plate, which is classified as type 1. The standard specifies a number of suitable electrode types in Table 4.6.1(B) that shall be chosen based on the steel type of the parent material. In this case, the welding electrode material chosen for mating of the system components is B-G49A 0U. A commercially available product of similar type is the Austmig ES6 G49A 3U wire electrode (Welding Industries of Australia 2023) which has an ultimate tensile strength of 570 MPa. This product slightly differs from the standard based on the impact testing temperature which is -30°C, rather than 0°C prescribed by the standard, however electrode materials with higher impact gradings are noted in the standard as also being acceptable (Standards Australia 2014b, p. 37).

Another property which has been identified as relevant to the system, and for which compliance must be achieved is the maximum leg length for fillet welds along an edge. Section 3.3.6 of AS/NZS 1554.1:2014 states that the maximum leg length of the fillet weld along an edge for material 6 mm and greater shall be the thickness of the material minus 1 mm (Standards Australia 2014b, p. 14). An example of a weld conforming to this clause is the weld between the top load plate and side restraint plates as shown in Figure 6.12.



(a) View of assembly weld between top load plate and side restraint plate which conforms to Section 3.3.6(b) of AS/NZS 1554.1:2014.

(b) View normal of assembly weld between top load plate and side restraint plate which conforms to Section 3.3.6(b) of AS/NZS 1554.1:2014.

Figure 6.12: Example of welds which conform to the requirements of edge fillet welds in AS/NZS 1554.1:2014

6.5 Finite Element Analysis

A finite element analysis of the assembly is required to determined any regions that may experience higher levels of stress than those which are prescribed by the relevant standards, and is also used to verify the maximum displacement of any component to ensure the appropriate limits are not exceeded.

Table 4 gave approximate values for the identified load cases. Now that the system has been modelled, a more accurate description of the applied forces can be calculated. The CoG of the assembly has been determined to be approximately 540 mm from the vertical faces of the side support sections, whilst the total mass was calculated to be 255 kg. With these values now finalised, the approximations can now be updated to give a more accurate description of the load cases which are shown in Table 5. Note the table includes the values derived from the static force analysis, and also the values used in the FEA study after the application of the multiplication factor given in Equation 2.

Table 5: Revised values of terms used to determine forces in previously discussed load cases.

Variable	Load Case 1	LC1 FEA Values	Load Case 2	LC2 FEA Values
a	0.635m	0.635m	0.635m	0.635m
b	0.422m	0.422m	0.422m	0.422m
c	0.540m	0.540m	0.540m	0.540m
d	—	—	1.6m	1.6m
e	—	—	1.58m	1.58m
f	—	—	0.342m	0.342m
W_{AC}	3140N	5232N	3140N	5232N
W_{SW}	2502N	4170N	2502N	4170N
W_{SS}	—	—	29430N	—
F_{A_x}	7925N	13208N	18689N	31149N
F_{A_y}	5641N	9401	—	—
F_{B_x}	7925N	13208N	18689N	31149N
F_{B_y}	—	—	1379N	2298N
F_C	—	—	7019N	11699N

The FEA study was completed using Creo Simulate using the values derived for Load Case 2, as this was identified as the most severe operating condition. Equations 3 and 4 describe the maximum stresses allowable in any of the components, which the FEA study was aimed at verifying. The weld material allowable stress has also been identified using Clause 9.8.2 of AS 3990-1993 which states that the equivalent stress in the weld must not exceed 0.57 σ_{UTS} (Standards Australia 2016a, p. 60). Table 6 below allows for quick reference when determining the magnitude of the allowable stresses in any component of the system.

Table 6: System component summary detailing the allowable stresses based on the relevant Australian Standards.

Component	Material	Allowable Stress, σ_{all} (MPa)
Primary Support Member	10 mm AS/NZS 3678-350	237.6
Main Support Plate	10 mm AS/NZS 3678-350	270
Backing Plate (Left)	10 mm AS/NZS 3678-350	270
Backing Plate (Right)	10 mm AS/NZS 3678-350	270
Vertical Side Plates	8 mm AS/NZS 3678-350	237.6
Top Load Plate	50x8 mm AS/NZS 3679.1-300	240
Lower SHS	100x100x5 mm AS/NZS 1163-C350	231
Upper SHS	50x50x4 mm AS/NZS 1163-C350	231
Side Restraint Plate	6 mm AS/NZS 3678-350	237.6
Centre Support Tabs	10 mm AS/NZS 3678-350	237.6
Weld Material	G49A 3U	324.9

For the analysis, an initial automatic meshing process was used, with subsequent control that limited the maximum element size to 10 mm applied to all components with the exception of the main support plate. This was done to ensure reasonable accuracy was achieved in areas where stresses were anticipated to change rapidly. Approximately 300 x 10^3 elements were created, with the analysis completing in under 10 minutes for each run on average.

A number of issues were noted after the first FEA study was conducted. Figure 6.13 shows very large von Mises stress in the region where the vertical side plates and the main support plate are joined. A high stress region was identified in the weld joining the backing plate to the lower SHS member (shown in Figure 6.14). The area where the primary support members are welded to the lower portion of the bottom SHS member also show high von Mises stress (Figure 6.15). Another region which showed excessive von Mises stresses is where the top load plate and backing plates are joined to the upper SHS member shown in Figure 6.16.

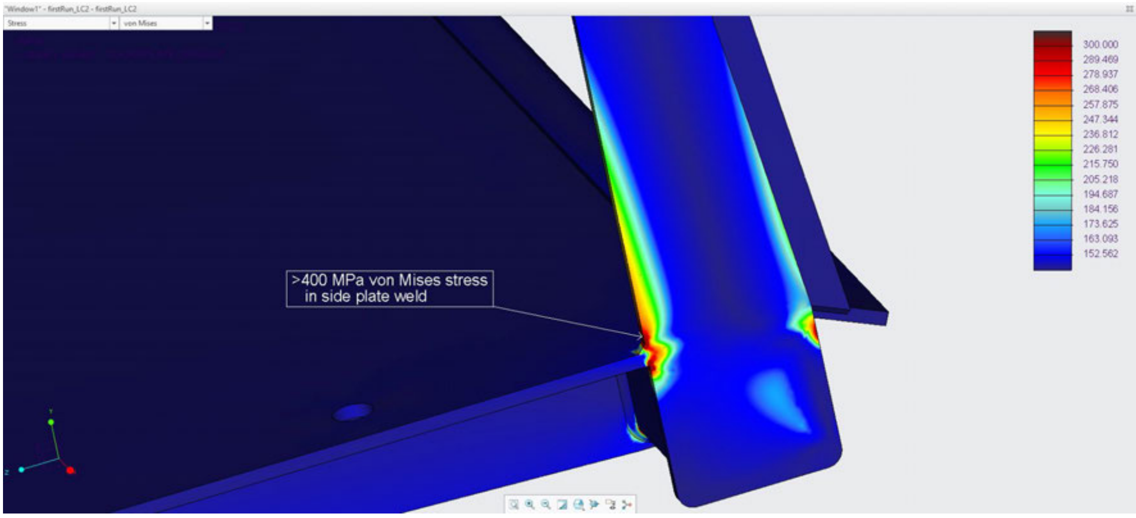


Figure 6.13: Region of high stress in the weld between the side plate and main support plate.

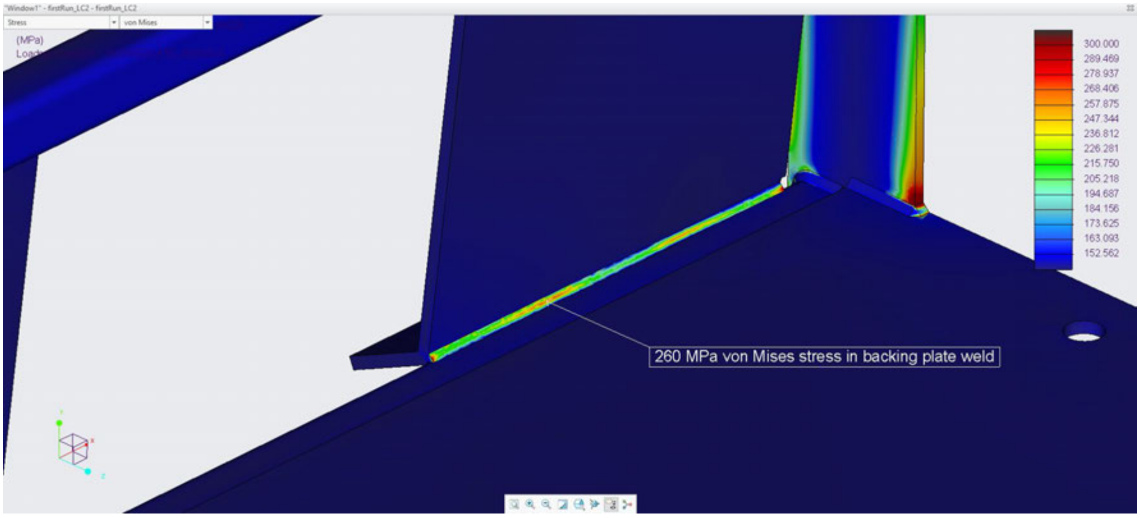


Figure 6.14: Back plate weld showing higher than allowable von Mises stress.

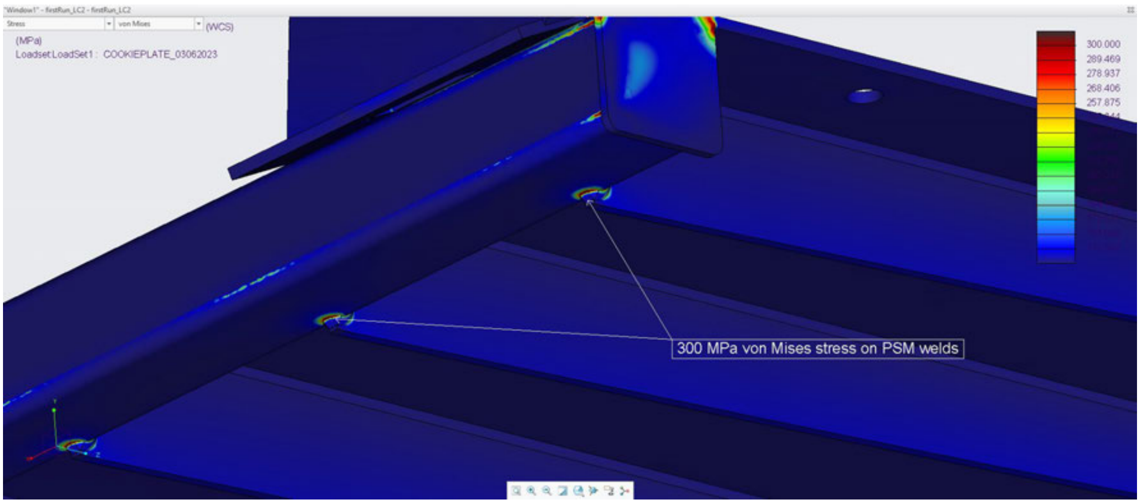


Figure 6.15: von Mises stress in excess of 300 MPa on lower sections of primary support member welds.

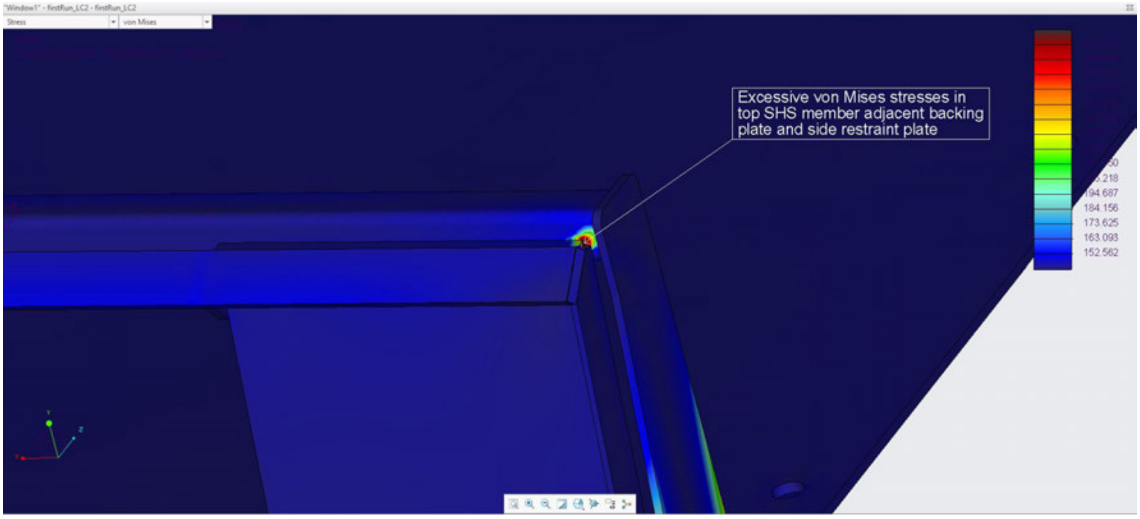


Figure 6.16: Excessive von Mises stress in the region surrounding the top of the backing plate and side restraint plates.

A number of significant modifications were made to the model to overcome the issues shown in Figures 6.13, 6.14, 6.15 and 6.16. The strength of the vertical side plates was increased by adding a reinforcing plate, effectively increasing the second moment of area, which enabled the assembly to better withstand the stresses associated with the applied bending moment (Figure 6.17). Very small areas of high stress are also shown in this figure, however they remain under the allowable limits.

The side restraint plates were able to be made redundant by incorporating their function into the modified vertical side plates. To achieve this, both the upper and lower SHS members were shortened and the main support plate was also slightly modified at allow for the change in geometry (also shown in Figure 6.17).

These modifications also aided in the reduction of the stresses in backing plate welds shown in Figure 6.18. A multi-run weld technique was also utilised to reduce the stress where the primary support members are welded to the lower SHS member (Figure 6.19).

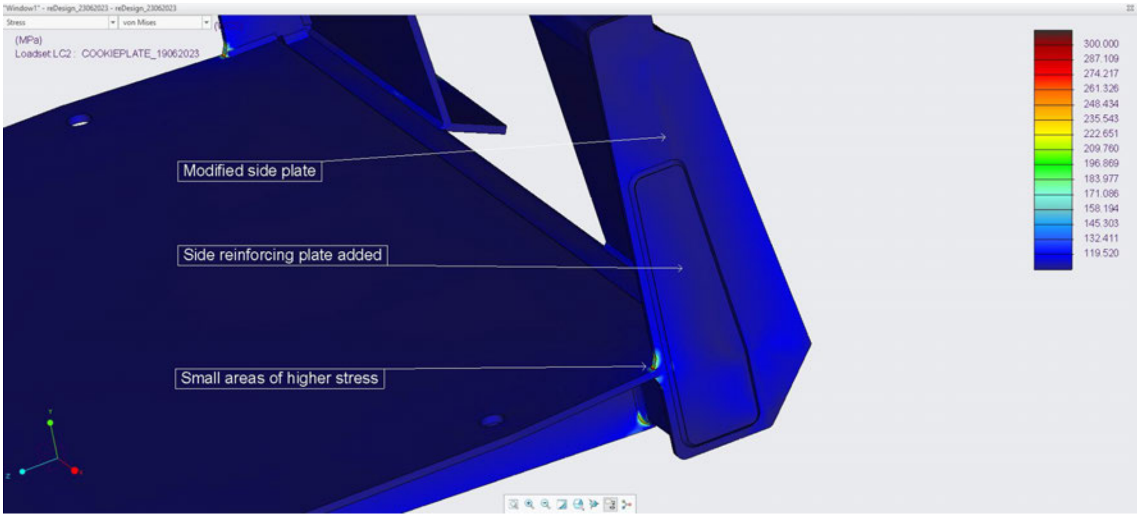


Figure 6.17: von Mises stress reduced with the addition of the side reinforcing plates.

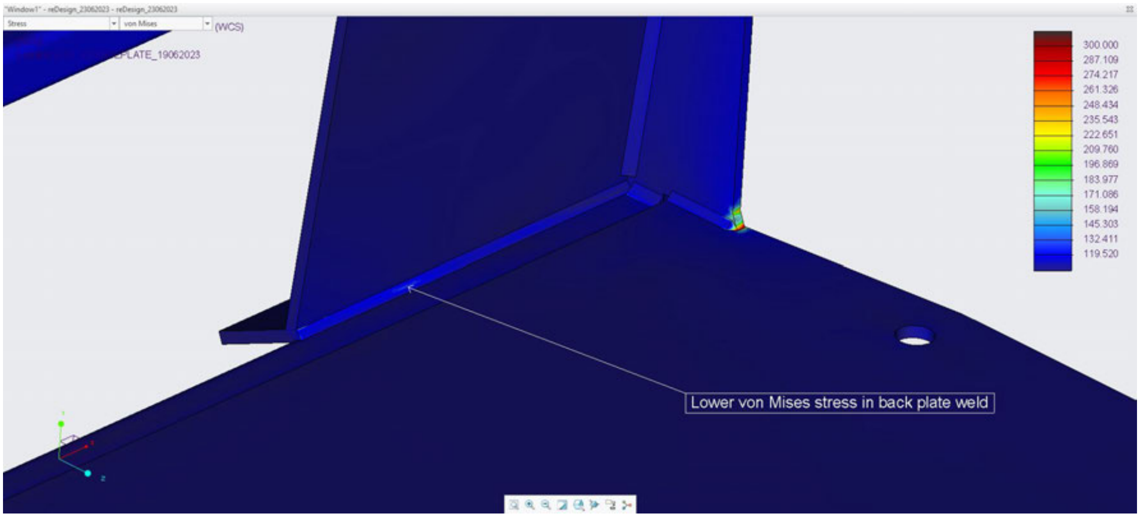


Figure 6.18: Back plate weld stresses reduced as a result of the modifications.

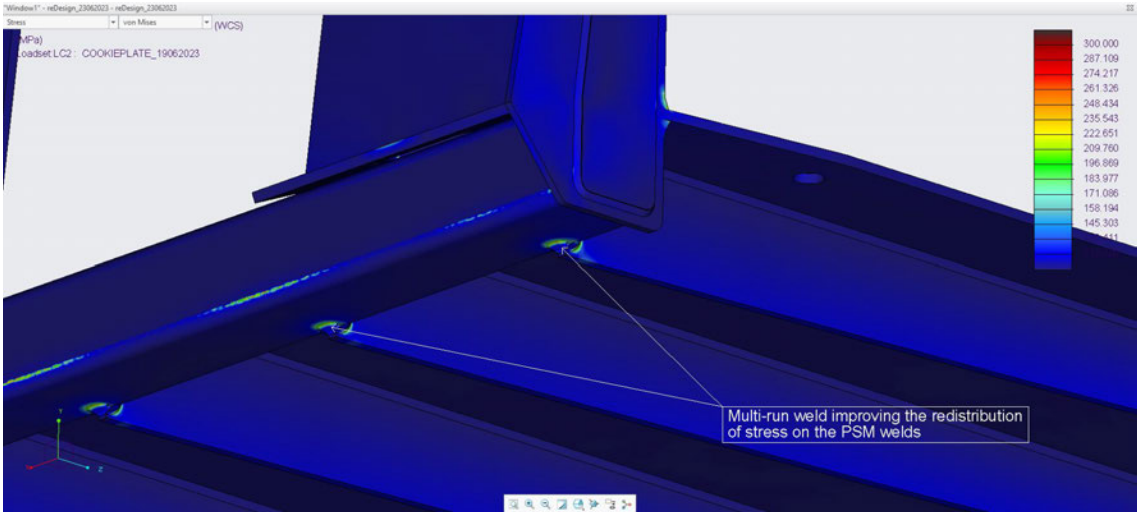


Figure 6.19: Weld stress reduced with by the inclusion of a multi-run weld on the primary support members.

The displacement was checked to ensure the requirements of Equations 5 and 6 were satisfied. A maximum displacement of 12.8 mm was determined previously for the ends of the primary support members using Equation 6. Equation 5 was used to determine a maximum displacement of 4.7 mm for the upper SHS member. Figure 6.20 shows both of these requirements have been satisfied.

