UNIVERSITY OF SKÖVDE

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BACHELOR DEGREE PROJECT

# Conceptual design of miniature vegetation cutter for demining activities in difficult terrain – an evaluation

Intended for the Chouf Mountains, Lebanon

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# Abstract

A conceptual design of a miniature vegetation cutter for use in minefields in southern Lebanon has been developed and the authors have evaluated its mechanical properties focusing on the stress in welded joints using the CAD software *Pro/ENGINEER Wildfire 4.0*. The conceptual design has been developed in the field in close cooperation with field staff from MAG Lebanon.

The requirements from the field specify that the cutter should have a cutting range of 80 cm, be equipped with adjustable covers, weigh less than 200 kg, be fitted to a commercial hydraulic excavator, and have the ability to cut vegetation, bushes and small trees including (olive) trees with a diameter up to 10 cm.

The miniature cutter is to be manufactured in the field and fitted to a commercial hydraulic excavator (Caterpillar 301.6C). It consists of a rotor on which eight cutting blades, alternatively chains, are attached in a helix formation. The cutter is protected by adjustable covers. When in operation, the rotor spins at 750 revolutions per minute whereby the blades cut through the shrubs and bushes in the cutter's path. The blades are mounted in a T-shape on arms, which are fastened to the rotor by a pin joint between two brackets (each) on the rotor. Blades and arms are to be welded together, as are the brackets to the rotor. These welded joints are the primary focus of the report. 3D CAD models have been created and analysed in *PTC Pro/ENGINEER Wildfire 4.0* to ascertain that the stress in the joints will not exceed the yield strength of the weld consumables, which should be 500 MPa. Ideally, the stress in the joints would be half the yield strength.

Type of bearings and a hydraulic motor have been selected for the cutter. Based on the specifications of the hydraulic motor an approximation of the forces acting on the weld joints in the case of an accidental stop (e.g. collision with a rock) has been calculated, and entered into the CAD software. Also, an approximation of the size (diameter) of branches the cutter would be able to tear apart in the case of branches getting stuck has been calculated and shown to be about 14.6 mm. Based on this, it is estimated that the cutter should be used only in areas where the shrubbery is of 20-30 mm in diameter. Considering this, and the relative light weight of the cutter, it is not likely that the cutter will be able to cut through the larger olive trees as requested, but it is considered that the tool still could be a valuable asset for mine clearing in Lebanon. In order to cut through thicker trees, it would be necessary to increase the power supply to the cutting system as well as the sturdiness of the cutting parts. Finding the required power and technical solutions for this demands further research which does not fit within the time frame for this report.

A preliminary weight approximation shows that the cutter will weigh roughly 170 kg, which falls below the limit of 200 kg and leaving some room for the bearings to be added.

The results from the stress analyses show that the stress in the welded joints falls well below the yield limit of 500 MPa, but not below 250 MPa. Still, the stress in all the welded joints is shown to be less than 300 MPa or at 40 % of the limit, which may still be acceptable. The end user will have to decide whether this is an acceptable safety margin before manufacturing the cutter and if it is not, measures will need to be taken to reinforce the weld joints and try to minimise the stress concentration in them.

# Preface

The authors would like to thank Åsa Wessel for opening the door to our field study in Lebanon.

Thanks also to the United Nations Mine Action Coordination Centre in South Lebanon for hosting us and for all their support. We would also like to thank the Swedish Rescue Service Agency, Norwegian People's Aid, and Danish Church Aid for taking us on so many field tours and patiently answering all our questions.

An especially big thank you to Christina Bennike and all of her staff at MAG Lebanon, as well as MAG International, for supporting us and the project and believing in us!

Last but not least, thanks to our supervisor Tobias Andersson for many inspiring and helpful conversations and meetings.

Hanna Risén and Emmily Sjölander

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# 1 Introduction

## 1.1 Background

### 1.1.1 Threat and consequence of landmines

Landmines are sometimes spoken of as "silent killers", since they hide underground and kill or maim its victims with no prior warning. They stay active for very long periods of time and therefore, unless they are removed, remain a threat to civilians and military personnel alike even long after the conflict during which they were laid has ended. They are indiscriminate weapons that do not distinguish between civilian and soldier, man or woman, adult or child, human or livestock or wild animals. They often block access to key points, such as water resources or power facilities, in communities and hinder the efficient development of livelihood, especially in the agricultural sector but also for example in tourism and construction or infrastructure. They also stop the safe return of refugees and IDP's to their home communities and make whole villages live in constant fear.

Aside from their humanitarian and socio-economic impact, landmines also have a considerable negative impact on the environment. Their use can trigger series of events that lead to adverse effects such as soil degradation, leakage of heavy metals to water sources and soil, deforestation, and possibly even effect the populations of entire species by altering food chains and natural habitats [1].

One study classifies the manners in which landmines lead to land degradation into five groups: access denial, micro-relief disruption, chemical contamination, loss of biodiversity and loss of productivity. All of these may have negative ecological effects, including overexploitation of non-contaminated land, damaging soil stability, interfering with the soils' capacity as a natural buffer between atmosphere, hydrosphere and living organisms in the exchange of micro-elements between them, and placing an additional burden on already threatened species [2].

### 1.1.2 Mechanical application in demining

While there are a number of stand-alone systems for mechanical mine clearance, what has come to be the most widely implemented role of machines within humanitarian mine action is ground preparation for subsequent manual clearance [3]. Ground preparation may or may not include detonation, destruction, or removal of mines [4], and is divided into different categories depending on level of ground penetration and/or removal of metal and tripwire. The scope of this report is limited to surface ground preparation, which includes the removal of vegetation and tripwire threat without penetrating the ground. It should however be noted that surveys and experience from the Chouf area show no evidence of tripwire contamination.

Vegetation removal was one of the first mechanical methods of ground preparation to be applied within humanitarian demining, and specialised machines for this have been used at least since 1995. For Mine Action programmes, the use of mechanical vegetation cutters can mean improved productivity compared to manual vegetation removal [3].

A one-year field study on a (large) non-intrusive[1] specialised machine used for vegetation cutting has been conducted by the HALO Trust, a British NGO, in Cambodia. This machine consists of a commercial vegetation cutter attached to a standard agricultural tractor. The tractor has been armoured to protect the operator. A 73.8 % average increase rate and a 50 % median increase rate were recorded. It was suggested that an intrusive machine could have brought a better increase rate. Similar results were obtained for a small, intrusive, remote-controlled machine for vegetation cutting (average increase rate 60 %, median 50 %), also used by the HALO Trust. The fact that this machine had to be remote-controlled meant limited vision and time lost when, for example, backing away from obstacles not seen in advance [3]. It stands to reason, then, that if a miniature machine like this could be operated manually, its contribution to increased productivity

1 *Non-intrusive* means that the machine is designed to work from a safe area, reaching into the contaminated area.

would be even greater. Increased productivity would be expected for any type of vegetation-cutting machines working in minefields in suitable conditions.

Some limiting factors for vegetation cutters, as well as other machinery used in humanitarian demining, would be the slope of the terrain and the condition of access roads, accessibility of spare parts and other material, as well as skilled mechanics and operators.

### 1.1.3 The landmine situation in Lebanon

Lebanon is a small nation of just over four million inhabitants and a total area of approximately 10,450 km$^2$. Bordering the Mediterranean Sea and situated between Israel and Syria, it has been subject to several conflicts during its short modern history – Lebanon only gained independence in 1943.

Since the Lebanese civil war in 1975-1990 until the end of the Israeli occupation in 2000, it has been estimated that around 550,000 landmines have been laid in Lebanon, affecting some 30 % of the population. The majority of the landmines still in the ground today are located in the proximity of the “Blue Line”, delineating the border with Israel, but landmines are prolific in several other parts of the country as well [5].

The province of Mount Lebanon, including county Chouf, is one of the areas most affected by antipersonnel (AP) landmines with one third of the contaminated land, preceded only by the provinces of Nabatieh and South Lebanon. It was estimated in the Landmine Impact Survey from 2004 that in all of Lebanon there was about 137 km$^2$ of contaminated land, affecting over one million individuals, whereas in Mount Lebanon the area of contaminated land totalled 50 km$^2$. In the area, more than 660 000 people were directly affected by the threat from landmines and unexploded ordnance [6].

The Lebanese Mine Action Centre (LMAC) estimated in May 2009 that a total area of 165 km$^2$ in Lebanon is still affected by landmines [7]. Out of this area, 20 km$^2$ belong to the Mount Lebanon province and approximately 250 suspected hazardous areas (SHA) are located in the south of Mount Lebanon.

A number of international organisations are conducting clearance work in South Lebanon, and of these the British NGO Mines Advisory Group (MAG) Lebanon is by far the largest with ten teams currently operating. Out of these ten teams, only one team is currently clearing landmines, while the remaining nine are doing battle area clearance (BAC), removing unexploded cluster munitions from the 2006 conflict with Israel. MAG has been working in Lebanon since 2000, conducting landmine and, since 2006, BAC of unexploded cluster munitions [8].

### 1.1.4 Landmines and related issues in Chouf, Mount Lebanon

The mines laid in the Chouf are known to be exclusively the so-called No. 4 mine. This is a plastic, pressure-actuated AP mine of Israeli origin containing 188 g of TNT. The operating pressure is eight kg and the measurements are 135 x 65 x 55 mm. It produces a blast explosion, which is to say that there is no fragmentation or shaped-charge to penetrate armour [9]. Figure 1 shows the exterior of the mine.

*Figure 1. A rendered-safe example of the Israeli No. 4 AP mine.*

MAG Lebanon is currently conducting humanitarian demining in county Chouf, part of Mount Lebanon, and since 2007 has cleared over 64,000 m$^2$ of land in the area [10]. In the Chouf, 19 SHA remain to be cleared, and the neighbouring counties of Aley, Baabda, and El Metn are still completely uncleared at the time of writing. Pending agreement from the LMAC, a number of the 214 SHA’s in these counties could be included in the work plan for 2010-2011.

An important crop in all of Lebanon is the olive, which includes both olive oil and table olives. Olive yield is cyclical: every other year is a low-yield year, every other a high-yield year. More mature trees, older than 25 years, also give higher yields. Crowther, in 2008, estimated that an average value of the olive crop per hectare is US$ 3,968 in low-yield years and US$ 8,694 in high-yield years [11]. Olive trees are being grown ubiquitously on the hillsides of the Chouf area, but due to the suspected landmine contamination many owners are unable to harvest and benefit from their return.

The clearance work in Mount Lebanon is fraught with issues. The minefields are so-called nuisance minefields, meaning that the pattern of the laid landmines is indiscernible or nonexistent, and large areas of land are not being used due to suspected contamination. This means that heavy vegetation, including the valuable olive trees, grows freely in the minefields and SHAs, which impedes clearance by making vegetation cutting necessary prior to the use of any clearance asset. In addition, the terrain is mountainous and steep, with much of the hillsides covered in terraced cultivations, see Figure 2. The target average clearance rate of the Mine Action Teams is set to 160 $m^2$ per team and operational day; however the teams are often slowed down significantly due to the need for vegetation cutting.

*Figure 2. Example of an overgrown terraced olive cultivation and minefield in the Chouf area.*

The use of mechanical assets for ground preparation increases the clearance rates and can lead to faster and safer clearance. However, rough terrain such as the small narrow terraces and steep hillsides makes it difficult or impossible for the vegetation-cutting machine currently owned by MAG Lebanon to access and/or operate in most of the areas. MAG Lebanon accordingly has identified the need for a miniature vegetation cutter to speed up the process of clearance.

## 1.2 Aim

### 1.2.1 Objective

MAG Lebanon seeks to address the problem by extending the existing mine clearance toolbox with a small vegetation cutting mechanical system. The system will consist of a specially developed tool for vegetation cutting fixed to a Caterpillar Mini Hydraulic Excavator model 301.6C (Canopy), small enough to operate in the narrow and inaccessible areas in Mount Lebanon.

The equipment will assist the Mine Action Teams with vegetation cutting in areas where the existing MAG vegetation cutter has no or limited access. The machine will be used for preparing access, creating minimal damage to forest, to unused remotely located SHA's and also for conducting the ground preparation without damaging the olive trees. The Caterpillar will be operated from the inside to provide a high level of control and visibility, and will be armoured to protect the operator.

The subject of this report is an evaluation of the proposed conceptual design of this vegetation cutter.

### 1.2.2 Specification of requirements

The design proposal will be presented according to the following specifications, which were requested directly by MAG Lebanon field staff:

- Ability to cut vegetation, bushes and small trees including (olive) trees with a diameter up to 10 cm
- Weight less than 200 kg
- Fixed to a Caterpillar 301.6C Mini Hydraulic Excavator (see specifications in Annex 1)
- Cutting range of 80 cm
- Adjustable cover

*The following points should be considered:*

- Detonation of the antipersonnel mine No 4.
- The power system (can the cutter be operated by the existing motor? – evaluation).
- The possibility to attach the tool in two positions (sideways or straight).
- Selection of material. Primarily depending on access in Lebanon.
- Rotational speed vs. working speed of machine.
- A spread-out and well-balanced placing of the blades to minimize vibrations.

*The result will be presented in a report which will include simple blueprints*

- The report shall provide a simple cost estimation.

*Delimitation*

- No armouring of the tool need be done.
- No in-detail planning is to be done.
    - For example: Only a selection of the type of bearings for the rotor need be made, with simpler calculations or reasoning to support the choice; a specific bearing does not need to be decided upon.

This report will treat only the design of the custom-made vegetation-cutting tool, thus the armouring of the canopy will not be discussed here.

## 1.3 Mode of procedure

The general design has been developed directly in the field, in immediate cooperation with MAG field staff and deminers, taking full advantage of their experience and knowledge as well as directly focusing on their requirements and needs. Continuous feedback has been given mutually and the design process has consisted of an open dialogue between the field personnel and the authors.

Data from Caterpillar and SSAB has been used to calculate available power to and ability of the cutter to withstand the possible impact and strain it may be subjected to. The design has then been modelled and analysed digitally using PTC *Pro/ENGINEER Wildfire 4.0* and its module *Pro/ENGINEER Mechanica Structure* in order to evaluate the strength of the construction in its proposed design.

# 2 Process and results

## 2.1 General design description

MAG Lebanon has a mechanical team, a Technical Field Manager Mechanical, and a well-equipped workshop where the vegetation cutter will be manufactured. All material will be obtain-ed from within Lebanon and the entire cutter will likely be manufactured from the same steel. The commercial steel Hardox© from SSAB has been used by MAG before and it was the hope of the technical team that it could be used for this piece of equipment as well.

### 2.1.1 Dimensions and function

The cutting tool consists of a tubular rotor with an inner radius of 35 mm, an outer radius of 50 mm, and a total length of ca 890 mm. The portion on which the knives are fastened (inside the adjustable covers, see below) will be roughly 840 mm long; the length of the rotor is extended 20 – 25 mm on each side to accommodate the two side plates on each side. Apart from this an additional length on each side may be necessary to fasten the two bearings needed. Figures 3, 4, and 5 overleaf show the various parts of the cutting tool.

