Universidad de Valladolid

ESCUELA DE INGENIERÍAS INDUSTRIALES

UNIVERSIDAD DE VALLADOLID

ESCUELA DE INGENIERIAS INDUSTRIALES

Grado en Ingeniería Mecánica

Diseño de un triturador de basura para conectar a una máquina compactadora de residuos

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TFG REALIZADO EN PROGRAMA DE INTERCAMBIO

TÍTULO: Design of a shredder attachment for a waste compactor

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Autor: **Sara Escudero**

Supervisor en la Universidad de destino: **Andrius Gedvila**

Idioma

X	Spanish
	Foreign (English)

Resumen

En este trabajo fin de grado se ha diseñado un triturador de basura para conectar a una máquina compactadora de residuos. El objetivo final de esta máquina novel es mejorar las operaciones en los vertederos a través del ahorro de espacio y la generación de más energía. Este objetivo se alcanza triturando la basura antes de compactarla, ambas operaciones con la misma máquina: compactador de residuos.

El dispositivo consta de dos trituradores hidráulicos conectados por una caja central que aloja dos motores radiales de pistones. Cada triturador es accionado por su propio motor y consta de dos ejes conectados por una trasmisión de engranajes.

La parte teórica presenta las características de los vertederos controlados, los compactadores de residuos y los trituradores de residuos hidráulicos. También explica las ventajas y desventajas de esta nueva tecnología, el proceso tecnológico, y los requerimientos de seguridad y medioambientales.

La parte de cálculo se centra en la carga que soporta la máquina y la transmisión de potencia, incluyendo engranajes, diseño de ejes, selección de rodamientos, chavetas, anillos de retención, motor y acoplamientos. Finalmente, se incluye el cálculo económico del dispositivo.

Tamaño del trabajo – 62 pág. De texto, 16 tablas, 25 fotos and 36 referencias.

Palabras clave: Vertedero, LFG, compactador de residuos, triturador de residuos hidráulico

VILNIUS GEDIMINAS TECHNICAL UNIVERSITY

FACULTY OF MECHANICS

DEPARTMENT OF MECHANICAL ENGINEERING

Sara, Escudero

DESIGN OF A SHREDDER ATTACHMENT FOR A WASTE COMPACTOR

Bachelor‘s degree final work (project)

Mechanical Engineering study programme, state code 612H33001

Machine Design specialisation

Mechanical Engineering study field

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VILNIUS GEDIMINAS TECHNICAL UNIVERSITY

FACULTY OF MECHANICS

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____2015______________

(Date)

Sara, Escudero

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Vilnius, 2015

(the document of Declaration of Authorship in the Final Degree Project)

VILNIUS GEDIMINAS TECHNICAL UNIVERSITY

Sara, Escudero, 20150117

Faculty of Mechanics

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DECLARATION OF AUTHORSHIP
IN THE FINAL DEGREE PROJECT

2015-06-04
(Date)

I declare that my Final Degree Project entitled "Design of a shredder attachment for a waste compactor" is entirely my own work. The title was confirmed on 2015-05-28 by Faculty Dean's order No. 114 me. I have clearly signalled the presence of quoted or paraphrased material and referenced all sources.

I have acknowledged appropriately any assistance I have received by the following professionals/advisers: Andrius Gedvila.

The academic supervisor of my Final Degree Project is Andrius Gedvila.

No contribution of any other person was obtained, nor did I buy my Final Degree Project.

SARA, ESCUDERO

(Signature) (Given name, family name)

Vilniaus Gedimino technikos universitetas

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Bachelor Degree Studies **Mechanics Engineering** study programme Bachelor Graduation Thesis 3

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Author: **Sara Escudero**

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Abstract

In this bachelor's degree final work was design a shredder attachment for a landfill compactor machine. The final goal of this new machine is improving landfill operation by saving space and generating more energy. This is fulfilled by shredding solid waste before compacting it, both operations with the same equipment: waste compactor.

The device consists of two hydraulic shredders connected by a central box that hold in two radial piston hydraulic motors. Each shredder is motioned by its own motor and consists of shafts connected by gear transmission.

Theoretical part reviews the controlled/sanitary landfills, waste compactors and hydraulic waste shredders characteristics. Besides, it is explained the advantages and disadvantages of the new technology, the technological process, and safety and environmental requirements.

Calculations part focus on the load supported and power transmission, including gears shafts design, selection of bearings, keys, retaining rings, motor and coupling. And economic calculations of the device.

Size of job – 62 pages of text, 16 tables, 25 pictures and 36 literature sources used.

Keywords: landfill, LFG, waste compactor, hydraulic shredder

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Introduction

[References: 3]

Nowadays solid waste management is a major issue in every developing country due to waste generation is increasing each year. Now we generate more than 2 billion tons of waste each year, and if we continue with the same waste generation pattern, by 2025 we could generate about 7 billion tons of waste. The major problem with this increasing amount of waste is the pollution that it generates which damages our environmental system.

For these reasons Governments are trying to stimulate good practices towards the waste management. The following graphic (figure 0-1) shows which the preferred methods in waste management are:

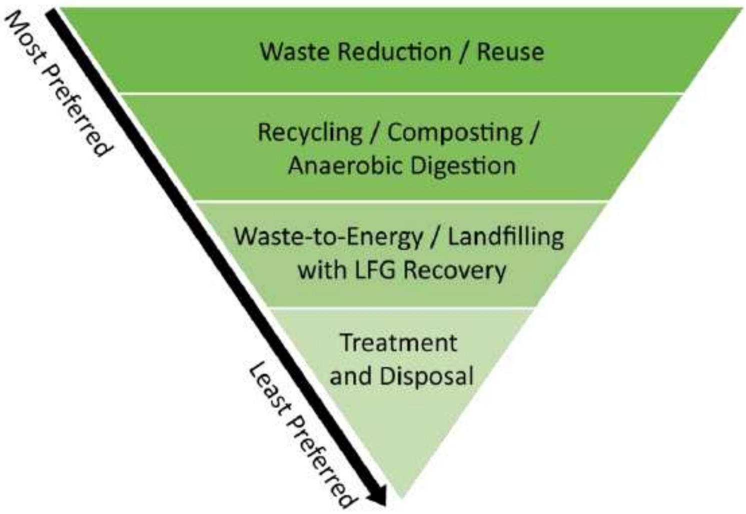


Figure 0-1: Preferred methods of waste management[3]

In this days only reuse and recycle are not enough for all the solid waste generation and the majority of solid waste still goes to landfills, therefore it is essential to improve techniques for the final disposal of waste. For instance, landfill gas (LFG) energy recovery is a technique that various countries are implementing. This system enables transforming an open dump into a controlled or sanitary landfill where gas energy can be collected and the risks to the environment are minimized.

1. Reference review and grounding of taken decisions

1.1. Controlled landfill

[References 3]

Controlled or sanitary landfills are the modern solution for the final disposal of waste. They have a lot of advantages against the open dumps; even they still are environmentally harmful. The following table (table 1-1) shows the differences of a controlled landfill against an open dump:

Table 1.1-1. Comparison of solid waste disposal sites[3]

Factor	Open dump	Controlled landfill
	Environmental factors	
Fires	Intentional burning common	Unlikely
Release of hazardous gases	Yes, if no collection exist	Yes, if no collection exist
LFG collection and control	Possible, poor collection efficiency expected	Likely
Unpleasant odours	yes	Minimal, if the right measures are taken to cover waste and control LFG
	Ground/Soil	
Topographical modification	Yes	Yes
Contamination (leachate)	Yes	Unlikely, depending on base conditions
	Water (surface and ground water)	
Channeling run off	No	Likely, depending on site conditions
Contamination	Likely	Minimal
	Flora and fauna	
Vegetative cover alteration	Yes	Yes
Changes in diversity	Yes	Yes
	Humans	
Health hazards	Yes	Potentially, depending on site conditions
Waste pickers	Yes	Unlikely
	Economics	
Decline of land value	Yes	Yes
Formal employment	No	Yes
Changes in land use	Yes	Yes

However, controlled landfills and sanitary landfills are more expensive than an open dump and also require expert design, skilled operators, and a proper management to guarantee its functionality.

The difference between the sanitary landfill and the controlled landfill is that the first one has a closer monitoring system. As a result, it is more expensive but also more efficient.

The working principle of a controlled landfill (figure1-1) is burying the solid waste, previously compacted, in layers in a lined and sealed pit (to prevent contamination of groundwater) and covering it with a plastic cover. In addition, at the end of each working day the trash must be covered with soil to prevent animals from digging up the waste and to avoid that odour waste and pathogens are spread by the wind.

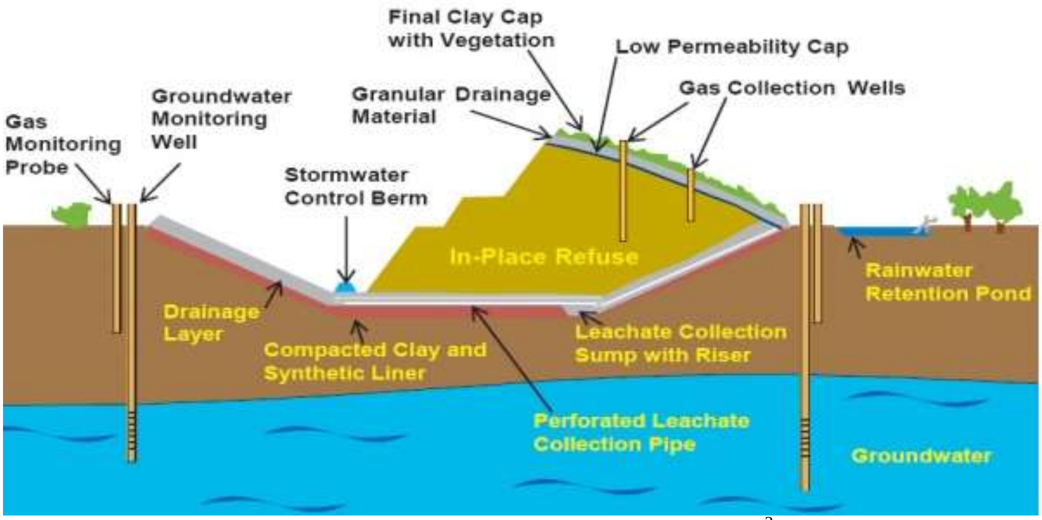


Figure 1.1-1: Elements in controlled landfills[3]

The area where the waste is being deposited, spread and compacted is known as the working face. I should be narrow enough so the waste can be compacted and covered rapidly, minimizing water infiltration, blowing litter and odours. The working face also should be gently sloped to prevent from landslides.

The waste is deposited in layers with 2 to 5 meters height (figure 1-2) because these heights prevent from landslides and also facilitate efficient compaction. The total depth of the pit must be more than 10 meters because it helps to a faster gas generation.

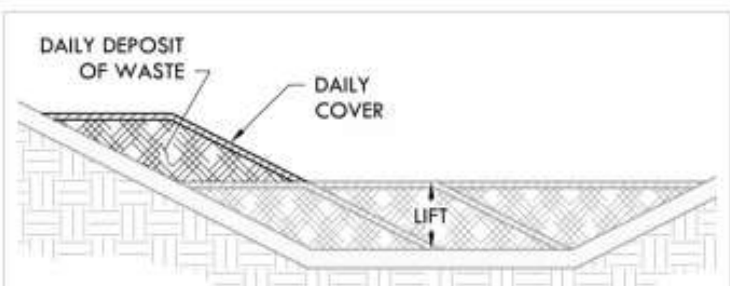


Figure 1.1-2: Layers in controlled landfills[3]

The following table (table 1-2) shows the impact that the different conditions of the sanitary landfill have on the gas recollection:

Table 1.1-2. Conditions that impact LFG Project Development[3]

Component	As Found Condition	LFG Generation	Amount of Methane in LFG	Collection Efficiency
Bottom Liner	None or Inadequate	No Impact	No Impact	Decreases
	Adequate	No Impact	No Impact	Increases
Leachate Collection and Removal System	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Final Capping	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Planned Filling Operations	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Compaction	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Daily and or Intermediate Cover	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Slopes	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases
Fire Control	None or Inadequate	Decreases	Decreases	Decreases
	Adequate	Increases	Increases	Increases

The density of the waste achieved by compaction has an effect on the potential gas energy that can be generated: the more waste mass, the more gas can be generated. Therefore, the more compaction, the more waste mass can be in the same volume. In addition, waste compaction also increases the anaerobic conditions necessary for the gas generation because it reduces the air pockets within the waste mass. Finally, waste compaction also limits the spread of fires.

1.2. Landfill compactor

[References: 3, 4, 5]

Nowadays waste compaction in landfills is executed by landfill compactors (figure 1.2-1). This equipment distributes and compact garbage on landfill sites (figure 1.2-2) to optimize site capacity, to reduce gas emissions and to stabilize the deposited material. The high operating weight of the machine combined with specially developed steel wheels crushes, kneads and compacts the garbage. From 3 to 5 passes of the compactor are needed in order to get an effective compaction of the garbage placed in layers of 0.3-0.5 meters.

Figure 1.2-1: Landfill compactor[5]

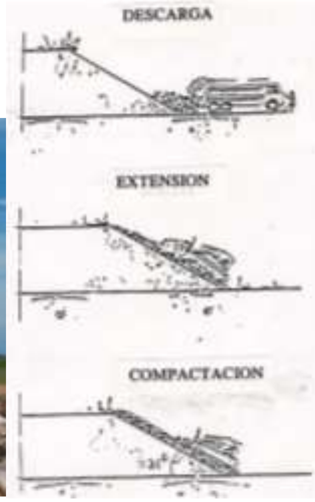


Figure 1.2-2: Download, distribution and compaction of garbage by landfill compactors[4]

The compactor design consists of front and rear frames connected by an oscillating/articulating joint. This provides even distribution weight and thus optimal traction. The fully enclosed frame prevents the ingress of unwanted material, and offers all drive components full protection.

Landfill compactors usually work with hydrostatic engines due to the high forces they have to perform and resist. In addition, the hydrostatic drive provides outstanding pushing power for dozing and distributing all types of waste.

In order to get a most efiicient compaction, garbage could be cut in small pieces before compacting it, reducing the air pockets in the whole volume. This could be done by a waste shredder attached in front of the landfill compactor.

1.3. Waste shredder

[References: 5, 6, 7, 8, 9, 10]

A waste shredder is a machine used for reducing the size of all kinds of material. Tearing material to reduce it to smaller particles depends on three actions:

Shearing: This action involves the actual cutting of material. As in scissors, shearing efficiency depends on the sharpness of the cutting edges working against each other and the tolerance of the space between them.

Tearing: This action involves pulling the material with such force that it comes apart. Some materials like fabric, soft metals, plastics, and tires, are more tearable than others.

Fracturing: Some materials are brittle, such as glass, hard plastics, and certain metals, and tend to be broken or shattered in a shredder when the cutters aren't sharp or are loose. Unlike tearing, when something breaks it releases energy explosively, sometimes propelling the shards upwards into the faces of the fascinated onlookers. Always wear eye protection.

Optimum Action: All three actions, shearing, tearing, and breaking are present when a shredder is being used. However, when the cutters are kept sharp and the tolerances tight, the dominant and most efficient reduction action should be shearing.

Shredders can be configured for each unique shredding application by choosing different number of rotating shafts (figure1.3-1), knife widths and knife hook design (figure 1.3-2) to achieve the exact shred size and production rate needed.

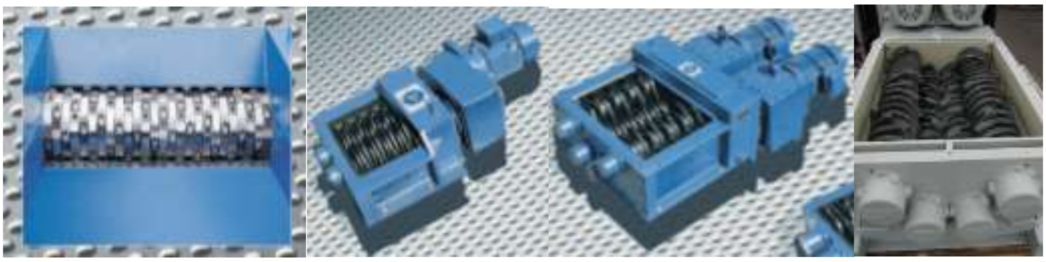


Figure1.3-1: Shredders with different number of rotating shafts[7 y 8]

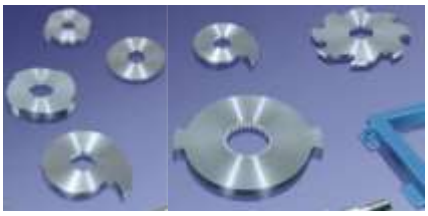


Figure 1.3-2: Blade shapes[7]

Blades can be integral (figure 1.3-2) or with inserts (figure 1.3-3). This last system is more economic because there is no need to replace the whole shaft, only the inserts. In addition, inserts can be used 4 times, each time with one corner. Even so, I am going to design a shredder with integral blades because of the modeling difficulty that the other solution presents. In any case, it could be done in a further studio of this project, since it is necessary only to change de design of the shaft and recalculate the loads for bearings, gears and motor.

Figure 1.3-3: Blades with inserts[9]

A common industrial shredder is compound of the cutting chamber, the motor and the feed hopper (figure 1.3-4). Material is fed through the hopper into the cutting chamber. Then, the material is shredded between the rotor knives and a fixed counterknife.

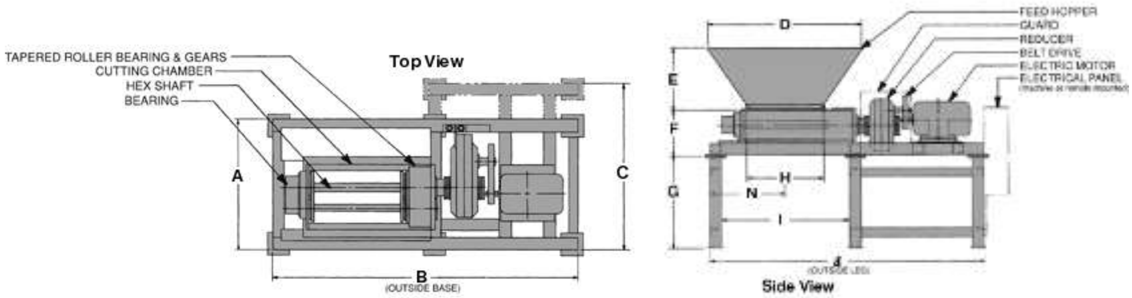


Figure 1.3-4: Common industrial shredder configuration[10]

Contrary to this design, the shredder purpose of this project has not feed hopper and the cutting chamber is vertical instead of horizontal, shredding the garbage that come up the front of the compactor equipment (Figure 1.3-5).

Figure 1.3-5: Landfill compactor equipped with a shredder attachment[5]

1.4. Search for analogues machines

[References: 11, 12, 13, 14]

In the current market it is not possible to find a machine equal as the one I propose in this project. Similar machines that I found are the followings:

- Vegetation cutter (figure 1.4-1): this machine shreds vegetation while it is moving forward. Contrary to my design, it only has one rotating shaft which would be not enough for the hard conditions of landfills.

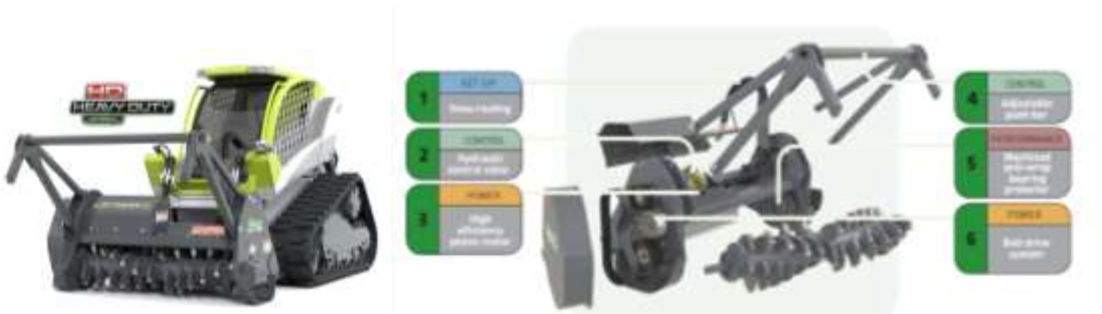


Figure 1.4-1: Vegetation cutter[11]

- Shred trucks (figure 1.4-2): This are trucks used in city garbage recolection. They collect garbage from containers and put it in the feed hopper. Then, they start to move while the shredder is working. Some of them also compact the garbage after shredding it.

Figure 1.4-2: Shred trucks[12 y 13]

- Mobile waste shredder (figure 1.4-3): It is an industrial shredder but portable by trucks. This is also a good alternative for improving landfill operations. Contrary to my design, it only shreds while it is stacionary.

Figure 1.4-3: Mobile waste shredder[14]

1.5. Justification of taking decision

The aim of this project is to add a waste shredder to a landfill compactor machine in order to improve the compaction efficiency in controlled landfills. Improving compaction efficiency means more methane is generated and more space is available.

The waste shredder will consist of an attachment to the compactor machine like dozer blades and buckets. It means the shredder could be joined or split from the machine and change for another attachment. Therefore, the original connections of the compactor must be maintained and the shredder connections must fit the compactors connections (Figure 1.5-1). In addition, it is required to conduct the hydraulic hoses to the hydraulic motor of the shredder and it would be required to calculate the new hydraulic power that the compactor machine must perform; as the 2 shredder motors need power and the weight to lift the attachment is changed.