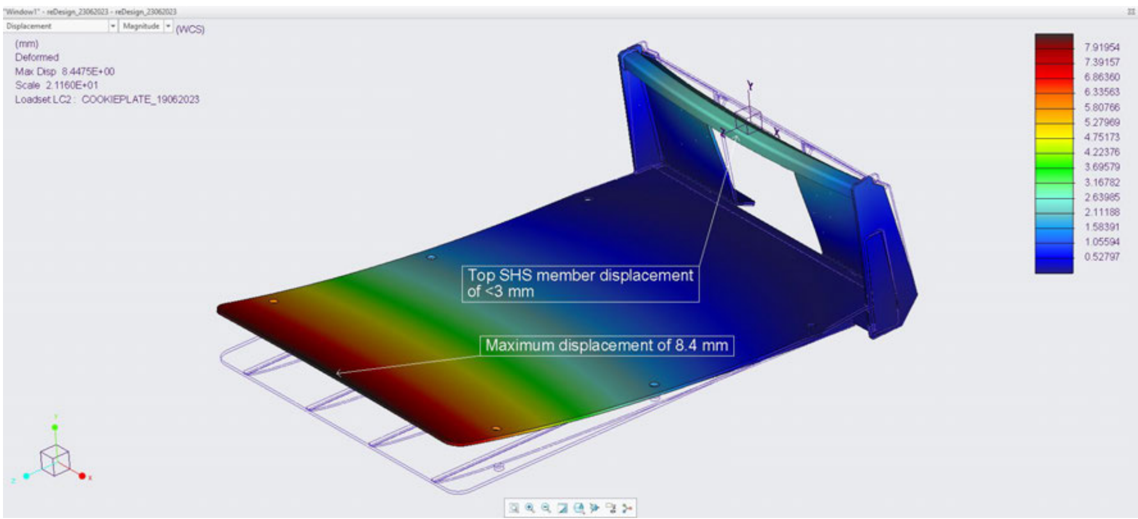


Figure 6.20: Maximum displacement of 8.4 mm at the end of the main support plate, and <3 mm at the centre of the top SHS member.

After the design was verified against Load Case 2, another FEA study was conducted for Load Case 1 to ensure the stresses and displacement were within the allowable limits. The loads used in the study were taken from Table 5 with a load used to model the AC placed where the CoG would be acting. Figure 6.21 shows a small region in which the von Mises stress appears to be excessive. This weld is designed to be continuous, however due to the geometry, issues arose whilst attempting to including a weld in this region with the FEA software. For this reason, engineering judgement was used in ignoring this small area of high stress.

Figure 6.22 shows the maximum displacement of 1.2 mm at the top SHS member, and 0.8 mm at the end of the main support plate.

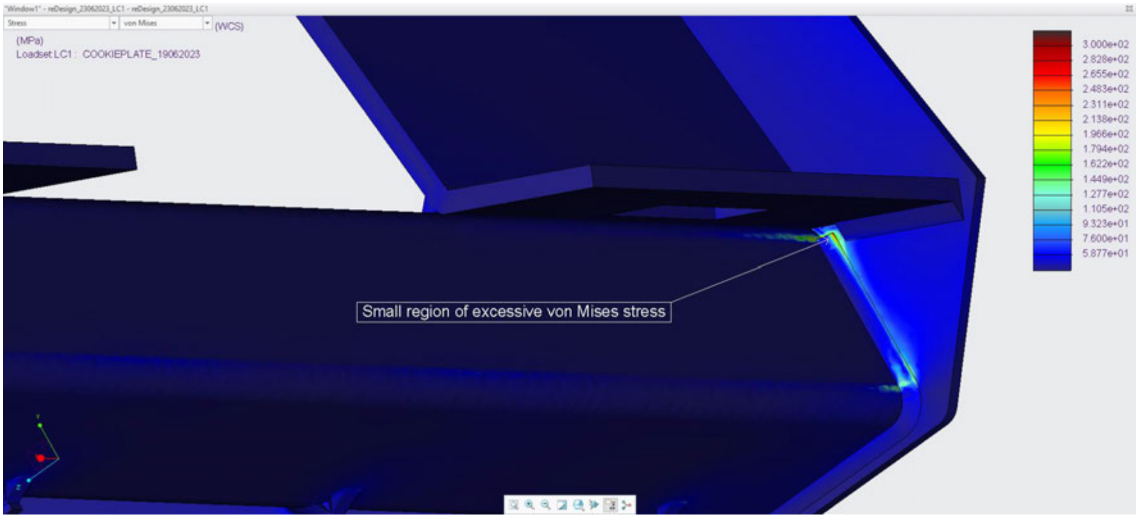


Figure 6.21: Excessive von Mises stress in a small region of the lower SHS member weld.

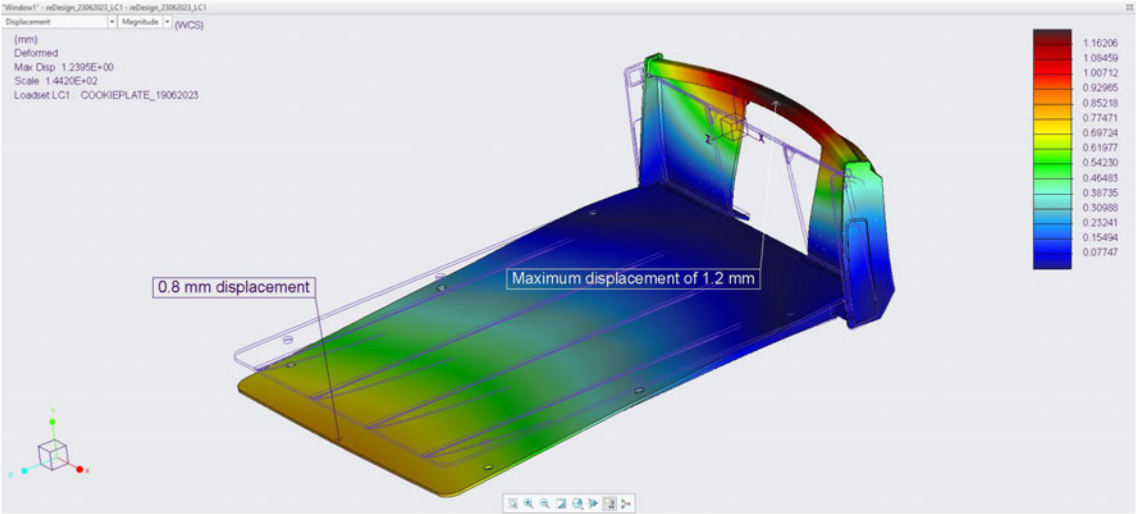


Figure 6.22: Maximum displacement of 1.2 mm at the top SHS member and 0.8 mm at the end of the main support plate

6.6 System Summary

A summary of the components and the materials with which they are fabricated is shown in Table 7 with views of the full assembly shown in Figures 6.23, 6.24 and 6.25 after making the necessary modifications discussed in the previous section. It is important to note that the material selection also complies with the requirement of AS/NZS 1554.1:2014 with regard to the permissible service temperature of the material. The standard states that the permissible service temperature of the material shall be less than the design service temperature which is determined by the Lowest One Day Mean Ambient Temperature (LODMAT) given by Figure B1 in the standard (Standards Australia 2014b, fig. B1). The location in which the system is intended to be used lies in close proximity to the 5°C LODMAT isotherm, indicating that the use of materials with low temperature impact properties is not required.

Table 7: System component summary detailing the materials and properties chosen for the assembly.

Component	Material	Qty	σ_Y (MPa)	Mass (kg)
Primary Support Member	10 mm AS/NZS 3678-350	4	360	36.6
Main Support Plate	10 mm AS/NZS 3678-350	1	360	154.8
Backing Plate (Left)	10 mm AS/NZS 3678-350	1	360	12.8
Backing Plate (Right)	10 mm AS/NZS 3678-350	1	360	12.8
Vertical Side Plates	8 mm AS/NZS 3678-350	2	360	10.1
Top Load Plate	50x8 mm AS/NZS 3679.1-300	1	320	3.55
Lower SHS	100x100x5 mm AS/NZS 1163-C350	1	350	16.1
Upper SHS	50x50x4 mm AS/NZS 1163-C350	1	350	6.05
Side Reinforcing Plate	6 mm AS/NZS 3678-350	2	360	2.52
Centre Support Tabs	10 mm AS/NZS 3678-350	2	360	0.15
Weld Material	G49A 3U	—	570 (UTS)	4
			Total Mass =	259 kg

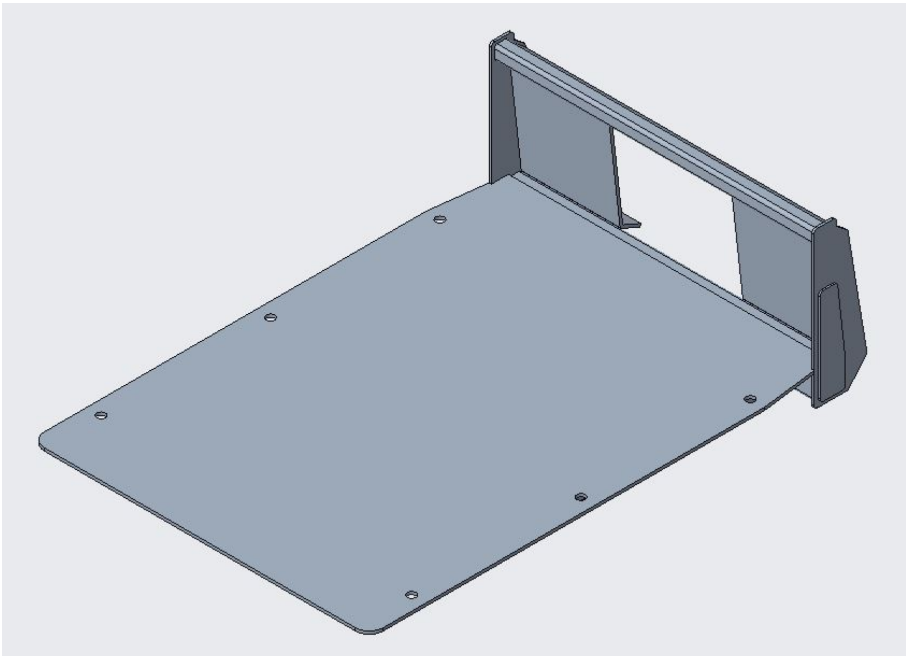


Figure 6.23: Front isometric view of the full assembly.

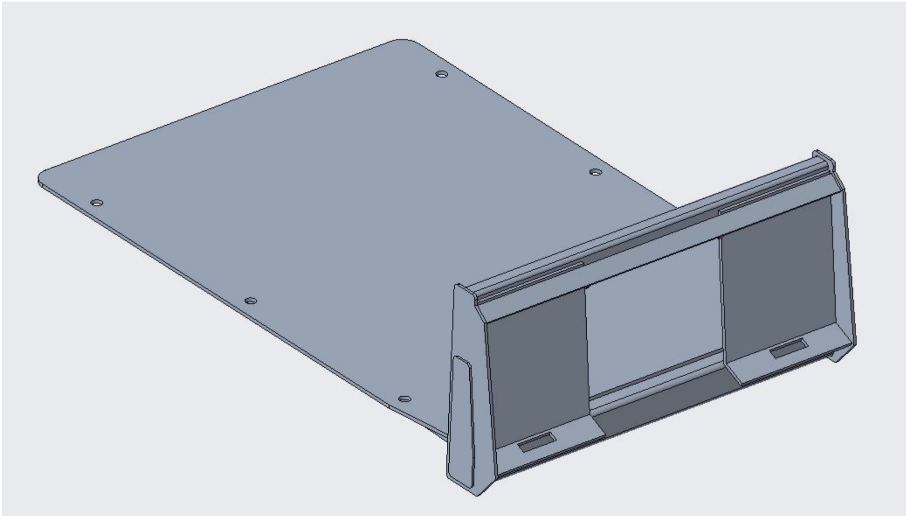


Figure 6.24: Rear isometric view of the full assembly.

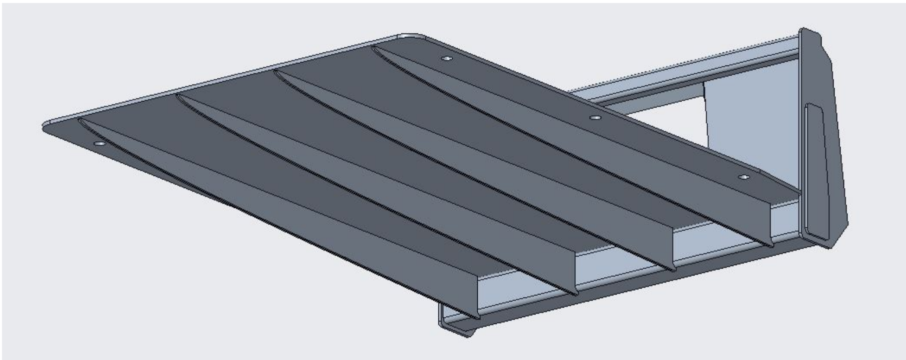


Figure 6.25: Underside isometric view of the full assembly showing the arrangement of the primary support members.

6.7 Failure Modes and Effects Analysis (FMEA)

The objective of the FMEA is to systematically review the individual components of the attachment with the intention of identifying the potential failure modes, the level to which each possible mode affects the operation of the system as a whole and to implement new controls as required to prevent each failure mode from occurring .

The methodology of the FMEA consists of the following (Bowles 2002):

1. Identify the ways in which the attachment can be used, both intentionally and

unintentionally.

2. Identify the hazards that exist from the above operation
3. Estimate the probability of those hazards occurring. This can be quantified with the use of a Table describing the possibility of the event occurring in a predetermined number of hours of operation as shown in Table 8
4. Estimate the consequences of the failure modes (Table 9)
5. Estimate the risk using a recognised risk matrix as shown in Table 10
6. Propose new methods for controlling the hazards
7. Implementation of the controls described above

Tables 8, 9 and 10 have been adopted from a method used in the author's place of work to identify the probability, severity and criticality of each unwanted event (Ashton Coal Operations Limited 2023).

Table 8: Method with which the probability and severity of an event is prescribed a letter to assist in determining the criticality.

A	Possibility of event occurring in 250 hours of operation
B	Possibility of event occurring in 500 hours of operation
C	Possibility of event occurring in 1000 hours of operation
D	Possibility of event occurring in 4000 hours of operation
E	Unlikely to occur over working life of the system

Table 9: Method with which the severity of an event is prescribed a number assist in determining the criticality.

1	Component fails to undetected unsafe system condition
2	Component fails to indicated unsafe system condition
3	Unsafe system if a second component fails simultaneously
4	System fails to safety (No hazard)

Table 10: Method with which the probability of an event is prescribed a letter to assist in determining the criticality.

Severity	A	B	C	D	E
1	1	2	3	4	5
2	6	7	8	9	10
3	11	12	13	14	15
4	16	17	18	19	20

The FMEA is included in Appendix D and has identified the failure modes that exist for each component. The most significant hazard that was identified that is applicable to all components is the overloading of the structure leading to catastrophic failure as a result of von Mises stresses exceeding the ultimate tensile strength of the material. The existing controls to mitigate this hazard which are common to all components included the confirmation of the results from the FEA study and chosen materials and welding techniques conforming to relevant Australian Standards. A number of additional controls were proposed which included the following:

- Implement a training package for the use of the system
- Attach a compliance plate to the implement which clearly states the safe working load
- Review the training package undertaken by operators to operate skid steer loader attachments
- Establish a Non-Destructive Testing (NDT) program which includes the periodic inspection of all welds by an accredited professional
- Confirm the mass of the implement with the use of appropriate weighing equipment
- Conduct a stability test conforming to AS ISO 22915:2018 and AS 2359.1:2019

6.8 Conclusion

The detailed design discussion has provided the results of a thorough analysis undertaken to describe the proposed system chosen from the initial design concepts. The decisions made to justify the design choices have been discussed, with in-depth analyses given on the critical components of the system.

The chosen material properties and the geometries which form the individual components have been discussed in detail, and then verified through the subsequent FEA studies. Slight modifications were made to improve the design where the initial FEA study showed excessive material stresses, which resulted in the redundancy of an original component, and aided in the simplification of the design.

The FMEA has described additional controls required prior to utilisation of the system to improve the safety of the design.

7 FABRICATION

As discussed in Section 3, a separate design analysis and subsequent fabrication of the chosen system is intended. This portion of the project has suffered significant delays due to factors including secondment of the author to more highly prioritised projects, operational challenges and difficulty in the engagement of third parties due to the relatively small scale of the design.

Some progress has been made however, with a number of quotes being obtained for the initial design and certification phase of the project.

Three quotes were obtained in total. One of the external companies provided an initial concept drawing accompanying their quote which provided a very interesting comparison with the author's proposed design. The external concept can be seen in Figures 7.1 and 7.2 (a general arrangement drawing is provided in Appendix G).

Figure 7.1: Concept design from external engineering consultant (Klinkhamer, B 2023, pers. comm., 18 July).

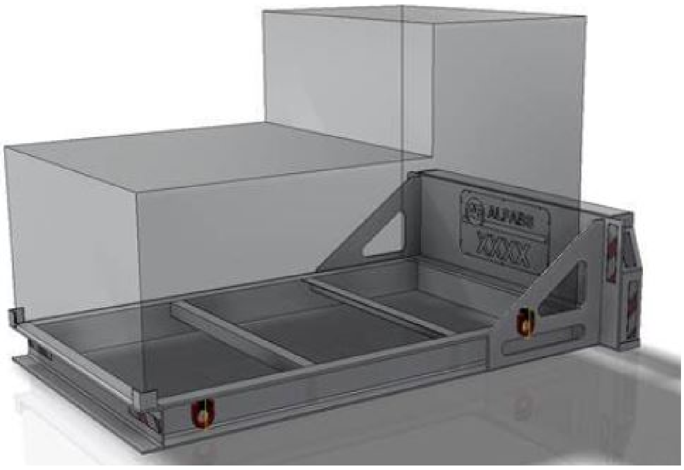


Figure 7.2: Concept design from external engineering consultant (Klinkhamer, B 2023, pers. comm., 18 July).

An important lesson in communication can be learnt from the inspection of the figures above. As the implement is designed to allow the AC units to be slid from the top of the belt starters over and onto the implement, the concept provided above would make this difficult due to the side restraints and corner supports.

Another important comparison is the open design shown in the above figures. Although much lower in mass, this design has the potential for pinch points in a number of areas between the channel sections and the AC unit. Again, inadequate communication could be attributed to this design choice. This could easily be rectified however. As the mass of this concept shown in Appendix G is stated as approximately 168 kg, the addition of a 6 mm plate to cover the open section would add approximately 75 kg bringing the total to 243 kg meaning it would be remain slightly under the mass required to maintain stability.

Arguably, the most interesting comparison between the author's concept and the one shown in Figures 7.1 and 7.2 is the use of commonly available sections with which the structure fabricated. This is common in heavy industry as the choice of these sections provides a method with which engineers can design robust structures with little cost as time is saved by removing the requirement of conducting rigorous calculations as were conducted by the author shown in Section 6.

The second quote received for the external design and certification phase was significantly more expensive than the first quote discussed above which was the main factor in the decision not to proceed with that consultant. The third company engaged for the certification and fabrication of the system was the only one of the three to come to site to gain further understanding of the problem. The author accompanied two engineering professionals on site and engaged in lengthy discussions on why the system was needed, what systems were already in place regarding the use of the skid steer loader and what the AC unit replacement process involved.

Several measurements were taken of the skid steer loader attachment point to confirm the required geometry would correlate to that which is described by the relevant Australian Standard (AS ISO 24410:2021) so the design engineers could be confident in the certification.

The decision to proceed with the third company was granted due to the rigour displayed in the collection of all required information. The author then had a spare AC unit transported from site to the contractor premises for the purpose of the third party mass and centre of gravity verification.

Regular contact was made with the engineering facility chosen to proceed with the fabrication. A concept drawing was provided (Figures 7.3, 7.4, 7.5 and in Appendix H) however the delay in receiving this information meant that not enough time was allowed for the completion of the fabrication and testing to have it included in this dissertation.

Figures 7.3, 7.4 and 7.5 show the similarities between the consultant's design and the author's own. Interestingly and completely independently, the consultant also chose to utilise support members below the main plate in a similar manner to the author, however only three members were used and were of a thicker material. Figure 7.5 also shows the simplicity in their design with regard to the constant section along their length. This is common and considered quite reasonable in heavy industry as time and cost are often important factors as small engineering firms compete for business.

The author is currently in discussion with the design engineer regarding the details for moving forward with the fabrication and certification of the system.