The rotor can be fitted with either eight T-shaped cutting knives to be used in heavier vegetation as a mulcher-like machine; or with eight chains to be used as a flail where the main task is cutting grass. The T-shaped cutting knives will be manufactured from two pieces (referred to as cutting arm and cutting blade) of steel plates of eight mm thickness welded together. The arm, which is 250 mm long by 40 mm wide, will then be fastened with a pin between two brackets of 10 mm thickness and ca 40 mm wide by ca 50 mm long. These pin joints will be fastened enough that the arms stay straight when the cutter is operating under easy conditions, but will give way if too much resistance is encountered. A limit for this will not be set here. The brackets will be welded to the rotor, positioning the knives in a helix pattern, at an angle distance of 45° from each other. The knives will thus cover 360° of the outer side of the cylinder with the aim of minimising the stress caused by impact during the running of the tool. The blade is 45 mm wide and 100 mm long, meaning that a total cutting width of 800 mm is achieved. A 3D model of the cutting parts as well as its blueprint is shown overleaf in Figure 3, and Figure 4 shows the 3D models of the knife and a bracket.

Detailed blueprints with dimensions are given in Annex 2.

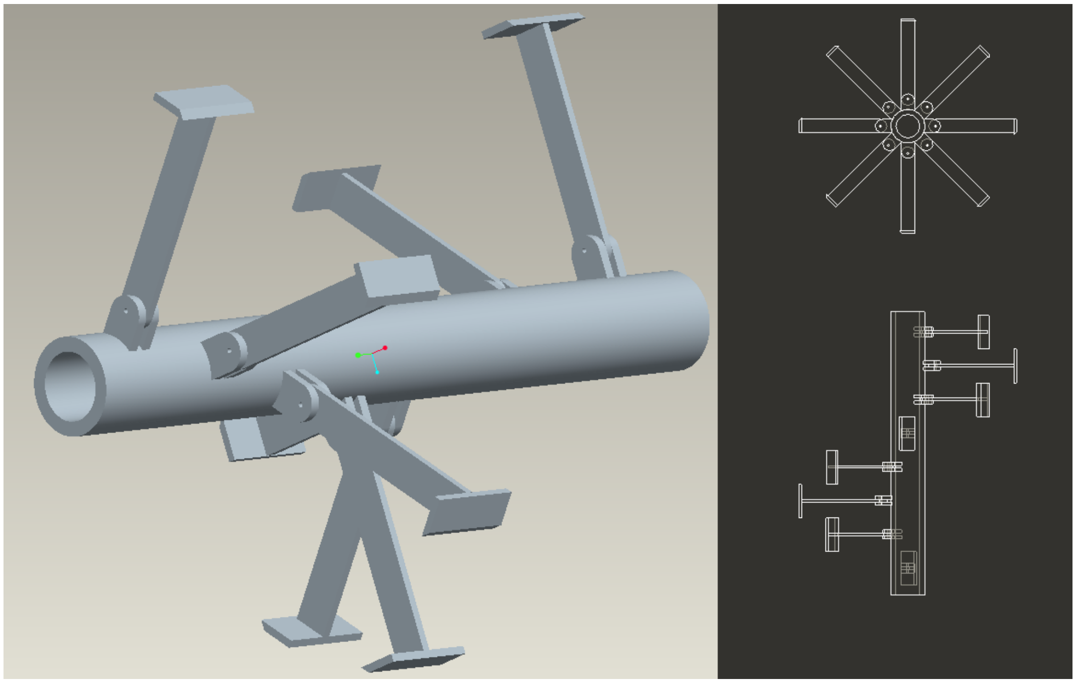

*Figure 3. 3D model and blueprint of cutting tool.*

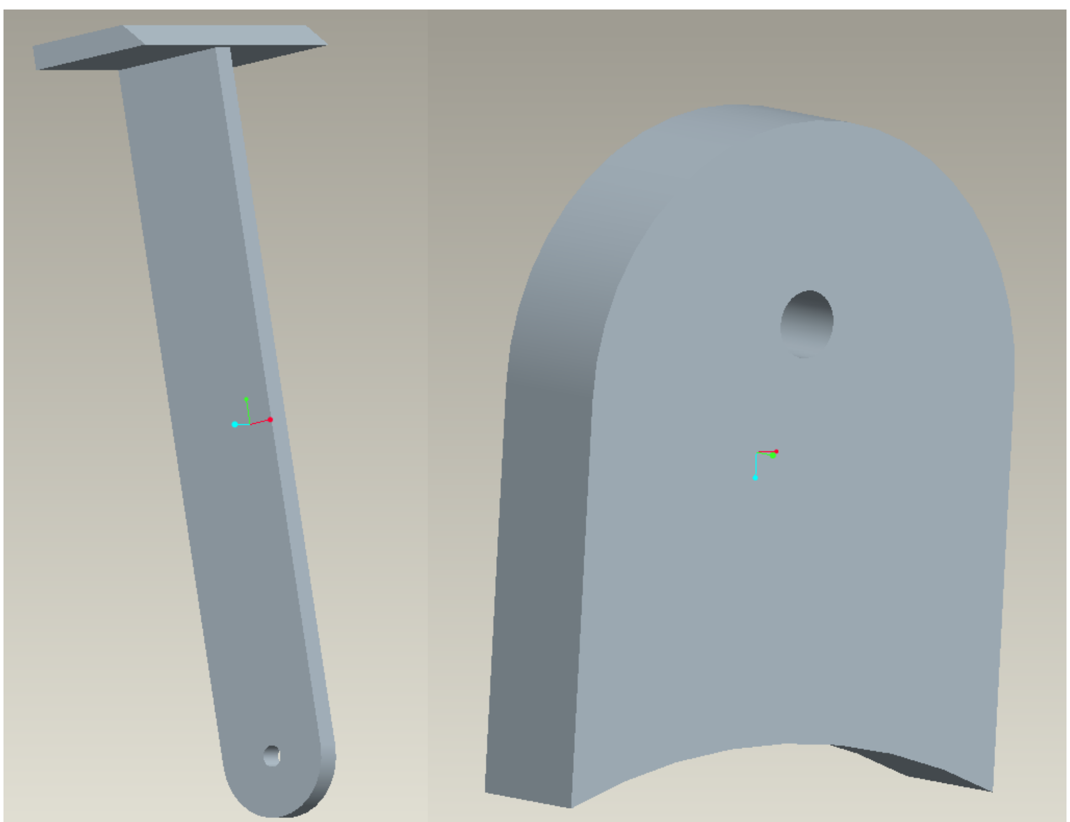

*Figure 4. 3D models of knife and bracket.*

The rotor spins around its axis at an angular velocity of 78.5 rad/s, or 750 rev/min (revolutions per minute). It is powered by the hydraulic system of the prime mover, meaning that a hydraulic motor will have to be obtained to transmit the rotation. The working speed across the surface will ideally be kept at 0.5 km/h, ensuring that all ground area is thoroughly covered and cut.

An adjustable cover prevents impaired visibility for the operator by hindering cut vegetation from flying uncontrollably. The adjustability also provides better adaption to different types of terrain. It is attained by fastening an outer cover to the outer side plates, and an inner cover to the inner side plates. The outer cover will be fixed, as the bracket for the hydraulic arm will be located on it, while the inner cover is manually adjustable by a simple winch mechanism. The inner side plates may for weight reasons be thinner than the outer depending on any bearing limitations, whereas the outer ones should be a little thicker to provide more stability. The same reasoning goes for the inner cover. The thickness of outer plates and cover can initially be set to eight mm, and the inner parts to six mm. Figure 5 shows the 3D model of the inner and outer cover, and Figure 6 shows the whole cutting tool inside the covers.

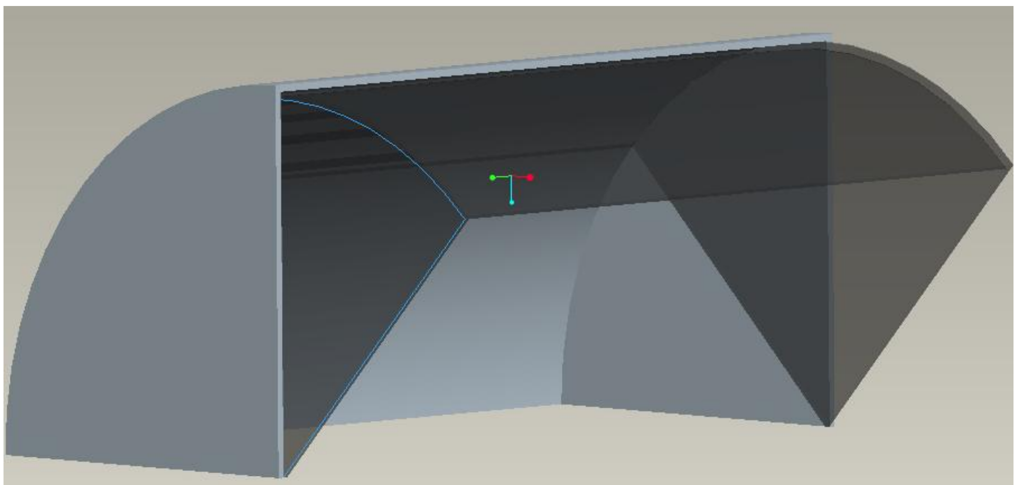

*Figure 5. 3D model of inner and outer covers.*

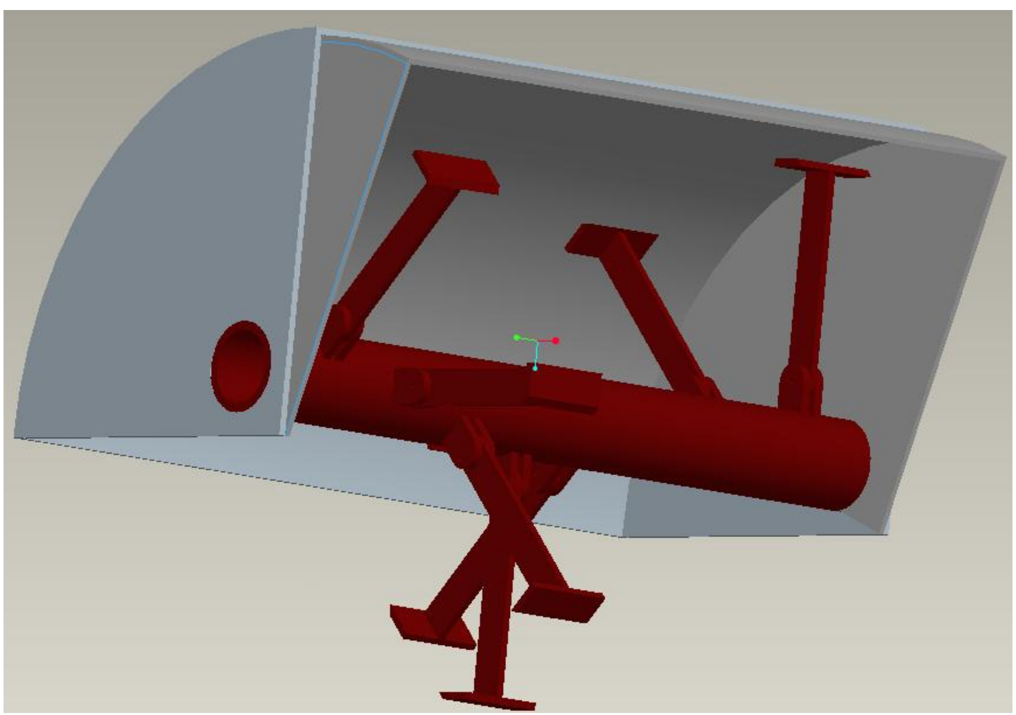

*Figure 6. Cutting tool with covers.*

When in operation, the vegetation cutter is fitted to the hydraulic arm of the excavator so that the cutting direction is perpendicular to the arm, i.e. the operator will be cutting from side to side rather than in a forward line. This will allow the operator greater control over the distance between the cutting tool and the prime mover, as well as give increased visibility. However, the cutter can also be fitted perpendicular to the arm to facilitate transportation.

Because the cutter is intended for use in low-threat areas primarily for area reduction purposes, the cutter itself will not be armoured. As the cutting tool is not intended to touch the ground, it is deemed unlikely that it will directly set off a landmine. Any explosion therefore is likely to occur far enough from the tool itself that damage to the cutter will be limited. To minimise downtime in case of an uncontrolled detonation it is suggested that both a cutter spare and spare parts such as extra blades and chains are kept on hand in the field.

### 2.1.2 Bearings

Four bearings will be needed for the tool, one for the rotation of both covers on each side. The relatively small diameter enables most bearings to be used. The bearing must be able to withstand the forces that result from the rotation of the cylinder. These forces are mainly radial. Due to the vibrations and uneven force impacts depending on the terrain, the bearing will also be exposed to some axial forces. The moment forces caused by the impact on the tool during operation are transmitted to the bearing and it is therefore necessary for the bearing to have the ability to handle moment forces. Paired single-row tapered roller bearings and four-point-contact angled bearings are suitable for moment forces as well as a high rotational speed. Since greater radial forces can be applied to a roller bearing, this type is preferable [12]. The bearing to be used in the tool is a single-row tapered roller bearing. Standard bearing material is sufficient, as the cutter will not be exposed to any type of liquid or other extraordinary conditions.

### 2.1.3 Material

Since Hardox©, which is readily available locally, was requested as the material to use, advice from representatives from SSAB in Sweden was sought. Because of the type of wear a vegetation cutter is subjected to (shock impact and an erosive environment thanks to sand and soil particles), harder steel is necessary especially for the cutting parts. The steel primarily recommended for the rotor, cutting blades, and arms is Hardox 450©, where the number 450 indicates the hardness of the steel measured in Brinell. If for some reason necessary, both Hardox 400© and Hardox 500© could likely replace Hardox 450© [13]. However, if required the adjustable cover and remaining parts may be manufactured from milder steel. This report uses properties of Hardox 450© where necessary, see Annex 4.

It is recommended that the steel used for the cutter is recycled and/or recyclable to as high a degree as possible – something which is often applied at the MAG workshop already. Also, damaged parts should be recycled or repaired by the user rather than be immediately discarded.

## 2.2 Hydraulic motor and torque capacity

### 2.2.1 Selection of hydraulic motor

To begin with, the cutting capacity of the extant hydraulic system must be verified. More power is required to cut shrubbery and bushes than to cut grass, wherefore the calculations are based on a wood-cutting scenario. It is assumed that the strike of the cutting blade on the branch will generally hit perpendicular to the grain of the wood, or close to it. Wood is an orthotropic material, being strongest in the direction parallel to the grain and weakest perpendicular to the grain [14].