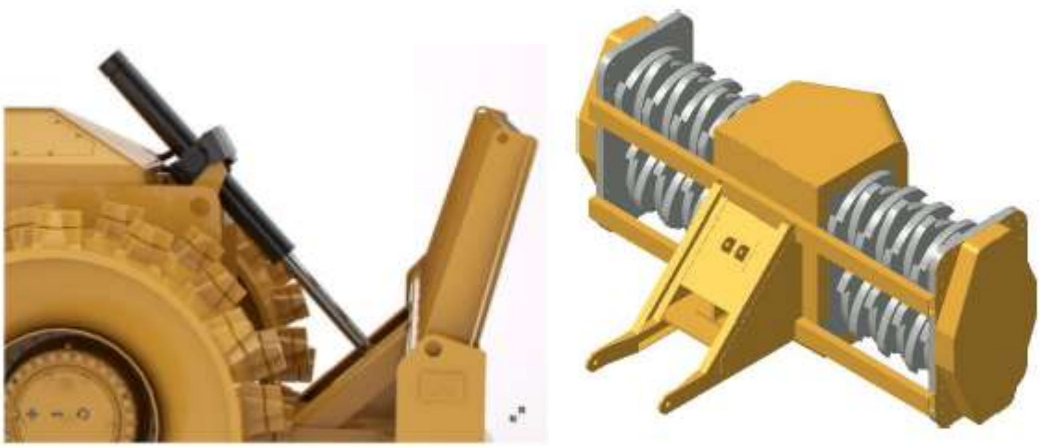


Figure 1.5-1: Landfill compactor connected to blade (left)[5], designed connection (right)

When the shredder is motionless, it works as a blade which pushes the garbage. When it is moving, the garbage in front of the shredder is trapped and cut in small pieces that fall down to the ground and are compacted by the wheels of the compactor equipment.

A slightly difference in the working of the shredder is worth to mention: the waste input will be from the front instead of the top like in the shredders I found in the market. Therefore, the 2 shredder's rotors will be in vertical instead of horizontal.

The shredder will be free from the back, bottom and upper parts to prevent from garbage been trapped and damage the machine. In addition, it will be very important to isolate the machine components, like bearings and gears, from the possible input of garbage. This will be fulfilled by using seals and covers.

The advantages and disadvantages of using this new machine in landfills are the following:

<u>Advantages</u>

- Reduction of the average size of the garbage → Reduction of the amount of air in the considered volume →
    - Methane generation increase.
    - More waste in the same volume → increase of the capacity of the landfill.
- Reduction in time compacting: few passes of the compactor are needed.

<u>Disadvantages</u>

- It requires specific equipment. However, it only would be a new attachment to an existing compactor machine.

2. Construction decisions and calculations

2.1. Compactor selection

[Reference: 5, 15]

In the market there are different companies that produce landfill compactors: CATERPILLAR, TANA, BOMAG, etc. The reference that I chose for this project is the model 826K (figure 2.1-1) from CATERPILLAR company.

Figure 2.1-1: Landfill compactor 826K from CATERPILLAR[5]

The principal characteristics are listed below:

- Principal dimensions:

  Length without any attachment: 7756 mm
  Width: 3800 mm
  High: 4568 mm

- Transmission: Cat Planetary Powershift transmission with Cat Torque Converter that eliminates torque converter losses while lowering system heat, improves travel speeds and increases fuel efficiency. It has 2 forward and 2 reverse speeds. The maximum forward speed is 6 km/h. That is the data that I will use for calculating the dynamic load which is applied to the shredder.

- Engine: Cat C15 ACERT.

  Power: 324 kW
  Maximum torque at 1300 rpm: 2005 Nm

Displacement: 15.2 L

Idle speed (min-max): 800-2300 rpm

- Hydraulic System: It is compound of the hydraulic steering system, the hydraulic brake system and the hydraulic lift cylinder system. To these standard systems it is necessary to add the hydraulic shredder system, which consists of 2 radial piston motors (its specifications are in chapter 2.4). The hydraulic schematic is in drawing MPfu-11.04.002.

  The specifications of the landfill compactor standard hydraulic system are the following:

  Pump flow at 1950 rpm: 117 L/min

  Main relief pressure: 24100 kPa

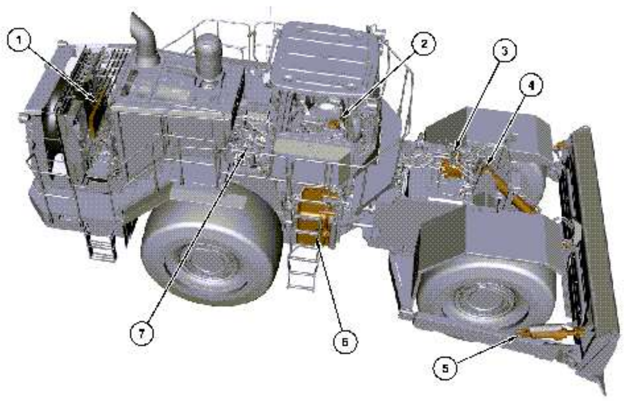
  Maximum supply pressure: 24100 kPa

  Lift system: Double acting cylinder

  Steering system – Circuit: Double acting – End mounted

  Steering system – Pump: Piston – Variable displacement

  The location of the components of the hydraulic system is shown in the following figure (figure 2.1-2):



1. Hydraulic oil cooler
2. Electrohydraulic control
3. Implement Valve
4. Lift cylinder
5. Tilt cylinder (not exist)
6. Hydraulic tank and filters
7. Implement pumps

Figure 2.1-2: Components of the hydraulic system in the landfill compactor[15]

2.2. Shredder design

[Refrences: 6, 10, 16]

The principles for choosing the suitable shredder are[6]:

1. Each type of material is best reduced by a certain type and configuration of shredder:

Different materials have their own physical characteristics which determine how they will react to the reduction process.

Ductile materials are not easily fractured but tend to tear into long strips. They are best reduced by shearing to ensure small particle size. Examples are cloth, rubber, soft plastics, paper, cardboard, or soft metals.

Friable Materials are materials that are easily fractured (the opposite of ductile) or broken into shards. Examples are stone, glass, cast metals, hard plastics, or wood. Shredded friable materials tend to come out as small pieces rather than the long strips.

2. "Grabbing" is the ability of a shredder to seize the material and pull it down into the cutters:

Grabbing is a function of the size and shape of the hook on the cutter and the size, as well as the weight and texture of the material coming into the shredder.

For instance a large, light, smooth object, like a plastic form, may be relatively easy to cut up but have a tendency to bounce or "float" on top of the rotating cutters. In this case it might be necessary to use a larger shredder or add a ram assist to push the material down into the shredder where it can be grabbed and pulled through.

A shredder should not grab more than it can shred. Some compressible materials, like carpeting or paper, can be grabbed too easily and can choke the shredder if too much is grabbed at a time. In such a case, one might reduce the hook height on the cutter so the shredder will only grab as much as it is designed for. One can also reduce the batch size or meter feed the shredder, controlling the rate of material going into the shredding table.

3. Shredder sizing:

The right sized shredder is important. It is recommended that shredders should be sized to need much less than 100% of its available power to do the job.

The reason for this is that as cutters dull with use, they require more power to shred. With an ample reserve power capacity, the performance of a shredder can be maintained for far longer before the cutters need to be replaced.

It is also recommended that shredders should be sized to grab only as much material as it has the power to shred. So besides model size and power capacity, considerations like cutter, shaft dimensions and material properties need to be taken into account.

4. Electric or hydraulic drive:

Electric Drives: Generally, electric shredders require less space, are easier to operate and maintain, and are more energy efficient than their hydraulic counterparts. They also tend to be less expensive. Electric shredders are appropriate and sufficiently powered processing many materials.

Hydraulic Drives: Hydraulic drives are often better for more heavy duty processing. They are also better for processing materials that experience frequent overloads from batch feeding. Hydraulic drives also offer better shockload protection from non-shreddables.

Final shredder selection:

The shredder chosen for this project is a two-shaft hydraulic shredder. The design is based on the model "Dual Shear M100H" from SSI company suitable for shredding very different sort of materials including mixed waste. The comparison of the specifications[16] of this model and the one that I design and calculate are in the following table (table 2.2-1):

Table 2.2-1: Comparison of reference and design models

Specifications	Reference shredder "Dual Shear M100H"	Design shredder
Drive	Hydraulic	Hydraulic
Nr. motors	1	1
Power range	112 - 149 kW	100 - 140 kW
Cutter thickness	50 mm	92.5 mm
Cutter material	4140HT	4140HT
Cutter diameter	530 mm	480 mm
Shaft diameter	188 mm	150 mm
Cutting length	1600 mm	1500 mm

Hydraulic shredder is more suitable for the aim of this project due to the hard work that is going to be put through it. The shredder must push about 17.5 tons of garbage in landfill operations (calculations of this figure are in chapter 2.3).

The shredder shafts will be 2 meters length aprox. (2.06 m for the driven shaft and 2.15 m for the driver shaft). The cutting part will be 1.5 m length with a hexagonal cross-section of 150 mm between faces. However, latest researches show that key shafts are more suitable for the purpose of a shredder because in case of wear it is only necessary to replace the key and not the shaft and the blades. The selected material is steel AISI/SAE 1095 normalized 293HB.

In two-shaft shredders hardened gears are used on each shaft to convey the high torque and low speed into the knives. The selected material is steel AISI/SAE 1050 hardened completely HB217. The gears calculations are in chapter 2.5.

The geometry of the blades is designed for the purpose of this machine: cutting mixed waste. The diameter is 875 mm and the selected material is steel 4140 HT. The geometry is specified in drawing MPfu-11.03.005.

Any overloads caused by foreign material are sensed by the PLC controller and cause the shaft rotation to stop and reverse direction to clear the foreign matter, and then resume. Shock loads caused by normal shredding operation are absorbed by a shock absorber drive train. These two components are not included in my design due to the difficulty that it implies; it could be taken into account in a further studio of this project.

The configuration of the attachment (figure 2.2-1) will consist in 2 identic shredders but mounted inversely. They will be connected by a central box which will hold the 2 motors (one for each shredder).

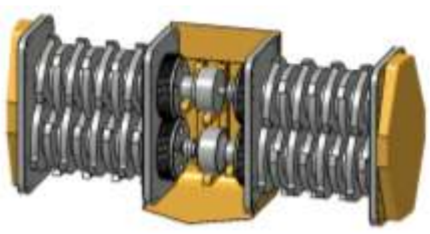


Figure 2.2-1: Shredder design and assembly.

The attachment including the frame will be 5 meters length and 2 m high. These dimensions have been proposed according the size of the possible blades that could be attached to the selected compactor, in particular the semi U-blade[5].

2.3. Load calculation

For estimating the load that the attachment must resist, I am going to calculate de volume of garbage that will be pushed by the front part of the attachment. For that, the necessary parameters are:

Total length of the attachment (l_T): 5 m

High of the attachment (h_T): 2 m

Length of the cutting shaft ($l_{shredder}$): 1.5 m

Diameter of the cutting blades (d_{blades}): 0.48 m

Approximate dimension of garbage being pushed (a): 3 m

The volume in front of the compactor will be:

$$V := l_T \cdot h_T \cdot a = 27\ \boldsymbol{m}^3 \tag{2.3-1}$$

Therefore, if the volume is 27 m^3 and the density of the non-compacted garbage[4] is 650 Kg/m3, we can conclude that the mass of garbage is:

$$m := V \cdot \rho = \left(1.755 \cdot 10^4\right)\ \boldsymbol{kg} \tag{2.3-2}$$

Then, if we want to know the weight of this amount of garbage:

$$W := m \cdot 9.8\ \frac{\boldsymbol{N}}{\boldsymbol{kg}} = 171.99\ \boldsymbol{kN} \tag{2.3-3}$$

We need to know the pushing force. For that, we are going to calculate the friction force. Supposing a friction coefficient of μfr = 1.5 (very rough surfaces).

$$F_r := W \cdot \mu_{fr} = 257.985\ \boldsymbol{kN} \tag{2.3-4}$$

This force is the static load. But, as the compactor is moving forward, this load is not going to be static. For this reason, it is necessary to calculate the dynamic load[1]:

$$F_d := F_e \cdot \left(1 + \sqrt{1 + \frac{\left(\frac{v_{compactor}}{\frac{\boldsymbol{m}}{\boldsymbol{s}}}\right)^2}{9.8\ \frac{F_e}{\boldsymbol{kN}}}}\right) = 516.163\ \boldsymbol{kN} \tag{2.3-5}$$

Being the compactor travel speed: $v_{compactor}$ = 7 kph

In order to make calculations, we need to know the force per meter. The front part of the attachment will measure 4.5 m, therefore the distributed force per meter will be:

$$Q := \frac{F_d}{l_T} = 114.703\ \frac{kN}{m} \tag{2.3-6}$$

Although this force is a parabolic distributed force, we are going to simplify it as 3 uniform distributed forces applied in the axis of the 2 shafts plus another one applied in the middle of both shafts. This last force will generate the torque which the shaft must resist.

$$q := \frac{Q}{3} = 38.234\ \frac{kN}{m} \tag{2.3-7}$$

Then, we need to know the torsion moment which is supported by the shaft. For modeling this, we are going to supposed that the previous force is applied in the end of the blades of the 2 shafts, causing a torsion moment in the 2 shafts. Therefore, the force applied in each shaft is the middle of the one that we calculated before:

$$T := \frac{q}{2} \cdot \frac{d_{blades}}{2} = 4.588\ \frac{kN \cdot m}{m} \tag{2.3-8}$$

The total torque applied to the shredder shaft will be the same as the torque transmitted by the gears:

$$T_T := T \cdot l_{shredder} = 6.882\ kN \cdot m \qquad T_G := T_T \tag{2.3-9}$$

The torque needed for moving the 2 shafts of the shredder will be:

$$T_m' := T_T \cdot 2 = 13.764\ kN \cdot m \tag{2.3-10}$$

The power needed for moving the 2 shafts of the shredder will be:

$$Hp' := T_m' \cdot n_{shredder} = 43.242\ kW \tag{2.3-11}$$

Being shredder speed revolutions[9] ($n_{shredder}$): 30 rpm

If we apply the efficiency coefficients for gears and bearings, we obtain the power and the torque that the motor must generate:

$$\eta_G := 0.94 \quad \eta_B := 0.99 \qquad \eta_T := \eta_G \cdot \eta_B^{\ 2} = 0.921$$

$$Hp := \frac{Hp'}{\eta_T} = 46.936\ kW \qquad T_m := \frac{Hp}{n_{shredder}} = 14.94\ kN \cdot m$$

(2.3-12 y 13)

2.4. Motor selection

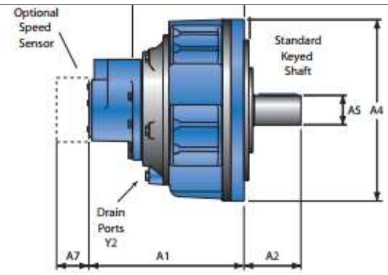
Previously, we calculate the power and torque that the motor must generate:

$$Hp := \frac{Hp'}{\eta_T} = 46.936\ \mathbf{kW} \qquad T_m := \frac{Hp}{n_{shredder}} = 14.94\ \mathbf{kN \cdot m} \qquad \text{(2.4-1 y 2)}$$

According to this figures, the selected motor is the product SMA1600 C1 XXX from the company ROTATORY POWER LIMITED[7]. It is a radial piston motor (eccentric configuration) for heavy-duty applications. Withstands high mechanical and hydraulic shock loads. The technical specifications are shown in the following table (table 2.4-1):

Table 2.4-1: Motor specifications[7]

Nominal displacement	1600 cc/rev
Moment of inertia	0.0487 Kg*m^2
Aprox. Dry weight	290 Kg
Max continuous power	140 kW
Max continuous speed	300 rpm
Max intermittent speed	160% Max continuous speed
Min speed	5 rpm
Max continuous pressure	350 bar
Max intermittent pressure	490 bar
Min return pressure	7 bar
Dimensions (mm):	
A1	449
A2	181
A4	545
A5	80



2.5. Gear calculation

Gear calculation has been made according the chapters 13 & 14 of Shigley's Mechanical Engineering Design 8th edition textbook[1].

The selected gears are spur gears with 20° of pressure angle.

The gear ratio is 1 because the speed of the 2 shafts is the same; the 2 gears will be identic.

The distance between the shaft's axes is 770 mm. Therefore, the gears have a pitch circle of:

Dp = 770 / 2 = 385 mm

The material is steel AISI 1050 normalized at 900ºC[18]:

Grade I, hardened completely 217HB

Ultimate strength: S_{ut} = 748 MPa

Creep strength: S_y = 427 MPa

The nomenclature of spur-gear teeth is shown in the following figure (figure 2.5-1):

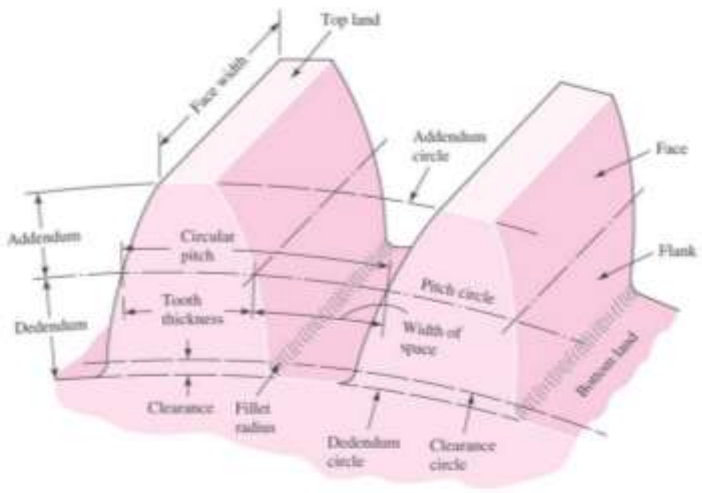


Figure 2.5-1: The nomenclature of spur-gear teeth[1]

The geometry of the gear is calculated as follow:

Modulus: m = (0.01 – 0.02) x Dp = 7.7 – 15.4 → m = 10 mm (2.5-1)

Dp = m x z → z = Dp / m = 770 mm / 10 = 77 teeth (2.5-2)

Circular pitch: p = (2 x π x Rp) / z = (2 x π x 385) / 77 = 31.426 mm (2.5-3)

Diametral pitch: P = z / Dp = 77 / 770 = 0.1/mm = 2.54/in (2.5-4)

Addendum/outside circle: Ra = Rp + m = 385 mm + 10 = 395 mm (2.5-5)

Dedendum/root circle: Rd=Rp–1.25 x m=385mm–1.25x10=372.5mm (2.5-6)

Clearance circle Rc = Rp – m = 385 mm – 10 = 375 mm (2.5-7)

Tooth thickness: t >= p/2 = 15.71 mm (2.5-8)

Addendum (head distance): a = 1/P = 1/0.1 = 10 mm (2.5-9)

Dedendum (root distance): b = 1.25/P = 1.25/0.1 = 12.5 mm (2.5-10)

Clearance (fillet radius): c = rf = b - a = 12.5 – 10 = 2.5 mm (2.5-11)

Figure 2.5-2: Pressure line[1]

Pressure line (measure from CATIA, figure 2.5-2): L = 53.748 mm

Contact ratio: mc=L/(p x cosΦ)=53.748/(31.42xcos20°)=1.82>1→ok (2.5-12)

Base circle: Rb = Rp x cosΦ = 385 x cos20° = 361.8 mm (2.5-13)

There will not be interference because Rc > Rb.