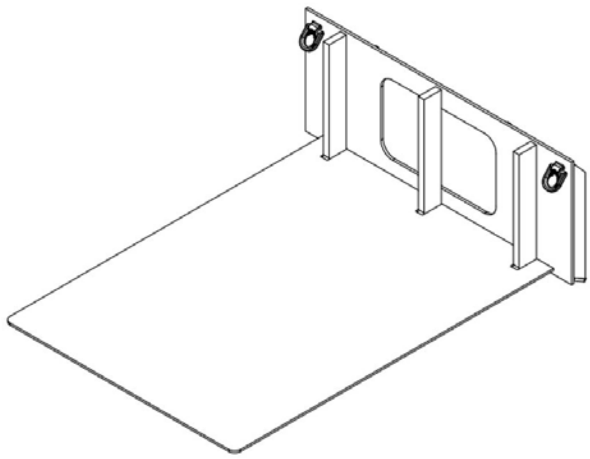


Figure 7.3: Top isometric view of concept design from external engineering consultant (Kohn, J 2023, pers. comm., 12 September).

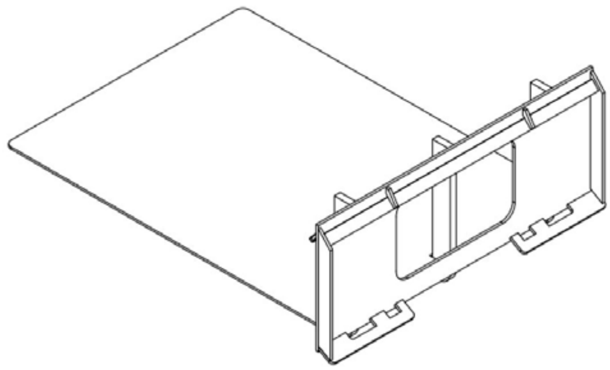


Figure 7.4: Rear isometric view of concept design from external engineering consultant (Kohn, J 2023, pers. comm., 12 September).

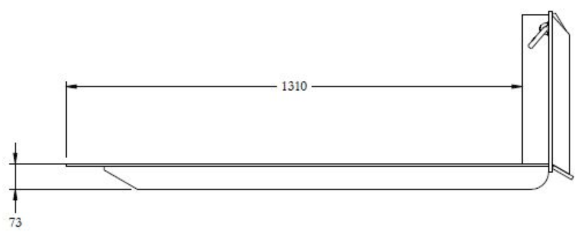


Figure 7.5: Side view of concept design from external engineering consultant showing support member geometry (Kohn, J 2023, pers. comm., 12 September).

8 DISCUSSION

Section 1.3 proposed a number of questions aimed at being answered through the research of suitable methods and the subsequent design of a system for the safe removal of the underground air conditioning units. Those questions are shown again below for clarity:

1. What are the appropriate design options for the problem discussed in Section 1.1?
2. What available materials are best suited for the fabrication of the system?
3. What is the recommended maximum displacement and maximum permissible stress for the components of the design for the intended application?
4. What factor of safety is applicable for the system?

The below is a discussion regarding the level to which the project has been able to meet the research objectives.

8.1 Design Options

A number of design concepts were proposed after the appropriate research and information gathering had been undertaken with regard to the problem discussed in Section 1.1. The information that was obtained on site was integral to enable the author to propose a small number of design options with the available systems and equipment already in place at the mine. A feasibility study had been planned to determine the most appropriate design, however research had determined that due to the relatively small nature of the project, a feasibility study could be conducted simultaneously during the conceptual design phase (Ertas & Jones 1996, p. 11). This allowed the author to limit the amount of time spent on designs that ultimately proved to be unsuitable for the problem, and focus more on concepts that were believed to better meet the design constraints.

Due to the physical dimensions of the environment in which the belt starter substations and AC units are positioned, the suitable design options were limited to the methods discussed in Section 4 which included an attachment which was designed to be secured to the front of an LHD, a skid steer loader attachment, and a monorail concept secured to the roof of the underground workings above the belt starter.

A number of high level design constraints were established early in the project which aided in determining the most suitable design option. The constraints were identified through a number of different avenues. The coal clearance system in an underground coal mine can consist of several individual conveyors, each requiring a dedicated belt starter substation installed at a different location. This information led to the establishment of the first constraint in which the chosen design would have to be suitable in each location. Without this constraint, multiple different systems may have been necessary. This situation is not ideal due to the added cost and the requirement for separate training packages for each system.

The second constraint proposed in Section 3.1.3 was adopted to ensure the chosen design limited the risk to the personnel involved in the replacement of the AC units. This project was intended to produce a system and method that would replace the existing procedure that had previously resulted in a mine worker suffering a serious hand injury, therefore it was critical that no additional hazards were introduced as a result of the change in methods. The third constraint was adopted for a similar reason, as lifting and load shifting methods used in the underground environment can present additional hazards and therefore should be avoided where possible.

The remaining constraints included the requirement to use common and available materials, the need for the mine workers current skill set to be sufficient, and the necessity for the design to adhere to all relevant Australian standards and legislation.

With the establishment of these requirements, it is believed that every feasible design concept was explored as discussed in Section 4, with those constraints aiding in the determination of the most suitable design option.

The plate style attachments were identified as the machinery available and commonly in use at an underground coal operation is designed so that many different styles of implements can be used for a number of applications. Due to the design of the attachment system, the implements on both the LHD and skid steer are easily interchangeable which creates a highly versatile fleet of mobile equipment. There are a large number of attachments currently in use for a wide variety of applications, however it was identified that due to the specialised nature of the task, a suitable and available attachment was unable to fit the requirements which necessitated the proposal for a novel implement design.

Although the design constraints required the minimising of lifting and load shifting methods, it was deemed appropriate to explore an option that used lifting equipment for the removal of the AC units. At the early stages of the project, it was unclear whether a machinery implement would be suitable for a number of reasons including physical size and access constraints, safe working capacity and stability issues. These concerns necessitated the proposal of an alternative method for further analysis in the event that the implement style concepts were deemed inappropriate due to the concerns discussed. This rationale is supported by Ertas and Jones (1996, p. 5) where they describe the need to explore alternative design options in the early phases of the project, as it can be more difficult to initiate changes when other approvals to proceed have been obtained.

The monorail method was discussed as a potential concept, and would use a similar philosophy as the longwall monorail system shown in Figure 2.5. This concept also presented a number of issues mainly in relation to the height constraints as previously discussed. However the issue that raised the most concern was the method with which the unit would remain under control when traversing from atop the belt starter to a position out in the roadway. Tag-lines are often used in lifting operations, however due to the floor gradient and the potential for lack of control, the use of a manual tag-line would introduce an additional hazard that may not be able to be reduced to a level that is as low as reasonably practicable.

A table was created (Table 3) to assist in determining the most suitable concept. This table helped quantify what was intuitively thought to be the most suitable option which was the skid steer loader attachment.

8.2 Material Choice

The only material that was thought to be suitable for the fabrication of this concept was structural steel due to the strength properties and ease of fabrication. Availability and economic reasons were also considered which made this material the most attractive. Another material which is becoming more popular in specialised applications is carbon fibre due to its strength to weight ratio. Although this material may have exhibited the properties desirable in an application which requires the minimisation of mass, it was never seriously considered due to the associated complex process with which carbon fibre components are manufactured, the lack of local carbon fibre manufacturing facilities and the relatively high cost involved in the production of these components.

Structural steel is available in many different grades, and the choice of grade with which each component was designed started with the identification of the most severe load case. A method was used to design the primary support members which would ensure the allowable stress was not exceeded, yet also minimised the material used in the fabrication of the component. This method led to the description of the primary support member geometry based on a number of factors including the allowable stress. The equation that gave a description of the geometry of the primary support members is Equation 28 (shown again below for clarity).

$$h = \sqrt{\frac{6(Px)}{nb\sigma_{all}}}$$

The three independent variables that were altered to give differing geometries were the width of the members, the number of members, and the allowable stress. The variables were changed with the intent of providing a component that would be able to serve its intended purpose, be compatible with other components in the system, and minimise mass due to the stability issues that had been identified. This process was of course an iterative one and is best described by visualising how the resulting geometry changed with each modification.

The initial grade of plate that was chosen was AS/NZS 3678-250, as through verbal discussions with other professional engineers, it was found to be the most commonly used in the industry. Intuitively, four members and a width of 10 mm plate were also used in the initial calculation which gave the geometry shown in Figure 8.1. Although the geometries are very similar, the height of the member at a distance of $x = 1.6$ m (closest to the skid steer loader attachment) changes appreciably when the allowable stress is changed with different grades of material.

A method with which the primary support members joined to the attachment subsystem was necessary. A design choice was made to weld the primary support members to a section of

AS/NZS 1163-C350 100x100x5 mm SHS used as the lower member of this subsystem. This meant that the AS/NZS 3678-350 plate with a height of 109 mm at the attachment end ($x = 1.6$ m) best suited the intended application.

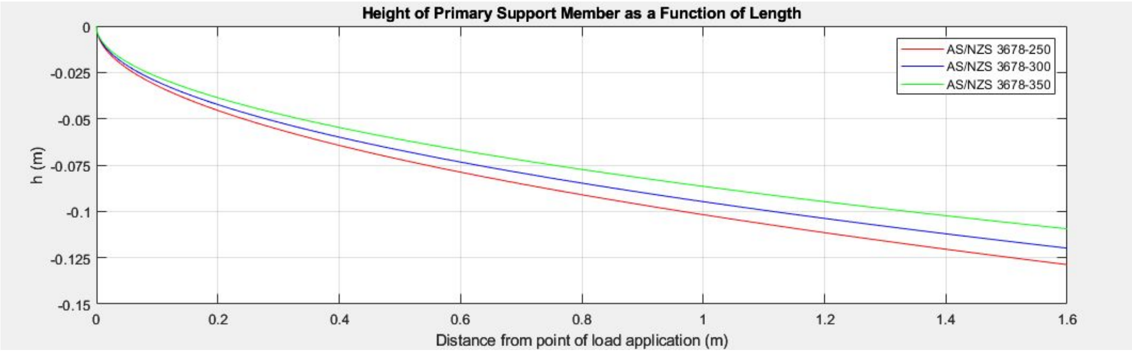


Figure 8.1: Differing geometries of the primary support members when changing the allowable stress with different grades of plate.

After the material grade had been chosen, a similar process was used to calculate and visualise the geometry buy changing the number of supports used and the width of the members (Figures 8.2 and 8.3 respectively).

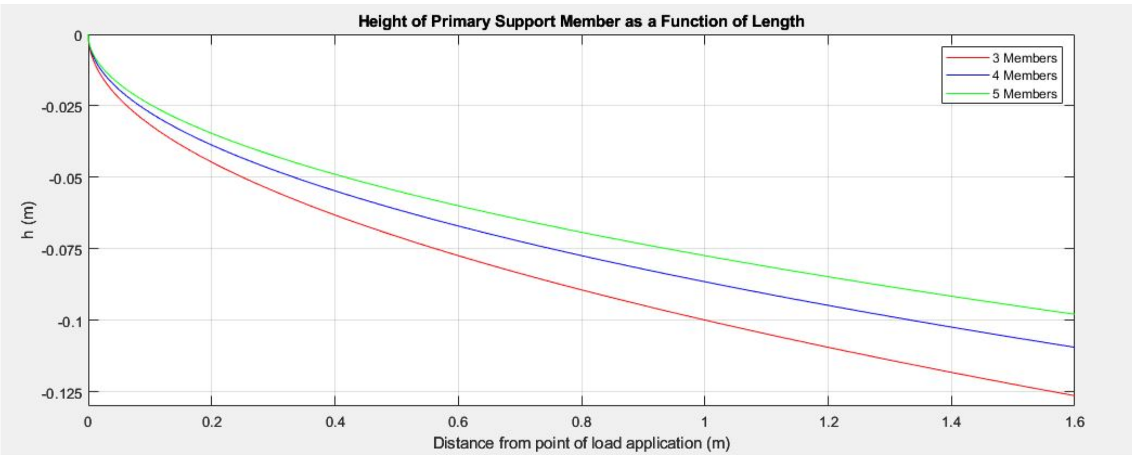


Figure 8.2: Differing geometries of the primary support members when changing the number of members used.

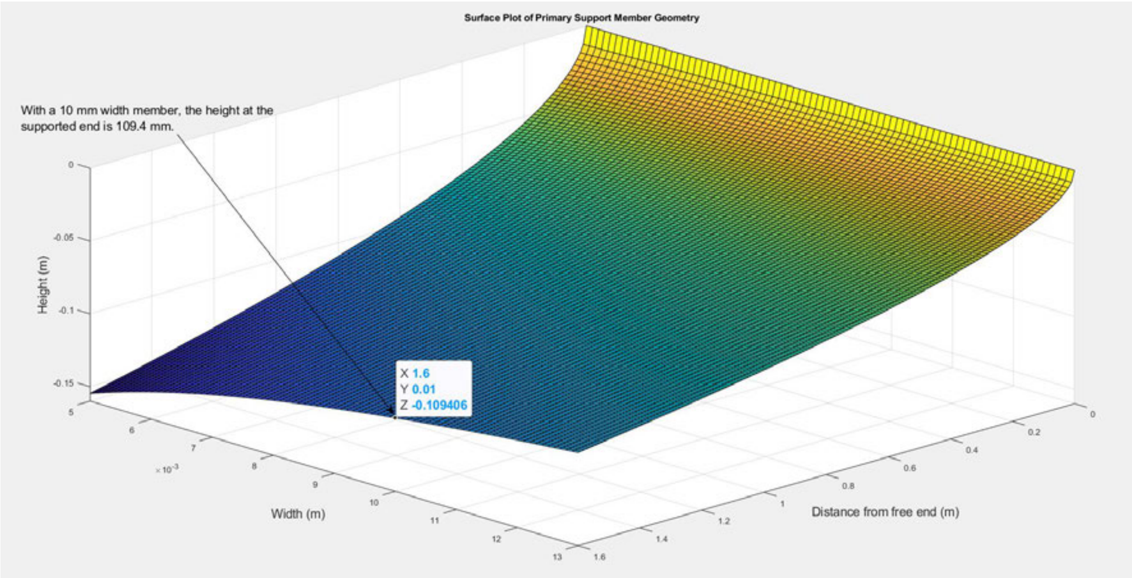


Figure 8.3: Differing geometries of the primary support members when changing the width of plate used for fabrication.

The material with the properties that complied with the allowable stress requirement for the primary support members was found to be AS/NZS 3678-350 10 mm structural steel plate, with the use of four members. For ease of manufacturing, this same grade and size plate was proposed for the design of the main support plate and both backing plates.

The materials chosen for the remaining components that made up the attachment assembly all adhered to the relevant Australian Standards, and included 50 x 8 mm AS/NZS 3679.1-300 flat bar for the top load plate, 50 x 50 x 4 mm AS/NZS 1163-C350 and 100 x 100 x 5 mm AS/NZS 1163-C350 for the upper and lower square hollow sections respectively.

8.3 Stress and Displacement

The recommended maximum displacement of the system was determined with research of the relevant Australian Standards. With the primary support function of the implement acting as a cantilever and the upper SHS member acting as a span type support, it was determined that the deflection limits given in AS 4100:2020 (Standards Australia 2020, p. 185) would be applicable. With the main cantilever support being 1.6 m in length, Equation 6 gave a limit of 12.8 mm, with which the system has been shown to comply.

The maximum displacement of the upper SHS member was shown to be less than 3 mm, which complies with the maximum displacement determined by Equation 5.

Table 11 summarises the recommended maximum, and actual deflection for the most severe load case of the two main areas for concern being the end of the main support plate and the mid point of the upper SHS member.

Table 11: Maximum recommended and actual deflection of the system components.

Component	Recommended Maximum (mm)	Actual (mm)	Equation
Main Plate	12.8	8.4	6
Upper SHS	4.7	2.7	5

As previously stated, the decision was made to use the same material for the main support plate that was used for the primary support members. This was decided for ease of manufacturing with the availability of modern manufacturing methods such as automated profile cutting that would allow the main support plate and primary support members to be cut from the same sheet of plate in one process. An option was explored however, using 8 mm AS/NZS 3678-350 for the main support plate to determine the effects on the structural rigidity under the same loading conditions. This option was investigated due to the necessity in the reduction of mass, which would aid in maintaining the stability of the skid steer loader.

An additional FEA study was conducted to determine the effects on the maximum deflection under load cases 1 and 2 (Figures 8.4 and 8.5 respectively) which showed that the deflection of the implement still remained within the allowable tolerances under both load cases.

The mass of the 8 mm plate was calculated to be 123.8 kg using a steel density of 7827 kg/m^3 (the default standard structural steel density in the Creo FEA software), a reduction of 31 kg compared with the aforementioned 10 mm plate. The use of the 10 mm plate is the preferred option due to the increased rigidity and still allows the system to fall within the calculated mass requirement to maintain stability, however if experimental verification found the system mass to be in excess of the theoretical calculations, or subsequent stability testing required a further reduction in mass, the option for the addition of an 8 mm main support plate still exists.

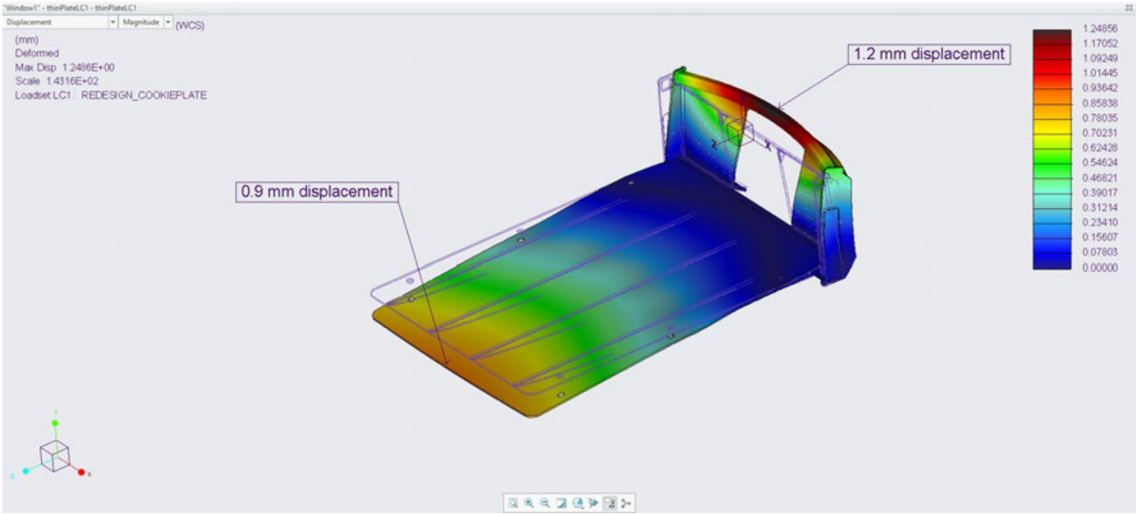


Figure 8.4: Maximum displacement of 0.9 mm and 1.2 mm displacement at the end of the main support plate and the top SHS member respectively, due to forces associated with load case 1 using an 8 mm main plate.

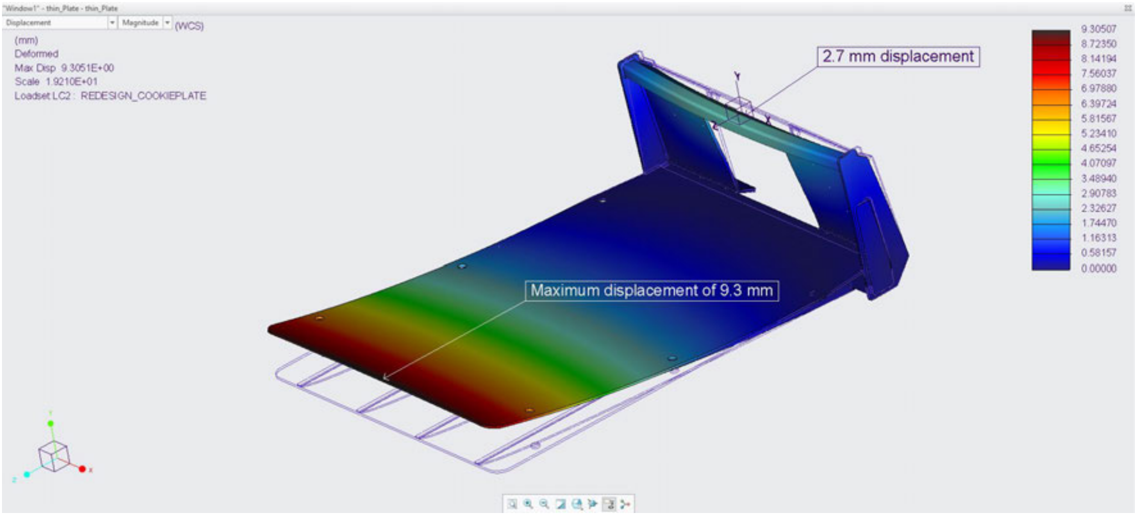


Figure 8.5: Maximum displacement of 9.3 mm due to forces associated with load case 2 using an 8 mm main plate.