According to the specifications of the Caterpillar (see Annex 1), the primary auxiliary circuits of the hydraulic system will deliver a flow of 34.4 l/min (litres per minute) at a (maximum) work pressure of 174 bar. In order to select a hydraulic motor, the displacement required to rotate the cutter at a speed of 750 rev/min using the available flow must first be determined. Using common hydraulic units, this is done by means of the following equation:

$$D = \frac{1000 Q \eta_V}{n} \tag{1}$$

where $D$ is the displacement in cm3/rev (cubic centimetres per revolution), $Q$ is the flow in l/min, $n$ is the rotational speed in rev/min, and $\eta_v$ is the volumetric efficiency which for this type of calculation can be approximated to 0.95.

Inserting the known numbers, a displacement value of roughly 43.6 cm3/rev is obtained. Looking at the Bosch Rexroth catalogue for external gear motors [15], the model G is seen to fulfil the criteria at hand. For this gear motor, the maximum continuous pressure is 180 bar while the minimum rotational speed is 500 rev/min, which both fall within the limitations for the cutter. The displacement of the motor selected must be the closest above 43.6 cm3/rev available, which here is 45 cm3/rev.

It can be seen from the catalogue [15] that the model G with the mentioned displacement will deliver a torque of roughly 100 Nm at the angular frequency of 750 r/min and a continuous pressure of 150 bar (see Annex 3 for diagram). It is here assumed that the pressure of the hydraulic system cited in the Caterpillar documents is the maximum and that the system will likely not always run at maximum capacity.

### 2.2.2 Capacity of the system

A worst-case scenario for the cutter is that a branch is not fully cut through by the blade, gets stuck and has to be pulled apart by the force of the rotational movement. Knowing the delivered torque, it is possible to calculate the branch diameter the cutter would be able to tear apart rather than cut apart. Branches are assumed to be of circular section, which is a reasonable approximation of the real-life situation. As the outer radius of the rotor will be 50 mm and the pin joint between the arm and bracket is here seen as being friction-free (i.e. the maximum resistance in the pin joint has been exceeded and the arm has folded), the lever arm $r$ in this case will be 50 mm. The system can be seen as static as the arm will momentarily stop when the wood gets stuck, even if only very briefly. The resulting force $F$ in the branch can be computed using equation 2:

$$M = Fr \tag{2}$$

Solving for $F$, the result is approximately 2 kN. This force is applied in the longitudinal axis of the wood, i.e. parallel to the grain. As this is where wood is the strongest it follows that more power is needed to tear the branch apart. Since it is not known which type of wood will be cut at any given time or place, an estimation of the tensile strength is necessary. Figure 7 shows the forces acting on a stuck arm.

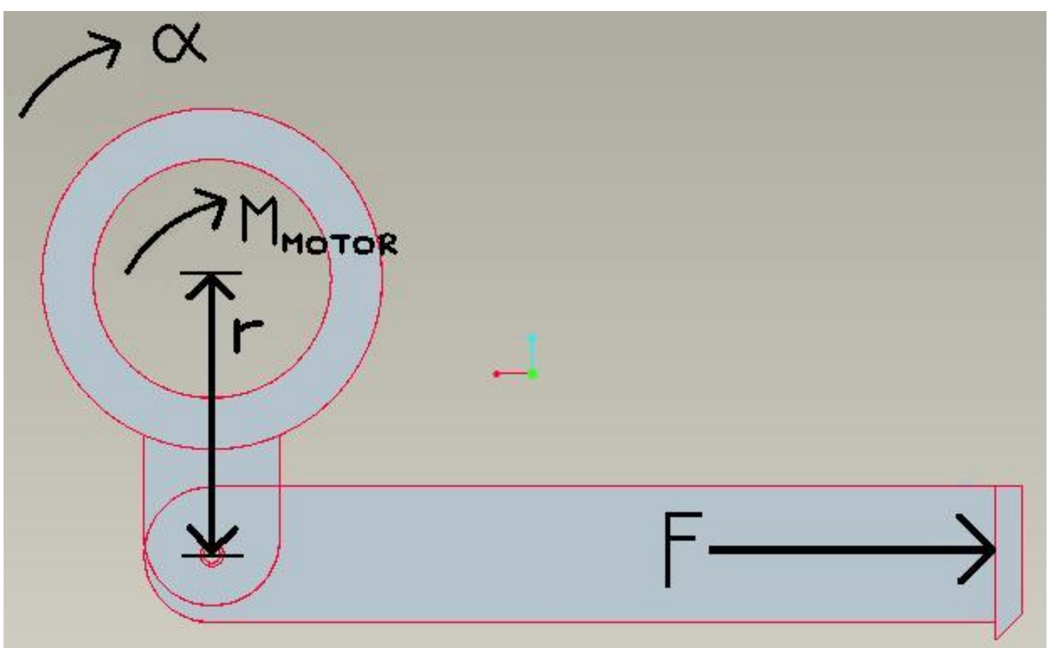


*Figure 7. Folded arm with acting forces, after collision.*

Going through property tables for various wood species used by the US Department of Agriculture Forest Service [14], it becomes clear that tensile strength parallel to grain is rarely listed in wood handbooks, but it is commonly accepted that the modulus of rupture for a given species can serve as a conservative estimate of the tensile strength. Green wood is weaker than dried wood and assuming a maximum tensile strength parallel to grain $\sigma_{max//}$ of 120 MPa should provide a satisfactory safety margin. Most wood species fall well below this value, and it is likely that so do shrubbery and bushes in the Middle East. In addition, considering the momentum of the cutter as well as the cutting action and weakening of the wood brought on by the shock impact, it is likely that the actual tensile strength limit $\sigma_p$ is only at about 10 per cent of the above, i.e. roughly 12 MPa.

Knowing the tensile strength limit $\sigma_p$, the applied force $F$, and assuming a circular branch section, the radius $r_w$ of the branch can be computed by solving the following equation for $r_w$:

$$\sigma_p = \frac{F}{\pi {r_w}^2} \tag{3}$$

The result obtained is 7.3 mm, which means that the branch thickness possible to pull apart corresponds to an undamaged branch with a radius of 7.3 mm, analogous to a diameter of 14.6 mm. With the branch being already partially cut through, this should be acceptable as long as the machine is not used where branches are thicker than 20-30 mm in diameter (based on the assumption that the blade will be able to cut through 15-25 mm of wood without major difficulty).

## 2.3 3D models and analyses

### 2.3.1 Preparatory calculations

Knowing the approximate capacity of the cutter, the next question is whether the welded joints (brackets to rotor and cutting blade to cutting arm) will be able to withstand the stress of a collision with an object that forces the rotor to a halt. This could be caused by for example a too-thick branch or the accidental striking of a rock.

To calculate the stress in the joints, CAD programme Pro/ENGINEER Wildfire 4.0 is used to draw up 3D models of the welded joints. The models are then analysed in Pro/ENGINEER Mechanica. While the actual cutter will be subject to shock impact in a dynamic setting, it is sufficient to use static analysis for the scenario envisioned here. In order to do so, the assumption is that the cutting arm can be seen as a beam subjected to a (static) bending load in the moment the rotation stops. The approximate stress at different locations can then be found by applying the "load" to the models and running the static analysis. This "load force" must be calculated manually.

In order to find the force corresponding to the bending load, the total torque affecting the system must be computed. The total torque $M_{tot}$ will be the sum of the torque from the motor, $M$, and the torque stemming from the moving parts' inertia, $M_i$.

$M$ of course is known, and $M_i$ can be calculated using the relation:

$$M_i = I_{tot}\,\alpha \tag{4}$$

where $I_{tot}$ is the moment of inertia of the cutter and $\alpha$ is the angular acceleration.

The angular acceleration will have to be approximated. Knowing the maximum rotational speed $\omega$, it can be assumed that the time elapsed before the rotor comes to a halt ($\omega$ = 0) is up to one second. This is just a rough approximation and the actual halting time may be shorter.

$$\alpha = \frac{\Delta\omega}{t} \tag{5}$$

It follows that the angular acceleration under these circumstances will be roughly 78.5 rad/s$^2$.

To find $I_{tot}$ it is necessary to look at one rotating component at a time, finding the moment of inertia for each part and finally adding them up, i.e.

$$I_{tot} = I_r + I_b + I_a \tag{6}$$

where $I_r$, $I_b$, and $I_a$ denote the moments of inertia of the rotor, the blades, and the arms respectively.

As the axis of rotation passes through the rotor's centre of mass, finding $I_r$ is relatively straightforward. It is done using the following equation for a hollow circular shaft:

$$I_r = \frac{m}{2}(r_i^2 + r_o^2) \tag{7}$$

where $r_i$ and $r_o$ denote the inner and outer radius of the rotor, respectively, and $m$ denotes the rotor's mass. Knowing the dimensions and that the density $\delta$ of the steel is 7850 kg/m$^3$, the mass is calculated to just under 28 kg. Rounding up to 30 kg, $I_r$ is thus 0.05588 kgm$^2$.

To find the moments of inertia of the blades and the arms, some simplifications are made as only an approximate figure is needed. Both the cutting arm and the cutting blade are considered rectangular, i.e. the sharpened edge of the blade and the rounded part near the pin hole, as well as the pin hole itself, of the cutting arm are omitted. Figure 8 and Figure 9 show the assumed shapes and denoted dimensions of the parts – note that they are not to scale.

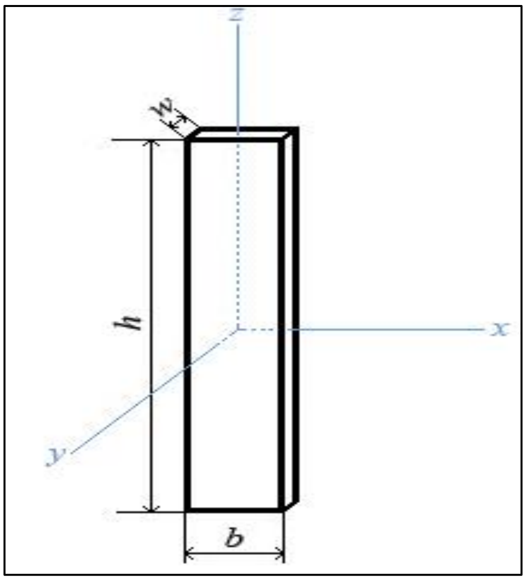


*Figure 8. Shape and denotations of cutting arm.*

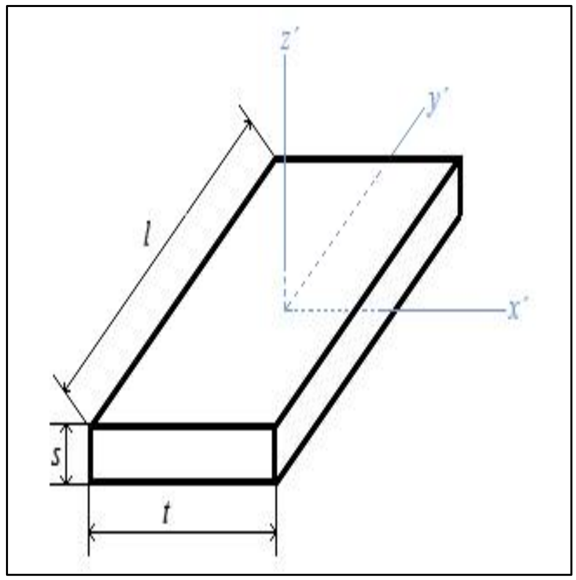


*Figure 9. Shape and denotation of cutting blade.*

Both the arm and the blade are of course rotating around the same axis as the rotor. Initially the moments of inertia rotating around the $y$- and $y'$-axes (centres of mass) will be found and the parallel axis theorem can then be used to find the actual moments of inertia. The moments of inertia are calculated from the parts' initial positions in rotation, i.e. before the pin joint gives way at the collision. This is done because at the collision only one arm will fold, making scant difference to the total, and because this original position is the one that, in a manner of speaking, carries the inertia. For the arm, the relation is as follows:

$$I_y = \frac{m}{12}(b^2 + h^2) \tag{8}$$

Similarly, for the blade:

$$I_{y'} = \frac{m}{12}(t^2 + s^2) \tag{9}$$

$I_y$ and $I_{y'}$ of course denote the moments of inertia around the $y$ and $y'$ axis respectively.

The parallel axis theorem defines a body's moment of inertia $I_{ax}$ about an axis parallel to but at a perpendicular distance from one through the body's centre of mass:

$$I_{ax} = I_{cm} + md^2 \tag{10}$$

where $I_{cm}$ is the body's moment of inertia about its centre of mass; $m$ is the body's mass; and $d$ is the distance between the two axes (i.e. from the centre of rotation to the axes $y$ and $y'$, separately). This distance is for the arm about 0.175 m and for the blade roughly 0.304 m. Figure 10 below shows the distances in a simplified sketch.

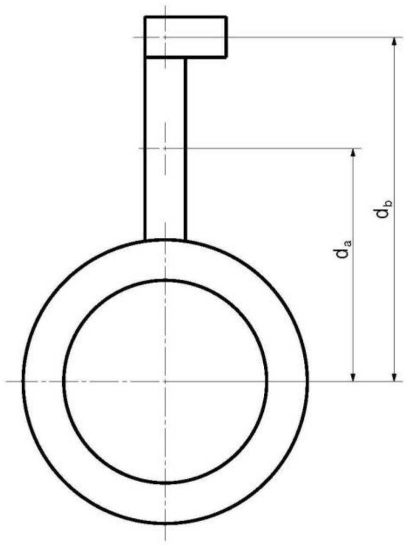


*Figure 10. Distances between the relevant axes of rotation, where $d_a$ is 0.175 m and $d_b$ is 0.304 m.*

Equations (8) and (10) for the arm yield a moment of inertia of 0.0192 kgm² for each arm. Since there are eight arms, the total moment of inertia for the arm component $I_a$ is 0.1539 kgm².

For the blade, equations (9) and (10) yield a moment of inertia of 0.0233 kgm². Times eight for the number of blades, the total moment of inertia for the blade component $I_b$ is 0.1860 kgm².

Rounding off, equation (6) then gives that $I_{tot}$ is approximately 0.4 kgm².

Inserting the results from equations (5) and (6) into equation (4), $M_i$ is shown to be around 31 Nm. Adding this to the torque from the motor, $M_{tot}$ is obtained for a value of 131Nm. At the maximum distance from the centre, i.e. at the outer radius of the cylinder 50 mm out, this total torque will correspond to a force load $F_l$ using the same principle as in equation (2). Solving for $F_l$, the result is close to 2 620 N. When applying the load to the 3D models the reality of the shock impact will be considered and the force doubled to 5 240 N to come closer to the real situation.

It is recommended that with Hardox 450©, welding consumables with a yield strength of up to 500 MPa be used [16]. The analyses are therefore conducted on the models to verify that the stress in the welded joints is well below this limit: preferably, with a safety factor of 2 (i.e. the measured stress should fall below 250 MPa in the welds).

### 2.3.2 Cutting blade weld

The model used in Pro/ENGINEER Wildfire 4.0 is a 3D model with solid elements. A very slim hole between the cutting arm and blade is used to simulate the weld, together with a rounded outside surface as shown in Figure 11. The hammer is modelled as one piece.