The forces applied to the gear teeth are calculated as followed:

The torque supported by the gear was calculated in the chapter 2.3:

$$T := 6.882\ kN \cdot m \quad (2.5\text{-}14)$$

The tangential force is obtained directly from the torque supported by the gear:

$$Wt := \frac{T}{Rp} = 17.875\ kN \quad (2.5\text{-}15)$$

The radial force is obtained through the pressure angle:

$$Wr := Wt \cdot \tan(\Phi) = 6.506\ kN \quad (2.5\text{-}16)$$

Lewis approximation:

Angular speed: n = 30 rpm

Circumferential speed:

$$V := n \cdot \frac{Dp}{2} = 1.21\ \frac{m}{s} \quad (2.5\text{-}17)$$

Dynamic factor (for cut or milled profile):

$$K_{vL} := \frac{6.1 + \left(\frac{V}{\frac{m}{s}}\right)}{6.1} = 1.198 \quad (2.5\text{-}18)$$

Face width:

$F = (3*p - 5*p) = (94.248 - 157.08)$ mm = 150 mm = 5.9 in (2.5-19)

Lewis geometry factor (from table 14-2 from Shigley's textbook[1])

According to number of teeth, z → Y = 0.435 (2.5-20)

Bending strain:

$$\sigma_f := \frac{K_{vL} \cdot Wt \cdot P}{F \cdot Y} = 32.827\ MPa \tag{2.5-21}$$

AGMA calculation:

- Dynamic factor:

$$K_v := \left(\frac{A + \sqrt{\left(\frac{V}{\frac{m}{s}} \right) \cdot 200}}{A} \right)^B = 1.21 \tag{2.5-22}$$

$$B := 0.25 \cdot (12 - Q_v)^{\frac{2}{3}} = 0.825 \qquad Q_v := 6 \tag{2.5-23}$$

$$A := 50 + 56 \cdot (1 - B) = 59.773 \tag{2.5-24}$$

- Size factor (if Kt < 1, then Kt =1):

$$K_s := 1.192 \cdot \left(\frac{\left(\frac{F}{in} \right) \cdot \sqrt{Y}}{(P\ in)} \right)^{0.0535} = 1.22 \tag{2.5-25}$$

- Overload factor (figure 14-17 from Shigley's textbook[1]):

Power source driven machine uniform & heavy impact): $K_o = 1.75$

- Distribution load factor:

$$K_m := 1 + C_{mc} \cdot (C_{pf} \cdot C_{pm} + C_{md} \cdot C_g) = 1.429 \qquad (2.5\text{-}26)$$

$C_{mc} := 1$ (Uncrowned teeth)

$$C_{pf}' := \frac{F}{10 \cdot Dp} - 0.0375 + 0.0125 \cdot \left(\frac{F}{in}\right) = 0.056 \qquad (2.5\text{-}27)$$

$$\frac{F}{10 \cdot Dp} = 0.019$$

If (F/10Dp) < 0.05 → (F/10Dp) = 0.05 →

$$C_{pf} := 0.05 - 0.0375 + 0.0125 \cdot \left(\frac{F}{in}\right) = 0.086 \qquad (2.5\text{-}28)$$

$C_{pm} := 1$

(separately assembly pinion with S1/S<0.175: 0.156/1.6=0.0975)

$$C_{md} := A_{md} + B_{md} \cdot \left(\frac{F}{in}\right) + C_{md} \cdot \left(\frac{F}{in}\right)^2 = 0.343 \qquad (2.5\text{-}29)$$

(table 14-9 for A, B, C values: open gears)

$A_{md} := 0.247$ $B_{md} := 0.0167$ $C_{md} := -0.765 \cdot 10^{-4}$

$C_g := 1$ (assembly condition)

- Rim thickness factor: (constant gear thickness) → $K_B = 1$
- Stress cycles factor: 10^7 cycles → $Y_N = 1$ & $Z_N = 1$
- Reliability factor: R=0.99 → $K_R = 1$
- Temperature factor: oil t[a] < 120ºC → $K_T = 1$
- Surface condition factor: there is no detrimental effect in the tooth surface finish

$C_f := 1$

- Elastic coefficient: (table 14-8 from Shigley's textbook[1])

Steel gear & pinion with Elasticity modulus 2*10^5 MPa

$$C_p := 2300 \cdot \sqrt{psi} = 190.98 \sqrt{MPa}$$

- Surface strength geometrical factor: (outside gears)

$$I := \frac{\cos(\Phi) \cdot \sin(\Phi)}{2 \cdot m_N} \cdot \frac{m_G}{m_G + 1} = 0.08 \tag{2.5-30}$$

Velocity ratio: $m_G = 1$

Load distribution ratio: (spur gears) $m_N = 1$

- Bending strength geometrical factor (figure 14-6 from Shigley's textbook[1]):

spur gears, 20º, full-size teeth, z = 77 → J = 0.46

- Hardness ratio factor: $m_G = 1$ & $m_N = 1$ → $C_H = 1$

Strength calculation:

Bending strength:

$$S_F := (0.533\ HB + 88.3)\ MPa = 203.961\ MPa \tag{2.5-31}$$

Contact strength:

$$S_C := (2.22 \cdot HB + 200)\ MPa = 681.74\ MPa \tag{2.5-32}$$

Strain calculation:

Bending strain:

$$\sigma_F := Wt \cdot K_o \cdot K_v \cdot K_s \cdot \frac{P}{F} \cdot \frac{K_m \cdot K_B}{J} = 95.649\ MPa \tag{2.5-33}$$

Contact strain:

$$\sigma_C := C_p \cdot \sqrt{Wt \cdot K_o \cdot K_v \cdot K_s \cdot \frac{K_m}{Dp \cdot F} \cdot \frac{C_f}{I}} = 509.298\ MPa \tag{2.5-34}$$

Bending and contact security factors:

$$n_F := \frac{S_F \cdot \frac{Y_N}{K_T \cdot K_R}}{\sigma_F} = 2.132 \qquad n_C := \frac{S_C \cdot \frac{Z_N \cdot C_H}{K_T \cdot K_R}}{\sigma_C} = 1.339 \tag{2.5-35 y 36}$$

Compare n_F with n_C^2: the threat is contact strain because $n_C^2 < n_F$

$$n_F = 2.132 \qquad n_C^{\ 2} = 1.792$$

2.6. Shaft calculation

The shaft calculation has been made according the chapter 6 of Shigley's Mechanical Engineering Design 8th edition textbook[1]. The bending diagrams have been obtained from the software MDR Fx from Valladolid University.

The shafts will be exposed to medium and alternate torsion and bending moments because the bending and torsion strains are not constant. I will consider that the medium and alternate moments have the same value.

$$F_m = \frac{F_{max}+F_{min}}{2} \qquad F_a = \frac{F_{max}-F_{min}}{2} \tag{2.6-1 y 2}$$

The theory that is going to be follow in order to determine the strength of the shafts is the ED-Goodman theory.

Selected material: Steel AISI 1095 Normalized at 900ºC[19].

Hardness: HB293

Ultimate strength: S_{ut} = 147 ksi

Creep strength: S_y = 72 ksi

Limit fatigue strength:

$$S_e' := 0.5 \cdot S_{ut} = 73.5\ ksi \tag{2.6-1}$$

Limit fatigue strength applying correcting factors:

$$S_e := k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_d \cdot k_f \cdot S_e' = 32.205\ ksi \tag{2.6-2}$$

Surface finish factor: machining or cold rolling

$$k_a := a \cdot \left(\frac{S_{ut}}{ksi}\right)^b = 0.719 \qquad a := 2.70 \quad b := -0.265 \tag{2.6-3}$$

Size factor:

$$k_b := 1.51 \cdot \left(\frac{d}{mm}\right)^{-0.157} = 0.712 \qquad 51\ mm < d < 254\ mm \tag{2.6-4}$$

Load factor: bending → k_c = 1

Temperature factor: tª=150ºC → k_d = 1.025

Reliability factor: 99% → k_e = 0.814

Miscellaneous factor: k_f = 1

2.6.1. Driven shaft calculation

Bending and torsion diagrams (figure 2.6.1-1):

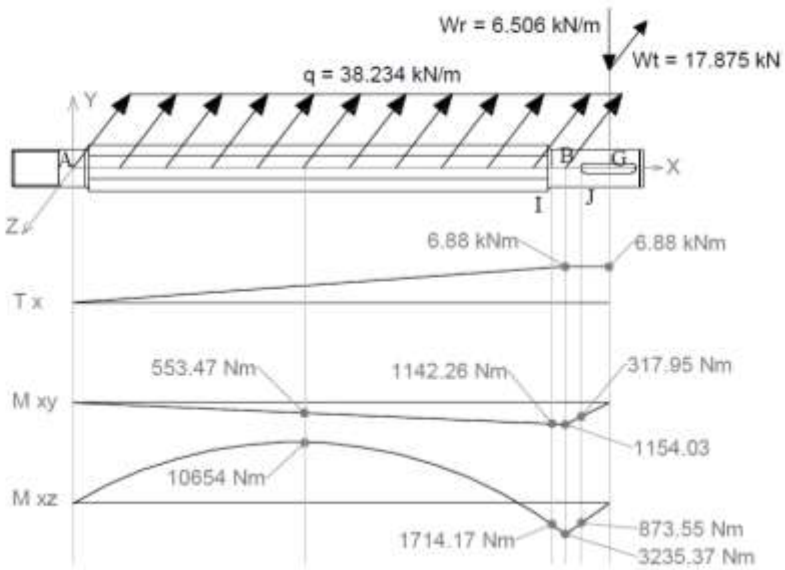


Figure 2.6.1-1: Bending and torsion diagrams.

Firstly, the static calculation with the maximum stress theory:

The maximum bending, shear and torsion values are:

$$M_{max} := 10668.3\ N \cdot m$$

$$V_{max} := 32609.3\ N$$

$$T_{max} := 6.882\ kN \cdot m$$

The maximum bending and shear values are localized in the part of the shaft with d = 0.12 m, which geometrical characteristics are the followings:

$$h := 0.12\ m$$

$$A := 0.01131\ m^2$$

$$I := 1.0179 \cdot 10^{-5}\ m^4$$

The bending strain meets criteria with a value for security facto of n= 1.5:

$$\sigma_{max} := \frac{M_{max} \cdot c}{I} = 9.121\ ksi \qquad \sigma_{max} < \frac{S_y}{2} = 1 \qquad (2.6\text{-}5\ y\ 6)$$

The shear strain meets criteria with a value for security facto of n= 1.5:

$$\tau_{max} := \frac{4 \cdot V_{max}}{3 \cdot A} = 0.558\ ksi \qquad \tau_{max} < \frac{S_y}{2 \cdot n} = 1$$

(2.6-7 y 8)

The torsional strain meets criteria with a value for security facto of n= 1.5:

$$\tau_{tmax} := \frac{T_{max} \cdot r'}{J'} = 2.942\ ksi \qquad \tau_{tmax} < \frac{S_y}{2 \cdot n} = 1$$

(2.6-9 y 10)

Secondly, the <u>fatigue calculation</u>:

The points that are going to be analyzed are:

- Shoulder with high bending and torsion moments applied in I
- Keyway with high bending and torsion moments applied in J

<u>Point I (x = 1.549 m)</u>

The diameters in the shoulder are:

$$d := 120\ mm = 4.724\ in \qquad D := 150\ mm = 5.906\ in$$

The bending moments obtain from the software are:

$$MI_{XY} := 1142.26\ N \cdot m = \left(1.011 \cdot 10^4\right)\ in \cdot lbf$$

$$MI_{XZ} := 1714.17\ N \cdot m = \left(1.517 \cdot 10^4\right)\ in \cdot lbf$$

Therefore, the total bending moment is:

$$M_aI := \sqrt{MI_{XY}^2 + MI_{XZ}^2} = \left(1.823 \cdot 10^4\right)\ in \cdot lbf \qquad M_mI := M_aI$$

(2.6-10 y 11)

And the torsion moment (obtained in chapter 2.3) is:

$$T_mI := 6.882\ kN \cdot m = \left(6.091 \cdot 10^4\right)\ in \cdot lbf \qquad T_aI := T_mI$$

(2.6-12 y 13)

Estimate K_f y K_{tf} (table 7-1 from Shigley's textbook[1]):

Fillet shoulder well round (r/d=0.1) → K_f = 1 .7 K_{tf} = 1.5

Security factor:

$$n_i' := \frac{16}{\pi \cdot d^3} \cdot \left(\frac{\left(4 \cdot (K_f \cdot M_aI)^2 + 3 \cdot (K_{ft} \cdot T_aI)^2\right)^{\frac{1}{2}}}{S_e} + \frac{\left(4 \cdot (K_f \cdot M_mI)^2 + 3 \cdot (K_{ft} \cdot T_mI)^2\right)^{\frac{1}{2}}}{S_{ut}} \right) = 0.311$$

$$n' := \frac{1}{n_i'} = 3.218$$

(2.6-14 y 15)

The material and size of the shaft are correct to support the load according ED-Goodman theory.

Now, the stress concentration factors are changed in order to obtain a more accurate result:

(Kt-Figure A-15-9, Ktt-Figure A-15-8, q-Figure 6-20 from Sigley's textbook[1])

$$r_I := 9\ mm = 0.354\ in \qquad \frac{D}{d} = 1.25 \qquad \frac{r_I}{d} = 0.075$$

Bending:

$$K_f I := 1 + qI \cdot (K_t I - 1) = 1.595 \qquad K_t I := 1.7 \qquad qI := 0.85$$

(2.6-16)

Torsion:

$$K_{ft} I := 1 + q_t I \cdot (K_{tt} I - 1) = 1.428 \qquad K_{tt} I := 1.45 \qquad q_t I := 0.95$$

(2.6-17)

Fluctuating stresses due to bending and torsion:

$$\sigma_a' I := \left(\left(\frac{32 \cdot K_f I \cdot M_a I}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_{ft} I \cdot T_a I}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 7.797\ ksi$$

(2.6-18)

$$\sigma_m' I := \left(\left(\frac{32 \cdot K_{ft} I \cdot T_m I}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_f I \cdot M_m I}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 8.744\ ksi$$

(2.6-19)

Security factors:

$$n_f I := \frac{1}{n_{fi} I} = 3.316 \qquad n_{fi} I := \frac{\sigma_a' I}{S_e} + \frac{\sigma_m' I}{S_{ut}} = 0.302$$

(2.6-20 y 21)

Check creep:

$$n_y I := \frac{S_y}{\sigma_a' I + \sigma_m' I} = 4.353$$

(2.6-22)

<u>Point J (x = 1.681 m)</u>

The diameter in this point is:

$$d := 120\ mm = 4.724\ in$$

The bending moments obtain from the software are:

$$MJ_{XY} := 317.95\ N \cdot m = (2.814 \cdot 10^3)\ in \cdot lbf$$

$$MJ_{XZ} := 873.55\ N \cdot m = (7.732 \cdot 10^3)\ in \cdot lbf$$

Therefore, the total bending moment is:

$$M_a J := \sqrt{MJ_{XY}^{\ 2} + MJ_{XZ}^{\ 2}} = (8.228 \cdot 10^3)\ in \cdot lbf \qquad M_m J := M_a J$$ (2.6-10 y 11)

And the torsion moment (obtained in chapter 2.3) is:

$$T_m J := 6.882\ kN \cdot m = (6.091 \cdot 10^4)\ in \cdot lbf \qquad T_a J := T_m J$$ (2.6-12 y 13)

Stress concentration factor (table 7-1: Mill keyway: r/d=0.02):

Bending:

$$K_f J := 2.2$$

Torsion:

$$K_{ft} J := 3$$

Fluctuating stresses due to bending and torsion:

$$\sigma_a' J := \left(\left(\frac{32 \cdot K_f J \cdot M_a J}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_{ft} J \cdot T_a J}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 15.386\ ksi$$ (2.6-18)

$$\sigma_m' J := \left(\left(\frac{32 \cdot K_{ft} J \cdot T_m J}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_f J \cdot M_m J}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 17.716\ ksi$$ (2.6-19)

Security factors:

$$n_{ft} J := \frac{\sigma_a' J}{S_e} + \frac{\sigma_m' J}{S_{ut}} = 0.598 \qquad n_f J := \frac{1}{n_{ft} J} = 1.672$$ (2.6-20 y 21)

Check creep:

$$n_yJ := \frac{S_y}{\sigma_a'J + \sigma_m'J} = 2.175 \tag{2.6-22}$$

Deflection and slope checkout: the values for deflection and slope have been obtained from the software MDR Fx by Valladolid University.

Slope in bearings positions: for roller bearings the value should be less than 0.026-0.052 rad.

$$\theta_A := \sqrt{\left(1.085 \cdot 10^{-3}\right)^2 + \left(6.017 \cdot 10^{-5}\right)^2} = 0.001\ \mathbf{rad} \tag{2.6-23}$$

$$\theta_B := \sqrt{\left(1.198 \cdot 10^{-4}\right)^2 + \left(1.203 \cdot 10^{-4}\right)^2} = \left(1.698 \cdot 10^{-4}\right)\ \mathbf{rad} \tag{2.6-24}$$

Deflection and slope in gear position: for spur gears without crown: < 0.005 rad & spur gears with P<10: 0.01 in.

$$\delta_G := \sqrt{\left(1.499 \cdot 10^{-4}\ \mathbf{m}\right)^2 + \left(2.78 \cdot 10^{-5}\ \mathbf{m}\right)^2} = \left(1.525 \cdot 10^{-4}\right)\ \mathbf{m} \tag{2.6-25}$$

$$\theta_G := \sqrt{\left(7.828 \cdot 10^{-4}\right)^2 + \left(1.702 \cdot 10^{-4}\right)^2} = \left(8.011 \cdot 10^{-4}\right)\ \mathbf{rad} \tag{2.6-26}$$

2.6.2. Motor shaft

Bending and torsion diagrams (figure 2.6.2-1):

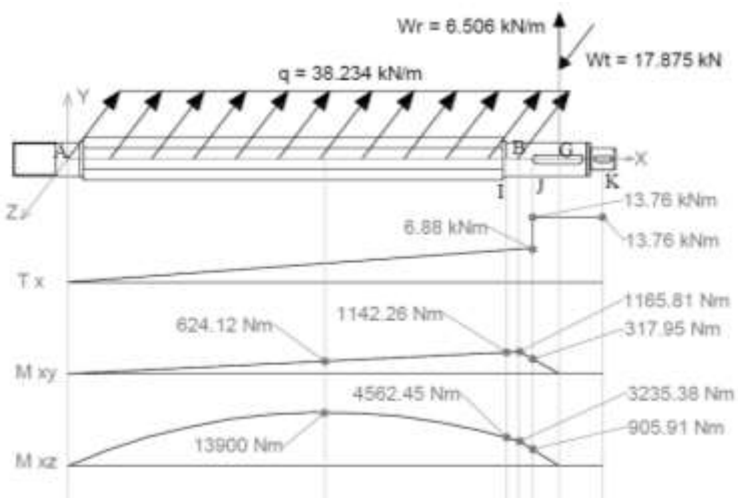


Figure 2.6.2-1: Bending and torsion diagrams in the motor shaft.

Firstly, the static calculation with the maximum stress theory:

The maximum bending, shear and torsion values are:

$$M_{max} := 13903.68\ \boldsymbol{N \cdot m}$$

$$V_{max} := 32609.3\ \boldsymbol{N}$$

$$T_{max} := 13.764\ \boldsymbol{kN \cdot m}$$

The maximum bending and shear values are localized in the part of the shaft with d = 0.12 m, which geometrical characteristics are the followings:

$$h := 0.12\ \boldsymbol{m}$$

$$A := 0.01131\ \boldsymbol{m}^2$$

$$I := 1.0179 \cdot 10^{-5}\ \boldsymbol{m}^4$$

The maximum torsion value is localized in the part of the shaft with d = 0.08 m, which geometrical characteristics are the followings:

$$d' := 0.08\ \boldsymbol{m}$$

$$r' := \frac{d'}{2} = 0.04\ \boldsymbol{m}$$

$$J' := \frac{\pi \cdot d'^4}{32} = \left(4.021 \cdot 10^{-6}\right)\ \boldsymbol{m}^4 \qquad (2.6\text{-}27)$$

The bending strain meets criteria with a value for security facto of n= 1.5:

$$\sigma_{max} := \frac{M_{max} \cdot c}{I} = 11.887\ \boldsymbol{ksi} \qquad \sigma_{max} < \frac{S_y}{2} = 1 \qquad (2.6\text{-}5 \text{ y } 6)$$

The shear strain meets criteria with a value for security facto of n= 1.5:

$$\tau_{max} := \frac{4 \cdot V_{max}}{3 \cdot A} = 0.558\ \boldsymbol{ksi} \qquad \tau_{max} < \frac{S_y}{2 \cdot n} = 1 \qquad (2.6\text{-}7 \text{ y } 8)$$

The torsional strain meets criteria with a value for security facto of n= 1.5:

$$\tau_{tmax} := \frac{T_{max} \cdot r'}{J'} = 19.858\ \boldsymbol{ksi} \qquad \tau_{tmax} < \frac{S_y}{2 \cdot n} = 1 \qquad (2.6\text{-}9 \text{ y } 10)$$

Secondly, the <u>fatigue calculation</u>:

The points that are going to be analyzed are:

- Shoulder with high bending and torsion moments applied in I
- Keyway with high bending and torsion moments applied in J
- Section with a high static torsion moment & diameter reduction in K

Point I (x = 1.549 m)

The diameters in the shoulder are:

$$d := 120\ mm = 4.724\ in \qquad D := 150\ mm = 5.906\ in$$

The bending moments obtain from the software are:

$$MI_{XY} := 1142.26\ N \cdot m = \left(1.011 \cdot 10^{4}\right)\ in \cdot lbf$$

$$MI_{XZ} := 4562.45\ N \cdot m = \left(4.038 \cdot 10^{4}\right)\ in \cdot lbf$$

Therefore, the total bending moment is:

$$M_aI := \sqrt{MI_{XY}^{\ 2} + MI_{XZ}^{\ 2}} = \left(4.163 \cdot 10^{4}\right)\ in \cdot lbf \qquad M_mI := M_aI$$

(2.6-10 y 11)

And the torsion moment (obtained in chapter 2.3) is:

$$T_mI := 6.882\ kN \cdot m = \left(6.091 \cdot 10^{4}\right)\ in \cdot lbf \qquad T_aI := T_mI$$

(2.6-12 y 13)

Estimate K_f y K_{tf} (table 7-1 from Shigley's textbook[1]):

Fillet shoulder well round (r/d=0.1) → $K_f = 1.7$ $K_{tf} = 1.5$

Security factor:

$$n_i := \frac{16}{\pi \cdot d^3} \cdot \left(\frac{\left(4 \cdot (K_f \cdot M_aI)^2 + 3 \cdot (K_{ft} \cdot T_aI)^2\right)^{\frac{1}{2}}}{S_e} + \frac{\left(4 \cdot (K_f \cdot M_mI)^2 + 3 \cdot (K_{ft} \cdot T_mI)^2\right)^{\frac{1}{2}}}{S_{ut}} \right) = 0.388$$

$$n := \frac{1}{n_i} = 2.576$$

(2.6-14 y 15)

The material and size of the shaft are correct to support the load according ED-Goodman theory.