The maximum permissible stress for each system component was similarly determined from the relevant Australian Standards. AS 3990-1993 - Mechanical Equipment - Steelwork was determined to be the relevant standard which is applicable to numerous applications including mining equipment using the working stress method (Standards Australia 2016a, p.6).

The allowable stress was determined from clause 5.2 of the above standard which requires the maximum stress in beams bent about the axis of minimum strength to be less than the yield stress of the material multiplied by a factor of 0.75 (Equation 3). The beams for which this clause was applicable were determined to be the top load plate and the main support plate. For beams bent about the axis of maximum strength such as the primary support members and vertical side plates, the maximum permissible stress as described by the same clause was the yield stress of the material multiplied by a factor of 0.66 (Equation 4) (Standards Australia 2016a, eqs 5.2(1), 5.2(2)). The upper and lower SHS members were symmetric about both of their cross-sectional axes, so a conservative approach was taken by analysing the stress in these members using the latter rule (Equation 4).

Table 6 in Section 6.5 summarised the magnitude of the allowable stress for each component and is shown here again for clarity.

Table 6. System component summary detailing the allowable stresses based on the relevant Australian Standards.

Component	Material	Allowable Stress, σ_{all} (MPa)
Primary Support Member	10 mm AS/NZS 3678-350	237.6
Main Support Plate	10 mm AS/NZS 3678-350	270
Backing Plate (Left)	10 mm AS/NZS 3678-350	270
Backing Plate (Right)	10 mm AS/NZS 3678-350	270
Vertical Side Plates	8 mm AS/NZS 3678-350	237.6
Top Load Plate	50x8 mm AS/NZS 3679.1-300	240
Lower SHS	100x100x5 mm AS/NZS 1163-C350	231
Upper SHS	50x50x4 mm AS/NZS 1163-C350	231
Side Restraint Plate	6 mm AS/NZS 3678-350	237.6
Centre Support Tabs	10 mm AS/NZS 3678-350	237.6
Weld Material	G49A 3U	324.9

The FEA study of the initial design identified a number of areas which were in excess of the permissible stress as shown in Figures 6.13, 6.14, 6.15 and 6.16. These high stress areas required the redesign of the attachment subsystem and resulted in a simplification with the side restraint plates being made redundant by reducing the overall width to the width described by AS ISO 24410:2021.

8.4 Factor of Safety

Two equations were identified as part of the literature review which proposed suitable methods with which the factor of safety of the system could be determined (Equations 7 and 8). However

Method 2 of the *Fully Rigid Design Method* given in AS 3990-1993 requires the system to be designed to withstand the anticipated loading conditions multiplied by a factor of 1/0.60. Section 5 of the same standard requires the magnitude of allowable stress to be determined by multiplying the yield stress by a factor of either 0.75 or 0.66 depending on whether the bending moment occurs about the axis of minimum or maximum strength (Standards Australia 2016a, p. 14; p. 22).

As multiplication factors are used on both the loading conditions and the allowable stress, a FoS can be determined by examining Equations 7 and 8 and incorporating the conditions imposed by the standard described above.

The significant strength of the material in Equation 7 can be taken as the yield strength, and the significant stress from the expected loads can be taken as the yield strength multiplied by either 0.75 or 0.66 (denoted in the following equation as ϕ). Similarly, the design overload in Equation 8 can be interpreted as the expected loads multiplied by 1/0.60. More precisely, the design overload refers to the load at which the design would be expected to fail, as in a buckling problem and may be significantly less than the yield stress. However, this term can be interpreted in this context to determine the minimum FoS if the system is shown to withstand those loading conditions.

$$FoS = \frac{significant\ strength\ of\ the\ material}{significant\ stress\ from\ expected\ loads} = \frac{Yield\ Strength}{(\phi)Yield\ Strength}$$

$$FoS = \frac{design\ overload}{normally\ expected\ load} = \frac{(1/0.60)Expected\ Load}{Expected\ Load}$$

As these conditions have both been incorporated into the design, the above equations can be multiplied together to give the overall FoS for members with bending moments about their axis of minimum and maximum strength (Equations 35 and 36 respectively):

$$FoS_{min} = \left(\frac{1}{0.75}\right)\left(\frac{1}{0.60}\right) = 2.22 \tag{35}$$

$$FoS_{max} = \left(\frac{1}{0.66}\right)\left(\frac{1}{0.60}\right) = 2.52 \tag{36}$$

It is therefore determined that provided the design is shown to withstand the loading conditions and the material stress remains below that which is described, the minimum FoS for the system can be taken 2.2. This value is in agreement with the description given by Juvinall and Marshek (2012, p. 276) in which it is stated that a FoS between 2 and 2.5 is applicable for systems with materials used in situations where the applied loads and subsequent stress can be adequately determined. The desirable safety factor of 2.5 discussed by Saldana-Robles et al. (2020, p. 12), although slightly higher due to the difference in application, is also in agreement with the values described.

A comparison can be made between the designed FoS and the results obtained by the FEA study. Very small areas of relative high stress were observed (approximately 300 MPa, see Figure 8.6) which would suggest a lower FoS than discussed above. However, it has been determined through engineering judgement that these stresses would not manifest in the physical design due to the complex geometry in which these stresses are detected.

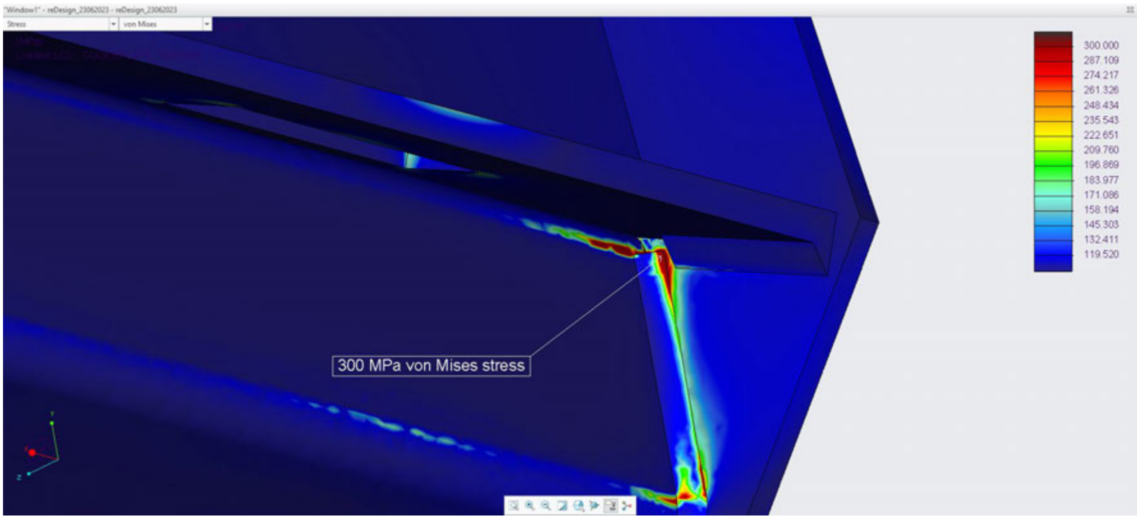


Figure 8.6: Very small region of 300 MPa von Mises stress in region adjacent vertical side plate and lower SHS weld.

A detailed visualisation of the stress on the lower edge of the primary support members can be seen in Figure 8.7. These members were designed as non-prismatic beams of constant strength. The peak of approximately 100 MPa (ignoring the stress in close proximity to the welded joint) coincides with the applied concentrated load used to model the AC unit, which describes the reason as to why the observed stress is not exactly constant as Figure 6.5 previously suggested.

Also worth noting is the rapid decline in stress at the free end of the support. This is explained by the load being applied over a short area as shown in Figure 8.8 as would reasonably be expected to occur in practice (as opposed to a concentrated point load).

Using this observed peak of 100 MPa and the allowable stress for this member of 237.6 MPa, Equation 7 provides a FoS of approximately 2.38 and suggests an agreement with the determination of the FoS between 2.0 and 2.5 discussed previously.

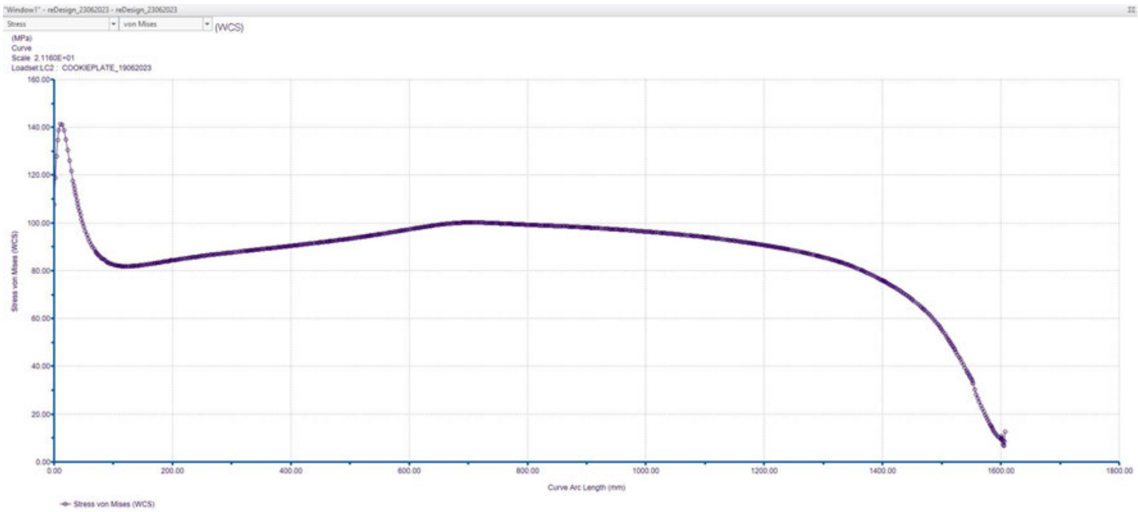


Figure 8.7: Graph of von Mises stress on lower edge of primary support members.

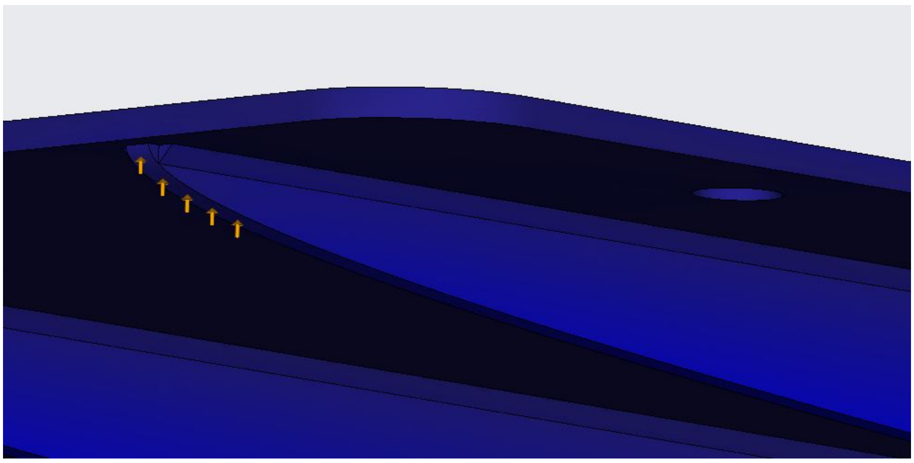


Figure 8.8: Load distributed over a small distance on the primary support member.

8.5 Conclusion

The preceding has provided a detailed discussion of the research questions that were proposed during the early stages of the project. The suitable design options, materials, functional requirements and factors of safety described suggest a design that would be suitable for the intended application.

9 FURTHER WORK

A number of requirements have been identified which have not yet been completed with regard to both the implementation of the system prior to use, and also the ongoing life cycle management. These aspects of the project are discussed in this section.

9.1 Stability Testing

Prior to the certification and use of the proposed system in the underground environment, stability testing needs to be undertaken as required by AS ISO 22915.1:2018 - Industrial Trucks - Verification of Stability and AS 2359.1:2019 - Powered Industrial Trucks - Part 1: General Requirements. This is intended to occur when the attachment is delivered to site, with the use of the available skid steer loader.

AS ISO 22915.1:2018 defines the requirements of the test procedures for determining and verifying the stability of industrial trucks. The procedure described requires the truck to be placed on a surface capable of tilting the truck. The standard also allows the use of other methods which provide the same result as the tilt table method. For example, the test performed on a slope of the same angle would be considered a suitable method in the absence of a tilt table (Standards Australia 2018).

The actual positioning of the trucks tipping axis relative to either the tilt table axis or slope of the ground on which the test is performed, and the required angle which must be achieved without tipping is given in AS ISO 22915.2:2021 - Industrial Trucks - Verification of stability. Part 2: Counterbalanced trucks with mast (Standards Australia 2021d). Prior to this test procedure being carried out on site, a thorough risk assessment must be carried out with the identified controls implemented to ensure the test can be performed safely and without incident.

It is anticipated that this will be the final test prior to the certification and implementation of the new system.

9.2 Asset Life Cycle Management

Once the system has been implemented, the asset will need to be managed in accordance with the site asset life cycle management system. Part of this process is intended to include the ongoing maintenance and inspection requirements as has already been identified as part of the FMEA process, where the need for 12 monthly NDT testing will be required to ensure the system remains safe to use.

Section 3 also described how the design process was part of a larger framework called the Product Realisation Process (ed. Davis 1998, p. 122). This process includes all phases of a new product's life cycle from initial consultation through to it's disposal. The formulation of the asset life cycle management plan which details these requirements will form part of the ongoing future work required for the system to remain compliant.

10 CONCLUSION

The project aims and objectives proposed in Section 1.2 have largely been achieved. The primary aim of the project was to design and test a system for the safe replacement of the underground AC units, and although the physical fabrication and testing of the system remains to be completed, the design and behaviour of a conceptual system that meets all relevant Australian Standards and legislation has been the presented.

The benefits of the introduction of the new system are obvious in the fact that a safer method with which the work is carried out is implemented. Real world consequences regarding the safety of workers have been identified, which is the reason for the outsourcing of the fabrication and certification of the system to an experienced design engineer. However a deep understanding of the subject matter has been presented with which the author has been able to collaborate with external service providers for the future production and testing of the proposed system.

The significance of the project has a number of aspects. The first, and arguably most obvious benefit, is the development of a method to carry out a real world task with a reduced risk to the workers involved. The underground coal industry is a heavily legislated sector with potentially severe penalties for those persons whom undertake business and are found to be negligent as a result of an incident. It is for this reason that rigorous engineering methods must be undertaken when designing a system that is intended for use in the coal industry, and as such, those methods should be able to be defended in the event that the system becomes part of an incident investigation.

The methods that have been proposed in this dissertation through the research of appropriate literature and collaboration with industry professionals, are believed to demonstrate the level of rigour which would be expected from a practising professional engineer who is tasked with undertaking a design project. As a result, it is the opinion of the author that the methods employed, especially with regard to engineering software validation techniques and geometry optimisation, should be undertaken and documented with each project similar in nature to the one described.

The research conducted as part of this project has demonstrated the appropriate techniques, design limitations, materials and factors of safety applicable for the design of a system which is required to conform to relevant industry standards. Further learnings in the adequate communication with external stakeholders throughout the process can be employed in future projects to gain a deeper understanding which may help in the realisation of a better product for the end user, whilst improving customer/supplier relationships.

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11.1 Legislation

Work Health and Safety Act 2011 (NSW)

Work Health and Safety Regulation 2017 (NSW)

Work Health and Safety (Mines and Petroleum Sites) Act 2013 (NSW)

Work Health and Safety (Mines and Petroleum Sites) Regulation 2022 (NSW)

12 Appendices

Appendix A - Project Specification, Timeline and RMP

ENG4111/4112 Research Project

Project Specification

For: Michael Moore

Title: The Design of a System for the Removal of Underground Air Conditioning Units

Major: Mechanical Engineering

Supervisors: Dr. Steven Goh

Nathan Reid, Ashton Coal Operations Limited

Sponsor: Ashton Coal Operations Limited

Enrolment: ENG4111 – ONL S1, 2023

ENG4112 – ONL S2, 2023

Project Aim: The design, build and testing of a new system for the safe installation and removal of underground air conditioning units.

Programme: Version 1, 14th February 2023

1. Determine the requirements of the project by examining relevant equipment details and information in relation to the installation and removal of the underground air conditioning (AC) units.
2. Review any existing systems and processes for underground material handling and all relevant Australian Standards and legislation that will apply to the design and testing of the system.
3. Review the limitations of existing onsite machinery regarding maximum operating grade, stability, safe working load and tractive effort.
4. Develop an appropriate design and model using PTC Creo.
5. Identify all loading cases and conduct FEA using PTC Creo.
6. Develop detail drawings which will allow for the manufacture of the system.
7. Engage with suppliers and have the prototype manufactured and certified.
8. Develop and implement a lifecycle management plan for the equipment.

If time and resources permit:

9. Conduct onsite testing of the system against all relevant testing standards.
10. Validate the design by removing and re-installing an AC unit in the underground environment.

The Design and Feasibility of a System for the Removal of Underground Air Conditioning Units

University of Southern Queensland

Project Start: Mon, 20/02/2023

	Semester 1																				Semester 2																			
Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
	20-Feb	27-Feb	6-Mar	13-Mar	20-Mar	27-Mar	3-Apr	10-Apr	17-Apr	24-Apr	1-May	8-May	15-May	22-May	29-May	5-Jun	12-Jun	19-Jun	26-Jun	3-Jul	10-Jul	17-Jul	24-Jul	31-Jul	7-Aug	14-Aug	21-Aug	28-Aug	4-Sep	11-Sep	18-Sep	25-Sep	2-Oct	9-Oct	16-Oct	23-Oct	30-Oct	6-Nov	13-Nov	20-Nov

TASK	START	END	PROGRESS
Information Gathering			
Measure and model relevant underground equipment	20-Feb-23	27-Feb-23	
Gather equipment details (AC unit weight, mounting details)	20-Feb-23	27-Feb-23	
Review Australian Standards and legislation, company standards	20-Feb-23	06-Mar-23	
Literature Review			
Identify relevant literature regarding existing equipment designs	27-Feb-23	13-Mar-23	
Review stress analysis literature	13-Mar-23	27-Mar-23	
Identify FEA techniques applicable to similar equipment	27-Mar-23	10-Apr-23	
Conceptual Design Phase			
Model a number of conceptual designs using CAD software	17-Apr-23	24-Apr-23	
Identify constraints relevant to each concept	24-Apr-23	01-May-23	
Conduct feasibility assessment for each concept	01-May-23	08-May-23	
Detailed Design			
Identify loading cases for preferred design choice	15-May-23	29-May-23	
Model design using CAD and perform FEA	29-May-23	12-Jun-23	
Create detail drawings	19-Jun-23	03-Jul-23	
Production			
Engage suppliers, manufacture prototype	10-Jul-23	28-Aug-23	
Operational Requirements			
Develop life cycle asset management plan	10-Jul-23	14-Aug-23	
Conduct operational risk assessment	10-Jul-23	10-Jul-23	
Develop safe work procedure	17-Jul-23	17-Jul-23	
Onsite Testing Phase			
Conduct on-site testing (machine compatibility, stability etc.)	04-Sep-23	11-Sep-23	
Test prototype by removing/installing AC unit underground	18-Sep-23	25-Sep-23	
Dissertation Preparation			
Prepare draft dissertation	27-Feb-23	30-Oct-23	
Modify and submit dissertation based on supervisor feedback	06-Nov-23	27-Nov-23	

USQ Safety Risk Management System

Safety Risk Management Plan – Offline Version			
Assessment Title:	The Design of a System for the Removal of Underground Air Conditioning Units	Assessment Date:	24/04/2023
Workplace (Division/Faculty/Section):	Faculty of Engineering and Surveying	Review Date:(5 Years Max)	24/04/2028

Context

Description:	
What is the task/event/purchase/project/procedure?	The design and testing of a system for the safe removal of underground air conditioning units
Why is it being conducted?	To adhere to the Work Health and Safety (Mines and Petroleum Sites) Act (2013) and Regulation (2022)
Where is it being conducted?	Student's residence, Ashton Coal Operations Limited, Camberwell, NSW
Course code (if applicable)	ENG4111/4112 — Chemical name (if applicable): N/A

What other nominal conditions?	
Personnel involved	Michael Moore
Equipment	Home PC Skid steer loader attachment (being designed)
Environment	Desktop design phase in residence On-site testing of system at mine site
Other	On-site testing environment will include flat, gravel surface
Briefly explain the procedure/process	Conduct literature review and design of system as proposed using Home PC Conduct on-site testing using skid steer loader and attachment adhering to relevant Australian Standards

Assessment Team - who is conducting the assessment?