The empty space between arm and blade is modelled as being 1.5 mm high, which is an exaggeration as in reality the surfaces will be much closer to each other. However, it is considered that the stress measured in the simulated weld will be more accurate due to the stress concentration in the relatively long and thin members. The material of the weld joint is assumed to be linearly elastic and isotropic.

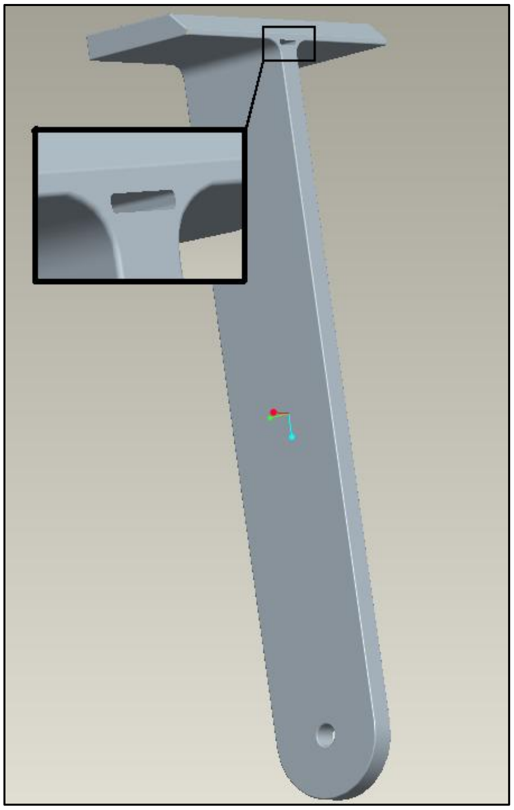

*Figure 11. 3D model of knife with simulated weld.*

To perform the analysis a simpler model of the blade is used. This is to minimise the sharp edges which might cause problems for the calculations performed by ProE. To simulate a load, acting in a way as accurate as possible, an edge is created. In other words, a small piece of the sharp chamfer is left out to enable the analysis, as can be seen in Figure 12.

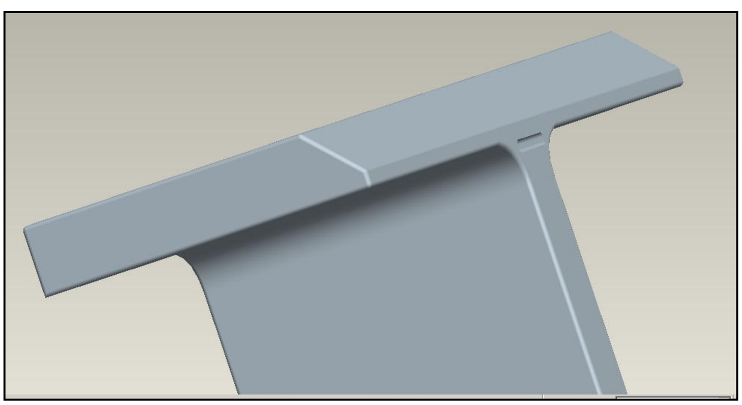

*Figure 12. 3D model of blade with a simplified edge.*

The analysis run is static with a force applied to the underside of the blade as seen in Figure 10, as anything getting stuck at the knife's edge will lead to a force acting on the knife in an "outward" direction, parallel to the arm. The load applied is 5 420 N, as calculated from the rotation speed as mentioned in the preparatory calculations. The model is constrained in all directions and rotations, through the pinhole where the knife is connected to the bracket (note that this restraint is also an approximation as in reality there will be a degree of flexibility in the pin joint). The load and constraints can be seen in Figure 13.

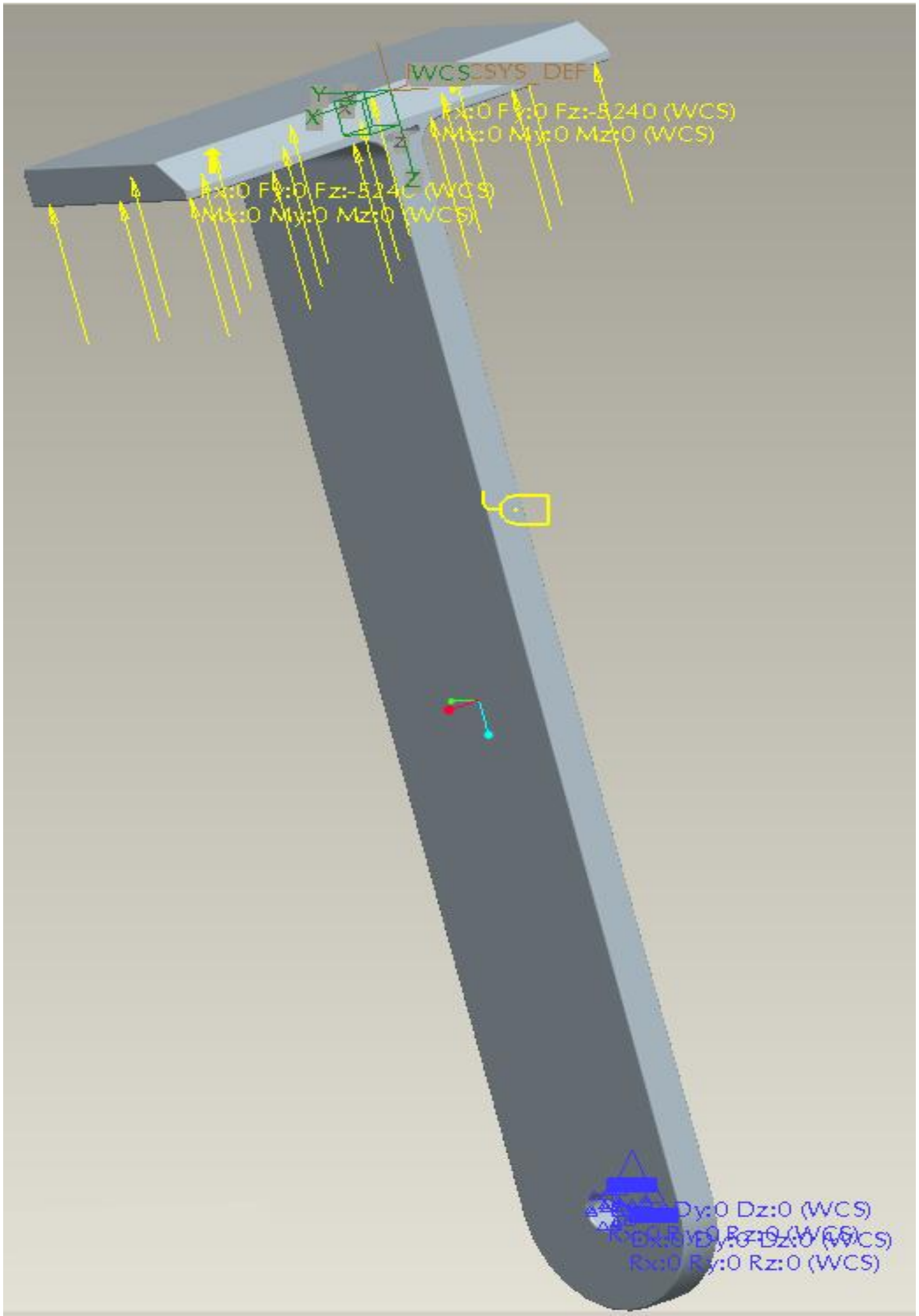


*Figure 13. Loads and constraints for the knife analysis.*

The material used in the model is Hardox 450©, whose properties can be seen in Annex 3.

A static analysis with solid elements is run. Figure 14 overleaf illustrates the mesh elements created on the knife. The number of solid elements created is 3 724.

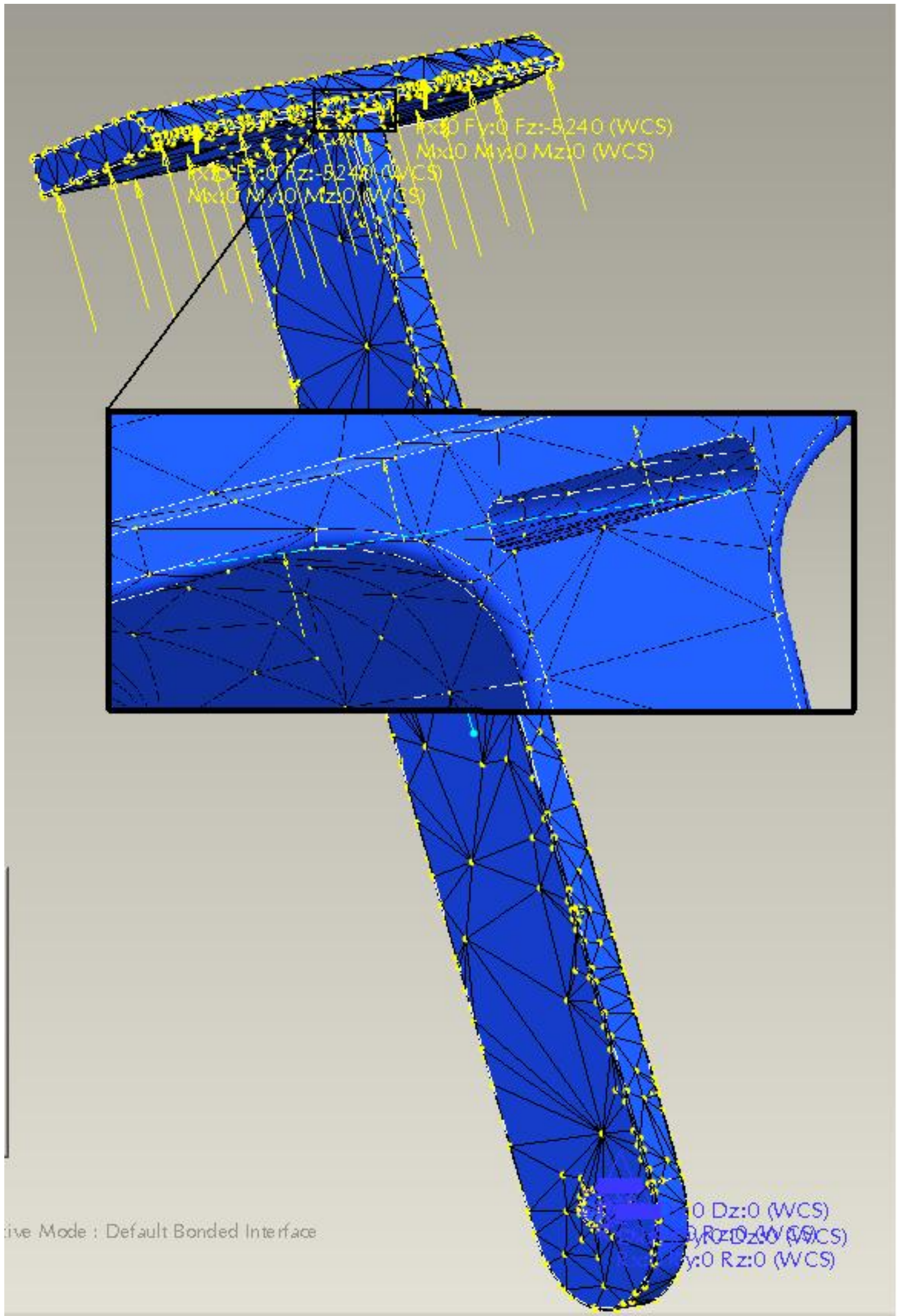


*Figure 14. The mesh elements on the knife blade*

A single-pass analysis is performed in order to obtain values for maximum stress according to von Mises. To compare results a multi-pass analysis is also performed and the convergence requirement is put to 15%, the polynomial order to 3 - 9.

### 2.3.3 Bracket weld

The model used is a 3D model with solid elements. When modelling the bracket weld, some approximations relative to the actual conditions must be made. Both bracket welds share equally the 5 240 N load. Rather than drawing up both brackets, one is considered enough as the stress in the individual weld is the focus. As a reminder, Figure 15 shows the actual design of the bracket/pin joint – compare to the model in Figure 16. The bracket and part of the cylinder on which it is fastened are modelled as one piece.

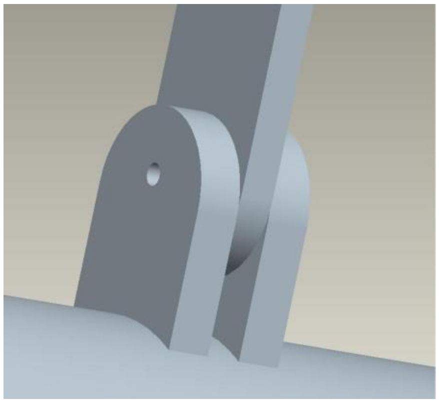

*Figure 15. Actual design of bracket/pin joint.*

To simulate the weld, a very slim hole between the bracket and the outer surface of the cylinder is created, so that the rounded-off sides of the bracket still connected to the cylinder surface represents the weld itself.

As with the weld simulated in the knife, the hole between the surfaces is an exaggeration.

Again, as the point of interest is the weld joint, it is not necessary to model the entire cylinder – the half-cylinder is sufficient to get an idea of the stress distribution. The weld joint is assumed to be linearly elastic and isotropic.

Figure 16 shows an image of the 3D model used, including a close-up of the weld hole.

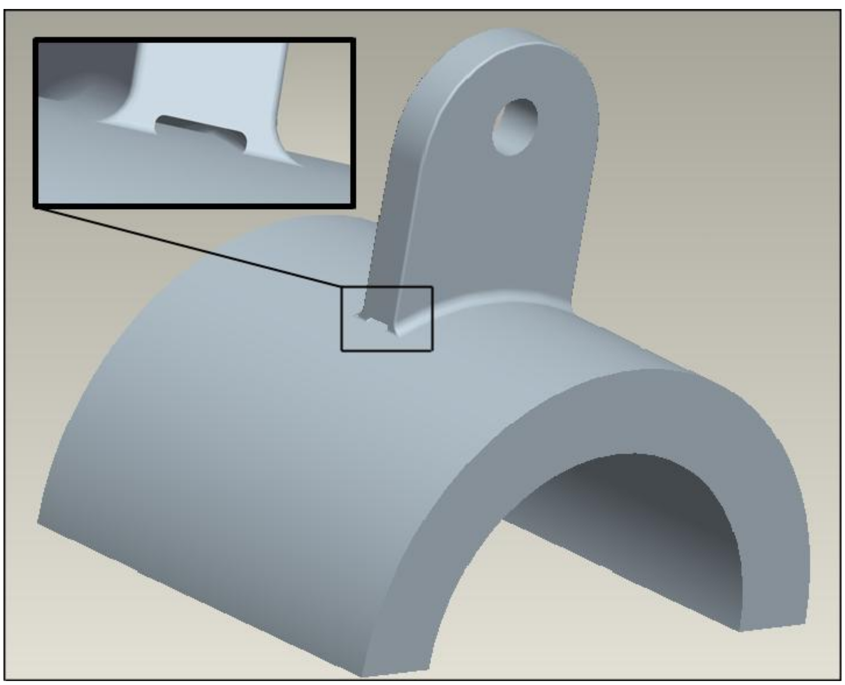

*Figure 16. 3D model of the bracket-cylinder; inlay close-up of weld.*

The model is constrained in all directions and rotations at the flat surfaces of the cylinder. The load applied (to each weld joint) is 2 620 N. This is calculated from the impact force during the rotation in the case stated in section 2.3.1 ovan. Constraints and loads are shown in Figure 17.