Now, the stress concentration factors are changed in order to obtain a more accurate result:

(Kt-Figure A-15-9, Ktt-Figure A-15-8, q-Figure 6-20 from Shigley's textbook[1])

$$r_I := 9\ mm = 0.354\ in \qquad \frac{D}{d} = 1.25 \qquad \frac{r_I}{d} = 0.075$$

Bending:

$$K_f I := 1 + qI \cdot (K_t I - 1) = 1.595 \qquad K_t I := 1.7 \qquad qI := 0.85 \qquad (2.6\text{-}16)$$

Torsion:

$$K_{ft} I := 1 + q_t I \cdot (K_{tt} I - 1) = 1.428 \qquad K_{tt} I := 1.45 \qquad q_t I := 0.95 \qquad (2.6\text{-}17)$$

Fluctuating stresses due to bending and torsion:

$$\sigma_a 'I := \left(\left(\frac{32 \cdot K_f I \cdot M_a I}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_{ft} I \cdot T_a I}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 9.697\ ksi \qquad (2.6\text{-}18)$$

$$\sigma_m 'I := \left(\left(\frac{32 \cdot K_{ft} I \cdot T_m I}{\pi \cdot d^3} \right)^2 + 3 \cdot \left(\frac{16 \cdot K_f I \cdot M_m I}{\pi \cdot d^3} \right)^2 \right)^{\frac{1}{2}} = 10.069\ ksi \qquad (2.6\text{-}19)$$

Security factors:

$$n_{ft} I := \frac{\sigma_a 'I}{S_e} + \frac{\sigma_m 'I}{S_{ut}} = 0.37 \qquad n_f I := \frac{1}{n_{ft} I} = 2.706 \qquad (2.6\text{-}20\ y\ 21)$$

Check creep:

$$n_y I := \frac{S_y}{\sigma_a 'I + \sigma_m 'I} = 3.642 \qquad (2.6\text{-}22)$$

<u>Point J (x = 1.681 m)</u>

The diameter in this point is:

$$d := 120\ mm = 4.724\ in$$

The bending moments obtain from the software are:

$$MJ_{XY} := 317.95\ N \cdot m = \left(2.814 \cdot 10^3\right)\ in \cdot lbf$$

$$MJ_{XZ} := 905.91\ N \cdot m = \left(8.018 \cdot 10^3\right)\ in \cdot lbf$$

Therefore, the total bending moment is:

$$M_a J := \sqrt{MJ_{XY}^{\ 2} + MJ_{XZ}^{\ 2}} = \left(8.497 \cdot 10^{3}\right)\ in \cdot lbf \qquad M_m J := M_a J \tag{2.6-10 y 11}$$

And the torsion moment (obtained in chapter 2.3) is:

$$T_m J := 6.882\ kN \cdot m = \left(6.091 \cdot 10^{4}\right)\ in \cdot lbf \qquad T_a J := T_m J \tag{2.6-12 y 13}$$

Stress concentration factor (table 7-1 from Shigley's textbook[1]): Mill keyway: r/d=0.02):

Bending:

$$K_f J := 2.2$$

Torsion:

$$K_{ft} J := 3$$

Fluctuating stresses due to bending and torsion:

$$\sigma_a' J := \left(\left(\frac{32 \cdot K_f J \cdot M_a J}{\pi \cdot d^{3}}\right)^{2} + 3 \cdot \left(\frac{16 \cdot K_{ft} J \cdot T_a J}{\pi \cdot d^{3}}\right)^{2}\right)^{\frac{1}{2}} = 15.386\ ksi \tag{2.6-18}$$

$$\sigma_m' J := \left(\left(\frac{32 \cdot K_{ft} J \cdot T_m J}{\pi \cdot d^{3}}\right)^{2} + 3 \cdot \left(\frac{16 \cdot K_f J \cdot M_m J}{\pi \cdot d^{3}}\right)^{2}\right)^{\frac{1}{2}} = 17.716\ ksi \tag{2.6-19}$$

Security factors:

$$n_{ft} J := \frac{\sigma_a' J}{S_e} + \frac{\sigma_m' J}{S_{ut}} = 0.598 \qquad n_f J := \frac{1}{n_{ft} J} = 1.672 \tag{2.6-20 y 21}$$

Check creep:

$$n_y J := \frac{S_y}{\sigma_a' J + \sigma_m' J} = 2.174 \tag{2.6-22}$$

<u>Point K (x = 1.681 m)</u>

The diameter in this point is:

$$d' := 80\ mm$$

In this point there is no bending moments, neither alternate torsion moment because the torque which the motor generates is continuous. Therefore:

$$M_aK := 0 \quad M_mK := 0 \quad T_aK := 0$$

$$T_mK := 13.764\ kN \cdot m = \left(1.218 \cdot 10^{5}\right)\ in \cdot lbf \quad T_aK := 0$$

The polar moment of inertia in this section is:

$$J := \frac{\pi \cdot d'^{4}}{32} = \left(4.021 \cdot 10^{-6}\right)\ m^{4} \tag{2.6-27}$$

The torsion stress and the security factor are:

$$\tau := \frac{T_mK \cdot \frac{d'}{2}}{J} = 19.858\ ksi \qquad nK := \frac{S_y}{2 \cdot \tau} = 1.813 \tag{2.6-28 y 29}$$

<u>Deflection and slope checkout:</u> the values for deflection and slope have been obtained from the software MDR Fx by Valladolid University.

Slope in bearings positions: for cylinder bearings the value should be less than 0.026-0.052 rad.

$$\theta_A := \sqrt{\left(1.416 \cdot 10^{-3}\right)^2 + \left(6.017 \cdot 10^{-5}\right)^2} = 0.001\ rad \tag{2.6-23}$$

$$\theta_B := \sqrt{\left(1.581 \cdot 10^{-3}\right)^2 + \left(1.203 \cdot 10^{-4}\right)^2} = 0.002\ rad \tag{2.6-24}$$

Deflection and slope in gear position: for spur gears without crown: < 0.005 rad & spur gears with P<10: 0.01 in.

$$\delta_G := \sqrt{\left(3.027 \cdot 10^{-4}\ m\right)^2 + \left(2.78 \cdot 10^{-5}\ m\right)^2} = \left(3.04 \cdot 10^{-4}\right)\ m \tag{2.6-25}$$

$$\theta_G := \sqrt{\left(1.718 \cdot 10^{-3}\right)^2 + \left(1.702 \cdot 10^{-4}\right)^2} = 0.002\ rad \tag{2.6-26}$$

2.7. Bearing calculation

The bearing calculation has been made according the chapter 11 of Shigley's Mechanical Engineering Design 8th edition textbook[1].

Shaft parameters:

$$n := 30\ rpm$$

$$d := 120\ mm = 4.724\ in$$

Bearing calculation parameters:

Bearing reliability: R = 99%

Bearing longlife (table 11-4 from Shigley's textbook[1]): 8h working equipment which are not full-time used.

$$L_{hr} := 20000\ hr$$

$$L_{ciclos} := 60 \cdot \frac{L_{hr}}{hr} \cdot \frac{n}{rpm} = 3.6 \cdot 10^{7} \qquad (2.7\text{-}1)$$

Weibull parameters:

$$L_{10} := 10^{6} \qquad x_0 := 0.02 \qquad \theta := 4.459 \qquad b := 1.483$$

Adimensional design longlife:

$$x := \frac{L_{ciclos}}{L_{10}} = 36 \qquad (2.7\text{-}2)$$

Load application factor (table 11-5 from Shigley's textbook[1]): equipment with moderate impacts.

$$a_f := 3$$

Bearing type parameter (roller-bearing):

$$a := \frac{10}{3}$$

Radial forces applied to the bearings (obtained from MDR Fx software):

<u>Driven shaft:</u>

$$Rd_{AY} := 0.73\ kN \qquad Rd_{BY} := 7.24\ kN$$

$$Rd_{AZ} := 28.6\ kN \qquad Rd_{BZ} := 50.5\ kN$$

$$Rd_A := \sqrt{Rd_{AY}^{\ 2} + Rd_{AZ}^{\ 2}} = 28.609\ kN \qquad Rd_B := \sqrt{Rd_{BY}^{\ 2} + Rd_{BZ}^{\ 2}} = 51.016\ kN$$

B bearing is the one with more load, therefore we make calculations with B bearing load:

$$Fd := Rd_B$$

Dynamic load classification:

$$Cd_{10} := Fd \cdot a_f \cdot \left(\frac{x}{x_0 + (\theta - x_0) \cdot (1 - R)^{\frac{1}{b}}} \right)^{\frac{1}{a}} = 707.357 \; kN \qquad (2.7\text{-}3)$$

Motor shaft:

$Rm_{AY} := 0.73 \; kN$ $\qquad$ $Rm_{BY} := 7.24 \; kN$

$Rm_{AZ} := 32.61 \; kN$ $\qquad$ $Rm_{BZ} := 10.69 \; kN$

$$Rm_A := \sqrt{Rm_{AY}^{\;2} + Rm_{AZ}^{\;2}} = 32.618 \; kN \qquad Rm_B := \sqrt{Rm_{BY}^{\;2} + Rm_{BZ}^{\;2}} = 12.911 \; kN$$

A bearing is the one with more load, therefore we make calculations with B bearing load:

$$Fm := Rm_A$$

Dynamic load classification:

$$Cm_{10} := Fm \cdot a_f \cdot \left(\frac{x}{x_0 + (\theta - x_0) \cdot (1 - R)^{\frac{1}{b}}} \right)^{\frac{1}{a}} = 452.261 \; kN \qquad (2.7\text{-}3)$$

Selected bearings: cylindrical roller bearings of single row from SKF company. The model is NJ 2324 ECM (figure 2.7-1) and its specifications[20] are:

Inside diameter: d = 120 mm

Outside diameter: D = 260 mm

Thickness: t = 86 mm

Dynamic load: C_{10} = 915 kN

Static load: C_0 = 140

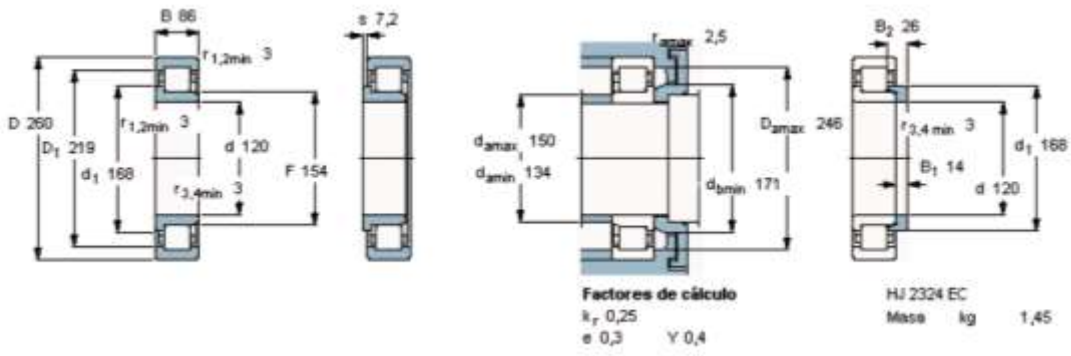


Figure 2.7-1: cylindrical roller bearings of single row NJ 2324 ECM[20]

2.8. Keyway calculation

Material selected: AISI 1045 Steel, quenched and tempered to 595HB[21]

$$S_{ut} := 1825\ MPa \qquad S_y := 1259\ MPa$$

<u>Gear keyway:</u> Key M32 x 18 x 180 Form A PL OV from G.L. Huyett[22]

- Input parameters:

  Transmitted torque: T_G = 6.882 kNm

  Shaft diameter: d = 120 mm

  Keyway length: l = 180 mm

- Selected keyway parameters:

  $$h := 18\ mm \qquad b := 32\ mm$$

- Force and strain over the keyway in the shaft surface:

  $$F := \frac{T_G}{\frac{d}{2}} = 114.7\ kN \qquad \sigma := \frac{F}{\frac{(h \cdot b)}{2}} = 398.264\ MPa \tag{2.8-1 y 2}$$

- Verification for failures due to crushing:

  $$n := \frac{S_y}{\sigma} = 3.161 \tag{2.8-3}$$

<u>Coupling keyway:</u> Key M22 x 14 x 70 Form A PL OV from G.L. Huyett[23]

- Input parameters:

  Transmitted torque: T_G = 6.882 kNm

  Shaft diameter: d = 80 mm

  Keyway length: l = 70 mm

- Selected keyway parameters:

$$h' := 14\ mm \qquad b' := 22\ mm$$

- Force and strain over the keyway in the shaft surface:

$$F' := \frac{T_G}{\frac{d'}{2}} = 172.05\ kN \qquad \sigma' := \frac{F'}{\frac{(h' \cdot b')}{2}} = \left(1.117 \cdot 10^{3}\right)\ MPa$$

(2.8-1 y 2)

- Verification for failures due to crushing:

$$n' := \frac{S_y}{\sigma'} = 1.127$$

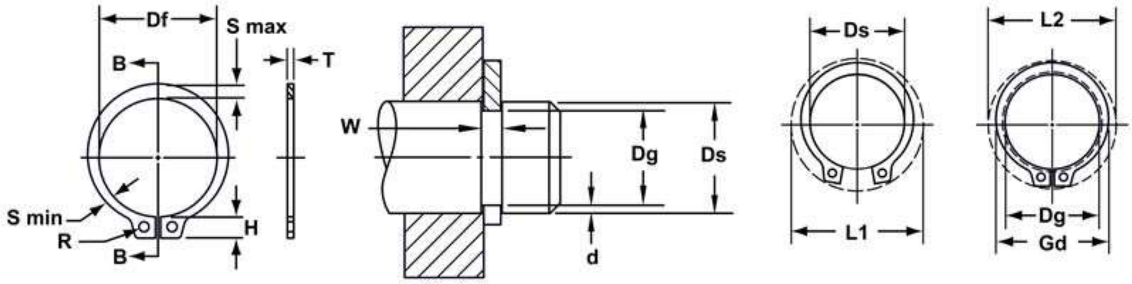
(2.8-3)

2.9. Retaining rings selection

Selected retaining rings: Daemar P/N DSH-120 external shaft ring. The specifications[25] are in the following table (table 2.9-1).

Table 2.9-1: Specifications of the Daemar P/N DSH-120, DSH-External shaft ring[25].

Shaft specifications	Ring specifications
Shaft diameter: D_S = 120 mm	Material: Stainless steel 420 hardened and temp.[24]
Groove diameter: D_G = 116 mm	Ring free diameter: D_f = 113 mm (tolerance +0.54 to -1.30 mm)
Groove depth: d = 2 mm	Ring thickness: t = 4 mm (tolerance: -0.10 mm)
Groove width: w = 4.15 mm	Trust load ring: L_R = 424 kN
Trust load groove: L_G = 123 kN	R.P.M. Limits: 4000 rpm



2.10. Coupling selection

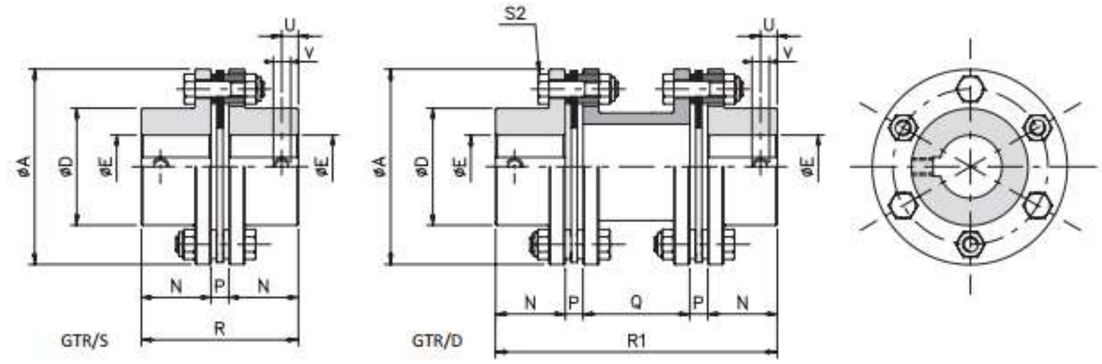
Selected couplings: GTR/S bore Φ80 H7 A1 bore Φ90 H7 A1.

The specifications[26] are in the following table (table 2.10-1).

Table 2.10-1: Specifications of the coupling GTR/S bore Φ80 H7 A1 bore Φ90 H7 A1[26].

Dimensions	Technical specifications
A = 206 mm	Nominal torque = 2600 Nm
E H7 mix = 80	Maximum torque = 5200

E H7 max = 90 Maximum speed = 5500 rpm
R = 199



3. Technological design and calculation

3.1. Analysis of the initial data of the technological process

The technological process of the whole product can be divided in 2 big separated and simultaneous steps: manufacturing and assembly of the connection product and manufacturing and assembly of the shredder box product.

The connection product is constructed from standard cold forming profiles (Stainless steel AISI 316L), simple cut plates (Stainless steel AISI 316L) and 2 custom design parts (Stainless steel AISI 316L): the connections to the hydraulic cylinder. These connections must be compatible with the compactor machine. Although I had not accesss to accurate information of how are in real these connections, I have made an approximate design of them from the information which I have found on-line[5]. They will be manufacture by milling and welding technologies. Finally, the assembly of this product is completely made with welding technology. The machine used will be the Railtrac 1000 from ESAB company[27], which is a multitask system for welding and cutting.. This welding process is critical because of the heavy load which the product must support.

The shredder box product is compound at the same time of 2 equal shredders but mounted inversely. Therefore, it is needed 2 shredder products for assembling the shredder box product. Aside from the aluminum 3003 alloy cover parts which are manufactured by stamping and fixed with M10 bolts, washers and nuts.

The shredder product is compound of the following no standard parts sorted by manufacturing process:

Table 3.1-1: Non-standard parts sorted by manufacturing process

CNC multi task machine: Mazak Integrer i-200[28]	**CNC machining center:** GF HSM 800 LP[29]	**Water jet cutting center:** ESAB Hydrocut LX 4000[30]	**Punching press:** Foremost 5B, RMT Toggle press[31]
Motor shaft (from hex. Rod)	Gear (from bar)	Blade (from plate)	Hex. Seal (2nd operation)
Driven shaft (from hex. Rod)		Blade spacer (from plate)	Distance plate (from plate)
Circular and hex. seal (from tube)		Bearing housing (from plate)	Inside cover (from plate)
Nut (from hex. Rod)			

The **hexagonal seal** is manufactured from the **circular seal** by using a punching press to generate the inside hexagonal section.

The punching machine is also used to manufacture the **distance plate** (t=1mm) and the **inside cover** (t=8mm), both with the same die.

The geometrical contour of the **blades** is highly complicated to do it with a milling machine, it would take long time. For this reason water jet cutting is the process chosen to manufacture the blades. In addition, this process saves a lot of metal waste generated by milling. Then, heat treatment is applied to the blades to give them more hardness as they would be exposed to high wear.

Blade spacer (t=92.5mm) and **bearing housing** (t=86mm) are also manufacture with water jet machine because their thickness is too large to use punching press.

Blade and blade spacer are rigidly connected by welding its edges. The blade spacer is fixed to the shaft by a pin which eliminated axial movement of the blades.

Gears are machine by shape milling. It is a slow manufacturing method, however, only 2 gears by product are needed and the tools used are easily found in every workshop. As the CNC machining center is only used for manufacturing 2 gears, time is not a critical issue because in that time the rest of the parts must be manufactured in the other machines (multi task, water jet & punching press machines).

3.2. Technological process design stages (included assembly)

The attached block diagram in drawing MPfu 11.04.003 shows how the technological process should occur.

Control operations must be performed after each stage of the technological process.

3.3. Design of the technological process of one part

The part selected for describing its technological process in detail is the motor shaft because it is one of the most solicited parts of this product. Not only it must resist bending stress, but also torsion stress. In addition, it must be design for placing the rest of the parts of the product, like bearings, gears, nuts, seals or blades.

The technological process selected for manufacturing the motor shaft is turning with a CNC Lathe machine. The principal specifications of the machine and the tools used are detailed in the Technological process drawing MPfu 11.04.001.

The initial stock is a hexagonal rod with 150 mm between faces and 2285.15 mm length. The selected material is steel AISI/SAE 1095 293HB.

The clamping device is a 3 jaw chuck for the first 2 operations and then the dead center is used for better clamping of the shaft because it is a long part. The first operation is a face turning for guaranteeing the perpendicularity between the face and the shaft axis. Once this specification has been fulfilled, then the dead center is drilled on the shaft.

Calculations of feeds and speeds for machining has been made with the recommendations of Sandvick Coromant by using CoroGuide 2.0 free software. Then, I have check all the calculations using the CoroKey 2010 guide[32], the thread guide and the parting off and grooving guides, and Microsoft Excel with the following formulas obtained from CoroKey 2010 guide:

Cutting speed (v_c): $$v_c = \frac{D_m \cdot \pi \cdot n}{1000} \quad \text{(m/min)} \tag{3.3-1}$$

Spindle speed (n): $$n = \frac{v_c \cdot 1000}{\pi \cdot D_m} \quad \text{(rpm)} \tag{3.3-2}$$

Power required (P_c): $$P_c = \frac{v_c \cdot a_p \cdot f_n \cdot k_c}{60 \cdot 10^3} \quad \text{(kW)} \tag{3.3-3}$$

Machining time (t): $$t = \frac{L}{f} \quad \text{(s)} \tag{3.3-4}$$

Roughness (R_a): $$R_a = \frac{f^2}{32 \cdot r_\varepsilon} \quad (\mu m) \qquad (3.3\text{-}5)$$

With:

D_m: diameter (mm) — f_n: cutting feed (mm/rev)

a_p: axial depth of cut (mm) — k_c: specific cutting force (N/mm^2)

The specific cutting force[33], k_c, can be explained as the force in the cutting direction needed to cut a chip area of 1 mm^2 that has a thickness of 1 mm. I choose a value of 2350 N/mm^2 according to the machining material and its hardness. Coromant guides and the following graphic support my decision.