Assessor(s)	M Moore
Others consulted:	

Eg 1. Enter Consequence

Eg 2. Enter Probability

Probability	Consequence				
	Insignificant No Injury 0-$5K	**Minor** First Aid $5K-$50K	**Moderate** Med Treatment $50K-$100K	**Major** Serious Injuries $100K-$250K	**Catastrophic** Death More than $250K
Almost Certain 1 in 2	M	H	E	E	E
Likely 1 in 100	M	H	H	E	E
Possible 1 in 1000	L	M	H	H	H
Unlikely 1 in 10 000	L	L	M	M	M
Rare 1 in 1 000 000	L	L	L	L	L

Recommended Action Guide

E =Extreme Risk – Task ***MUST NOT*** proceed

H=High Risk – Special Procedures Required (See USQSafe)

M=Moderate Risk – Risk Management Plan/Work Method Statement Required

L=Low Risk – Use Routine Procedures

Eg 3. Find Action

Step 1	Step 2	Step 2a	Step 2b	Step 3			Step 4				
Hazards: From step 1 or more if identified	***The Risk:*** What can happen if exposed to the hazard without existing controls in place?	***Consequence:*** What is the harm that can be caused by the hazard without existing controls in place?	***Existing Controls:*** What are the existing controls that are already in place?	***Risk Assessment:*** Consequence x Probability = Risk Level			***Additional controls:***	***Risk assessment with additional controls:***			
				Probability	**Risk Level**	**ALARP? Yes/no**		**Consequence**	**Probability**	**Risk Level**	**ALARP? Yes/no**
Design Phase											
• Prolonged seating at PC during design phase	• Muscle soreness • Eye strain	Insignificant	• Regular stretching • Frequent breaks • frequent hydration	Unlikely	Low	Yes		Select a consequence	Select a probability	Select a Risk Level	Yes or No
• Gathering information from AC units	• Cuts/abrasions • Lifting/Slinging • Crush injuries • Equipment Damage	Minor	• PPE • Hand awareness • Trained and appointed operator • Correct and rated lifting equipment • Establish drop-zones when lifting • Barricading/demarcating of work area • Equipment pre-use inspections	Unlikely	Low	Yes		Select a consequence	Select a probability	Select a Risk Level	Yes or No
Testing Phase											
• Unloading attachment for delivery to site	• Equipment Damage • Vehicle interaction • Muscle strain • Back injuries • Slippery surfaces • Cuts/abrasions	Minor	• Trained and appointed operator • Barricading/demarcating of area • Plan travel route • Assess and drive to the conditions • 3-point contact when required • PPE	Unlikely	Low	Yes or No		Select a consequence	Select a probability	Select a Risk Level	Yes or No

Step 1	Step 2	Step 2a	Step 2b	Step 3			Step 4				
			• Check for, and remove any sharp edges if required								
• Testing of attachment using skid steer loader	• Vehicle roll over • Vehicle/personnel interaction • Falling objects	Minor	• Adhere to AS 2359 for testing of stability • Trained and appointed operator • Use of operator restraints as designed • Barricading/demarcating of area to appropriate radius • Secure test load (AC unit) using appropriate tie down equipment • Use slow, deliberate movements of machine functions	Possible	Moderate	No	• Test attachment first using lower dummy mass	Minor	Unlikely	Low	Yes
Operational Usage											
• Removing AC unit underground using attachment	• Vehicle roll over • Vehicle/personnel interaction • Explosive atmosphere • Equipment Damage • Electrical Energy • Hand Injuries	Minor	• Trained and appointed operator • Use of operator restraints as designed • Barricading/demarcating of area to appropriate radius • Secure load (AC unit) using appropriate tie down equipment • Use slow, deliberate movements of machine functions • Vehicle mechanical, electrical and operational pre-use inspections	Unlikely	Low	Yes		Select a consequence	Select a probability	Select a Risk Level	Yes or No

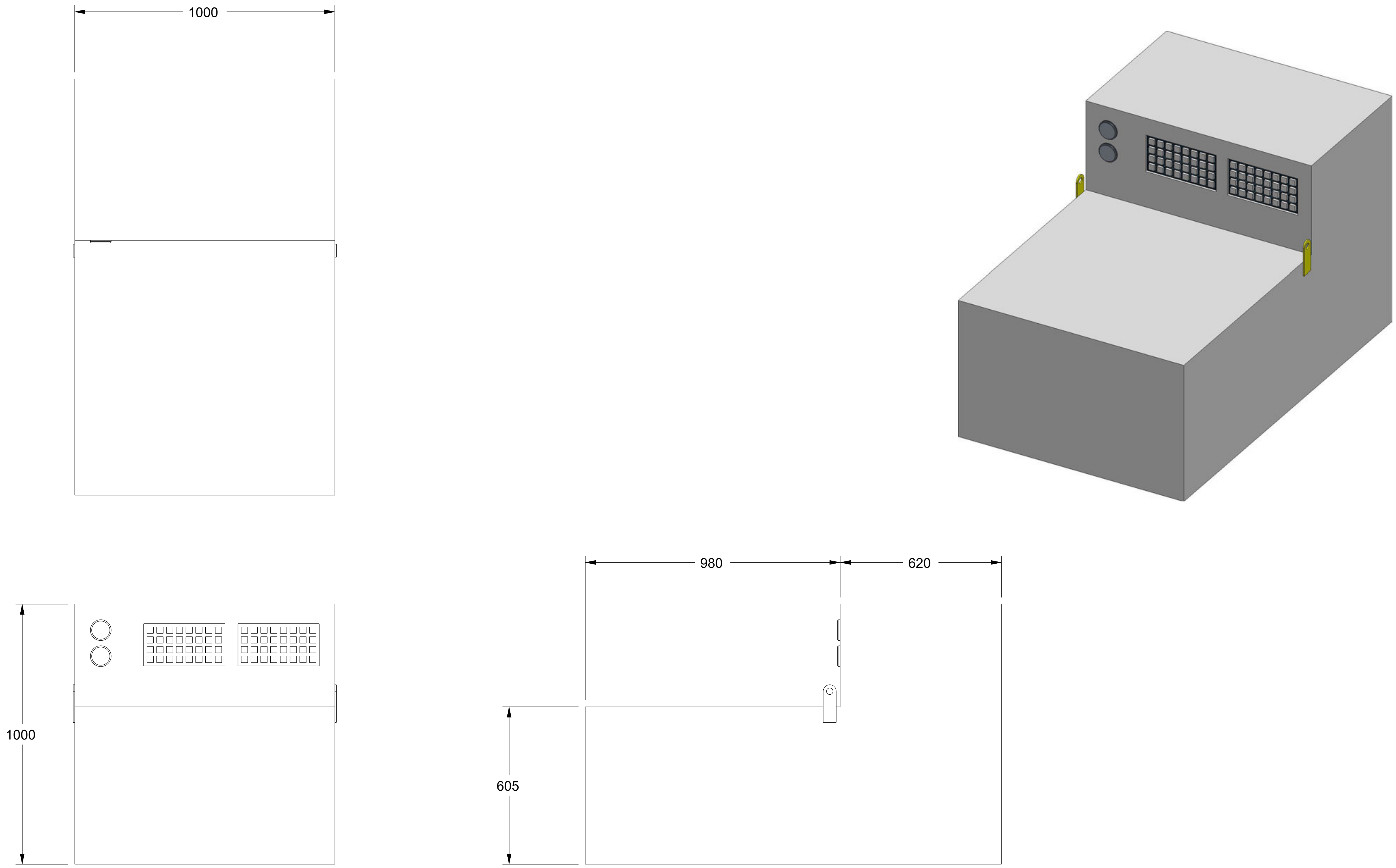
Step 1	Step 2	Step 2a	Step 2b	Step 3			Step 4				
			• Correct equipment isolation using site procedures • Positive communication when manoeuvring AC unit • Use of correct tooling and packing when positioning AC unit								
		Select a consequence		Select a probability	Select a Risk Level	Yes or No		Select a consequence	Select a probability	Select a Risk Level	Yes or No

Step 5 - Action Plan (for controls not already in place)			
Additional controls:	*Resources:*	*Persons responsible:*	*Proposed implementation date:*
• Use of lower mass item for initial stability testing	• Any item with known mass lower than 300 kg	• M Moore	• When attachment is delivered to site
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.
			Click here to enter a date.

Step 6 - Approval			
Drafter's name:	Michael Moore	Draft date:	24/04/2023
Drafter's comments:			
Approver's name:		Approver's title/position:	
Approver's comments:			
I am satisfied that the risks are as low as reasonably practicable and that the resources required will be provided.			
Approver's signature:		Approval date:	Click here to enter a date.

Appendix B - AC Unit General Assembly Drawing

REF	REVISION	BY	CKD	DATE



ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED
ALL PLATE AS PER AS/NZS 3678:2016 - 250
ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300
ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

ALL EDGES TO BE DEBURRED

MAT'L:

MASS: 320 kg

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE AC UNIT GENERAL ARRANGEMENT

DRAWN: M MOORE

DATE: 12/8/2023

SCALE 1:16

DRG.NO AC-001

A3

Appendix C - Derivations of Equations Describing the Vertical Displacement of the Primary Support Members

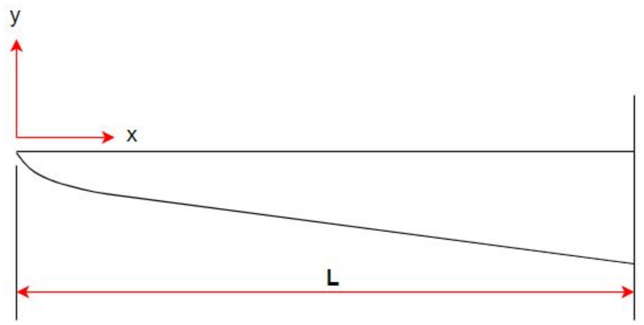


Figure 12.1: Diagram depicting primary support member for displacement equation.

Equation for the Elastic Curve:

$$\frac{d^2y}{dx^2} = \frac{M(x)}{EI}$$

Where:-

$$\begin{aligned}
M(x) &= \frac{P}{n}x \\
I_{xx} &= \frac{1}{12}bh^3 \\
h &= \left(\frac{6Px}{nb\sigma_{all}}\right)^{1/2} \\
\therefore I_{xx} &= \frac{1}{12}b\left[\left(\frac{6Px}{nb\sigma_{all}}\right)^{1/2}\right]^3 \\
&= \frac{1}{12}b\left(\frac{6Px}{nb\sigma_{all}}\right)^{3/2} \\
&= \frac{b}{12}\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}x^{3/2}
\end{aligned}$$

$$
\begin{aligned}
\frac{M(x)}{I} &= \frac{\frac{P}{n}x}{\frac{b}{12}\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2} x^{3/2}} \\
&= \frac{12\frac{P}{n}}{b\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}} x^{-1/2}
\end{aligned}
$$

Substituting into the equation for the elastic curve:

$$
\begin{aligned}
E\frac{d^2y}{dx^2} &= \frac{M(x)}{I} \\
E\int \frac{d^2y}{dx^2}dx &= \frac{12\frac{P}{n}}{b\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}} \int x^{-1/2}dx \\
E\frac{dy}{dx} &= \frac{12\frac{P}{n}}{b\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}} \left(2x^{1/2} + C_1\right)
\end{aligned}
$$

Let:- $x = L$, $\frac{dy}{dx} = 0$

$$
\begin{aligned}
\therefore 0 &= 2L^{1/2} + C_1 \\
C_1 &= -2L^{1/2}
\end{aligned}
$$

Integrating again:

$$
\begin{aligned}
E\int \frac{dy}{dx}dx &= \frac{12\frac{P}{n}}{b\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}} \int \left(2x^{1/2} - 2L^{1/2}\right) dx \\
Ey &= \frac{12\frac{P}{n}}{b\left(\frac{6P}{nb\sigma_{all}}\right)^{3/2}} \left(\frac{4}{3}x^{3/2} - 2L^{1/2}x + C_2\right)
\end{aligned}
$$

Let:- $x = L$, $y = 0$

$$
\begin{aligned}
\therefore 0 &= \frac{4}{3}L^{3/2} - 2L^{1/2}L + C_2 \\
C_2 &= -\frac{4}{3}L^{3/2} + 2L^{3/2} \\
&= \frac{2}{3}L^{3/2}
\end{aligned}
$$

$$
\therefore y = \left[\frac{12\frac{P}{n}}{Eb\left(\frac{6P}{nb\sigma_{all}}\right)^{\frac{3}{2}}} \right] \left(\frac{4}{3}x^{3/2} - 2L^{1/2}x + \frac{2}{3}L^{3/2} \right)
$$

Appendix D - Failure Modes and Effects Analysis (FMEA)

Component Potential Failure Mode/Hazard	Exisiting Controls	F	S	C	Additional Controls
1. Main Support Plate					
1.1 - Excessive von Mises stresses experienced which causes plastic failure of Main Support Plate	1.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	- Training package to include SWL of implement. - Compliance plate attached clearly stating SWL.
	1.1.2 - Material chosen conforms to AS/NZS 3678:2016.				
1.2 - Member undergoes excessive amount of deflection.	1.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
1.3 - Tie-down holes fail plastically	1.3.1 - Hole positions conform to Section 9.6.2 of AS 3990:1993	D	3	14	Nil - Existing controls considered adequate.
2. Primary Support Members					
2.1 - Excessive von Mises stresses experienced which causes plastic failure of Primary Support Members	2.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	2.1.2 - Design of members as non-prismatic beams of constant strength based off of recognised theoretical analysis.				
	2.1.3 - FEA study shows complete agreement with theoretical analysis.				

	2.1.4 - Material chosen conforms to AS/NZS 3678:2016.				
2.2 - Members undergo excessive amount of deflection.	2.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
2.3 - Primary Support Members undergo localised buckling	2.3.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	D	3	14	Nil - Existing controls considered adequate.
	2.3.2 - Members fully welded along lengths conforming to AS/NZS 1554.1:2014				
3. Vertical Side Plates					
3.1 - Excessive von Mises stress due to applied loads causes plastic failure	3.1.1 - Re-inforcing plates added to vertical side plates to increase section modulus of component.	C	3	13	Nil - Existing controls considered adequate.
	3.1.2 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.				
	3.1.3 - Material chosen conforms to AS/NZS 3678:2016.				
3.2 - Member undergoes excessive amount of deflection.	3.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.

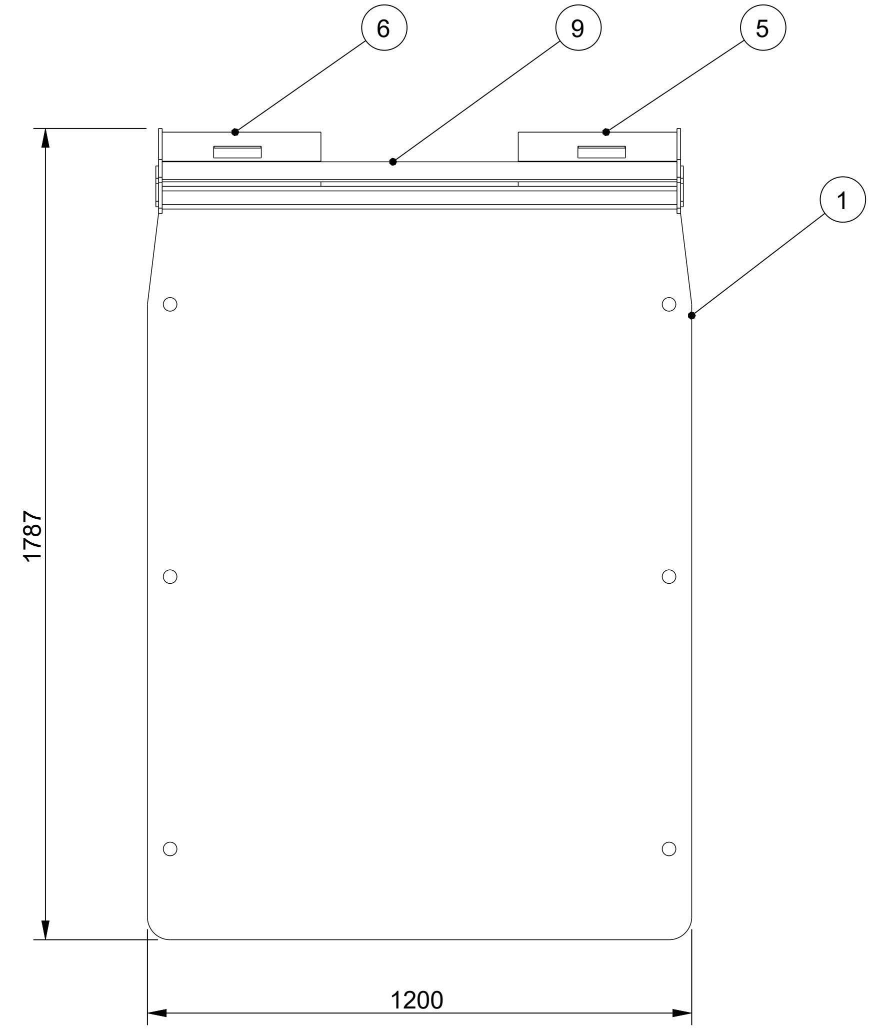
4. Top SHS Member					
4.1 - Excessive von Mises stress due to applied loads causes plastic failure	4.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	4.1.2 - Material chosen conforms to AS/NZS 1163:2016.				
4.2 - Member undergoes excessive amount of deflection.	4.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
5. Bottom SHS Member					
5.1 - Excessive von Mises stress due to applied loads causes plastic failure	5.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	5.1.2 - Material chosen conforms to AS/NZS 1163:2016.				
5.2 - Member undergoes excessive amount of deflection.	5.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
6. Backing Plates					
6.1 - Excessive von Mises stress due to applied loads causes plastic failure	6.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	6.1.2 - Material chosen conforms to AS/NZS 3678:2016.				
6.2 - Member undergoes excessive amount of deflection.	6.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.

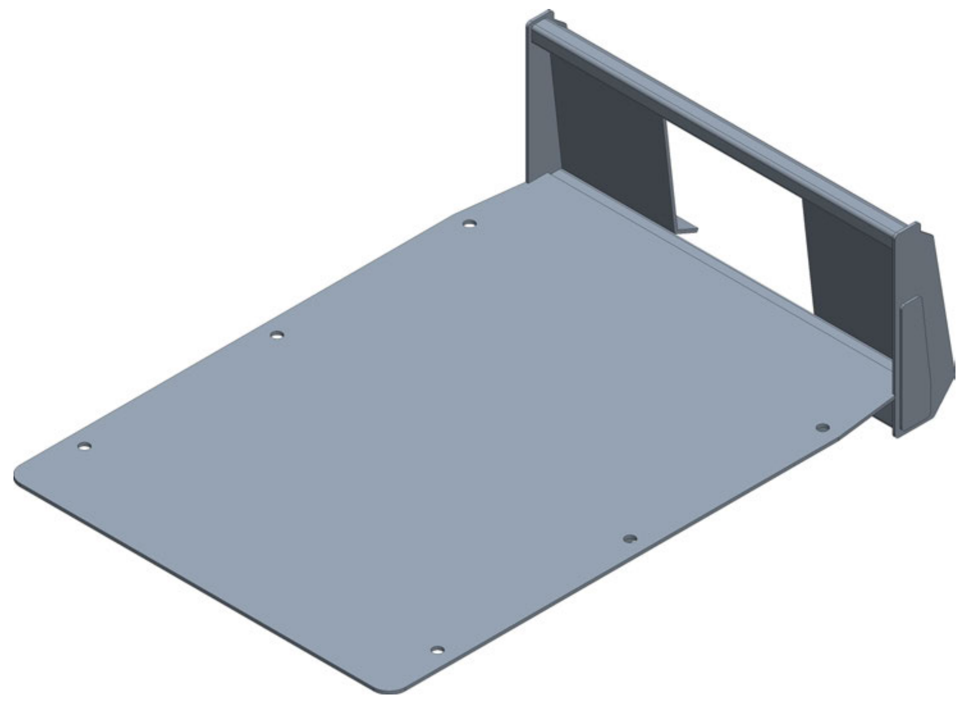
7. Side Reinforcing Plates					
7.1 - Excessive von Mises stress due to applied loads causes plastic failure	7.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	7.1.2 - Material chosen conforms to AS/NZS 3678:2016.				
7.2 - Member undergoes excessive amount of deflection.	7.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
8. Centre Support Tabs					
8.1 - Excessive von Mises stress due to applied loads causes plastic failure	8.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	8.1.2 - Material chosen conforms to AS/NZS 3678:2016.				
8.2 - Member undergoes excessive amount of deflection.	8.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
9. Top Load Plate					
9.1 - Excessive von Mises stress due to applied loads causes plastic failure	9.1.1 - FEA study conducted which verifies component is able to withstand applied loads as per AS 3990:1993.	C	3	13	Nil - Existing controls considered adequate.
	9.1.2 - Material chosen conforms to AS/NZS 3679.1:2016.				