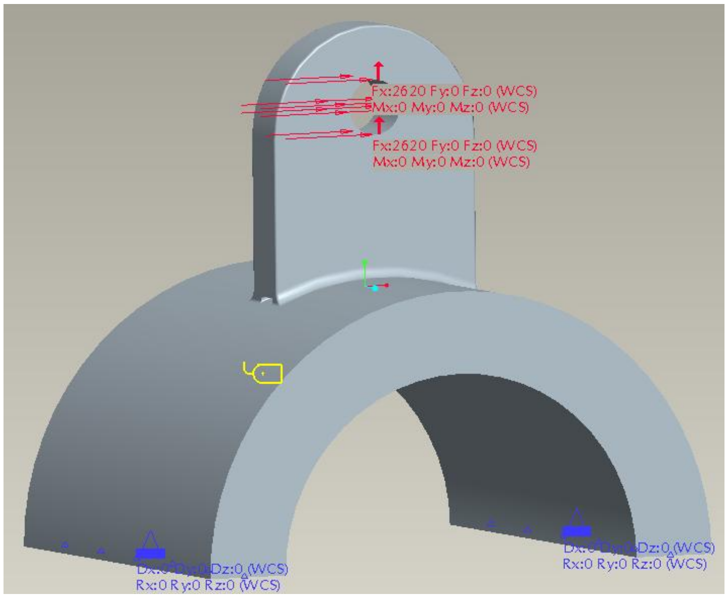


*Figure 17. Loads and constraints for the bracket-cylinder analysis.*

As previously stated, the material used in the model is Hardox 450©, whose properties can be seen in Annex 3.

Figure 18 shows the mesh created for the analysis before it is run. The number of solid elements is 1 741.

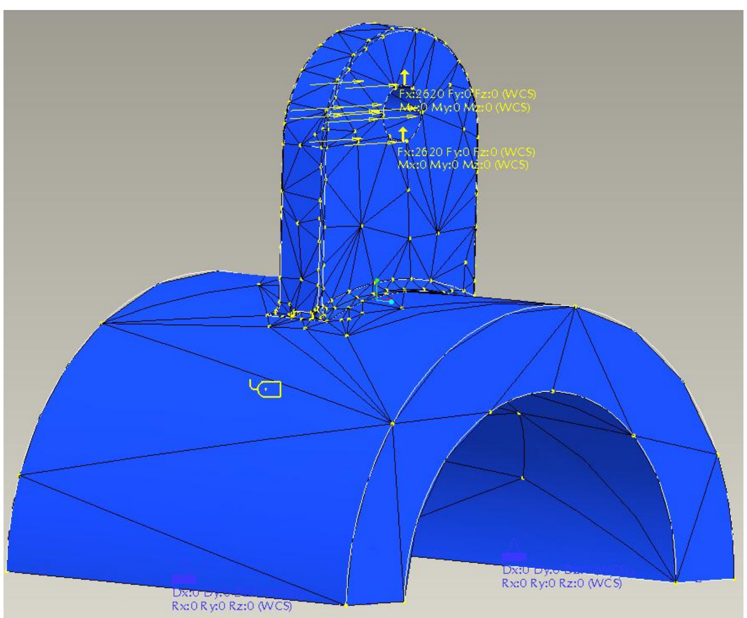


*Figure 18. The mesh elements on the bracket-cylinder.*

As with the cutting blade weld, a single-pass analysis is performed in order to obtain values for maximum stress according to von Mises. To compare results a multi-pass analysis is also performed and the convergence requirement is put to 15%, the polynomial order to 3 -9.

### 2.3.4 Results of the analyses

In the blade weld, the single-pass analysis shows that the maximum stress in the weld according to von Mises is 235 MPa. Figure 19 illustrates the stresses according to von Mises. The multi-pass analysis gives a value of stress according to von Mises of 230 MPa, with a convergence of 1.2%.

The single-pass and multi-pass analyses respectively show a maximum principal stress in the weld is 272 MPa and 270 MPa with a convergence of 2%. This means that according to the von Mises method, the stress in the blade weld falls below the limit of 250 MPa, but when looking at the principal stress the limit is slightly exceeded. Both values are however well below the absolute maximum of 500 MPa.

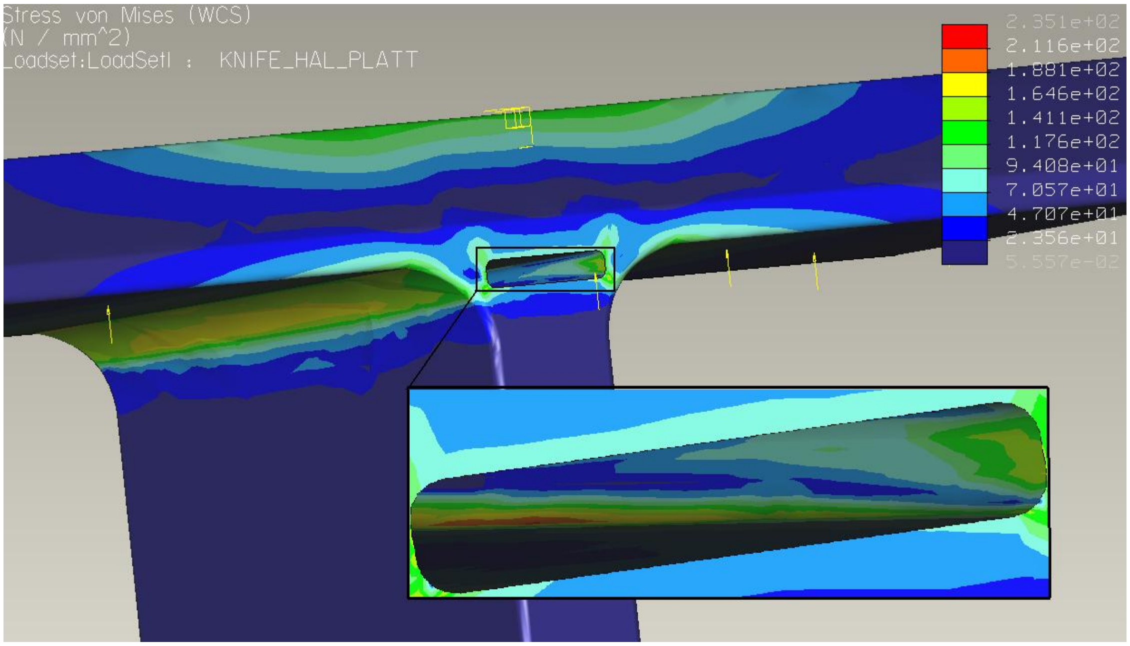


*Figure 19. Stress von Mises results for the knife.*

For the bracket weld, the results of the single-pass analysis show that the highest stress according to von Mises will be 280 MPa, whereas the maximum principal stress is 276 MPa. Figure 20 below shows the von Mises stress from the single-pass analysis. The multi-pass results give a maximum stress of 273 MPa for both von Mises (convergence 0%) and principal stress (convergence 0.2%). The stress in the bracket welds then will exceed the 250 MPa limit while still well below 500 MPa.

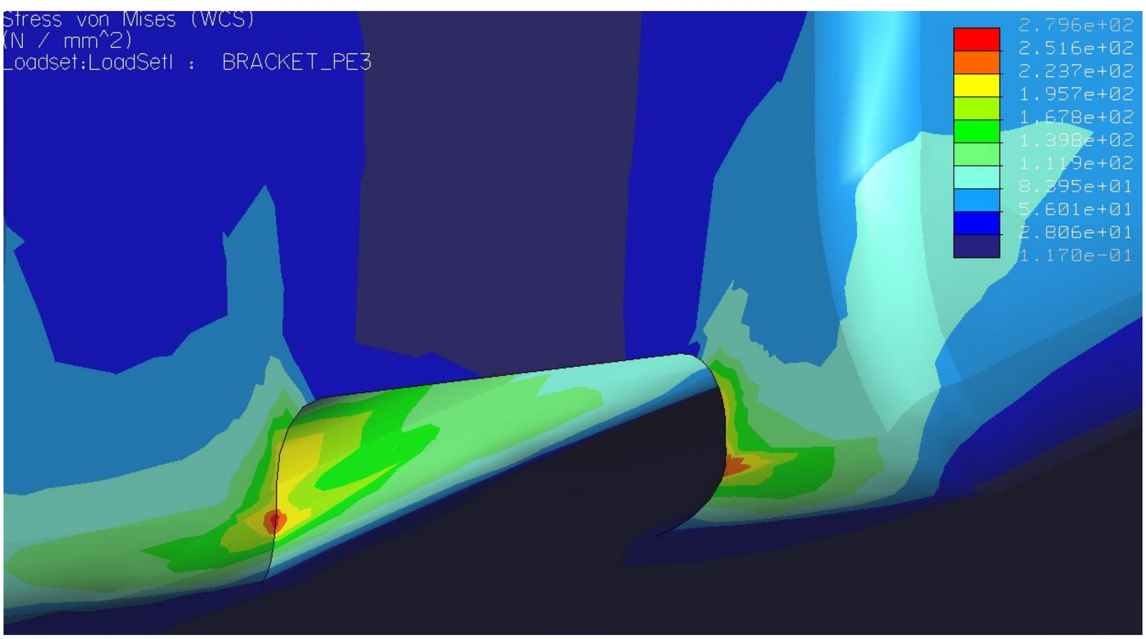


*Figure 20. Stress von Mises results for the bracket weld.*

## 2.4 Preliminary cost estimate and weight check

An initial estimation of cost has been made regarding the material and work required to manufacture a prototype of the vegetation cutter. The price estimates are based on MAG Lebanon's previous transactions with suppliers in Lebanon, as disclosed by the Technical Field Manager Mechanical and the Site Supervisor Mechanical, and should in no way be taken as final [17]. Table 1 below provides the preliminary overview. Prices are given in US dollars.

Assumptions for this cost estimate are that only the rotor, the brackets, the blades and the arms will be manufactured from Hardox 450© while the remainder of the metal components will consist of "everyday" mild steel with no closer specification. As steel is sold per kg, an approximate weight for the various parts has been computed using the dimensions given in the table and in the general description above. For the sake of simplicity, both types of steel are in this chapter taken to be of density 7 850 kg/m$^3$. The four side plates needed can be cut out of two steel sheets (mild steel), each of 0,5 x 1 m side, eight mm thick for the outer plate and six mm thick for the inner. The covers will also be bent from steel sheets.

The hydraulic motor and the bearings are omitted at this point as no quotes have been obtained for these. Table 1 shows the cost estimate, and it also shows the total weight of the tool, which is below the limit of 200 kg leaving some room for the bearings and the brackets for attaching the tool to the hydraulic arm.

**Table 1. Cost estimate, in US dollars, for manufacture of cutting tool.**

| | *Dimensions [m]* | | | *Quantit* | *Total weight [kg]* | *Price/kg* | *Total price* |
|---|---|---|---|---|---|---|---|
| *The tool* | | | | | | | |
| Outer side plates | 0,500 | 1,000 | 0,008 | 1 | 31,4 | $1,20 | $ 37,68 |
| Inner side plates | 0,500 | 1,000 | 0,006 | 1 | 23,6 | $1,20 | $ 28,26 |
| Outer cover | 0,869 | 0,685 | 0,008 | 1 | 37,4 | $1,20 | $ 44,86 |
| Inner cover | 0,852 | 0,676 | 0,006 | 1 | 27,1 | $1,20 | $ 32,55 |
| Rotor | | | | 1 | 30,0 | $8,50 | $ 255,00 |
| Brackets | 0,040 | 0,050 | 0,010 | 16 | 2,5 | $8,50 | $ 21,35 |
| Blades | 0,100 | 0,045 | 0,008 | 8 | 2,3 | $8,50 | $ 19,22 |
| Arms | 0,250 | 0,040 | 0,008 | 8 | 5,0 | $8,50 | $ 42,70 |
| Winch | | | | 1 | | | $ 150,00 |
| (Motor) | | | | | 10,1 | | |
| (Bearings) | | | | | | | |
| **Total** | | | | | **169,4** | | **$ 631,62** |
| | | | | | | | |
| *Consumables* | | | | | *Quantity* | *Price* | *Total pric* |
| Bending | | | | | 2 | $3,00 | $ 6,00 |
| Cutting | | | | | | | $ 120,00 |
| Welding rods and grinding discs | | | | | | | $ 25,00 |
| Paint | | | | | | | $ 40,00 |
| | | | | | | | |
| **Total** | | | | | | | **$ 191,00** |
| | | | | | | | |
| **Grand Total** | | | | | | | **$ 822,62** |

# 3 Discussion

As seen above, the results from the analyses show that the stresses in all the welded joints exceed 250 MPa, with the exception of the von Mises stress in the blade joint. However, the stresses still fall well below the absolute maximum of 500 MPa and only exceed the stipulated limit by about 10%. They all fall below 300 MPa, which while not at half the absolute maximum is at 3/5 of it. This means that the safety margin, if no changes in the design are made, would be at 40% rather than 50%. This could perhaps be acceptable, considering that these calculations are based on extremes and high approximations. The user will need to decide whether the 40% safety margin is acceptable before manufacturing the cutter. If it is not, measures will need to be taken to reinforce the weld joints and try to minimise the stress concentration in them.

Of course, it must be kept in mind that the cutter will be subject to continuous wear in the shape of shock impact, erosion, heat etc, and that this static analysis conducted does not consider any of these. It is accepted that the cutting tool will need regular replacement of the blades and knives, or chains when they are used. However, the results indicate that the tool has a promising capacity to withstand singular and rare collisions with objects it cannot cut through, despite the stress exceeding 250 MPa. Still, as with any machine, the continuous wear will likely weaken the resistance of the welded joints with time. If a collision should occur shortly after a blade and knife exchange the chance that the blade joint is damaged is smaller relative to a collision when the system has not been serviced as recently. Where the bracket joints are concerned, these will not be replaced as often and so will be more likely to break the longer the cutter has been operational at the time of a collision.

Another factor to take into consideration is that due to the sideways attachment of the cutter, the risk of hitting an object large and hard enough to actually stop the rotational movement is significantly reduced compared to working in a forward movement. This in turn means that the main reason for wear of the welded joints is the continuous shock impact from cutting through smaller branches.