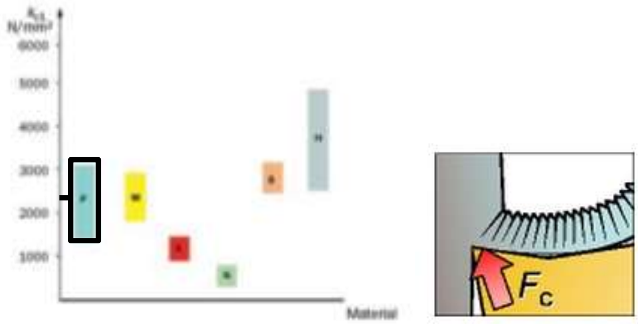


Figure 3.3-1: Specific cutting force graphic[33]

The recommended cutting speed given by Coromant guides are for a reference material and hardness. However, the cutting speed depends on the machining material and its hardness. As a result, the reference values must be corrected according the selected machining material. As the material selected is Steel 293HB and the reference material in Coromant guides is 220HB, the factor will be 0.62.

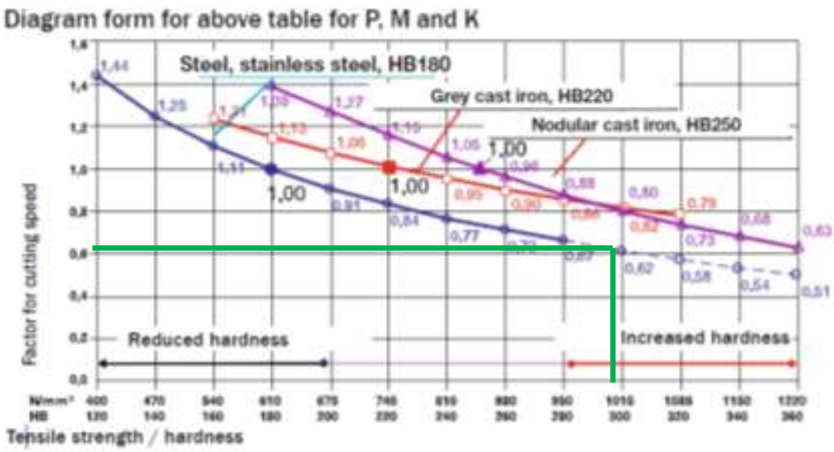


Figure 3.3-2: Factor for cutting speed diagram[32]

The last operation is very critical because after parting off the shaft, it will fall down. Therefore, shape and geometrical tolerances must be controlled.

4. Operation safety and environmental requirements

[Reference: 36]

4.1. Safety requirements

Use:

The machine is suitable for shredding mixed solid waste of various sizes.

The operator must be trained and competent. He must carefully follow all the regulations for driver's cabin safety (bulletproof glass, protection guards, pollen filters, rollbar, protection against tipping over).

When operating the machine, make certain that no one is standing within the machine's operating radius. Moreover, lights must be installed to indicate the machine's presence on site as well as everything needed to ensure that the unit is used in utmost safety.

Only the specific use and configuration indicated for the machine are admissible.

Noise, vibrations:

The vibrations produced by a well maintained machine do not constitute an operator health hazard.

Breakdowns can cause excessive vibrations; these must be immediately noted and eliminated to prevent compromising machine reliability and/or damaging operator health.

Safety:

Do not operate equipment unless guards and safety shields are in place.

Never allow anyone within machine's operating radius while it is in operation.

Do not stop or start suddenly when going uphill or downhill. Avoid operation on steep slopes. Be alert for holes in terrain and other hidden hazards. Always drive slowly over rough ground. Reduce speed on slopes and in sharp turns to prevent tipping or loss of control.

Stop shredder and tractor immediately upon striking an obstruction. Turn off engine, inspect shredder and repair any damage before resuming operation.

Disengage power to shredder and stop engine before dismounting from compactor, before making any repairs or adjustments, transporting, or unclogging shredder. Never run shredder with rotor shaft out of balance.

Handling:

These operations must be performed by personnel using equipment and personal protection equipment in compliance with the current accident prevention and work safety laws.

Transport:

This machine must be transported using an appropriate vehicle for the particular model. The machine must be secured with belts or with other appropriate means to prevent it from shifting positions while the vehicle is in motion.

Handling, lifting, and unloading:

Always lift the machine using an adequate hoist (rated for the weight of the machine). Before moving the machine, visually check it for any unstable parts which could cause a hazard.

Application on machine hydraulic:

Before performing any operations, make certain that the power, oil capacity, and weight of the compactor machine comply with the ones of the shredder. To insure stability, the weight of the unit must be proportional to the weight of the operating machine.

Connecting the hydraulic hoses:

When attaching the quick-release couplings, make certain that they are hooked up to the appropriate connections. The first time they are installed on the machine the air must be purged.

Fitting machine to operating machine compactor hydraulic system:

Make certain that the feed and return hoses are the correct diameter and that they are rated for the pressures involved and the required oil flow. Always check system pressure with an adequate pressure gauge.

Make absolutely certain that the connections are not restricted in any way. Never make looping connections.

Piston motors must have an anti-cavitation valve installed between the hydraulic circuit delivery and return hoses. This allows the pump to take up oil from the return hose and into an open circuit preventing it from running dry, which could cause damage.

For the system to function well, check the degree of oil filtration; it must never be less than *20>=75 microns absolute.* Check the filters every 500 hours of operation.

Maximum acceptable temperature inside the hydraulic circuit is 70/80C.

Never exceed a pressure of 25 P.S.I. bar in the return line.

Operation:

All operations described in this section should only be executed by personnel who have been trained in the operation of the compactor and shredder.

Start-up:

1. Check that the unit has been installed and adjusted correctly.
2. Set the machine on the ground.
3. Start up the hydraulic system gradually; i.e. starting at the minimum and increasing to between 4/5 and maximum revolution before you start work.

Forward speed:

Before starting any work, check the ground for any foreign objects, which may cause damage, danger or pollute the area if hit by the machine.

Make certain that there is no one in the working area.

Aligning the machine with the ground:

With the machine lowered, check that its full width is resting on the ground.

Never load the attachment with anything but the weight of the machine.

Turning with the machine connected:

To prevent undue strain never make turns with the machine resting on the ground. To make a turn, lift the machine slightly, make the turn, straighten up, then lower the machine and continue working in the set position.

Disconnecting the machine from the compactor:

Before disconnecting the machine from the compactor:

1. Stop the shredder machine.
2. Set the machine on the ground and turn off the compactors engine, making certain that it is completely stable.
3. Operate the spool valves in all directions to drain the oil into the tank and remove any residual pressure in the hoses.

When the machine is disconnected, always place it in the special housing and protect it from soil and dirt, which could cause serious damage.

Maintenance:

All maintenance operations must be performed with machine set on the ground, the compactor engine off, the key removed from the ignition and using the appropriate personal protection equipment in compliance with the laws in force. Make certain that no one approaches or enters the compactor while maintenance operations are being performed.

Lubrication:

Good machine maintenance requires that the unit be regularly lubricated with the proper amount of grease and oil. Before starting to lubricate the machine, clean all lubrication points with a rag to prevent introducing impurities through the lubrication points using a grease gun with a clean nozzle.

Oil leaks:

If any of the machine components leak oil, immediately recover the oil and repair the component. Leaking oil onto the ground causes pollution.

Sharpening the blades:

For the machine to function properly, the blades must be sharpened whenever they are excessively worn.

Storage:

Before storing the machine at the end of the season, it must be cleaned. For this purpose it is best to use with a jet of steam both on the outside and inside the body. Remove all residues of mulch and then dry and lubricate the machine. Do not store in a damp place or exposed to the elements. Never clean the machine and its parts while they are running. The machine must be off and isolated from the compactor.

4.2. Environmental requirements

Manufacturing:

Using environmentally friendly manufacturing equipment and avoiding long-distance transportation of the parts will decrease pollution.

Use:

While using the projected device, pollution came from the energy consumed and greases/lubricants disposal.

Grease or lubricants disposal must be according to the law.

The compactor machine has an efficiency-eco-mode that allows the compactor going in auto-shift when is not operating at maximum torque conditions, optimizing performance and thus saving fuel.

Dismantling, demolition, and disposal:

When the machine or any of its components are broken, worn or have reached the end of their expected lifespan and can no longer be used or repaired, they must be scrapped and disposed of in an appropriate manner. When the machine is worn out and has become obsolete, deliver it to a specialized demolition company for disposal.

Never leave parts of the machine in the environment and always dispose of oil through specialized contractors.

5. Economic calculation

5.1. Project cost calculation

In this section it is calculated the cost of producing one unit of the product purpose of this project, including the product design expenses and the means needed for manufacturing and construction.

Table 5.1-1: Standard parts prices

Nr.	Item	Standard	Note	Q.	Price (Eur.)	Sum. (Eur.)
1	120 steel external basic duty retaining ring	ANSI B27 7M 3AM1		4	1.75	7.00
2	Key M32 x 18 x 180 Form A	ISO-2492		4	15.00	60.00
3	Key M22 x 14 x 70 Form A	ISO-2492		2	15.00	30.00
4	M16x65 steel grade a hexagon head bolt	ISO 4014		80	3.00	240.00
5	M16 steel grade A hexagon nut	ISO 4032		80	0.30	24.00
6	16x30 steel grade a normal series plain washer	ISO 7089		80	0.15	12.00
					Sum:	**373.00**

Table 5.1-2: Catalogue parts prices

Nr.	Item	Material	Note	Q.	Price (Eur.)	Sum. (Eur.)
1	Piston radial motor: ROTATORY POWER SMA1600 C1 XXX	-		2	325.00	650.00
2	Roller bearing: SKF NJ 2324 ECM	-		8	30.00	240.00
3	Rigid coupling: GTR/S bore Φ80 H7 A1 bore Φ90 H7 A1	-		2	58.00	116.00
					Sum:	**1006.00**

Table 5.1-3: Stock parts cost

Nr.	Item	Material	Note	Q.		Price (Eur.)		Sum. (Eur.)
1	Hex. Rod	Steel AISI 1050 Norm.	F150 mm	4.60	m	12.00	Eur/m	55.20
2	Tube	S. Steel 420 hard. and temp.	Φ190/120 mm	0.18	m	9.00	Eur/m	1.62
3	Bar	Steel AISI 1095 Norm.	Φ800 mm	1.00	m	7.00	Eur/m	7.00
4	Plate	Steel 414HT	t= 92.5 mm	0.01	m^2	30.00	Eur/m^2	0.36
5	Plate	Steel 414HT	t= 86 mm	9.60	m^2	25.00	Eur/m^2	240.00

6	Plate	Aluminium 3003 alloy	t= 8 mm	0.01	m^2	10.00	Eur/m^2	0.12
7	Plate	Aluminium 3003 alloy	t= 1 mm	0.01	m^2	6.00	Eur/m^2	0.07
8	Plate	Aluminium 3003 alloy	t= 3 mm	20.00	m^2	7.50	Eur/m^2	150.00
9	□ Cold forming profile	Stainless steel AISI 316L	200x100x10 mm	11.00	m	20.00	Eur/m	220.00
10	□ Cold forming profile	Stainless steel AISI 316L	100x100x10 mm	4.00	m	15.00	Eur/m	60.00
11	C Cold forming profile	Stainless steel AISI 316L	200x100x10 mm	3.00	m	18.00	Eur/m	54.00
12	L Cold forming profile	Stainless steel AISI 316L	200 x200 x10 mm	2.80	m	22.00	Eur/m	61.60
13	L Cold forming profile	Stainless steel AISI 316L	700x500x100 mm	0.50	m	50.00	Eur/m	25.00
14	Plate	Stainless steel AISI 316L	t= 30 mm	1.80	m^2	18.00	Eur/m^2	32.40
15	Plate	Stainless steel AISI 316L	t= 3 mm	1.40	m^2	10.00	Eur/m^2	14.00
16	Plate	Stainless steel AISI 316L	t= 10 mm	0.80	m^2	15.00	Eur/m^2	12.00
17	Rectangular rod	Stainless steel AISI 316L	250x150 mm	1.30	m	8.00	Eur/m	10.40
18	H Cold forming profile	Stainless steel AISI 316L	800x450x30 mm	0.50	m	80.00	Eur/m	40.00
							Sum:	**983.772**

The total cost of the supplies is 2362.77 Eur.

Transportation cost could be calculated as the 10% of the supplies price:

$$P_T = P_s * 0.1 = 236.28 \text{ Eur.} \quad (5.1\text{-}1)$$

Therefore, the total cost of supplies placed in the workshop will be:

$$C_s = P_s + P_T = 2599.05 \text{ Eur.} \quad (5.1\text{-}2)$$

The necessary equipment for manufacturing and construction is going to be hired for 1 month. Except tools and consumables which are purchased. The approximate costs in a month of the equipment are in table 5.1-4.

Tools depreciation period is 2 years and tools will be used 1 month. Therefore, the tools depreciation cost in a month is:

$$\mathrm{Dt} = \frac{\mathrm{Pt}}{2\ \mathrm{years} * 12\ \mathrm{months}} = 16.76\ \mathrm{Eur} \quad (5.1\text{-}3)$$

Where: Dt is the depreciation cost and Pt is the purchased cost.

Table 5.1-4: Equipment

Nr.	Equipment hiring (including transportation)		Price (Eur./year)
1	CNC multi task machine: Mazak Integrer i-20027		2000.00
2	CNC machining center: GF HSM 800 LP28		2000.00
3	Water jet cutting center: ESAB Hydrocut LX 400029		3000.00
4	Punching press: Foremost 5B, RMT Toggle press30		1500.00
5	Welding center: ESAB Railtrac 1000		1000.00
6	Inspection tools and instruments		2500.00
	Equipment purchasing (including transportation)		
7	Tools and consumables		400.00
8	Tools depreciation		16.67
		Sum:	**12416.67**

The approximate costs of the manufacturing tasks for the design parts are in table 5.1-5. Heat treatment and painting are carried out by sub-contractors.

Table 5.1-5: Manufacturing tasks

Nr.	Task		Price (Eur.)
1	Turning		1500
2	Milling		1500
3	Water jet cutting		4000
4	Press cutting		100
6	Heat treatment		500
8	Drilling		800
10	Painting		200
		Sum:	**8600**

Although the control and power systems have not been the developed in this project, the approximate expenses of them are in table 5.1-6.

Table 5.1-6: Control and power equipment

Nr.	Item		Price (Eur.)
1	Electrical box		300
2	Power cables		200
3	Control box		100
4	Switches		100
5	Electrical wires		200
6	Optical sensors		700
7	Time relays		200
8	control box labels, warning notices		50
9	Contact group, sockets, cables channels		100
		Sum:	**1950**

Consumables needed for the well performing of the machine are in table 5.1-7.

Table 5.1-7: Consumables

Nr.	Item	Price (Eur.)
1	Gear lubricants	50
2	Greases for bearings	25
4	Hydraulic fluid	50
	Sum:	**125**

Construction and assembly works: In order to design and assemble the entire structure will require the designer, an engineer, technician, automation engineer, turner, miller, welding technician, an electrician and a mechanic. The cost associated to these professionals is in the table 5.1-8. The works will be completed within a month.

Table 5.1-8: Wages

Nr.	Function	Hourly wage (Eur.)	Daily working hours (h)	Monthly wages (Eur.)
1	Designer	20	8	3200
2	Constructor	15	8	2400
3	Manager	15	8	2400
4	Technician	10	8	1600
5	Automation engineer	10	8	1600
6	Electrician	10	8	1600
Total:				12800
Tax 24%:				3072
Sum:				**15872**

Preparation of the workplace: Before beginning any manufacturing or assembly task, it is necessary to prepare the workplace for the production of a new product. The cost of this task is in the table 5.1-9.

Table 5.1-9: Preparation of workplace costs

Nr.	Item	Price (Eur.)
1	Coordination costs	600
2	Workshop tuning	1000
3	Machines tuning	100
4	Personal protective equipment	400
5	Unanticipated additional costs	3000
	Sum:	**5100**

The entire cost of the project will consist of:

- Standard, catalogue and stock parts costs (table 5.1-1, table 5.1-2, table 5.1-3)
- Equipment hiring costs (table5.1-4)
- Manufacturing tasks costs(table5.1-5)
- Control and power equipment costs (table5.1-6)
- Consumables costs (table5.1-7)
- Wages (table5.1-8)
- Preparation of workplace cost (table5.1-9)

The project total cost is:

C = 2599 + 12400 + 8600 + 1950 + 125 + 15872 + 5100 = **46662.72 Eur.** (5.1-4)

The net profit will be approximately the 20% of the total cost:

N = P * 0.20 = 9332.54 Eur. (5.1-5)

Therefore, the market price of the product will be:

P = C + N = 55995.26 Eur. (5.1-6)

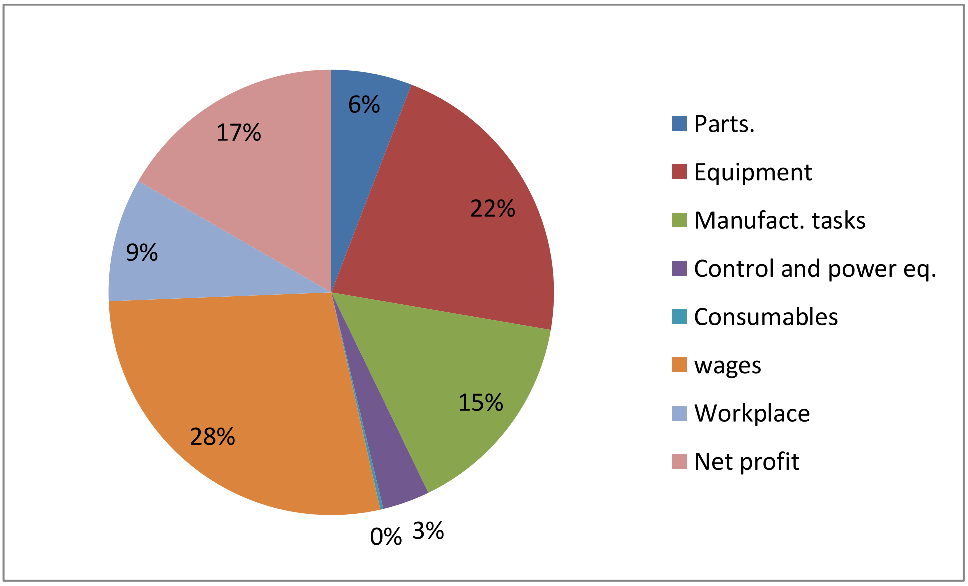


Figure 5.1-1: Manufacturing cost graphic.

5.2. Break-even point calculation

[References: 4, 34, 35]

Device payback period is the period of time when the costs associated with the purchase of one unit of the product are equivalent to the savings obtained.

In this case the break-even point depends on the amount of garbage generated by a city or region. Because of that, I am going to make the calculations for a specific region, for example: Spain.

In Spain the amount of garbage generated each year is about 10 million Tons[4], 60% goes to landfills[4], and there are approximately 3700 operating landfills[34], 125 of them are controlled landfills[34]. Therefore, the tons per year that one landfill process is about:

$$m = (10*10^6 \text{ Tons/year}) * 0.6 * / 3700 = 1621.6 \text{ Ton/year} \qquad (5.2\text{-}1)$$

Density of the garbage without compaction and with compaction is:

$P_{wc} = 650 \text{ Kg/m}^3$ $P_c = 1000 \text{ Kg/m}^3$

The new technology developed on this project increases compaction even more. Therefore, compaction increased around 10%: $P_c' = 1100 \text{ Kg/ m}^3$

Volume of gas generated by the landfill is: $V_G = 375 \text{ m}^3\text{/Ton}$. And, as we know from the reference review, higher compaction generates more gas. Therefore, the volume of gas generated increases around 10% using the new technology:

$V_G' = 412.5 \text{ m}^3\text{/Ton}$

Energy recovery from gas is about 20 MJ/ m^3. But, due to leaks, only 70% is profitable. Therefore, energy obtained is: $E = 14 \text{ MJ/ m}^3$

Knowing that the price of MJ in Spain is about 0.0392 Eur/MJ and the price of land in Spain is about 70 Eur/ m^2, it is possible to estimate the landfill cost per year before and after the introduction of the new technology.

Landfill annual operating cost is about 2 million Eur/year (OC)[35]. This figure is not going to vary with the new technology.

Price of energy per cubic meter is:

$$P_E = E * P_{MJ} = 14 \text{ MJ/ m}^3 * 0.0392 \text{ Eur/MJ} = 0.55 \text{ Eur/ m}^3 \qquad (5.2\text{-}2)$$

Depth of the landfill will be about 10 m. Therefore, price of land per m^3 will be:

$$P_L = (70 \text{ Eur/ m}^2) / (10\text{m}) = 7 \text{ Eur/ m}^3 \qquad (5.2\text{-}3)$$

Landfill cost per year will be the difference between the cost of the land and the cost of the energy recovered.

Landfill cost per year with current technology: (5.2-4)

C_{before} = OC + (m / P_c * P_L) – (m * V_G * P_E) = (1621.6 Ton/year / 1000 Kg/m^3 * 7 Eur/ m^3) – (1621.6 Ton/year * 375 m^3/Ton * 0.55 Eur/ m^3) = 1.665.556,4 Eur/year

Landfill cost per year with new technology: (5.2-5)

C_{after} = OC+ (m / P_c' * P_L) – (m * V_G' * P_E) = (1621.6 Ton/year / 1100 Kg/m^3 * 7 Eur/ m^3) – (1621.6 Ton/year * 412.5 m^3/Ton * 0.55 Eur/ m^3) = 1.632.109,8 Eur/year

Savings:

$S = C_{after} - C_{before}$ = 33446.5 Eur/year (5.2-6)

Product market price is:

P = 55334.73 Eur.