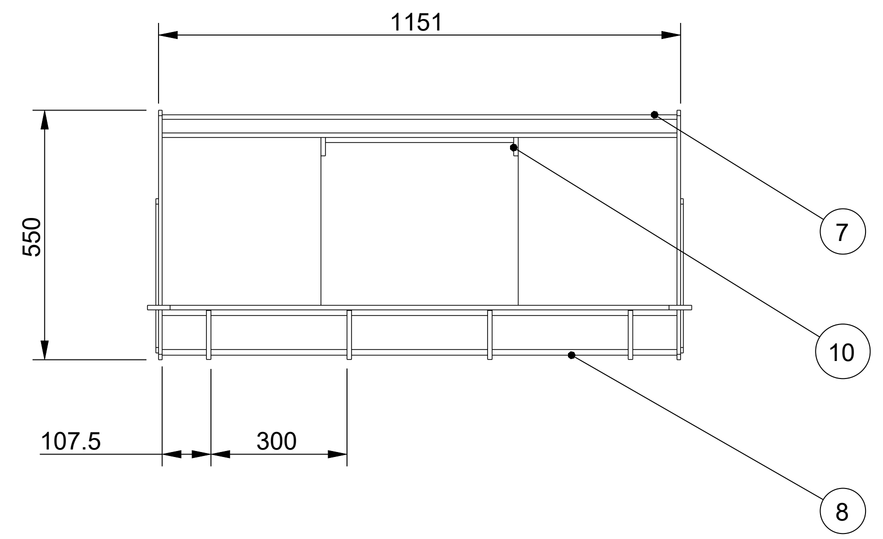
9.2 - Member undergoes excessive amount of deflection.	9.2.1 - Maximum deflection verified by FEA study which conforms to AS 4100:2020.	C	3	13	Nil - Existing controls considered adequate.
10. Attachment System					
10.1 - Coupling mechanism on skid steer detaches from implement attachment system.	10.1.1 - Attachment geometry conforms to AS ISO 24410:2021.	E	2	10	- Review training package undertaken by operators to operate skid ster loader attachments.
	10.1.2 - Load cases based on application of working force description given by AS ISO 24410:2021.				
11. Welding					
11.1 - Failure of welds due to undetected cracking or fatigue.	11.1.1 - All welding of components conforms to AS/NZS 1554.1:2014	C	2	8	- 12 monthly NDT inspection by accredited professional.
	11.1.2 - Attachment system does not undergo heavy cyclic loading and is not intended for highly frequent use.				
12. General					
12.1 - Failure of any component due to corrosion.	12.1.1 - Implement to conform to AS 2312.1:2014.	D	3	14	Nil - Existing controls considered adequate.
12.2 - Impact damage from other machinery	12.2.1 - Reflective stripes to be to appropriate areas on implement.	C	2	8	Nil - Existing controls considered adequate.
12.3 - Instability caused by the mass of the implement and AC unit.	12.3.1 - Mass of system designed with consideration of the rated load of the skid steer loader.	C	2	8	- Confirm mass by weighing implement. - Conduct stability test as per AS ISO 22915:2018 and AS 2359.1:2019.

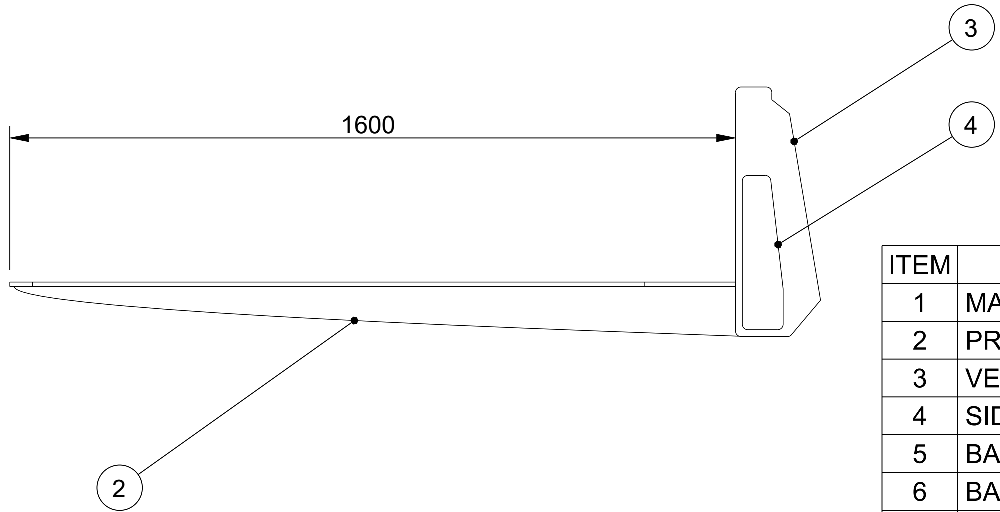
Appendix E - Working Drawings

REF	REVISION	BY	CKD	DATE









ITEM	DESCRIPTION	MATERIAL	QTY	DRAWING NO.
1	MAIN SUPPORT PLATE	AS/NZS 3678-350	1	SSA-PSC-01
2	PRIMARY SUPPORT MEMBER	AS/NZS 3678-350	4	SSA-PSC-01
3	VERTICAL SIDE PLATE	AS/NZS 3678-350	2	SSA-AC-01
4	SIDE REINFORCING PLATE	AS/NZS 3678-350	2	SSA-AC-01
5	BACKING PLATE - LH	AS/NZS 3678-350	1	SSA-BPLH-01
6	BACKING PLATE - RH	AS/NZS 3678-350	1	SSA-BPRH-01
7	UPPER SHS MEMBER	AS/NZS 1163 C350	1	SSA-AC-01
8	LOWER SHS MEMBER	AS/NZS 1163-C350	1	SSA-AC-01
9	TOP LOAD PLATE	AS/NZS 3679.1-300	1	SSA-AC-01
10	CENTRE SUPPORT TABS	AS/NZS 3678-350	2	SSA-AC-01