It should also be noted that a collision leading to a rotational halt will put a tremendous strain on the hydraulic motor if it is not switched off immediately. For this reason, the possibility of equipping the chosen motor with a safety valve to disconnect the motor if the resistance is too high should be investigated. With a valve of this type, the risk of twisting damage to the rotor in case it gets stuck would also be eliminated.

Regarding the capacity of the system to pull apart branches, it is quite obvious from the above calculations that the cutter will not be able to tear apart trunks of olive trees with a diameter of 10 cm. Whether or not the machine will be able to cut through these trunks has not been tested, but given its relative low weight this does not seem very likely. However, despite the fact that initially it was requested that the machine be able to cut smaller olive trees, as these are a valuable resource in Lebanon and a very important source of income for the landowners, it will likely not be a common occurrence that the owners will wish to have the trees cut down. Rather they will have to be protected when clearing the minefields – this is usually the case. Even one single olive tree to be harvested can make a considerable difference in income for a family, and considering the amount of time an olive tree requires to reach optimum maturity/profitability it is all the more understandable that owners are often unwilling to part with the trees. While there may of course be other trees of larger diameter in the SHAs, it is likely that the ability of the cutter will still suffice to achieve the desired increased clearance rate as the major part of vegetation in the hills consists of shrubbery and the cutter is small enough to be able to manoeuvre between the larger part of thicker trunks. In order to cut through thicker trees, it would be necessary to increase the power supply to the cutting system as well as the sturdiness of the cutting parts. Finding the required power and technical solutions for this demands further research which does not fit within the time frame for this report.

Concerning smaller branches getting stuck in the cutter, this is not likely to occur often or at all given the speed of rotation, the sharpened blade, the relative thin-ness of shrub branches, and the fact that the branches are generally struck at the angle where they will be the weakest. If despite this the branches should get stuck, it is likely that only a small portion of the branch will remain to cut through provided a) that proper maintenance and care of the blade is ensured so that it is kept sharp; and b) that the machine is not being used to cut branches of too-large dimensions. The cutter should be able to handle branches of a diameter of 20-30 mm without any problems. This size can probably be considered "normal" for bush vegetation.

There is some concern that the single helix of cutting arms/blades around the rotor may lead to poor balancing of mass, which could result in frequency vibration and rotational issues, as well as additional structural loads and difficulties manoeuvring the tool. One solution to this would be to attach an extra helix of cutting arms and blades placed exactly opposite to the original helix, i.e. to use in total 16 cutting arms in pairs to make up a double helix. This way the tool will be better balanced, but the weight may become an issue. Further investigation is needed to ascertain this as well.

No calculations have been executed for the chain option, but it can be stated that the same maximum force will affect the bracket weld in the case of a chain getting stuck badly enough that the rotation stops. This means that the bracket weld is as likely to withstand a forced stop when the rotor is equipped with chains as it is when the blades are attached.

# 4 Conclusion

Despite the fact that the vegetation cutter cannot be shown to be able to cut down smaller olive trees, its capacity to handle bushes and shrubbery still make it a valuable asset in the difficult terrain of Mount Lebanon.

Determining the power requirements of a similar machine for cutting larger trees and designing the system for this could be an interesting subject of further research.

Before the manufacture of this cutting tool, closer investigation of bearings and hydraulics must be conducted with the aim to determine the exact and optimal dimensions required. The user must also decide whether a 40% safety margin rather than a 50% one is acceptable. The authors' opinion is that it is.

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# Annex 1: 301.6C, 301.8C Mini Hydraulic Excavators


| | 301.6C | 301.8C |
|---|---|---|
| Naturally Aspirated Diesel Engine | | |
| Net Power (ISO 9249) at 2400 rpm | 13.5 kW/18.4 hp | |
| Operating Weight | | |
| With canopy | 1610 kg | 1680 kg |
| With cab | 1720 kg | 1785 kg |

**Annex 1**

Brochure from Caterpillar:
*301.6C, 301.8C Mini Hydraulic Excavators* ©Caterpillar 2006

# Caterpillar® 301.6C and 301.8C Mini Hydraulic Excavators

*Designed to deliver reliable performance, versatility and ease of operation.*

## Performance and Versatility

**High digging forces and fast cycle times,** ensure that the Cat C-Series machines deliver the productivity that customers demand.

**Auxiliary lines and connectors fitted as standard,** mean that the Cat 301.6C and 301.8C come 'ready to work'.

**Hammer (one-way) auxiliary lines** fitted as standard.

**The dozer blade float function** enables more efficient clean up and landscaping operations.

**Choice of standard or long stick** enables matching of machine to application.

**High rotation bucket angle** combines good spoil retention and flat back trench characteristics.

## Comfort and Ease of Operation

**Cab and canopy options** are available. The C-Series cab provides a very spacious and comfortable work area with a high feature level as standard.

**Impressive legroom,** air circulation and well positioned, low effort controls give even greater operator comfort.

**Suspension seat as standard.**

**Automatically applied swing lock** aids machine transportation.

**Optional fan and sun blind package** offers further operator comfort.

*Engineered Durability*
*The heavy-duty bodywork and clean lines of the Caterpillar 301.6C and 301.8C ensure long life and reduce maintenance cost in tough, demanding applications.*


2

## Low Operating Costs

**Robust, reliable and easily serviced,** are key attributes of the C-Series design.

**Protected front linkage lines** are routed out of harms way, above the boom and stick.

**A tough counterweight and extra thick side panels** protect against impact damage.

**A large rear door** enables excellent access, whilst low maintenance front linkage pivot joints and well placed grease points reduce time taken for daily checks.

**The Cat S•O•S points** assist easy oil sampling for preventative maintenance checks.

**500 hour engine oil change period** helps to keep operating costs down.

## Customer Support

**Caterpillar dealers** offer unmatched customer support with excellent equipment management services and fast parts availability, resulting in maximum uptime and minimum repair costs.

**Equipment maintenance and management services** help optimize performance, reliability and profit.

**Annex 1**

Brochure from Caterpillar:
*301.6C, 301.8C Mini Hydraulic Excavators* ©Caterpillar 2006

## Operator Station

*Class leading space and high feature levels lead to a comfortable and productive environment.*

**The large, spacious cab** of the Caterpillar C-Series offers excellent all round visibility and flat glass throughout for easy and inexpensive replacement.

**Superior operator legroom** and door width offer even greater comfort and ease of use.

**'Eye line' instrumentation** provides the operator with a clear view of the machines monitoring system, ensuring fuel level, water temperature, and warning lights are easily seen. A service interval indicator is also included.

**Dozer lever mounted two speed travel control** provides easy operation and increased foot room.

**Dozer float** enables easy ground levelling for landscaping and finishing.

**Adjustable wrist supports** increase operator comfort and reduce fatigue.

**Automatically applied swing park brake** which aids machine transportation.


## Work Tools

*Caterpillar's range of buckets and hydraulically powered tools, matched to optimize machine performance.*

**Caterpillar offers a wide range of work tools** to increase the machines versatility even further. These tools are designed to get the best out of the machine and deliver excellent value through high productivity and long life.

Tools on offer include:
- Digging Buckets
- Ditch Cleaning Buckets
- Hydraulic Hammers
- Tilt Ditch Bucket
- Augers

**Mechanical Quick Coupler.**
The Caterpillar Quick Coupler makes tool changes fast and easy. Maximizing performance and compatible with all standard work tools.






## Serviceability

*Designed to keep 'down-time' to a minimum, with easy access and convenient locations for all daily checks.*

**A large 'swing out' rear door** allows access to major components and service points within the engine enclosure.
- Air filter
- Engine oil check and fill
- Engine oil filter element
- Radiator reservoir tank, check and fill
- Vertically mounted, spin-on fuel filter/water separator
- Windshield washer tank

**A sight gauge** for hydraulic oil makes fluid level checks convenient.

**S•O•S oil sampling valve** allows easy sampling of the hydraulic fluid for preventative maintenance.

**500 hour oil and filter change period** reduces operating costs and machine down time.

**Clean sided front linkage** with concealed hosing reduces damage and machine down time.

## Customer Support

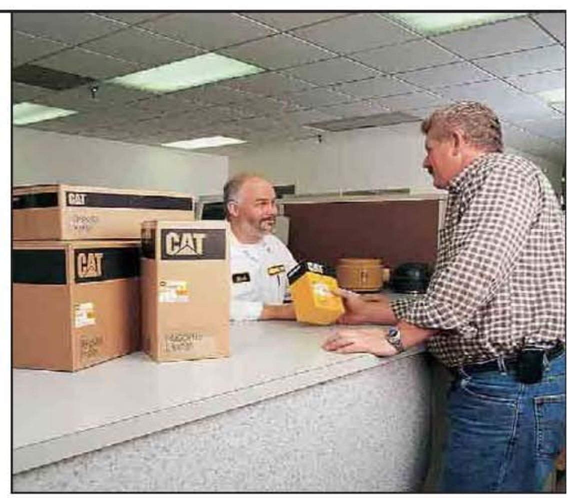
*Complete services provided by the world's largest dealer network.*

**Unmatched dealer support** with excellent equipment management services and fast parts availability – most within 24 hours – provide maximum uptime and minimum repair costs.

**Equipment maintenance services** to help optimize machine performance and reliability. Services include:
- Customer Support Agreements
- S•O•S oil analysis
- Maintenance contracts
- Scheduled technical inspections

**Equipment management services** to help optimize profits. Services include:
- Machine and work tool selection
- Rental and leasing
- Purchasing and financing
- Owning and operating cost management
- Extended warranties

For a complete explanation of these services please call your local Caterpillar dealer.



## Engine

Mitsubishi L3E.Naturally aspirated, watercooled, 4 stroke, 3 cylinder diesel engine.

| Ratings at 2400 rpm | | |
|---|---|---|
| Gross power | 14.0 kW | 19.0 hp |
| Net power | 13.5 kW | 18.4 hp |

**Dimensions**

| | |
|---|---|
| Bore | 76 mm |
| Stroke | 70 mm |
| Displacement | 952 $cm^3$ |

- All engine horsepower (hp) are metric including front page.
- Net power rating per ISO 9249 and EEC 80/1269
- Meets EU Stage II emission requirements

## Hydraulic System

**Pumps/Pressures/Circuits**

| | |
|---|---|
| Pumps: Two piston, one gear-type (max. delivery) | |
| Tandem | 2 x 19.8 lpm |
| Single | 1 x 14.6 lpm |
| Operating pressures | |
| Equipment | 206 bar |
| Travel | 206 bar |
| Swing | 174 bar |
| Auxiliary circuits | |
| Primary | 34.4 lpm at 174 bar |
| Secondary | 14.6 lpm at 174 bar |

**Digging Forces**

| | |
|---|---|
| Stick, standard | 9.9 kN |
| Stick, long | 8.8 kN |
| Bucket | 15.4 kN |

- Auxiliary lines for hammer operation are standard
- Two-way auxiliary lines for work tools such as augers and tilt ditch bucket are optional

## Swing System

| | |
|---|---|
| Machine swing speed | 10.5 rpm |
| Boom swing system with cast swing post | |
| Left (without stop) | 90° |
| Left (with stop) | 54° |
| Right | 50° |

- Automatic swing brake, spring applied, hydraulic release
- Centralized lubrication

## Service Refill Capacities

| | Liters |
|---|---|
| Cooling system | 4.0 |
| Engine oil | 4.1 |
| Fuel tank | 22 |
| Hydraulic tank | 24 |
| Hydraulic system | 37 |

## Sound Levels

Operator sound pressure level is 83 dB(A) for cab and 83 dB(A) for canopy builds when measured per ISO 6396 (dynamic).

## Electrical System

- 40-amp alternator
- 12-volt, 500 CCA, 35 amp/h maintenance free battery
- Sealed electrical connectors

## Weights*

With bucket, operator, full fuel and auxiliary lines.

| | 301.6C | 301.8C |
|---|---|---|
| Canopy (kg) | 1610 | 1680 |
| Cab (kg) | 1720 | 1785 |

* Weight depends on machine configuration.

## Undercarriage

H-shaped, fabricated frame.

- Fabricated design gives high durability
- Tapered roller frame reduces accumulation of material in the tracks
- Track tension adjustment is accomplished through grease-filled cylinders
- 301.8C variable width undercarriage for increased stability

## Blade

| | 301.6C | 301.8C |
|---|---|---|
| Width | mm | mm |
| Standard | 980 | 980 |
| Extended | – | 1340 |
| Height of blade | 225 | 255 |
| Dig depth | 230 | 250 |
| Lift height | 180 | 195 |

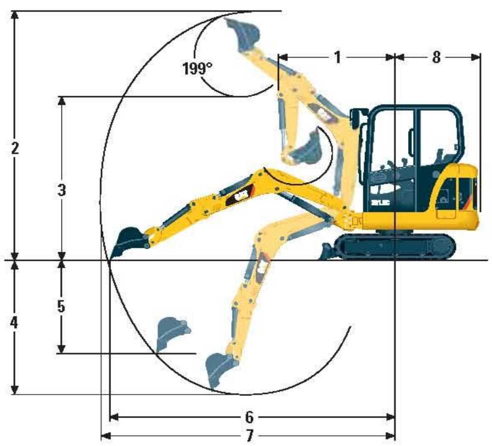
- Blade flotation function provides easier operation
- Replaceable, hardened, wear resistant cutting edge
- 301.8C has pin-on blade extensions to match width to extended undercarriage

## Travel System

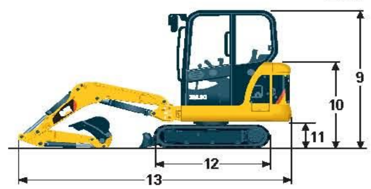
| | |
|---|---|
| Travel speed | |
| High | 4.5 km/h |
| Low | 2.0 km/h |
| Traction force (max.) | |
| High Speed | 7 kN |
| Low Speed | 15 kN |
| Gradeability (max.) | 30° |

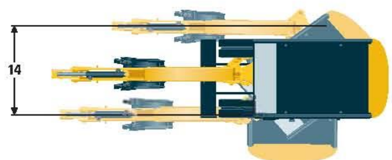
- Each track is driven by one independent 2-speed motor
- Drive modules are integrated into the roller frame for total protection
- Straight line travel when tracking and operating the front linkage simultaneously

## Dimensions and Lift Capacities




| | Standard Undercarriage mm | Extended mm |
|---|---|---|
| Track and blade width | 980 | 1340 |
| Gauge width | 750 | 1110 |
| Shoe width | 230 | 230 |

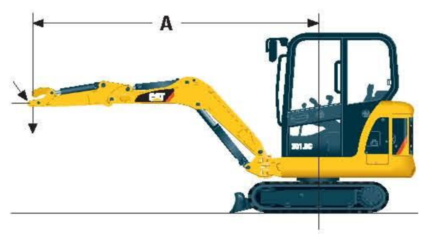



| | Standard Stick mm | Long mm |
|---|---|---|
| **1** | 1550 | 1560 |
| **2** | 3350 | 3460 |
| **3** | 2360 | 2480 |
| **4** | 2080 | 2320 |
| **5** | 1620 | 1810 |
| **6** | 3610 | 3800 |
| **7** | 3700 | 3890 |
| **8** | 1090 | 1090 |
| **9** | 2190 | 2190 |
| **10** | 1365 | 1365 |
| **11** | 435 | 435 |
| **12** | 1575 | 1575 |
| **13** | 3695 | 3695 |
| **14** | 975 | 975 |

With tracks level on the ground

| **Lift Capacities at Ground Level** | Over front | Over side | Over front | Over side |
|---|---|---|---|---|
| **A** Lift Point Radius (mm) | 2000 | | 3000 | |
| Standard Undercarriage | | | | |
| Blade down (kg) | *787 | 362 | *450 | 219 |
| Blade up (kg) | 515 | 329 | 300 | 201 |
| Extended Undercarriage (301.8 only) | | | | |
| Blade down (kg) | *782 | 580 | *445 | 334 |
| Blade up (kg) | 524 | 536 | 303 | 311 |




* Limited by hydraulic capacity rather than tipping load.
The above loads are in compliance with SAE hydraulic excavator lift capacity rating standard SAE J1097 and they do not exceed 87% of hydraulic lifting capacity or 75% of tipping capacity.
The excavator bucket weight is not included on this chart. Lifting capacities for standard stick.