Number of years needed for recover the machine cost is:

Y = P / S = 1.65 years (5.2-7)

However, it would be necessary more than one unit of the product to get an efficient compaction. Therefore, if we consider that 3 units are purchased:

Y = (P*3) / S = 4.96 years (5.2-8)

Therefore, the break-even point is in 5 years since the introduction of the new technology.

The following figure (figure 5.2-1) shows the break-even point:

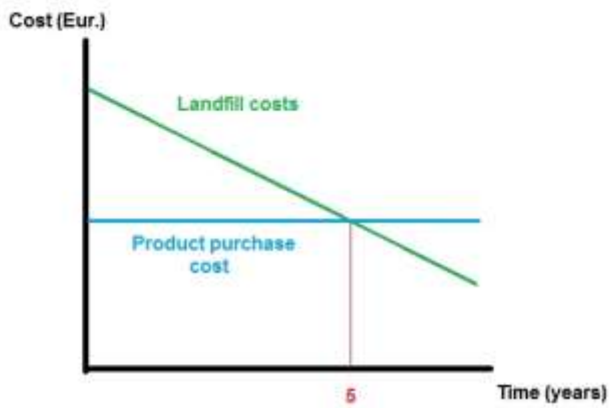


Figure 5.2-1: Break-even point.

This calculation has not taken into account the possible variation in land prices or in energy prices throughout the years. Therefore, the break-even point could be in less than 5 years in the case that land prices or energy price increase, possibilities that are not far from reality.

Conclusions

The initial conception of this project comes from the idea of trying to improve landfill operations. Although landfills are not the even close of been environmentally friendly, they are and will continue been used as final disposal of waste because recycling is not enough for the total solid waste generation. However, landfills have been improved by minimizing contamination and generating energy gas, these are called controlled or sanitary landfills. While waste compaction is important in any landfill because it saves space, in gas energy recovery landfills it is crucial because more compaction means more gas generation.

For all the above reasons, shredding garbage before compacting will significantly improve landfill operation. This system is already implemented in landfills through stationary shredders machines: solid waste is first shredded in a shredding center and then transported to the final place in the landfill and compacted. However, if distribution, compaction and shredding were carried out at the same time, it would be more economically efficient. From this point, the idea of designing a shredder attachment for a compactor machine came out.

The device consists of two hydraulic shredders connected by a central box that hold in two radial piston hydraulic motors. Each shredder is motioned by its own motor and consists of shafts connected by gear transmission.

After design and calculations have been made, many possible improvements have come to light. For example, SSI shredder company has changed his shaft design from hexagonal to key-shat arguing that in case of wear it is less expensive to change just a key than to change the shaft and the blades. Another possible improvement could be changing the integral blades for blades with inserts which are less expensive to replace than the integral blade in case of wear, solution by UNTHA company.

Referring economics, the calculated payback period is 5 years. However, in the case that land prices or energy price increase, possibilities that are not far from reality, this period could be reduced. Closer studio of the materials and technology process expenses could reduce the payback period as well.

Table list

Picture list

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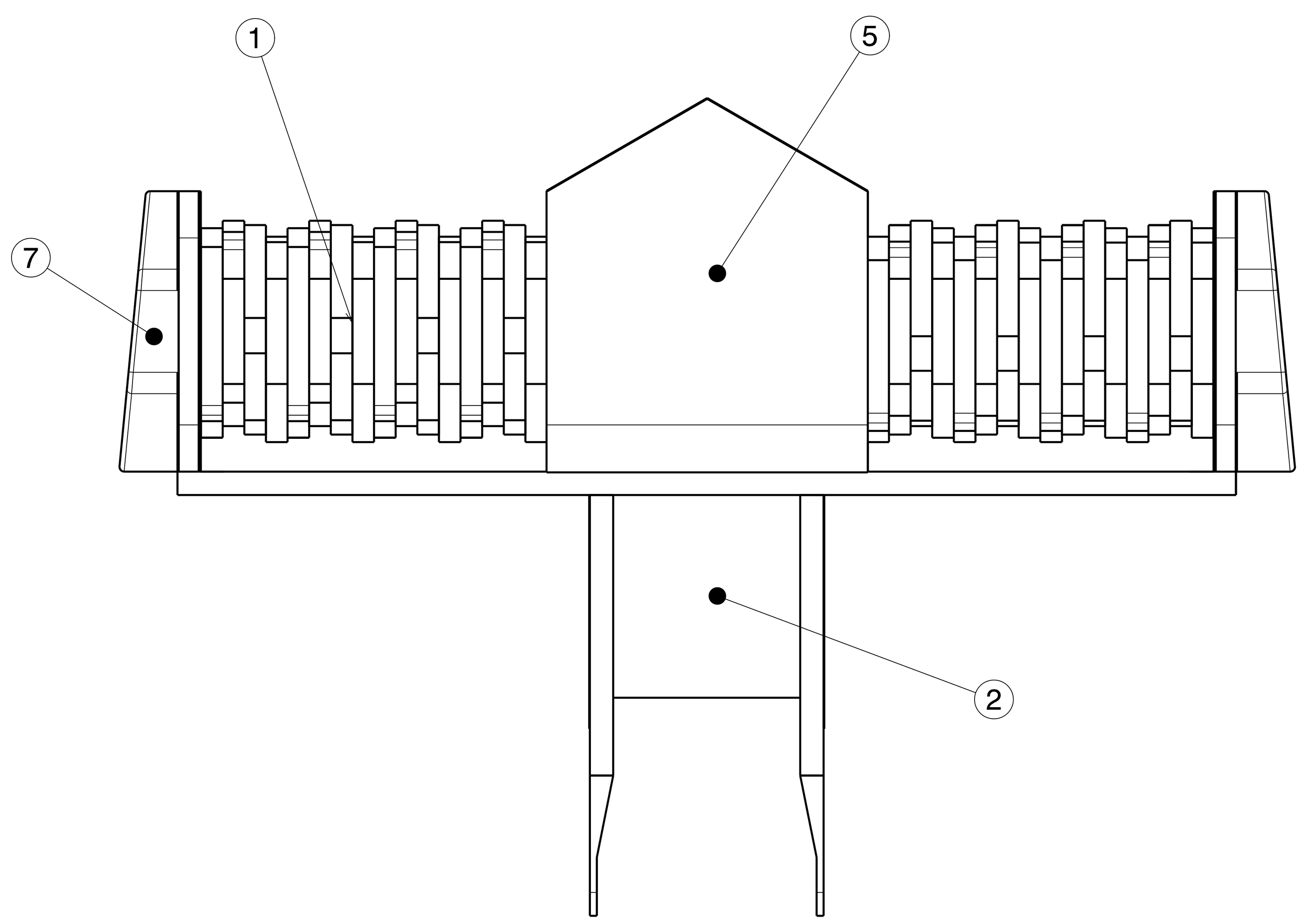
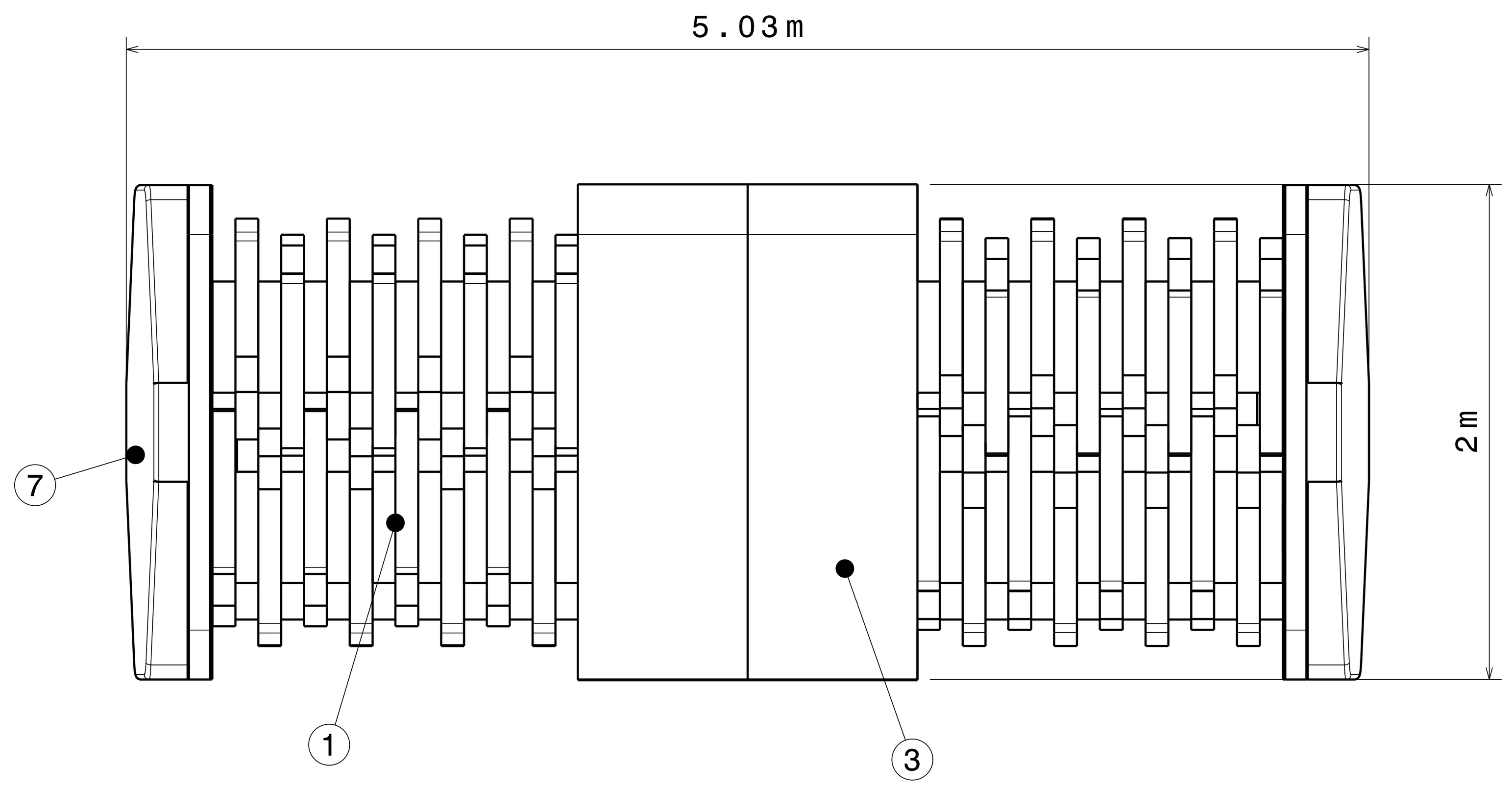
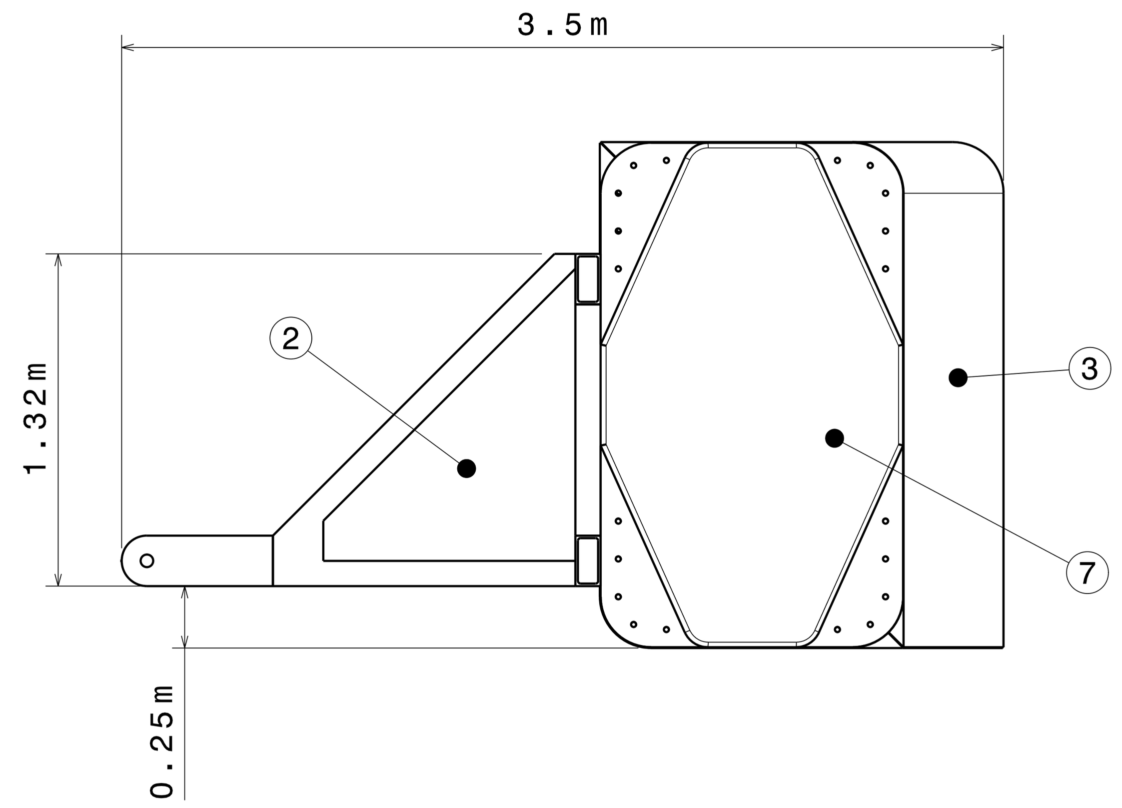
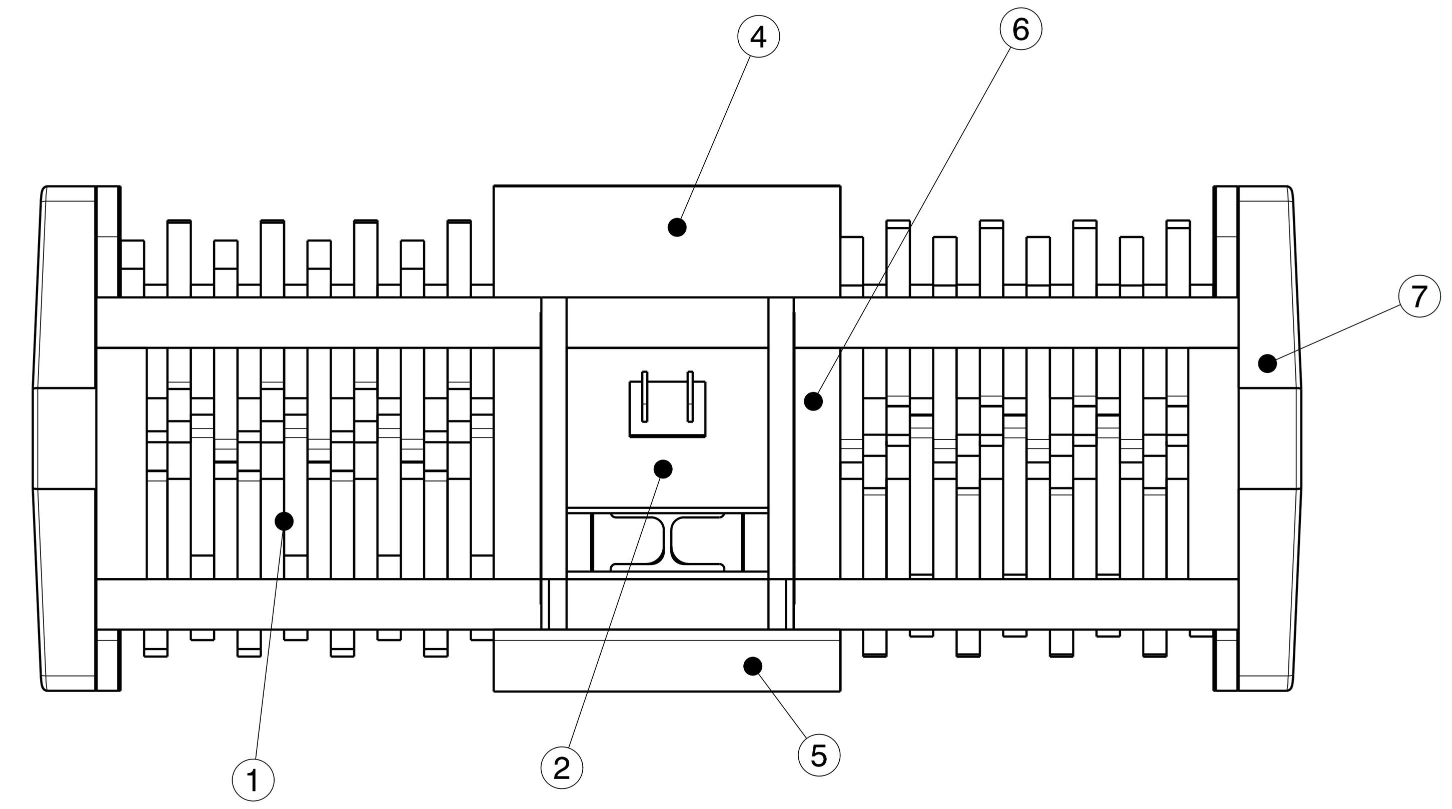
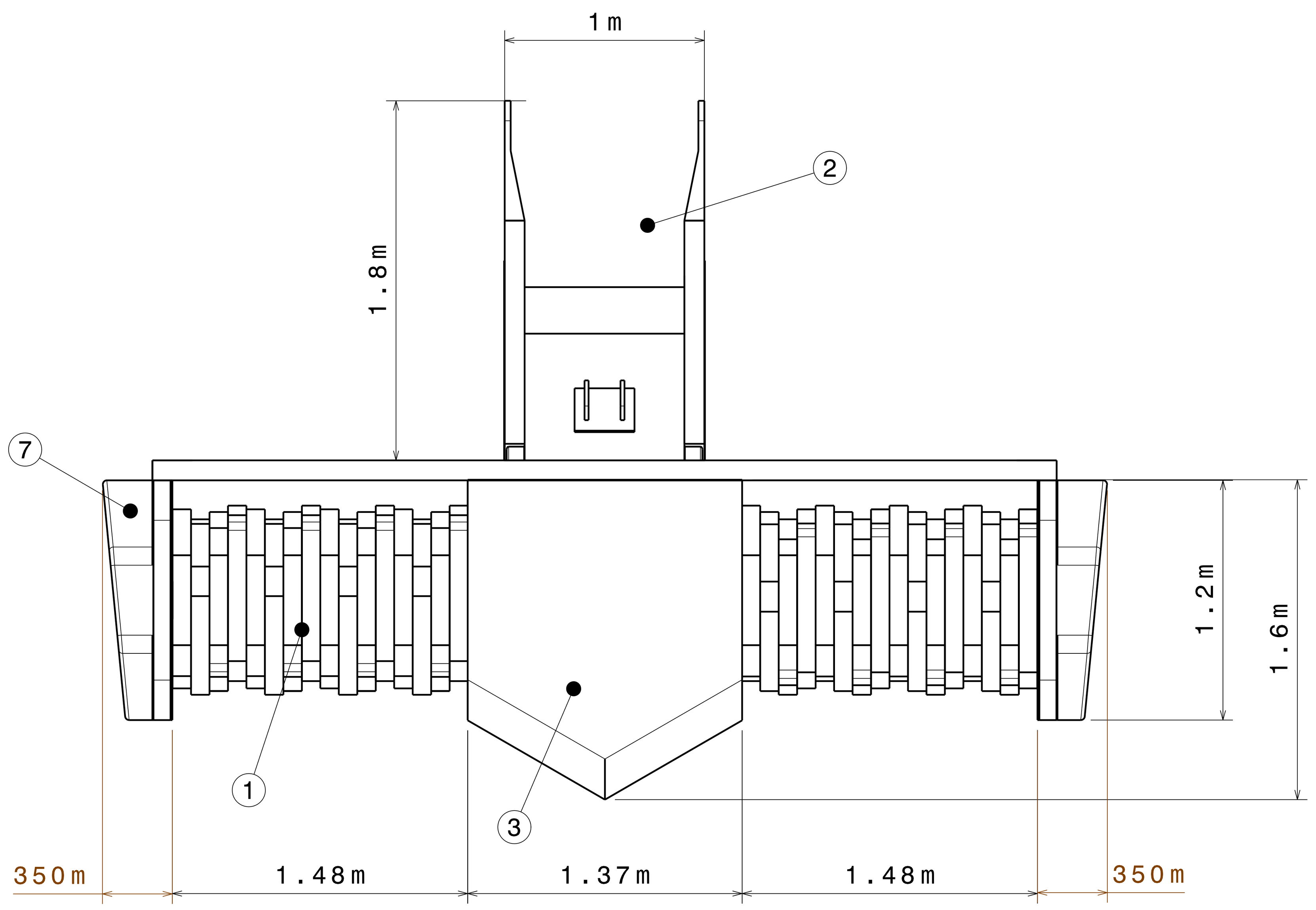


Right and front views of waste compactor with the shredder attachment
Scale: 1:50