UniSQ

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED

TOLERANCES:
±0.5 mm UNLESS OTHERWISE STATED
±1° UNLESS OTHERWISE STATED

ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED

ALL PLATE AS PER AS/NZS 3678:2016 - 350

ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300

ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

ALL EDGES TO BE DEBURRED

AS1100

MAT'L: AS SHOWN

FINISH: OR AS SHOWN 12.5

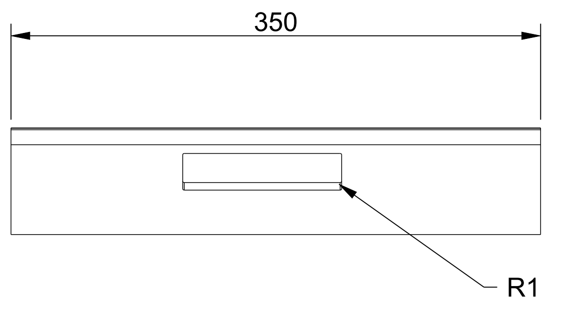
MASS: 259 kg

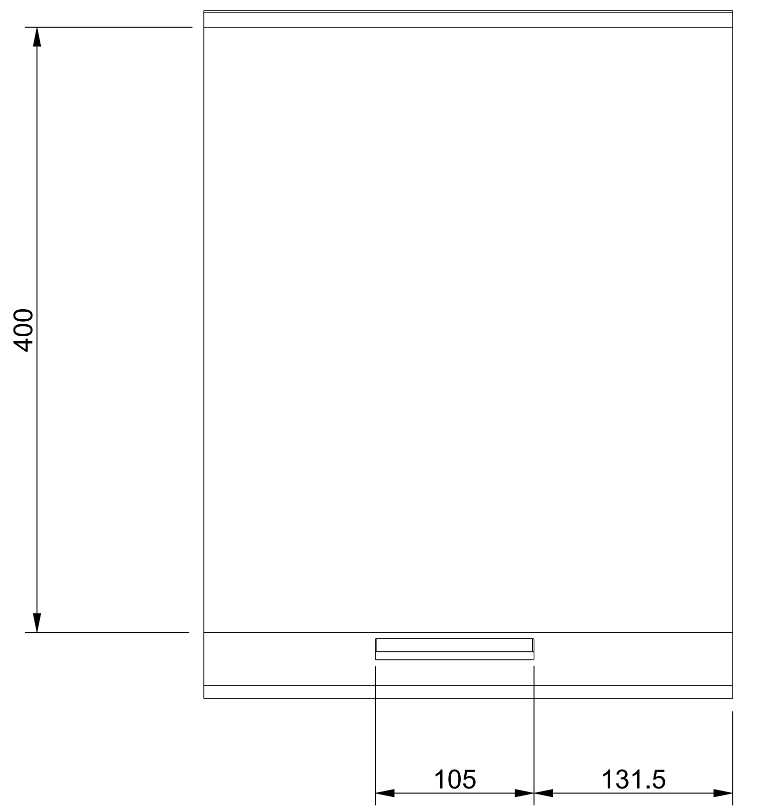
UNIVERSITY OF SOUTHERN QUEENSLAND

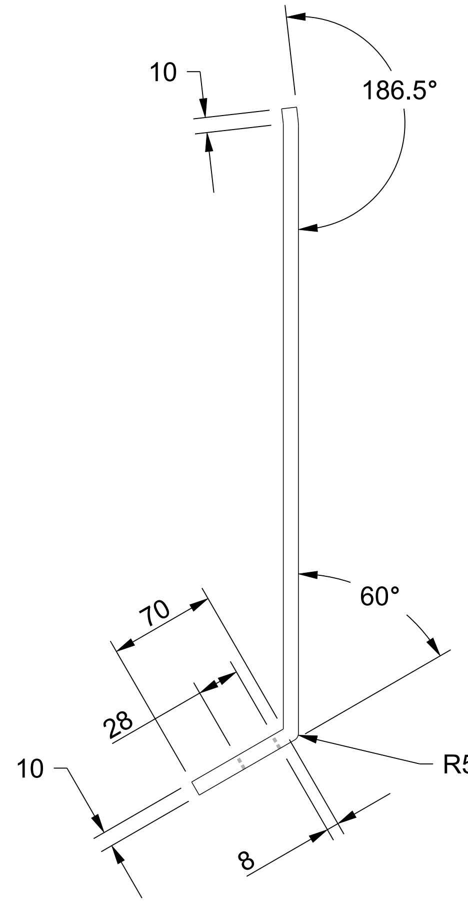
TITLE
SKID STEER LOADER IMPLEMENT

DRAWN: M MOORE	SCALE 1:24	DRG.NO SSA-0001	A3
DATE: 1/7/2023	SHEET: 1 of 6		

REF	REVISION	BY	CKD	DATE







UniSQ

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED

TOLERANCES:
±0.5 mm UNLESS OTHERWISE STATED
±1° UNLESS OTHERWISE STATED

ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED

ALL PLATE AS PER AS/NZS 3678:2016 - 350

ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300

ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

ALL EDGES TO BE DEBURRED

AS1100

MAT'L: AS/NZS 3678:2016-350

FINISH: OR AS SHOWN 12.5

MASS: 12.8 kg

REF ASSY No. SSA-0001

DRAWN: M MOORE

DATE: 1/7/2023

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE BACKING PLATE - LH

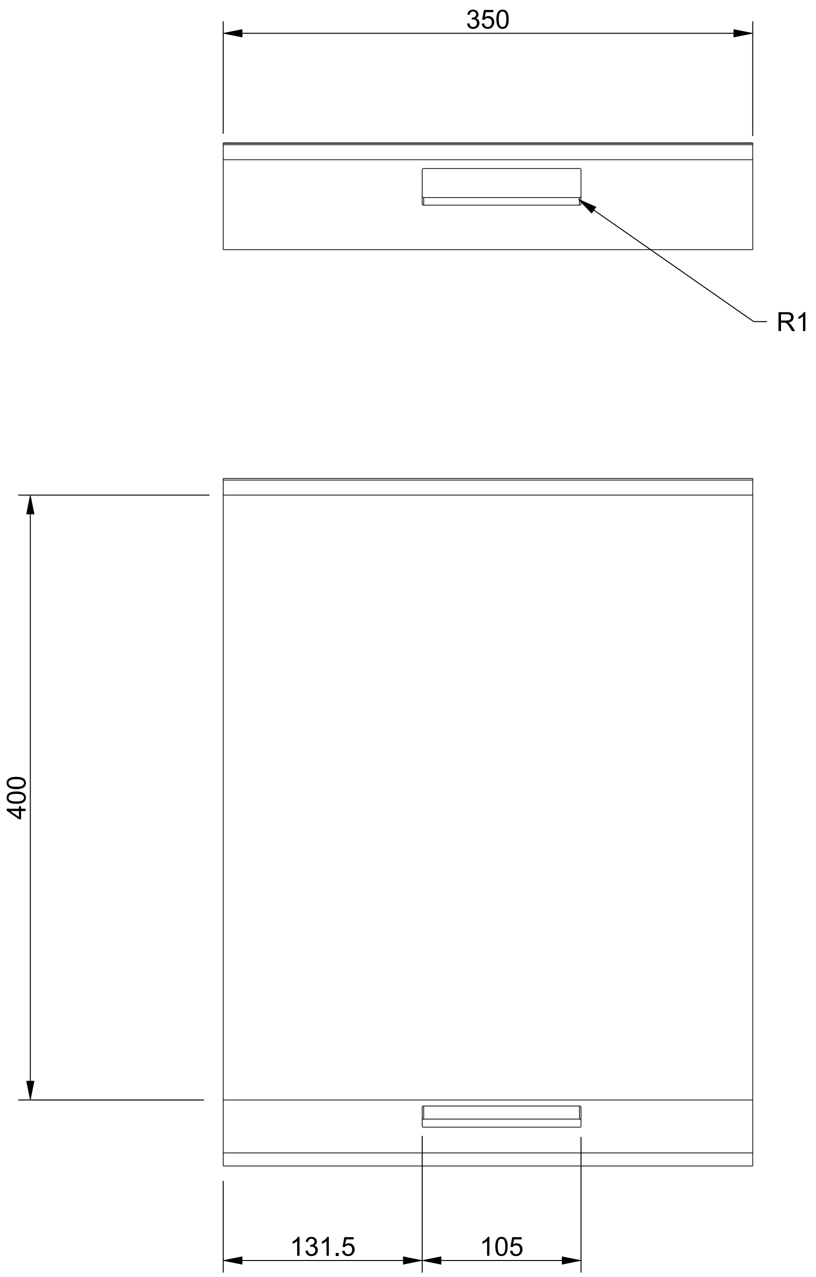
SCALE 1:5

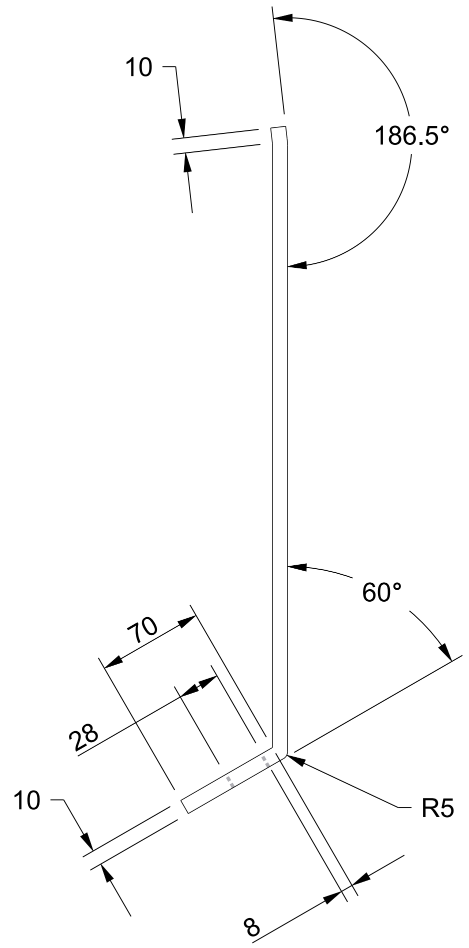
SHEET: 2 of 6

DRG.NO SSA-BPLH-01

A3

REF	REVISION	BY	CKD	DATE





UniSQ

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED

TOLERANCES:
±0.5 mm UNLESS OTHERWISE STATED
±1° UNLESS OTHERWISE STATED

ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED

ALL PLATE AS PER AS/NZS 3678:2016 - 350

ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300

ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

ALL EDGES TO BE DEBURRED

AS1100

MAT'L: AS/NZS 3678:2016-350

FINISH: OR AS SHOWN 12.5

MASS: 12.8 kg

REF ASSY No. SSA-0001

DRAWN: M MOORE

DATE: 1/7/2023

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE

BACKING PLATE - RH

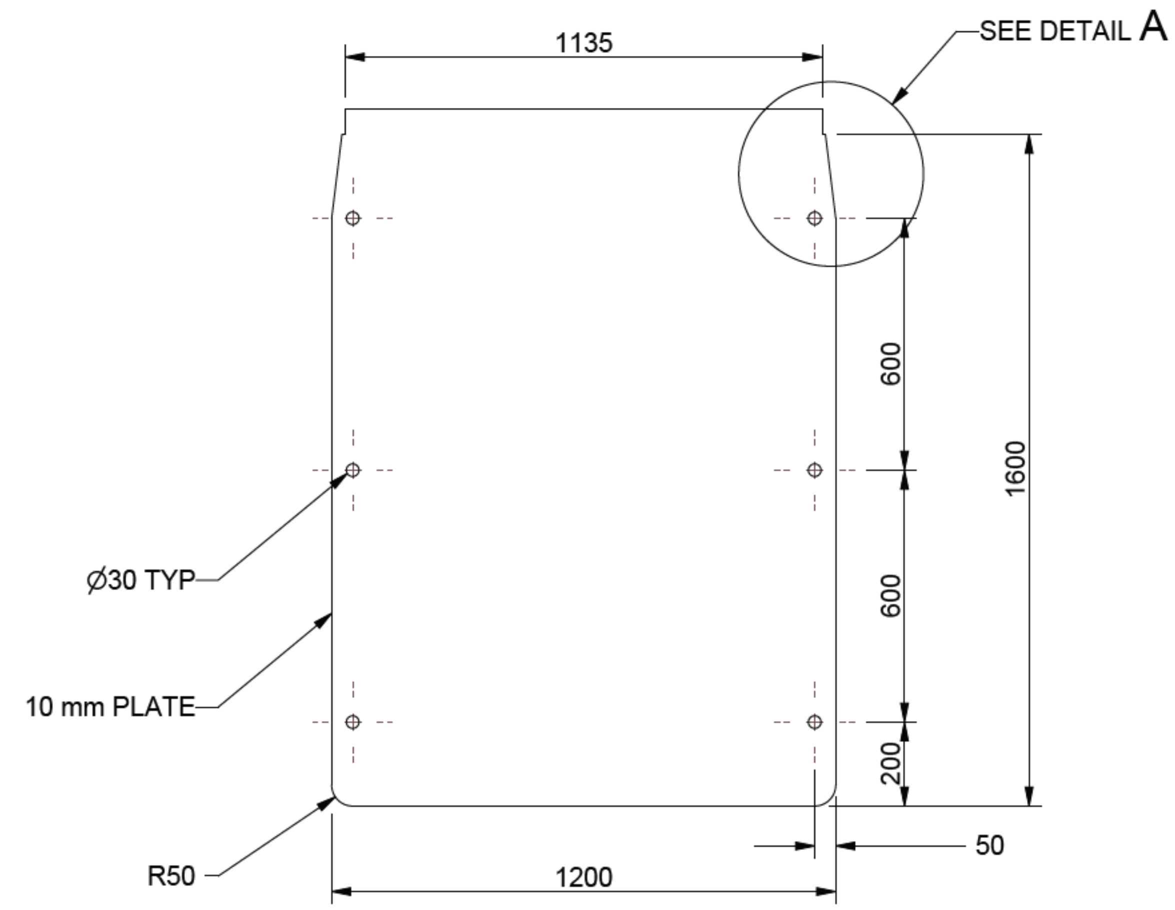
SCALE 1:5

SHEET: 3 of 6

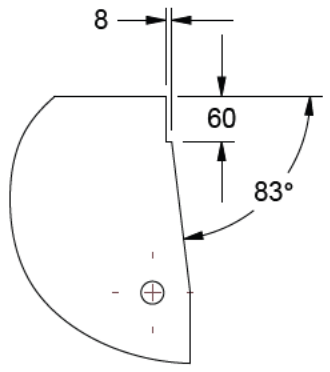
DRG.NO SSA-BPRH-01

A3

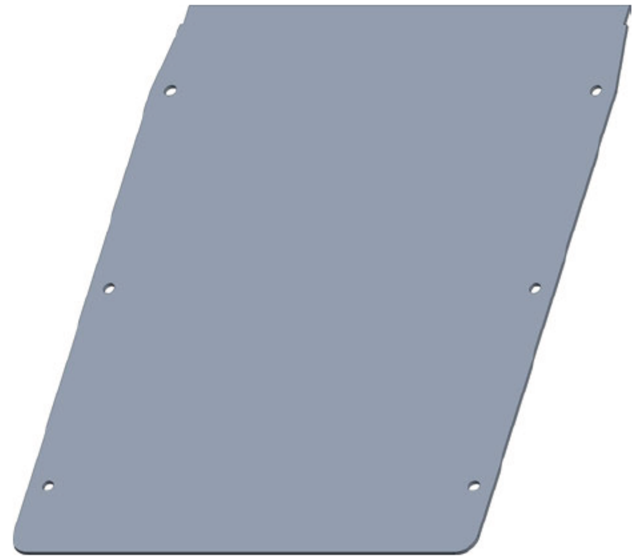
REF	REVISION	BY	CKD	DATE



SCALE 1:18

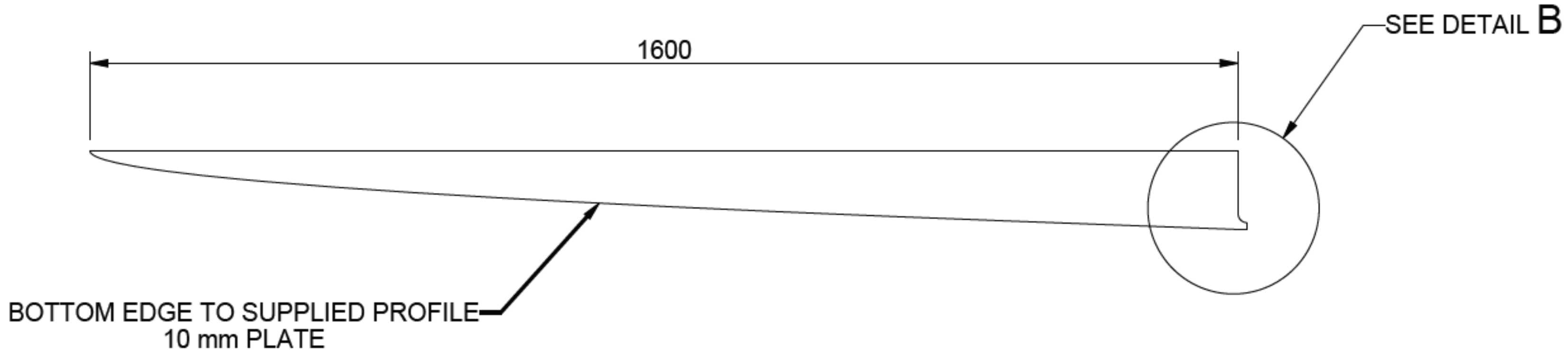


DETAIL A
SCALE 1:10



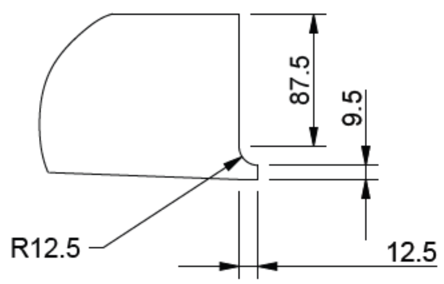
MAIN SUPPORT PLATE

(154.8 kg)

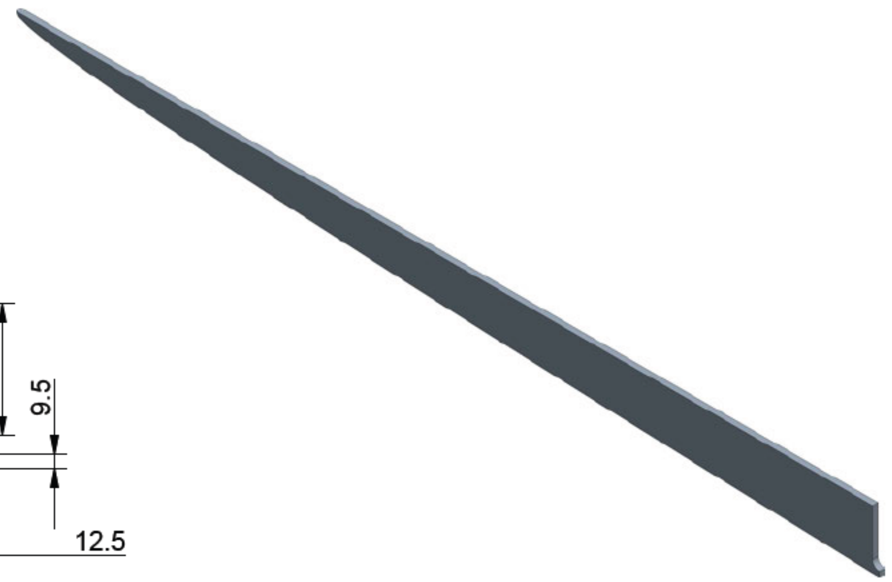


PRIMARY SUPPORT MEMBER (4 off)

(9.1 kg each)



DETAIL B
SCALE 1:5



UniSQ

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED
TOLERANCES:
±0.5 mm UNLESS OTHERWISE STATED
±1° UNLESS OTHERWISE STATED

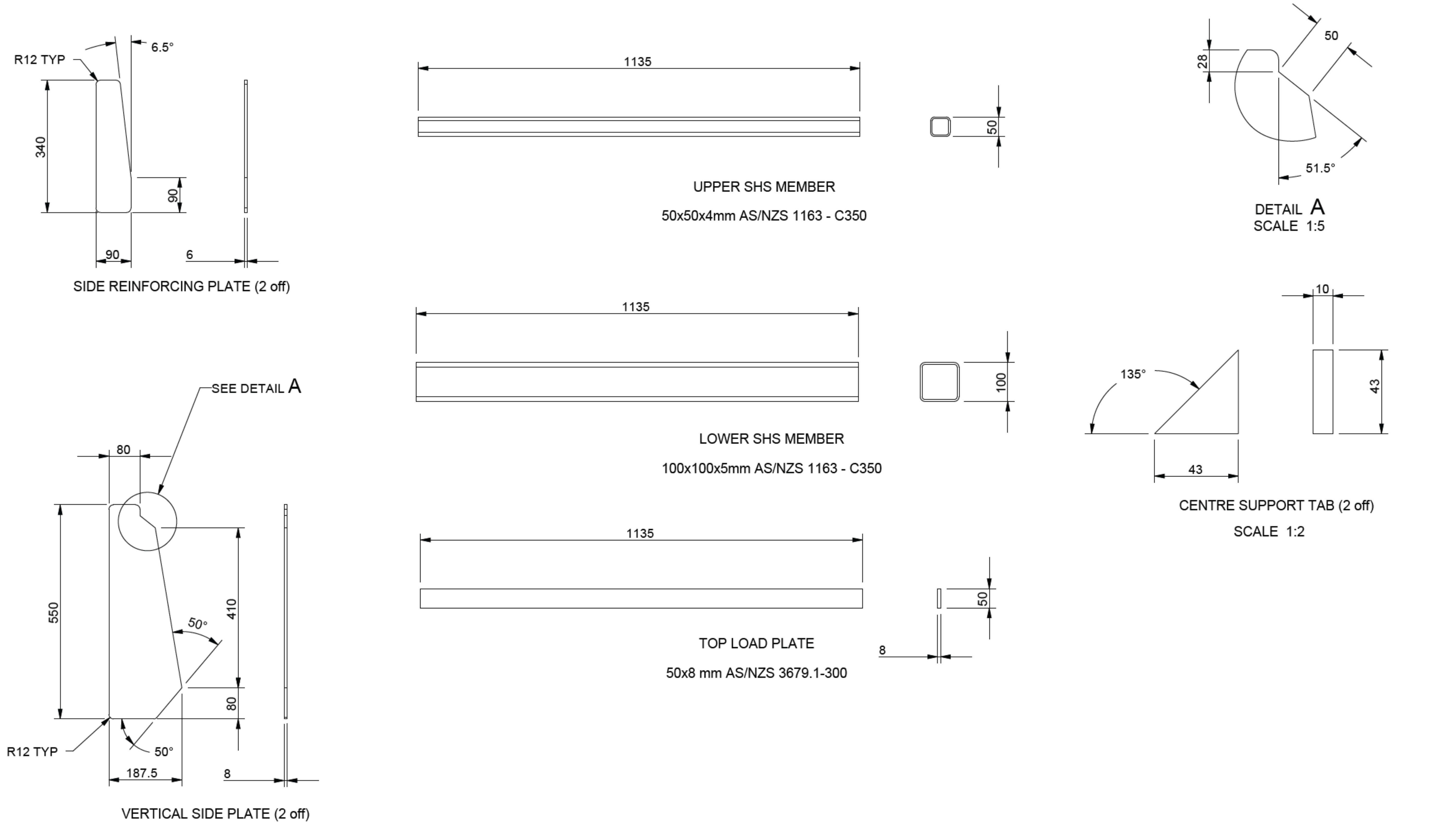
ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED
ALL PLATE AS PER AS/NZS 3678:2016 - 350
ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300
ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

ALL EDGES TO BE DEBURRED
AS1100
MAT'L: AS/NZS 3678:2016-350
FINISH: OR AS SHOWN 12.5
MASS: AS SHOWN

UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
PRIMARY SUPPORT COMPONENTS

DRAWN: M MOORE
DATE: 1/7/2023
SCALE 1:10 OR AS SHOWN
SHEET: 4 of 6
DRG.NO SSA-PSC-01
A3

REF	REVISION	BY	CKD	DATE



UniSQ

ALL DIMENSIONS IN mm
UNLESS OTHERWISE NOTED

TOLERANCES:
±0.5 mm UNLESS OTHERWISE NOTED
±1° UNLESS OTHERWISE NOTED

ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED
ALL PLATE AS PER AS/NZS 3678:2016 - 350
ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300
ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - C350

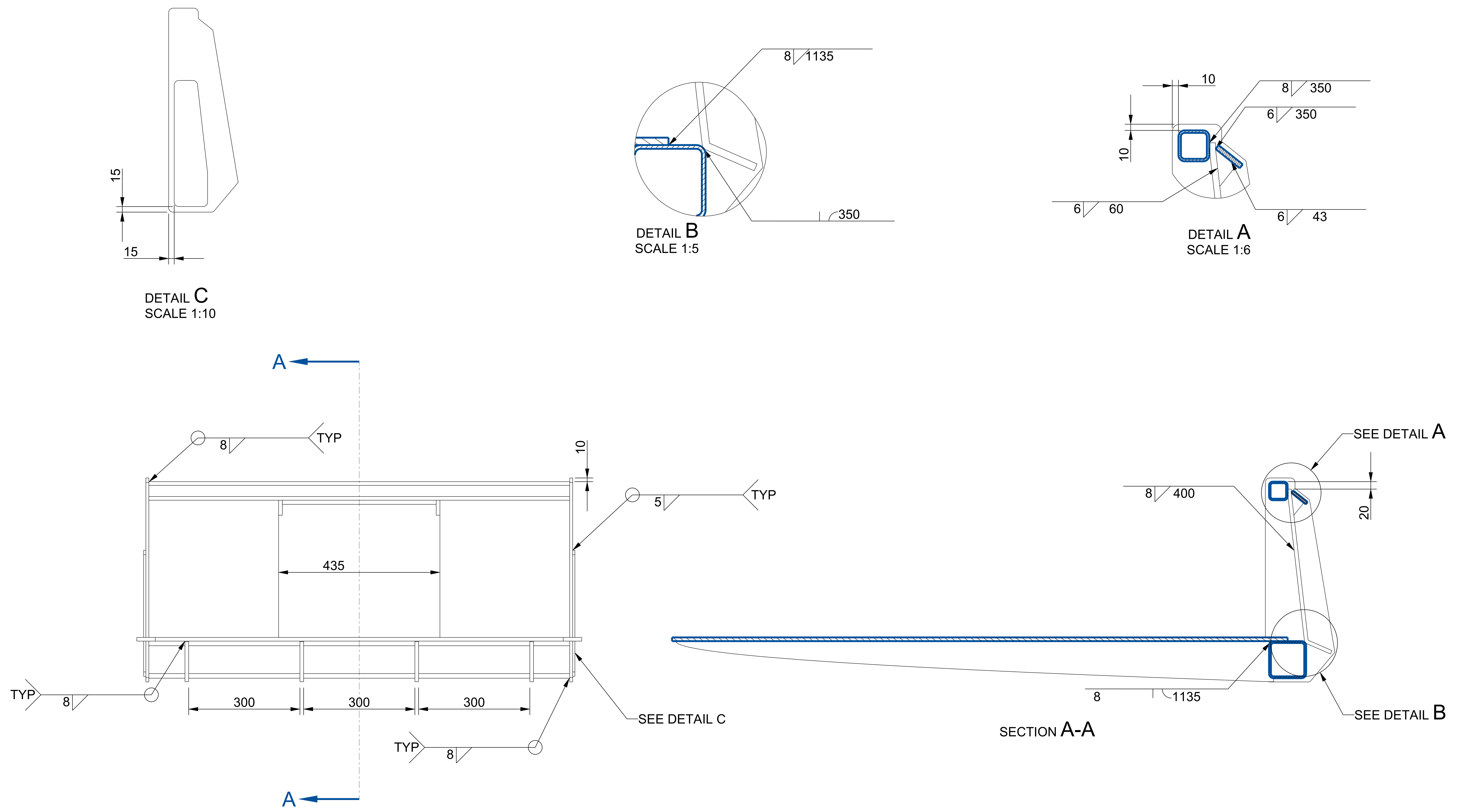
ALL EDGES TO BE DEBURRED
MAT'L:
MASS:

AS1100
FINISH: OR AS SHOWN 12.5 ✓

REF ASSY No. SSA-0001
DRAWN: M MOORE
DATE: 2/7/2023

UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
ATTACHMENT COMPONENTS
SCALE 1:10 OR AS SHOWN
SHEET: 5 of 6
DRG.NO SSA-AC-01
A3

REF	REVISION	BY	CKD	DATE



ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED
TOLERANCES: ±0.5 mm UNLESS OTHERWISE STATED ±1° UNLESS OTHERWISE STATED

ALL WELDS 8mm CONTINUOUS FILLET TO AS1554 UNLESS OTHERWISE NOTED
ALL PLATE AS PER AS/NZS 3678:2016 - 350
ALL BARS AND SECTIONS AS PER AS/NZS 3679.1:2016 - 300
ALL HOLLOW SECTIONS AS PER AS/NZS 1163:2016 - 350

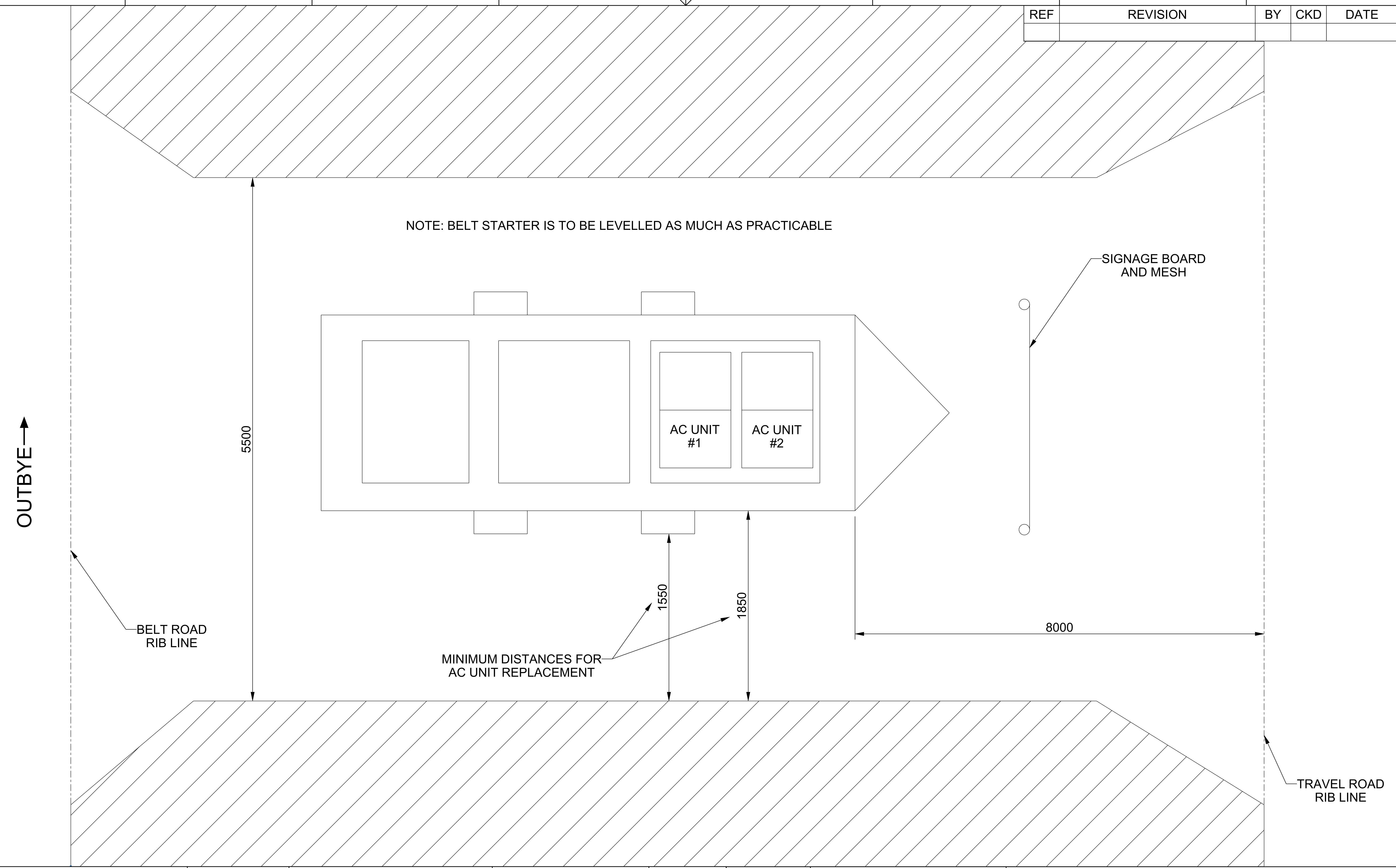
ALL EDGES TO BE DEBURRED
AS1100
FINISH: OR AS SHOWN 12.5
MAT'L:
MASS: AS SHOWN

REF ASSY No. SSA-0001
DRAWN: M MOORE
DATE: 2/7/2023

UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
SKID STEER ATTACHMENT WELDMENT DRAWING
SCALE 1:16 OR AS SHOWN
SHEET: 6 of 6
DRG.NO SSA-WELD-01
A3

Appendix F - Belt Starter Cut-Through Arrangement

REF	REVISION	BY	CKD	DATE



ALL EDGES TO BE DEBURRED

AS1100

MAT'L:

MASS:

ALL DIMENSIONS IN mm UNLESS OTHERWISE NOTED

REF ASSY No.