## Standard Equipment

Standard equipment may vary.
Consult your Caterpillar dealer for specifics.

- Adjustable wrist rests
- Alternator
- Automatic swing parking brake
- Auxiliary hydraulic circuit valve, controls and 'one way' (hammer) lines to the stick
- Auxiliary line quick couplers
- Boom cylinder guard
- Cab mounted work light
- Canopy, FOPS ISO 10262 (Level I) and TOPS ISO 12117
- Coat hook and cup holder
- Dozer blade with float function
- Floor mat
- Gauges or indicators for fuel level, engine coolant temperature, hour meter, engine oil pressure, air cleaner, alternator and glow plugs, service interval
- Horn
- Hydraulic oil cooler
- Lockable storage box
- Low maintenance linkage pin joints
- Maintenance-free battery
- Mobile phone holder and power point
- Rubber track, 230 mm wide
- Seatbelt, 50 mm
- Standard stick
- Suspension seat, vinyl covered
- Travel pedals
- Two-speed travel (dozer lever mounted)

## Optional Equipment

Optional equipment may vary.
Consult your Caterpillar dealer for specifics.

- 'Two way' (auger) auxiliary lines to the stick
- Biodegradable hydraulic oil
- Boom check valve
- Boom mounted light
- Cab, fully glazed with FOPS per ISO 10262 (Level 1), TOPS per ISO 12117, heater/defroster, interior light, windshield wiper and washer
- Control pattern changer
- Ecology drain valve for hydraulic tank
- Ecology drain valve for engine oil
- Fan and sun blind installation
- Front screen guard for cab and canopy
- Immobilizer system
- Long stick
- Mechanical quick coupler
- Mirrors, for cab and canopy
- Radio/CD player
- Radio installation kit
- Seat belt, 75 mm wide
- Suspension seat, fabric covered
  - Standard
  - High back
- 'Thumb ready' stick
- Tool kit
- Travel alarm
- Work tools

## 301.6C and 301.8C Mini Hydraulic Excavators

For more complete information on Cat products, dealer services, and industry solutions, visit us on the web at www.cat.com

Materials and specifications are subject to change without notice. Featured machines in photos may include additional equipment. See your Caterpillar dealer for available options.

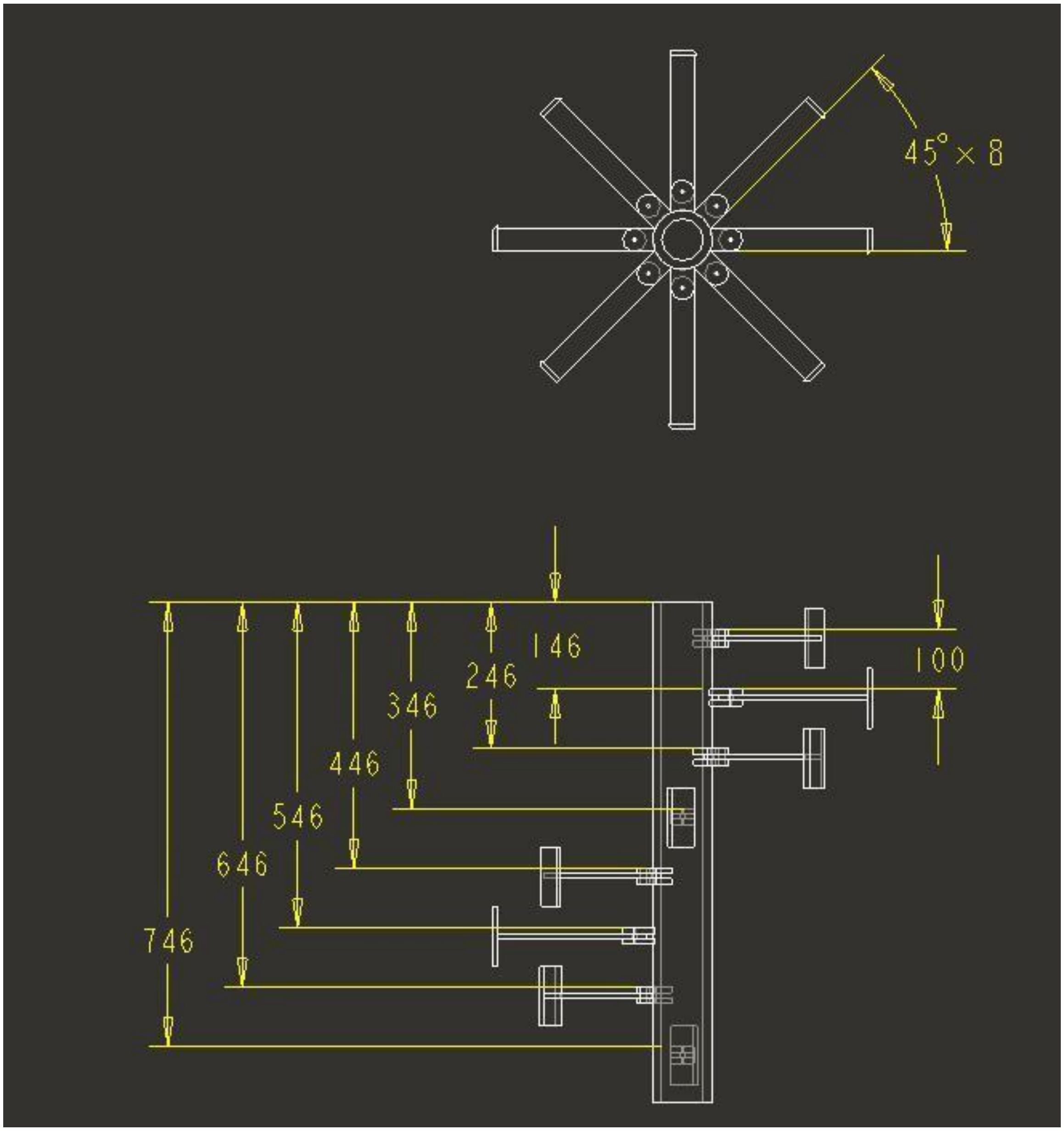


HEHH3291-1 (12/2006) hr

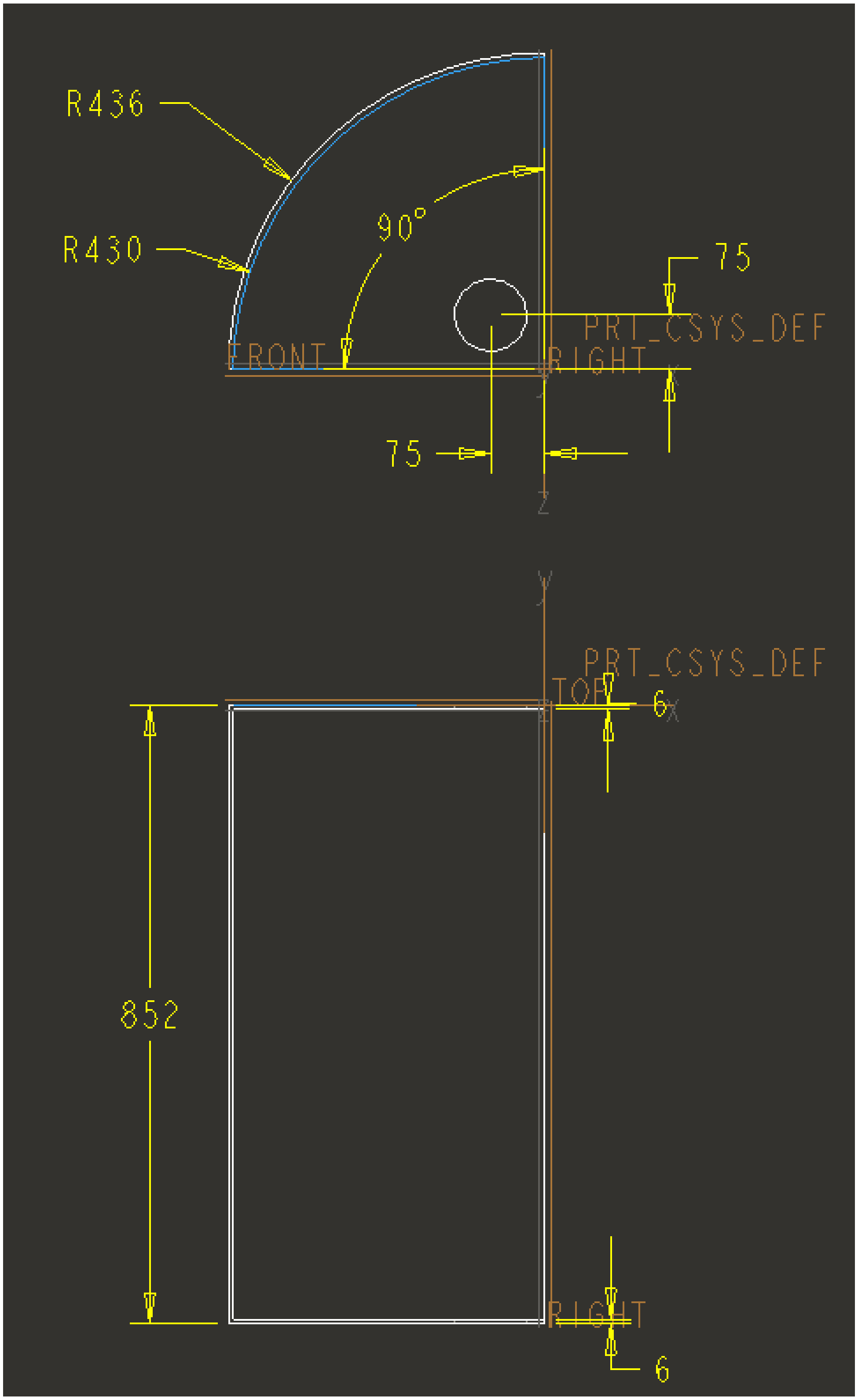
CATERPILLAR®

## Annex 2: Simple blueprints

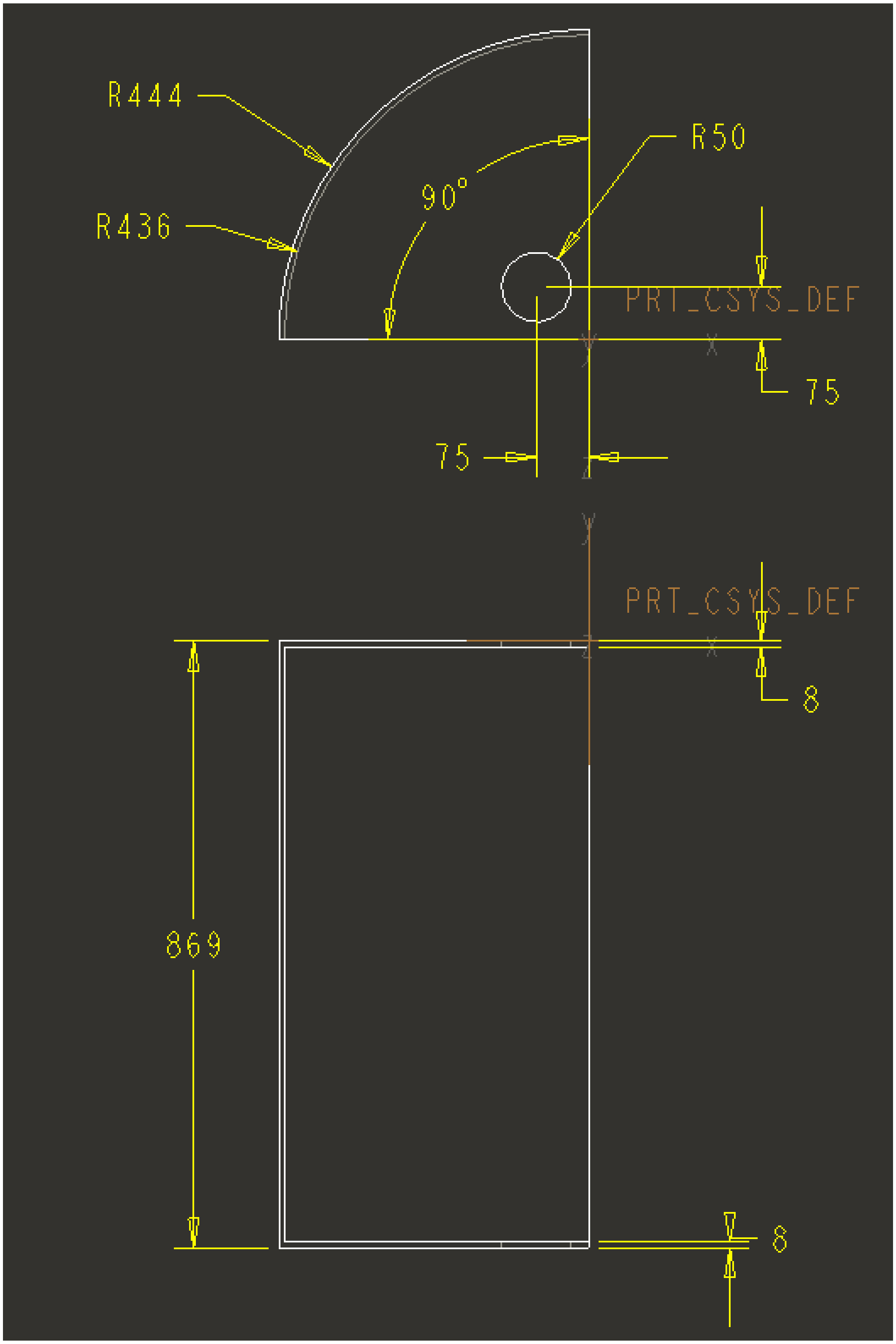
Dimensions in mm.