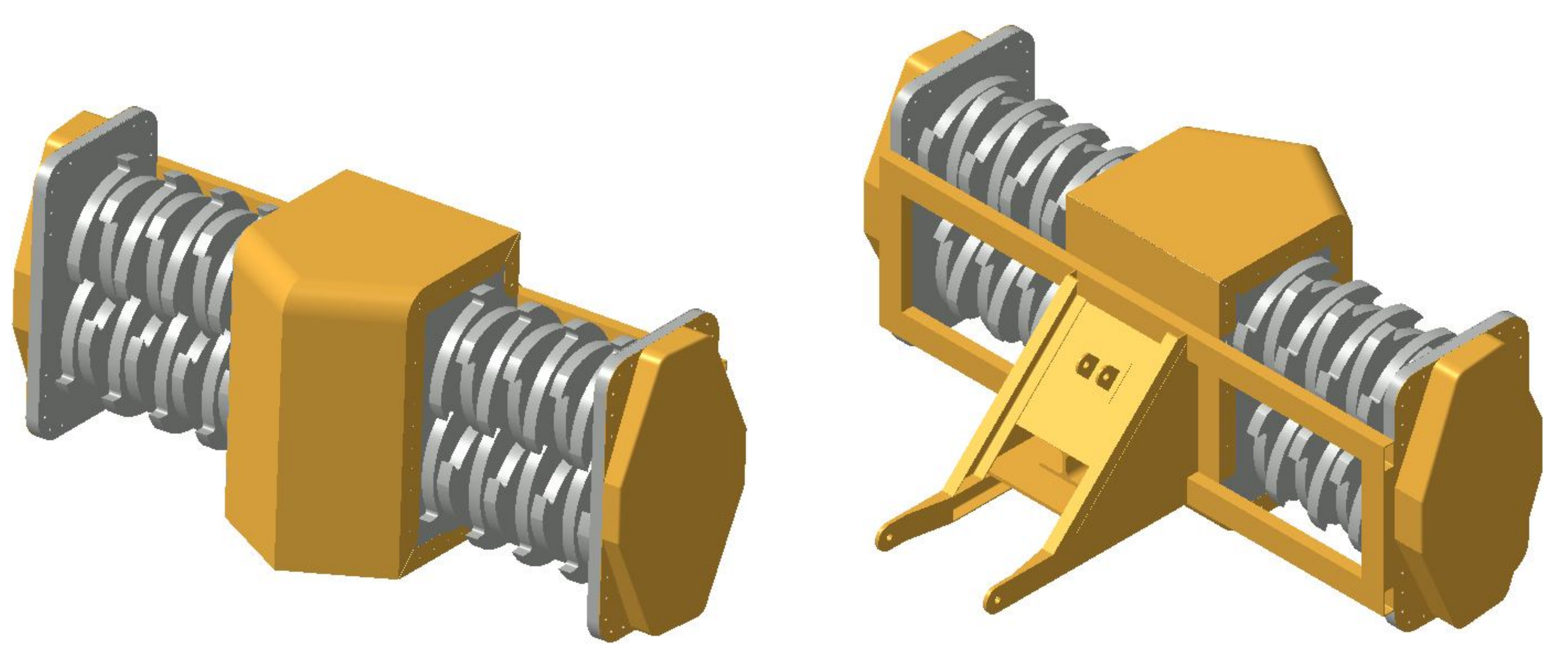








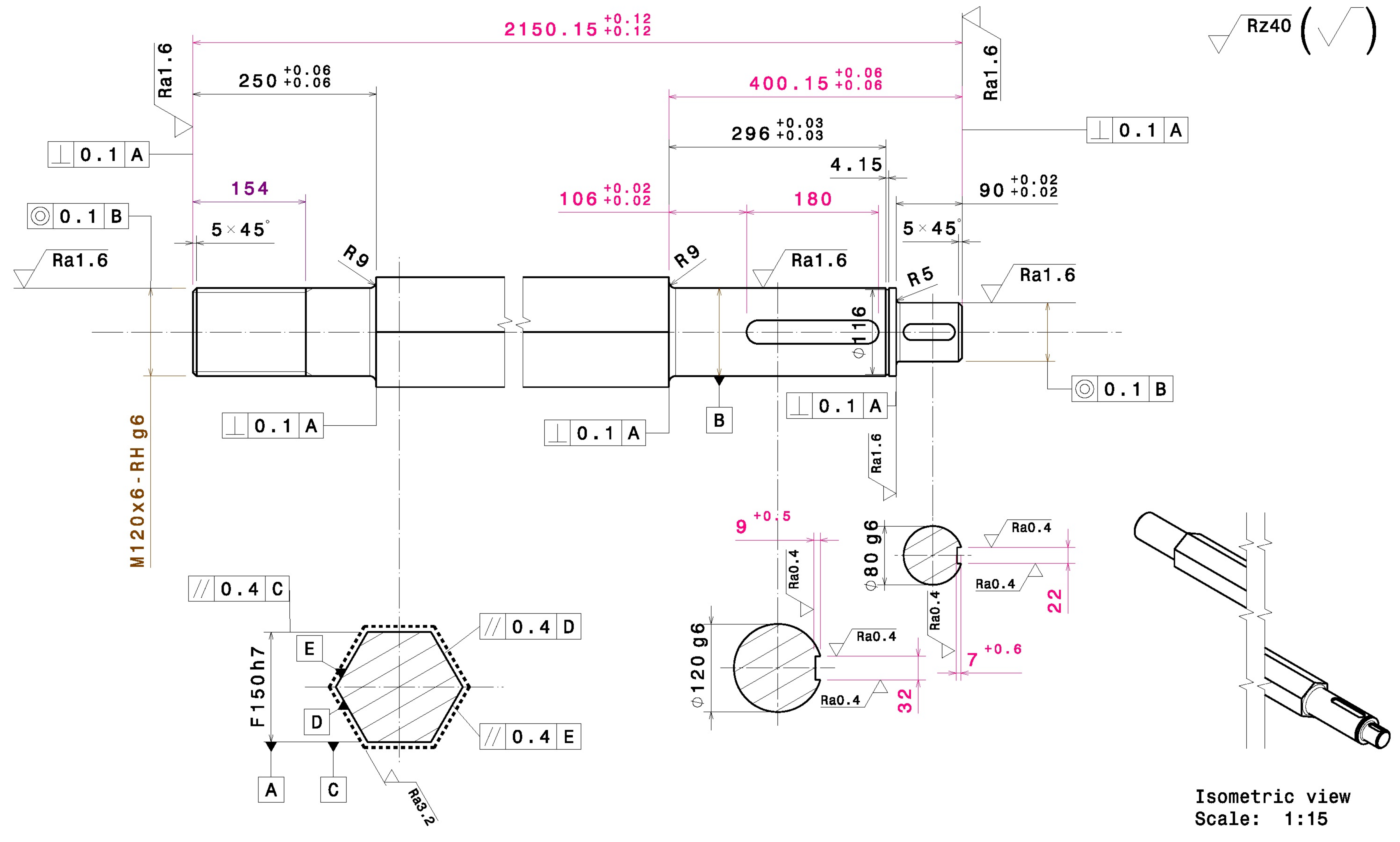
Isometric views of the shredder attachment
Scale: 1:40



1. General dimmensions = 5 x 3.5 x 2 m
2. Total weight of product = 289.5 kN
3. Total power required = 95 kW
4. Rotational speed = 30 rpm
5. Maximum pressure = 980 bar

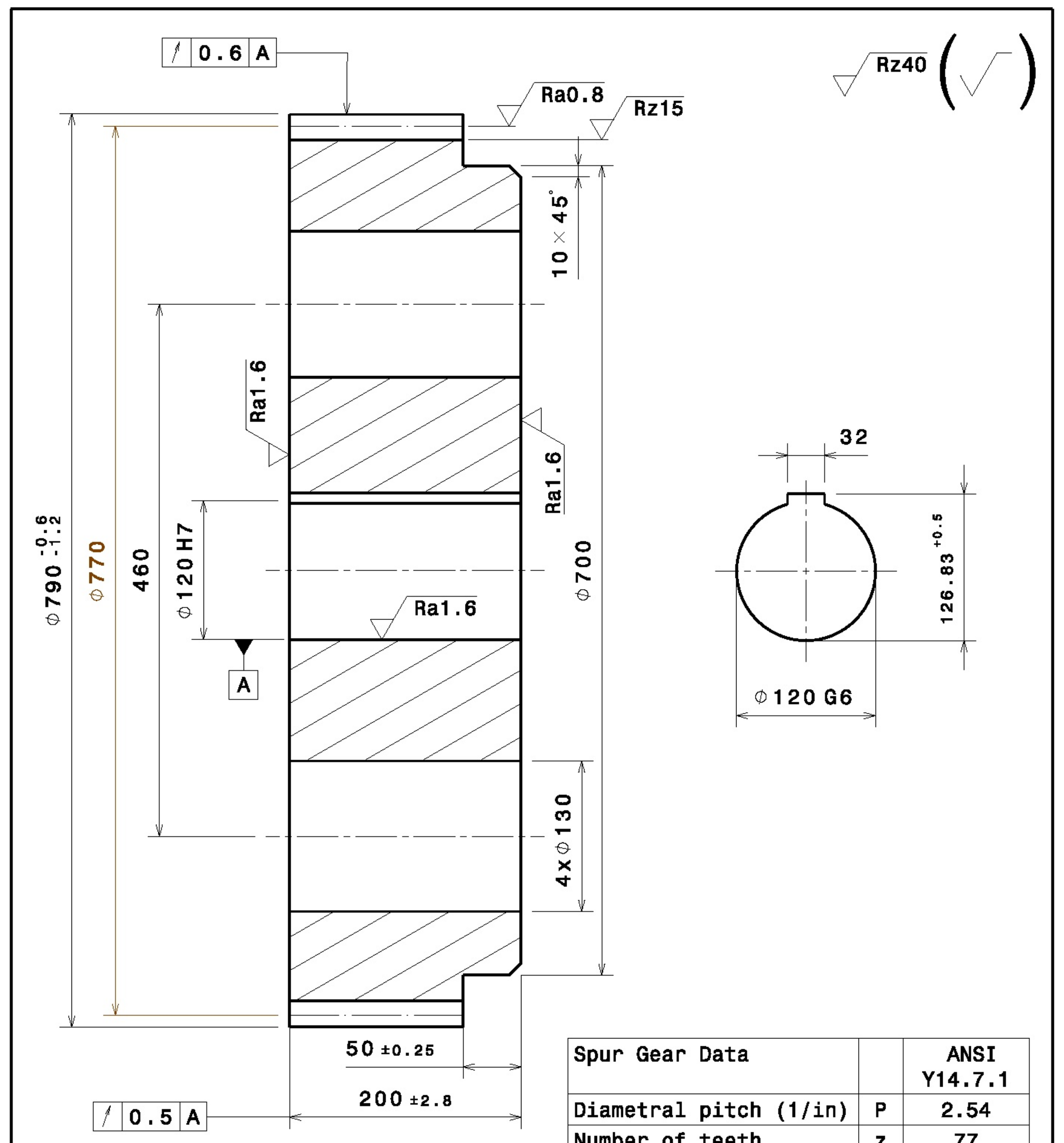
Reference N°.		Other information:	Material:	Scale: 1:25
Responsibility: Dprt. of Mech.	Consulted by:	Document type: General drawing		Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Shredder attachment	Drawing N°. MPfu 11.01.001	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 1/12

Reference N°:	Other information:	Material:	Scale: 1:10
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Assembly drawing	Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Shredder attachment assembly	Drawing N°: MPfu 11.02.001
	Checked by: Andrius Gedvila		Rev.: B · Date: 2015.06.04 · Lang.: en · Sheet: 2/12



Isometric view
Scale: 1:15

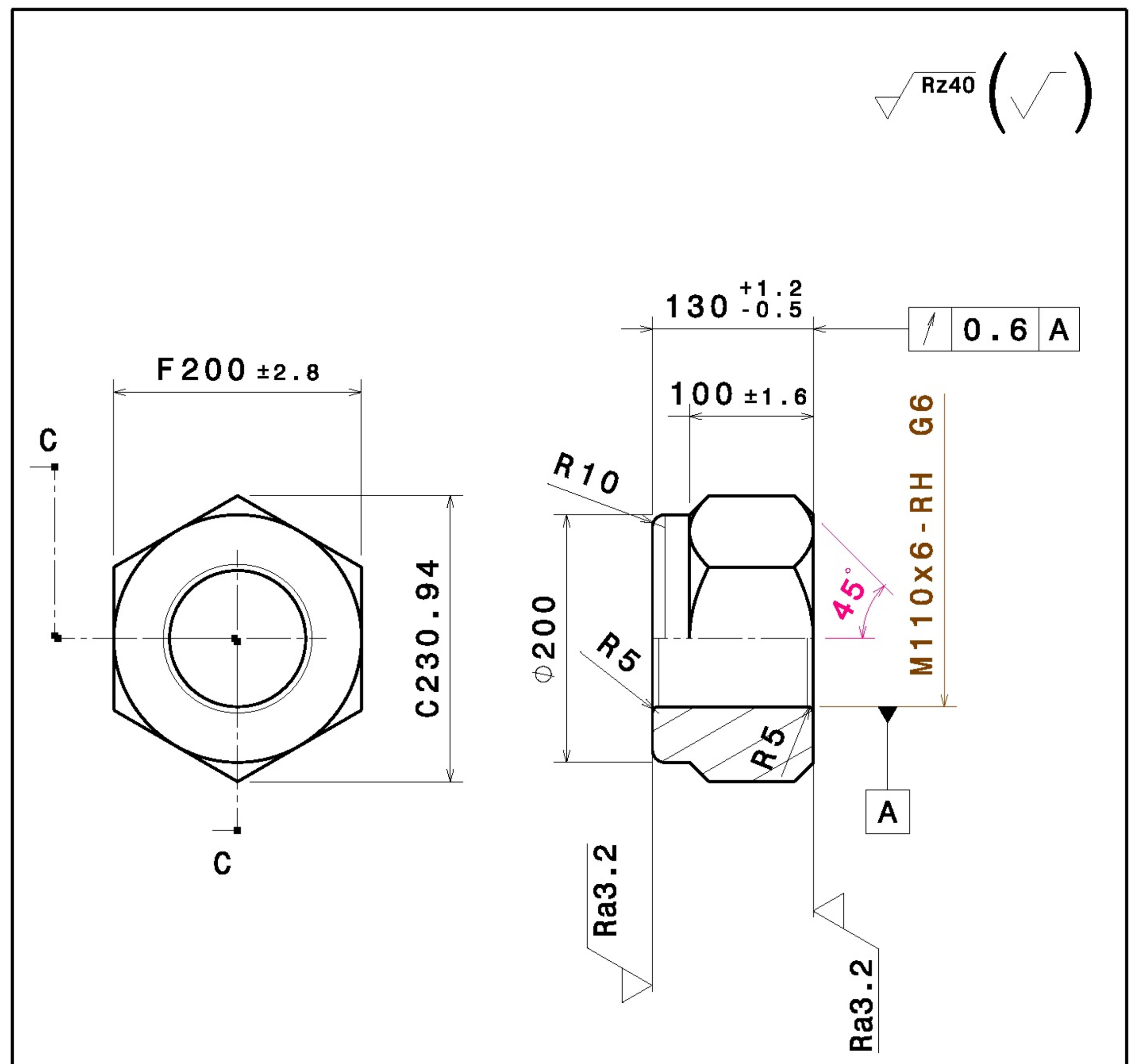
Reference Nº.	Other information:	Material: Steel AISI 1095 Normalized at 900ºC		Scale: 1:5
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Motor shaft	Drawing Nº.: MPfu 11.03.001	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 3/12



Spur Gear Data		ANSI Y14.7.1
Diametral pitch (1/in)	P	2.54
Number of teeth	z	77
Pressure angle		20°
Pitch diameter	Dp	770
Base circle diameter	Db	723.564
Circular pitch	p	31.416
Circular thickness	t	15.708
Root diameter	Dd	745

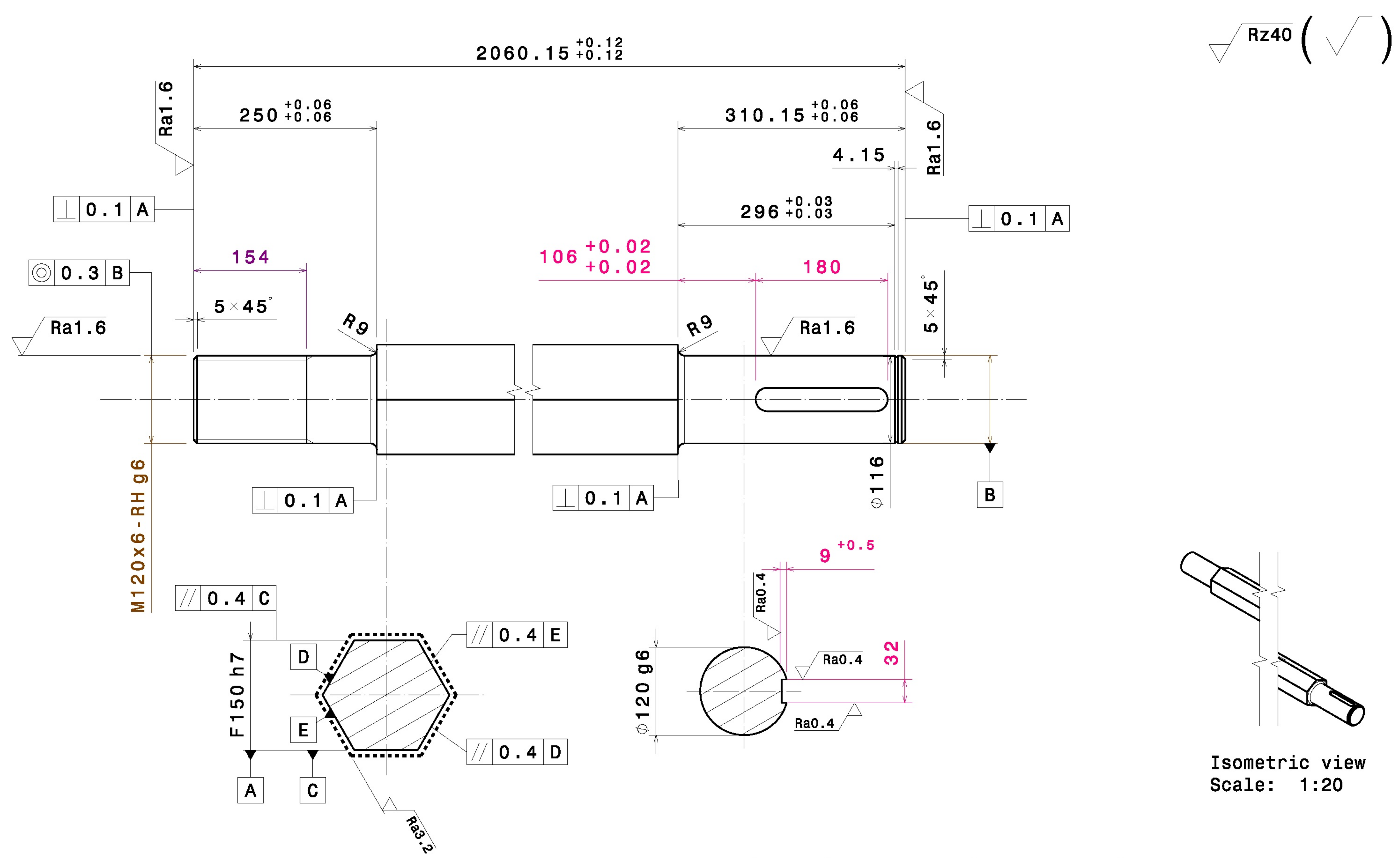
1. Dimensions and tolerances per ASME Y14.5-2009.
2. AISI 1050 Normalized HB 217.
3. Sharp edges round off.