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE

BELT STARTER C/T ARRANGEMENT

DRAWN: M MOORE

DATE: 9/7/2023

SCALE NOT TO SCALE

SHEET: 1 of 1

DRG.NO YAN-BSCT-A1

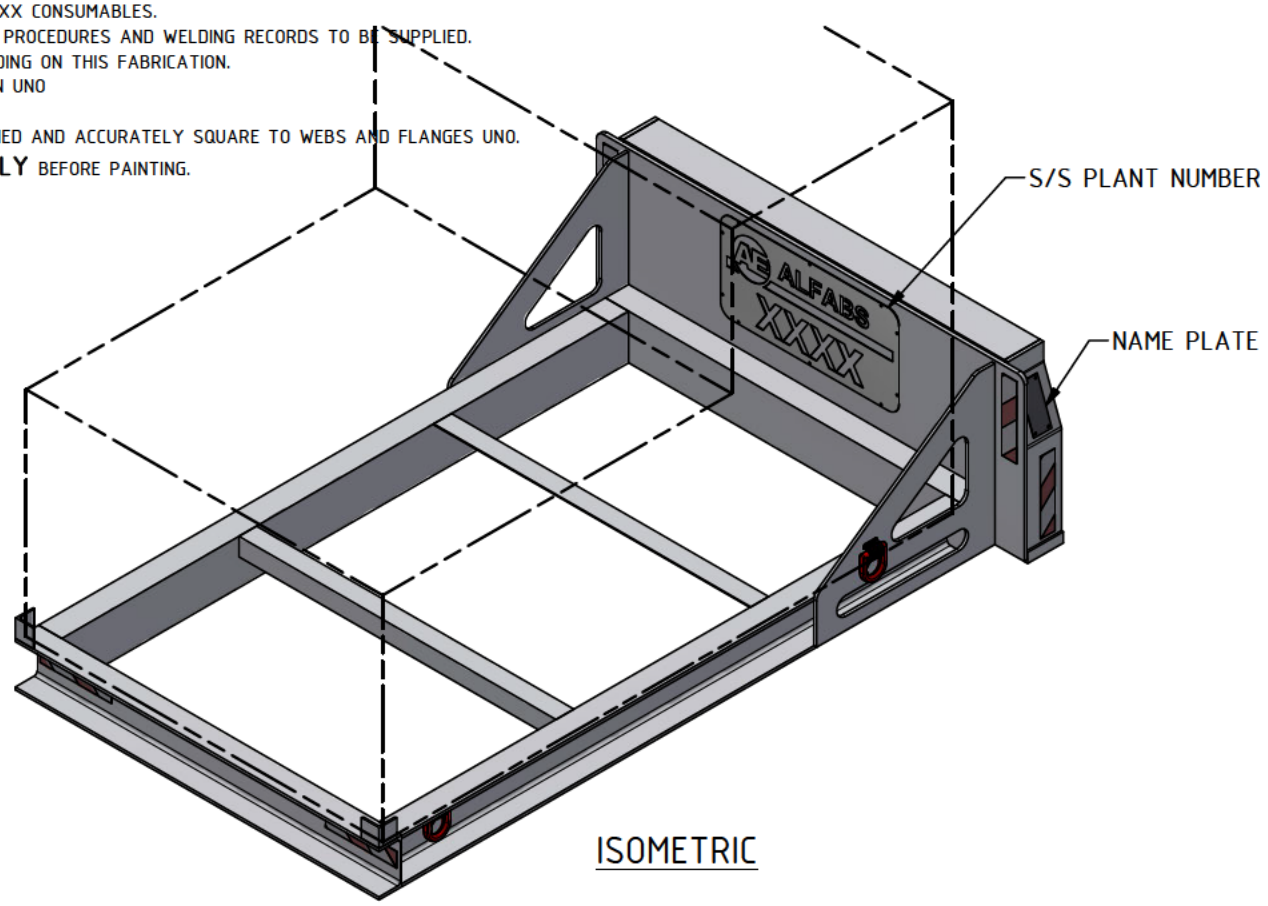
A3

Appendix G - External Concept General Arrangement

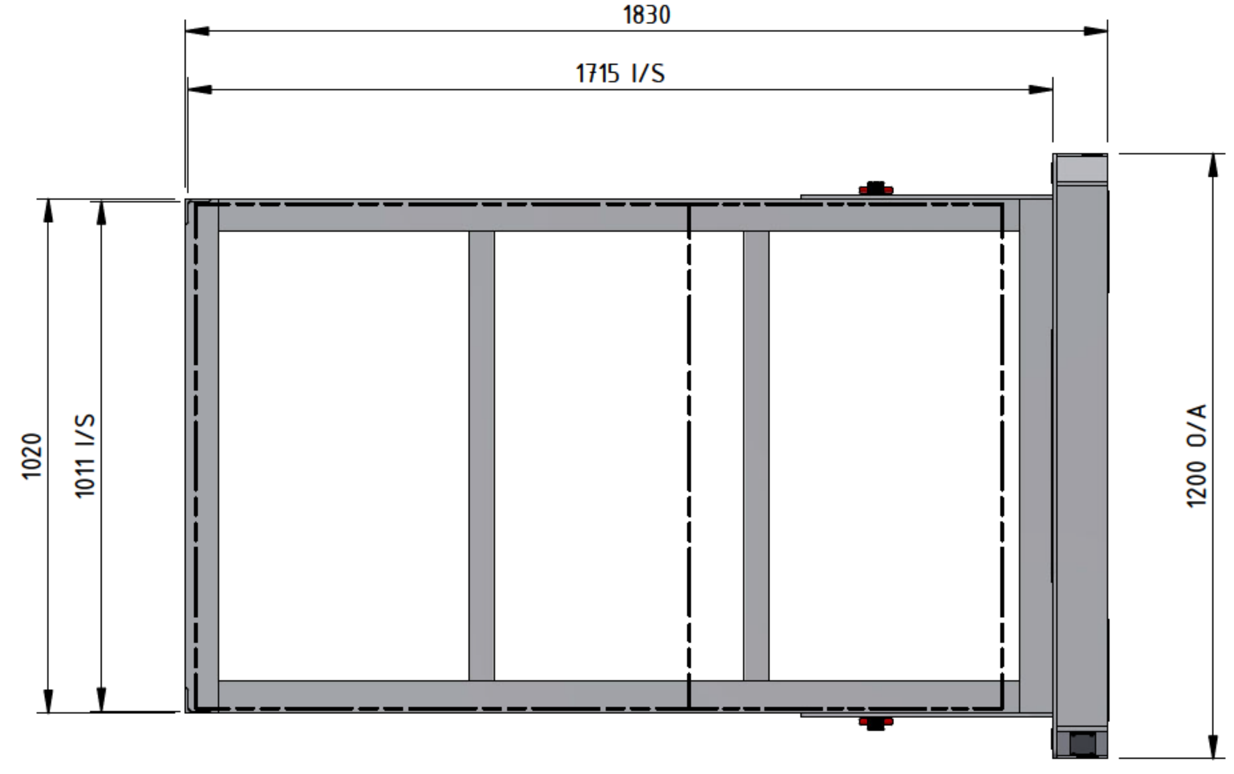
(Klinkhamer, B 2023, pers. comm., 18 July)

FABRICATION NOTES:

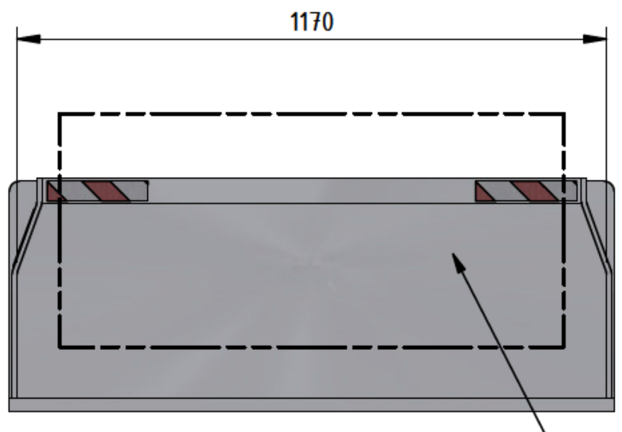
- MATERIAL TO BE FREE FROM DEFECTS. MATERIAL CERTIFICATES REQUIRED.
- PROFILED EDGES TO BE CLEAN AND UNIFORM BEFORE WELDING.
- WELDING TO AS1554.1 SP, USING E41XX OR E48XX CONSUMABLES.
- ONLY PREQUALIFIED WELDS TO BE USED, WELD PROCEDURES AND WELDING RECORDS TO BE SUPPLIED.
- ONLY QUALIFIED WELDERS TO UNDERTAKE WELDING ON THIS FABRICATION.
- ALL BUTT WELDS SHALL BE FULL PENETRATION UNO
- ALL FILLET WELDS SHALL BE CONTINUOUS UNO
- ALL STIFFENERS TO BE ACCURATELY POSITIONED AND ACCURATELY SQUARE TO WEBS AND FLANGES UNO.
- ENSURE THAT MATING PARTS MATE **PROPERLY** BEFORE PAINTING.
- IF IN DOUBT, ASK!



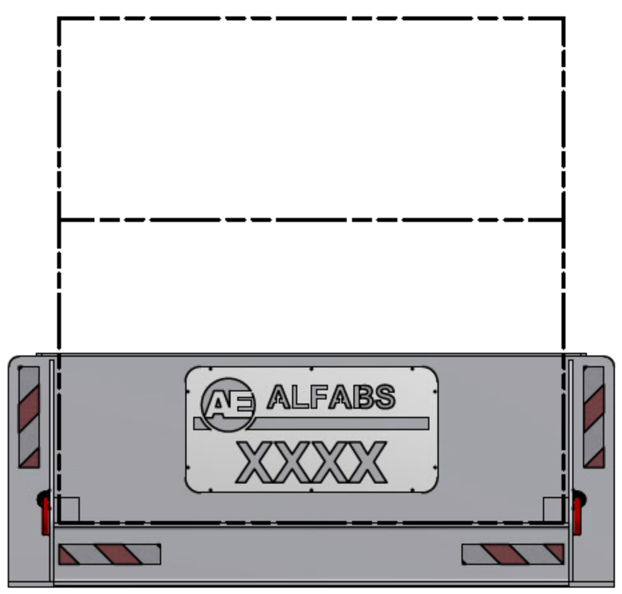
ISOMETRIC



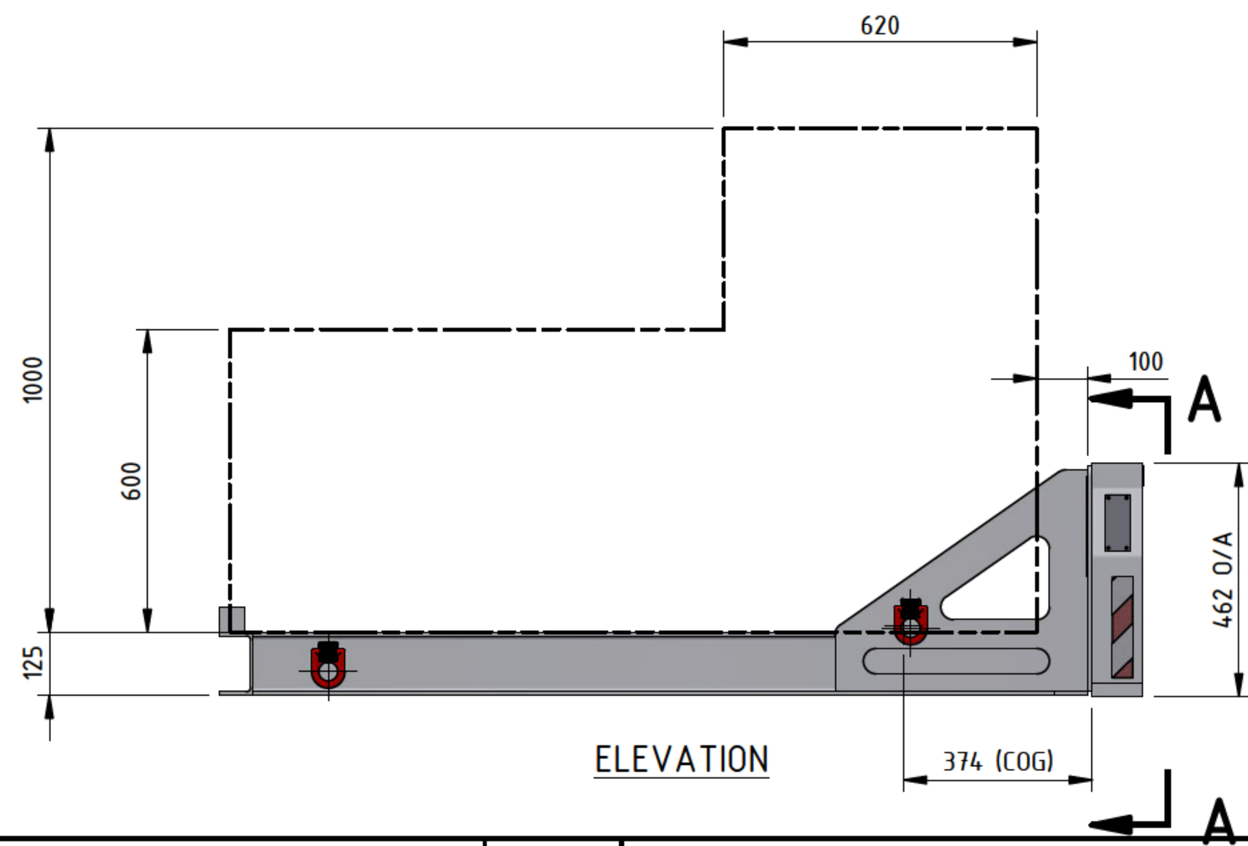
PLAN



SECTION A-A



FRONT VIEW



ELEVATION

REV	DATE	AMENDMENT DETAILS	BY	CHKD
0	18/07/2023	ISSUED FOR DISCUSSION	BK	

AE ALFABS
MINING EQUIPMENT PTY LTD
ABN: 18 099 383 650
152 MITCHELL AVENUE - KURRI KURRI NSW 2327
P. 1300 ALFABS - F. +61 2 49362913
E. mining@alfabs.com.au - W. www.alfabs.com.au

FINISHED ITEM MASS: 168.24 kg

MANUFACTURING DRAWING

SCALES 1 : 15
DRAWN BK
CHECKED
APPROVED

REF. DRAWING.
DRAWING TITLE: AIR CONDITIONER POD

1ST ISSUE:
SIZE A3
DRAWING No. AE-ASH-MACP-001
REV 0

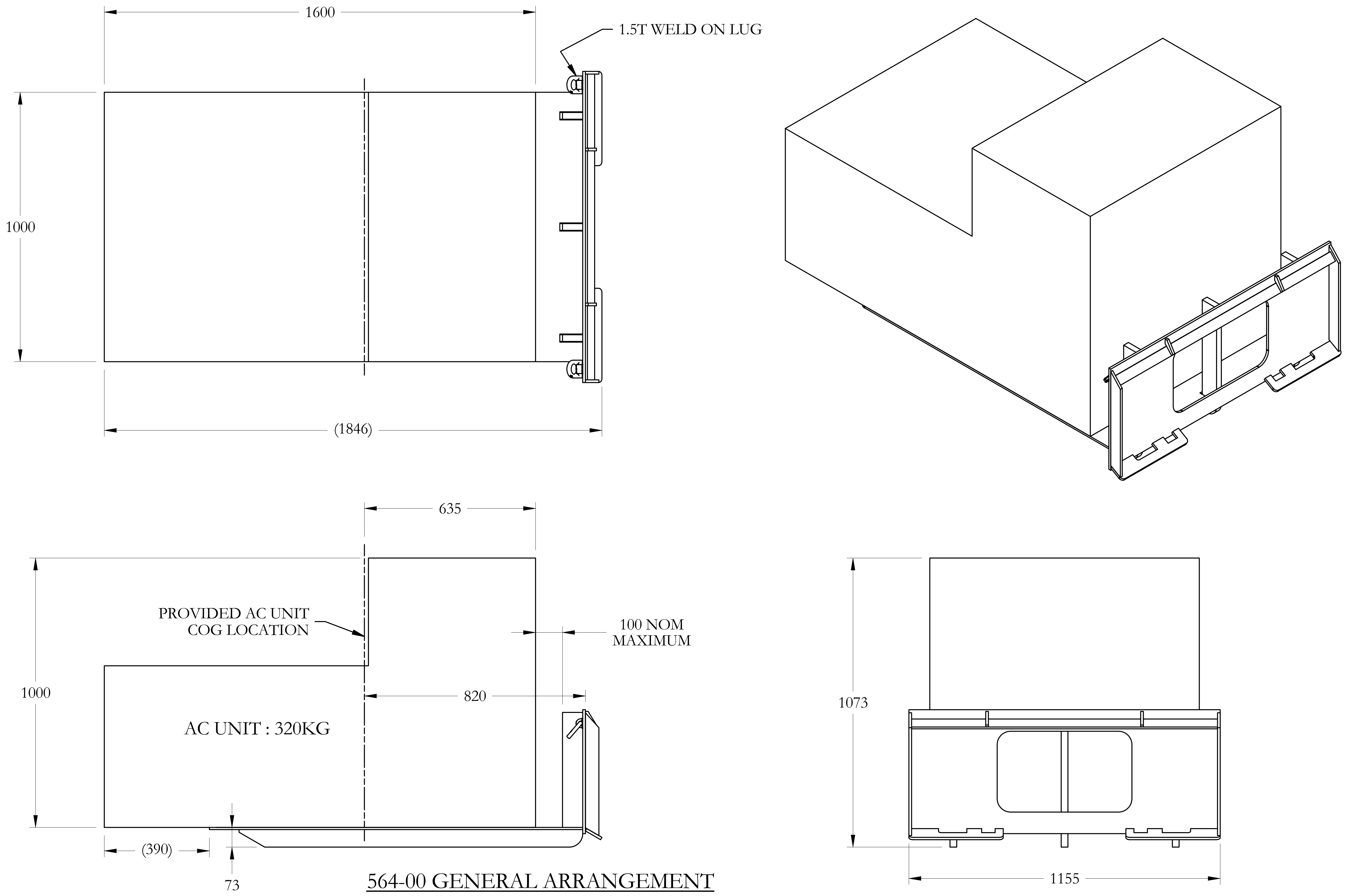
D:\Vault Workspace\Designs_PROJECTS\ASHTON\MUSTANG AIR CONDITIONER POD\AE-ASH-MACP-001.idw

Appendix H - External Concept General Arrangement

(Kohn, J 2023, pers. comm., 12 September)

REVISION HISTORY

REV	DESCRIPTION	DRAWN BY	DATE
A	ISSUED FOR COMMENT	JKOHN	12-Sep-23



564-00 GENERAL ARRANGEMENT

solid ENGINEERING	TOLERANCE UNLESS OTHERWISE SPECIFIED: WHOLE NUMBERS ±0.50 1 DECIMAL PLACE ±0.10 2 DECIMAL PLACE ±0.05 ANGLES ±0.5° ALL DIMENSIONS IN mm UNLESS NOTED OTHERWISE	DRAWING PRACTICE TO AS1100	DEBUR AND BREAK SHARP EDGES DO NOT SCALE ALL WELDS 6mm CONTINUOUS FILLET TO AS1554 GP UNLESS OTHERWISE NOTED TIGHTEN ALL BOLTS TO MANUFACTURERS SPECS ALL STEELWORK TO BE TO AS3679/300 UNO ALL HOLLOW SECTIONS TO BE AS1163 GRADE C350L0 UNO	MATERIAL : WEIGHT : 193 kg	DESIGNED: SOLID DRAWN: JKOHN 12-Sep-23 CHECKED: CUSTOMER APPROVED :	TITLE: AC UNIT & COOKIE PLATE GENERAL ARRANGEMENT PROJECT: AC COOKIE PLATE PART No: N/A DRG No: 564-00 REV: A SHEET SIZE A3