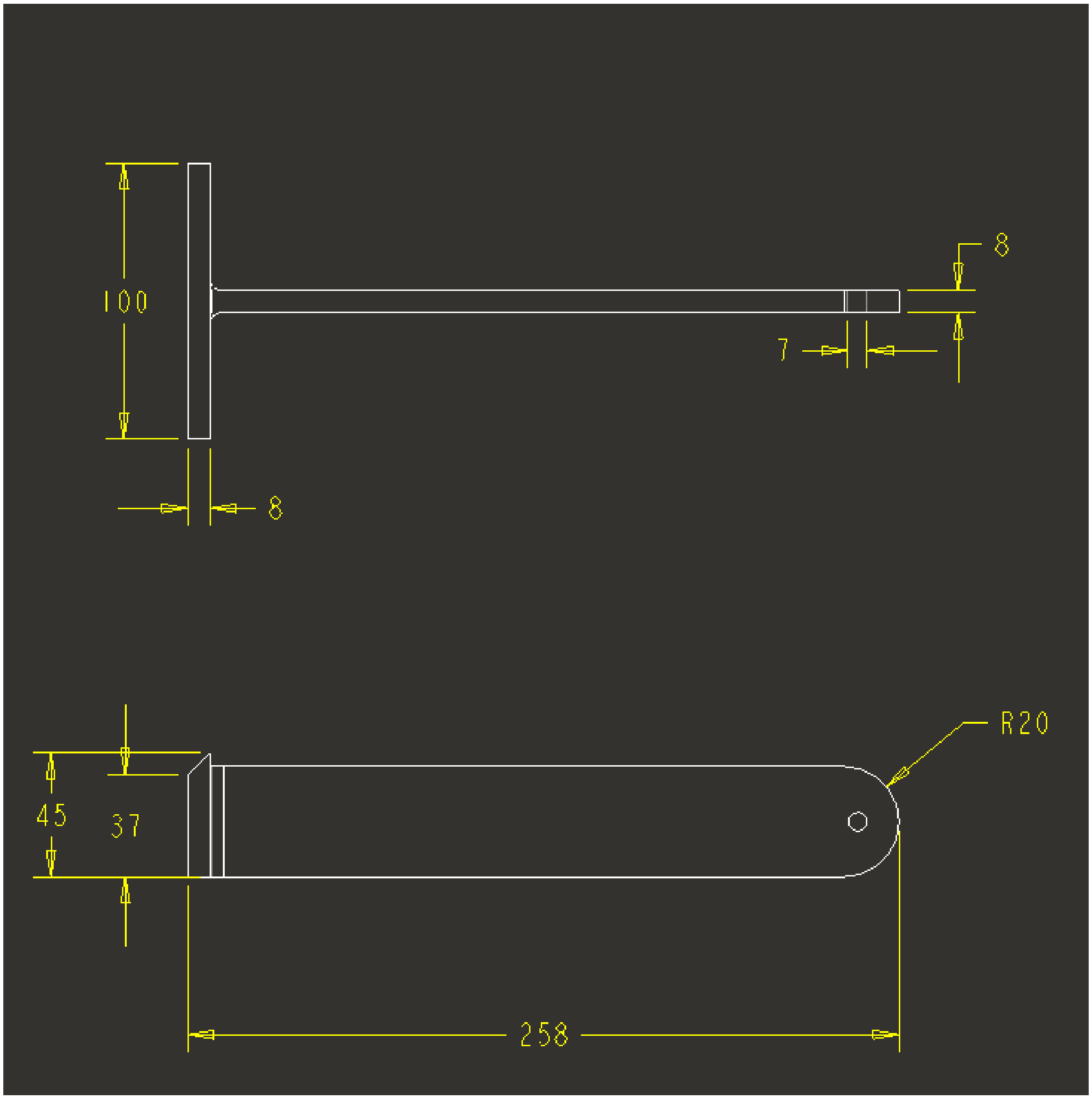

*A. Blueprint of rotor*



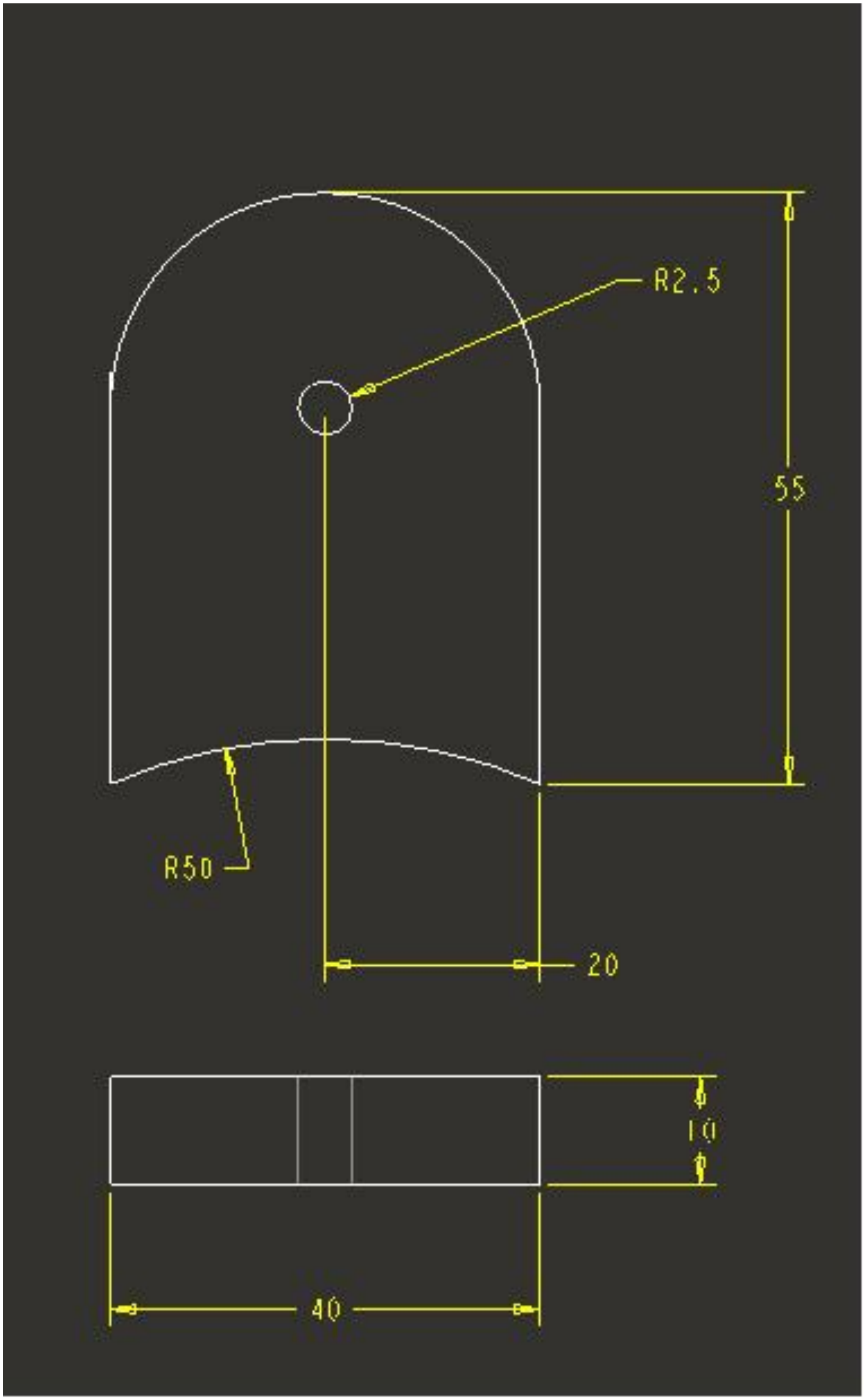

*B. Blueprint of inner cover*




*C. Blueprint of outer cover*




*D. Blueprint of knife and blade*

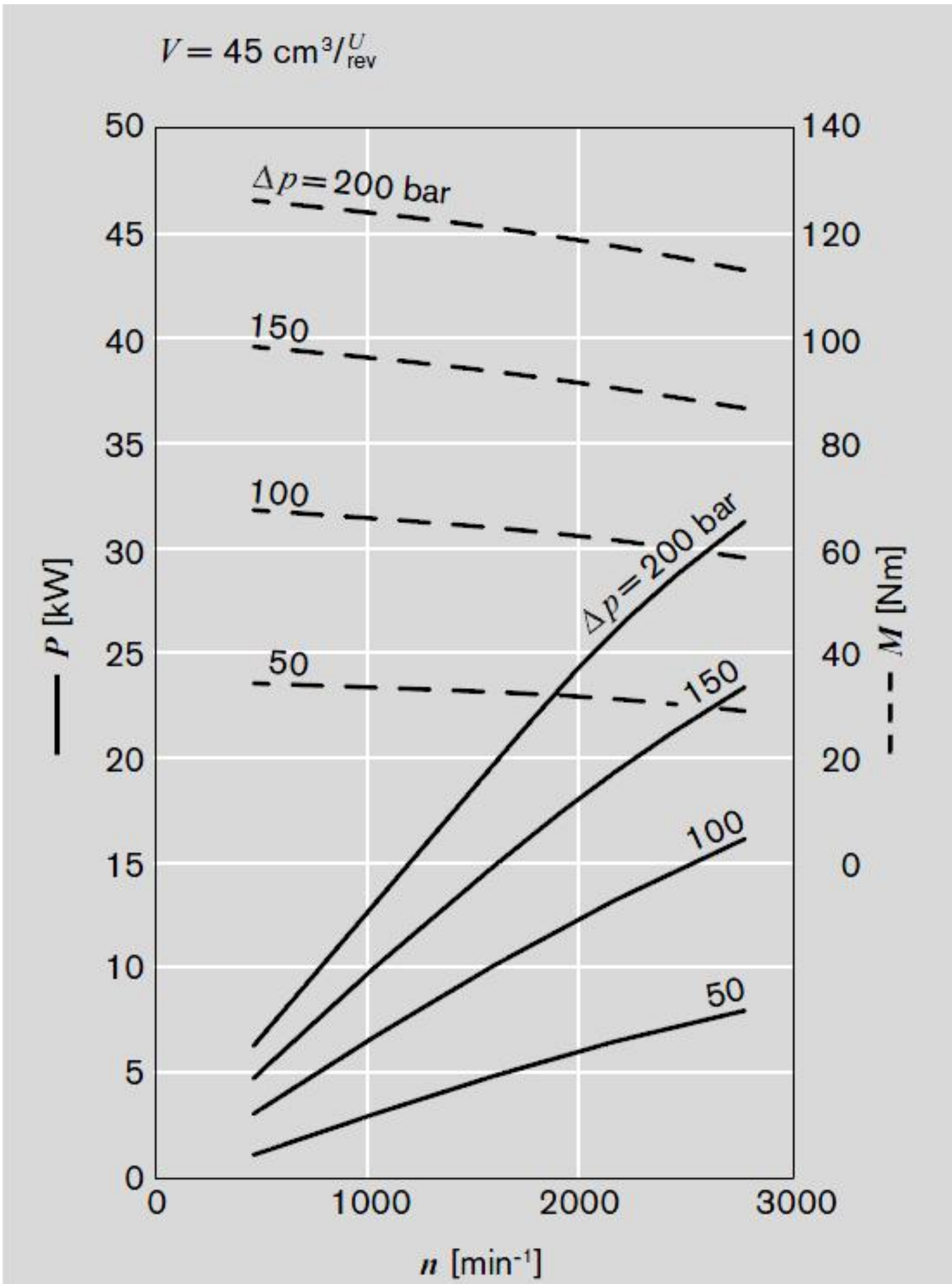



*E. Blueprint of bracket*

## Annex 3: Hydraulic motor torque diagram

Diagram showing torque $M$ delivered at a given pressure $\Delta p$ and angular frequency $n$ for the Bosch Rexroth external hydraulic gear motor model G with a displacement $V$ of 45 cm³/rev.

Adapted from the catalogue *External Gear Motors RE 14 026/05.09*. ©Bosch Rexroth AG 2009.



# Annex 4: Material properties of Hardox 450©

HARDOX® WEAR PLATE

Version 2008-05-10
Sid. 1 (2)

## Data Sheet: HARDOX 450

### ABRASION RESISTANT PLATE

HARDOX 450 is an abrasion resistant plate with a hardness of 450 HBW, intended for applications where demands are imposed on abrasion resistance in combination with good cold bending properties. HARDOX 450 offers very good weldability.

**Applications**

Dumper bodies, containers, crushers, sievers, feeders, measuring pockets, skips, cutting edges, conveyors, buckets, knives, gears, sprockets, etc.

**Chemical Composition** (ladle analysis)

| Plate thickness mm | C max % | Si max % | Mn max % | P max % | S max % | Cr max % | Ni max % | Mo max % | B max % | CEV typv. | CET typv. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 3*) - (8) | 0,21 | 0,70 | 1,60 | 0,025 | 0,010 | 0,25 | 0,25 | 0,25 | 0,004 | 0,41 | 0,30 |
| 8 - 20 | 0,21 | 0,70 | 1,60 | 0,025 | 0,010 | 0,50 | 0,25 | 0,25 | 0,004 | 0,47 | 0,34 |
| (20) - 40 | 0,23 | 0,70 | 1,60 | 0,025 | 0,010 | 1,00 | 0,25 | 0,25 | 0,004 | 0,57 | 0,37 |
| (40) - 50 | 0,23 | 0,70 | 1,60 | 0,025 | 0,010 | 1,40 | 0,25 | 0,60 | 0,004 | 0,59 | 0,36 |
| (50) - 80 | 0,26 | 0,70 | 1,60 | 0,025 | 0,010 | 1,40 | 1,00 | 0,60 | 0,004 | 0,72 | 0,41 |

*) Plate thickness below 4 mm only after special agreement.

$$CEV = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Cu + Ni}{15}$$

$$CET = C + \frac{Mn + Mo}{10} + \frac{Cr + Cu}{20} + \frac{Ni}{40}$$

The steel is grain refined.

**Hardness**

HBW
425-475

**Mechanical Properties**
Typical values for 20 mm plate thickness

| Yield strength $R_e$ MPa | Tensile strength $R_m$ MPa | Elongation $A_5$ % |
|---|---|---|
| 1200 | 1400 | 10 |

**Impact Properties**
Typical value for 20 mm plate thickness

| Test temperature °C | Impact energy Charpy-V, longitudinal J |
|---|---|
| -40 (40 F) | 40 |

**Testing**

Brinell hardness, HBW according to EN ISO 6506-1, on a milled surface 0,5-2 mm below plate surface per heat and 40 tons. Tests are made for every variation of 15 mm in the thickness of plates from the same heat.

**Delivery Conditions**

Q.

**SSAB Oxelösund AB, 613 80 Oxelösund, Sweden,**
**+46 155 25 40 00, www.hardox.com, www.ssabox.com**



SSAB OXELÖSUND