Reference Nº.	Other information:	Material: Steel AISI 1050 Normalized at 900ºC		Scale: 1:5
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Gear	Drawing Nº.: MPfu 11.03.001	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 5/12



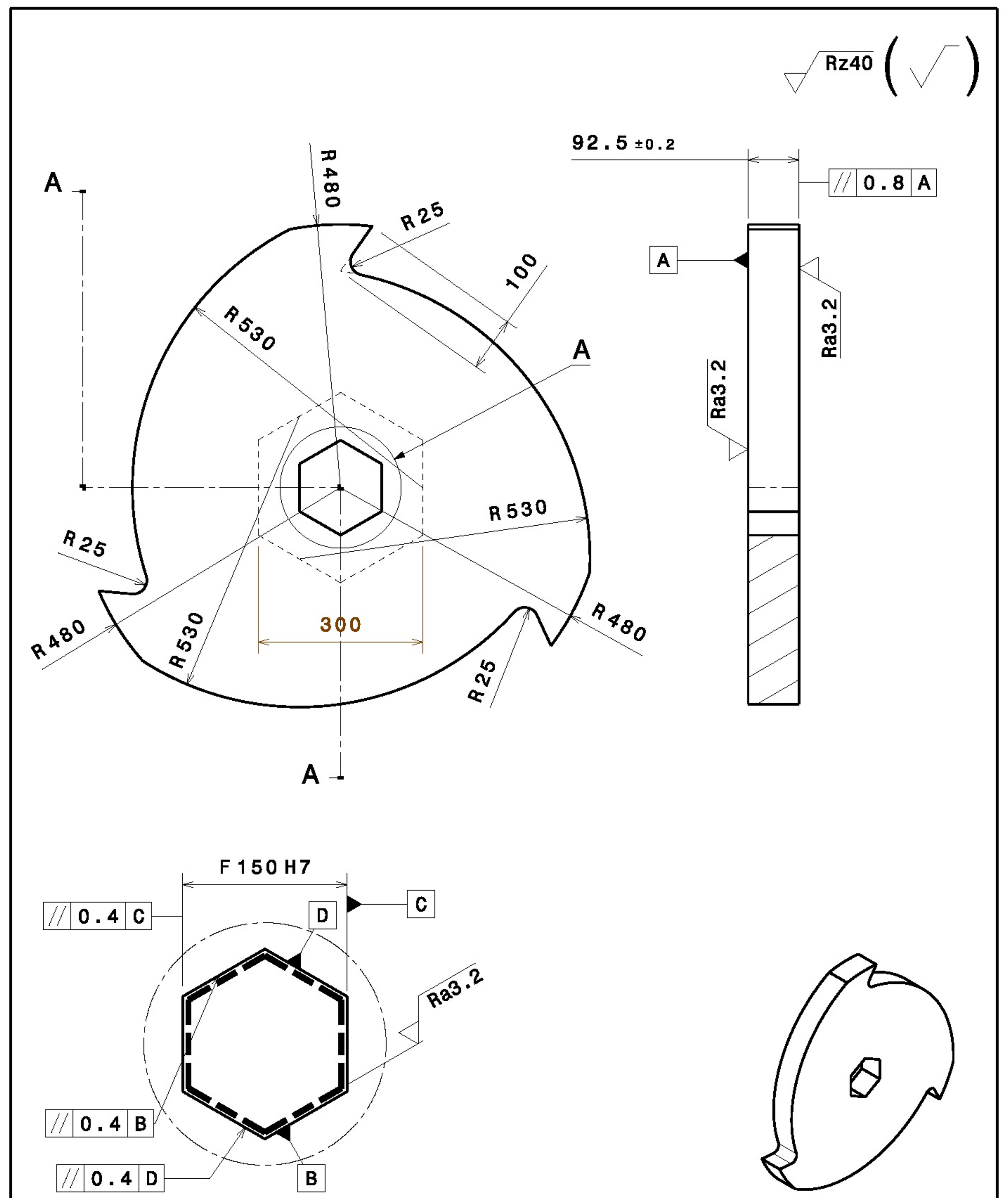
1. Sharp edges rounded off

Reference Nº.	Other information:	Material: Steel AISI 1050 Normalized at 900ºC		Scale: 1:5
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Nut	Drawing Nº.: MPfu 11.03.005	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 6/12



Isometric view
Scale: 1:20

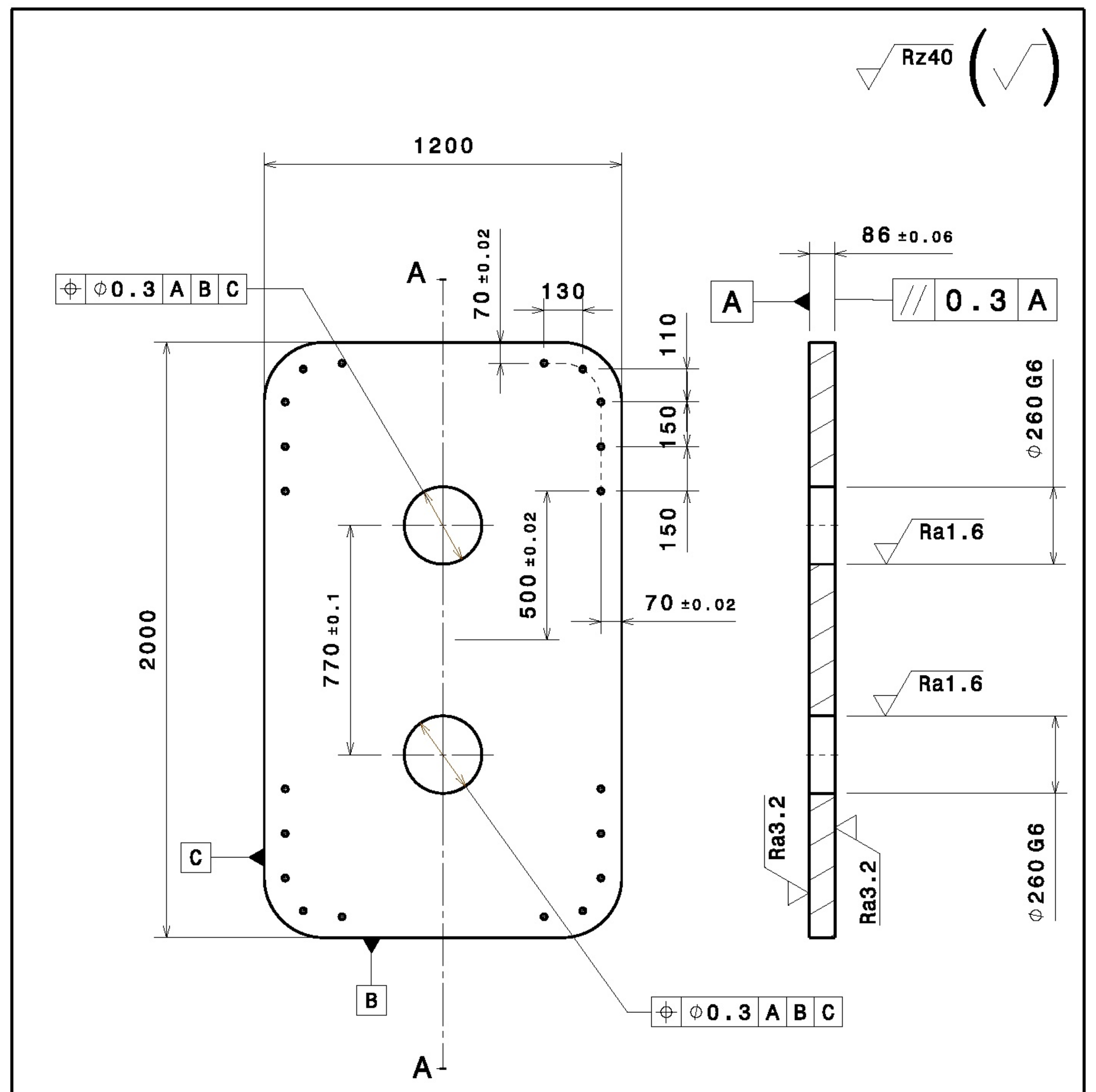
Reference Nº.	Other information:	Material: Steel AISI 1095 Normalized at 900ºC		Scale: 1:5
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Driven shaft	Drawing Nº.: MPfu 11.03.002	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 4/12



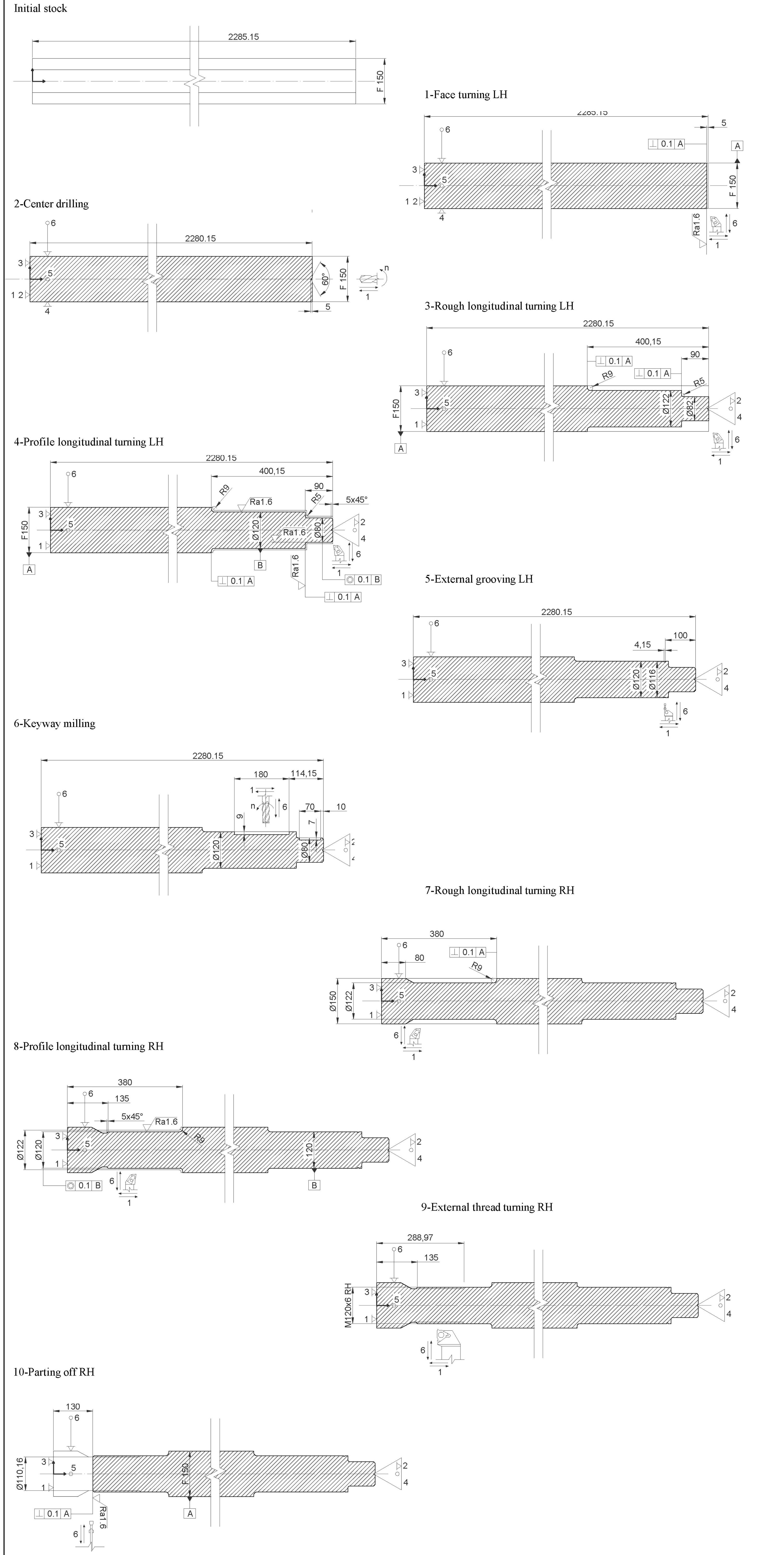
Detail A
Scale: 1:5

Isometric view
Scale: 1:20

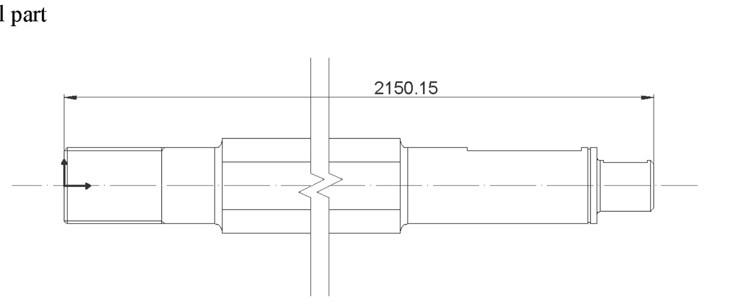
Reference Nº.	Other information:	Material: Steel 414HT		Scale: 1:10
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Blade	Drawing Nº.: MPfu 11.03.006	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 7/12



Reference Nº.	Other information:	Material: Steel 414HT		Scale: 1:20
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Part drawing	Document status: Educational	
Owner: VGTU MPfu-11	Drawn by: Sara Escudero	Title: Bearing housing	Drawing Nº.: MPfu 11.03.007	
	Checked by: Andrius Gedvila		Rev. B / Date: 2015.06.04 / Lang. en	Sheet: 8/12



Final part



Manufacturing sheet

Part name:	Motor shaft
Part material:	AISI/SAE 1095 293HB
Stock:	Hex. Rod: section F150, length 2285.15
Clamping device:	3 jaw chuck and dead center
Machine:	Multitask 01- Medium (8-12" chuck)
n min =	1 rpm
n max =	5000 rpm
P max =	22 kW
T max =	500 Nm

Nr. 1 **Operation:** Face turning LH

T1 **Insert:** CNMG 19 06 24-PR GC4325 **Tool:** DCLNL 2525M 19

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	150	-	5	0.359	950	268.470	18.875	10.56	1.68

Nr. 2 **Operation:** Center drilling

T2 **Tool:** 860.1-0510-019A1-PM 4234

Pass	D (mm)	L (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)
-	5	0	0.174	10800	170	2,49	0.00

Nr. 3 **Operation:** Rough longitudinal turning LH

T1 **Insert:** CNMG 19 06 24-PR GC4325 **Tool:** DCLNL 2525M 19 **L (mm)** = 400.15 **L (mm)** = 90

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	150	140	5	0.5	412	181.115	17.734	116.55	3.26
1	140	130	5	0.5	578	235.940	23.102	83.08	3.26
1	120	110	5	0.5	530	183.062	17.925	20.38	3.26
1	110	100	5	0.5	618	194.052	19.001	17.48	3.26
1	100	90	5	0.5	618	174.647	17.101	17.48	3.26
							47.432	4.25 min	

Nr. 4 **Operation:** Profile longitudinal turning LH

T1 **Insert:** CNMG 19 06 24-PR GC4325 **Tool:** DCLNL 2525M 19 **L (mm)** = 400.15 **L (mm)** = 90

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	130	120	2.4	0.359	1060	399.408	13.478	63.09	1.68
	90	80	2.4	0.359	1060	266.272	8.986	14.19	1.68
							11.232	1.29 min	

Nr. 5 **Operation:** External grooving LH

T3 **Insert:** N123F2-0265-0002-GF 1125 **Tool:** LF123F10-2525B **L (mm)** = 4.15

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	120	118	1	0.09	448	165.993	0.585	6.18	0.11
1	118	116	1	0.135	388	141.325	0.747	4.75	0.24
							0.666	10.93 min	

Nr. 6 **Operation:** Keyway milling

T4 **Tool:** 2S342-1111-038-PA 1730

1st keyway dimmensions (mm):

Depth	Width	Length	Radius
9	32	180	16

Pass AE	Pass AP	AE (mm)	AP (mm)	fz (mm/tooth)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)
2	3	10.7	3	0.0632	5360	187	9.01	50.00

2nd keyway dimmensions (mm):

Depth	Width	Length	Radius
7	22	70	11

Pass AE	Pass AP	AE (mm)	AP (mm)	fz (mm/tooth)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)
1	2	10.7	3.5	0.0632	5300	185	7.10	15.00

Nr. 7 **Operation:** Rough longitudinal turning RH

T5 **Insert:** CNMG 19 06 24-PR GC4325 **Tool:** DCLNL 2525M 19 **L (mm)** = 300

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	150	140	5	0.5	412	181.115	17.734	116.55	3.26
1	140	130	5	0.5	578	235.940	23.102	83.08	3.26
							20.418	3.33 min	

Nr. 8 **Operation:** Profile longitudinal turning RH

T5 **Insert:** CNMG 19 06 24-PR GC4325 **Tool:** DCLNL 2525M 19 **L (mm)** = 300

Pass	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	130	120	2.4	0.359	1060	399.408	13.478	63.09	1.68

Nr. 9 **Operation:** External thread turning RH

T6 **Insert:** 266RG-22MM01A600M 1125 **Tool:** 266RFG-2525-22 **Label:** M120x6 RH **L (mm)** = 154

Step	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)
1	120	118.97	0.516	6	451	168.475	20.429	3.41
2	118.968	118.12	0.426	6	451	167.269	16.745	3.41
3	118.116	117.34	0.39	6	451	166.164	15.229	3.41
4	117.336	116.74	0.3	6	451	165.315	11.655	3.41
5	116.736	116.23	0.253	6	451	164.598	9.786	3.41
6	116.23	115.79	0.222	6	451	163.969	8.554	3.41
7	115.786	115.38	0.201	6	451	163.400	7.718	3.41
8	115.384	115.01	0.185	6	451	162.876	7.081	3.41
9	115.014	114.67	0.172	6	451	162.389	6.564	3.41
10	114.67	114.35	0.162	6	451	161.930	6.165	3.41
11	114.346	114.04	0.153	6	451	161.497	5.807	3.41
12	114.04	113.75	0.145	6	451	161.086	5.489	3.41
13	113.75	113.47	0.139	6	451	160.692	5.249	3.41
14	113.472	113.21	0.133	6	451	160.316	5.011	3.41
15	113.206	112.95	0.128	6	451	159.953	4.811	3.41
16	112.95	112.7	0.124	6	451	159.602	4.651	3.41
Nr. Pass								
16	120	112.7	-	6	451	-	140.944	54.63

Nr. 10 **Operation:** Parting off RH

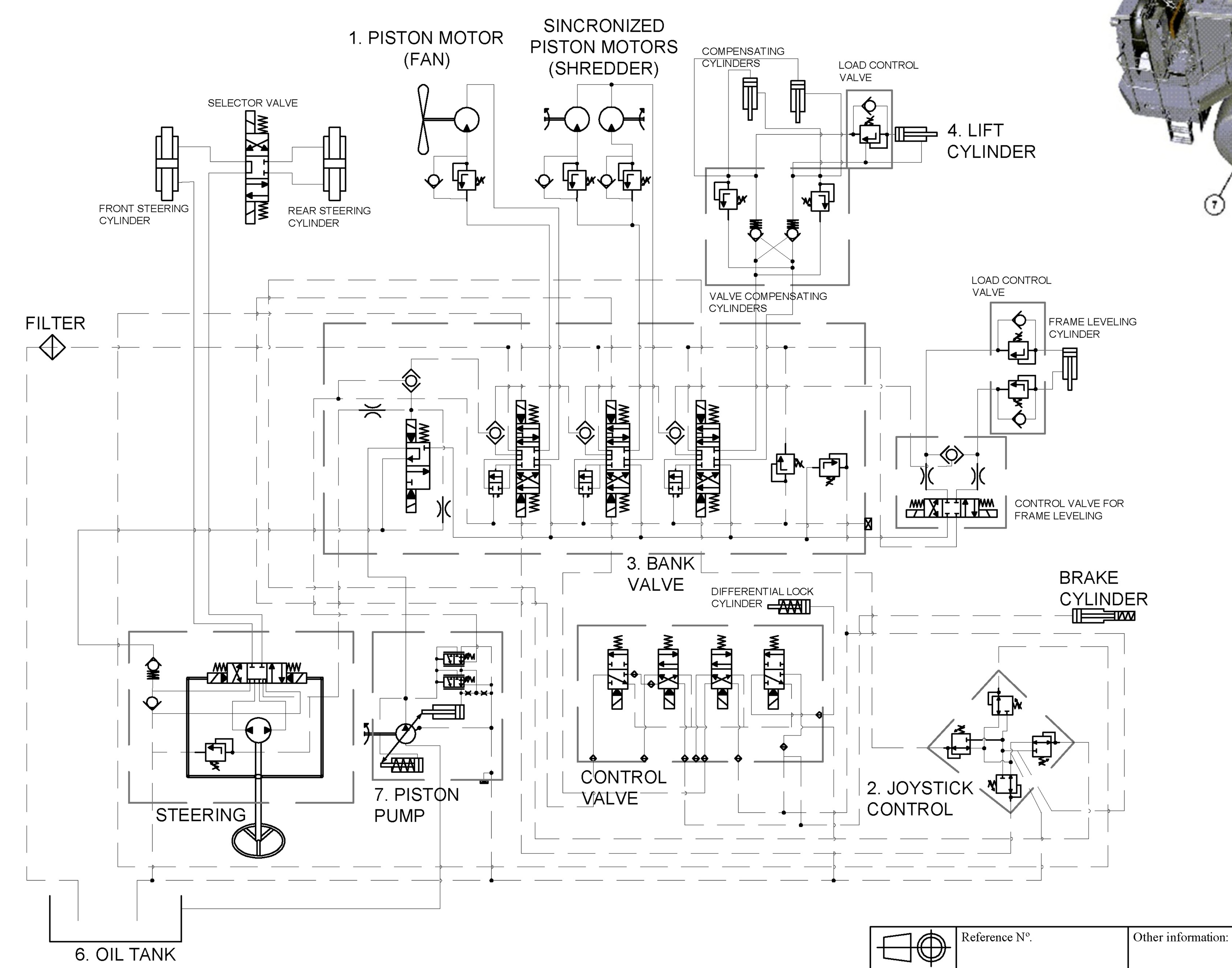
T7 **Insert:** QD-NK0600-0004-CR 1125 **Tool:** QD-NN2K60C25A

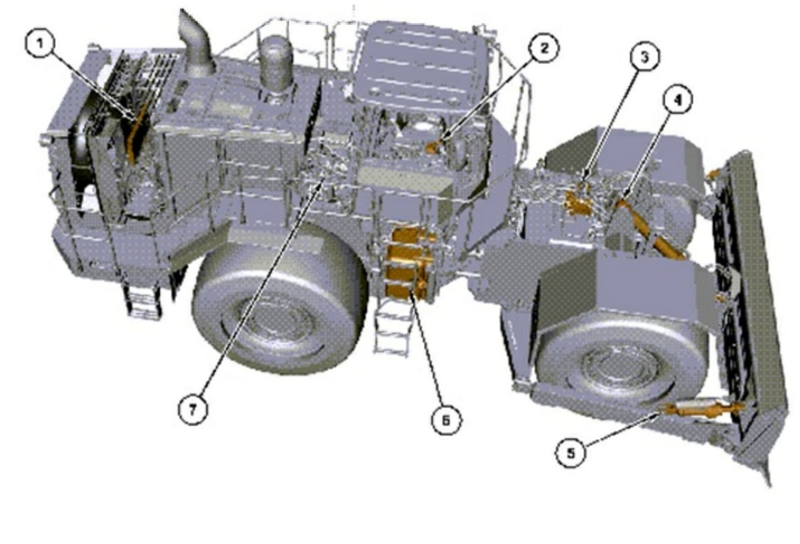
Step	Di (mm)	Df (mm)	ap (mm)	f (mm/rev)	n (rpm)	Vc (m/min)	Pc (kW)	t (s)	Ra (µm)
1	120	3	6	0.18	350	132	5.710	7.00	0.42
2	3	0	6	0.045	5000	132	2.160	0.00	0.03
							3.935	7.00 s	

Tools used

Nr.	Designation	Operation
T1	Insert: CNMG 19 06 24-PR GC4325 Tool: DCLNL 2525M 19	1, 3, 4
T2	Tool: 860.1-0510-019A1-PM 4234	2
T3	Insert: N123F2-0265-0002-GF 1125 Tool: LF123F10-2525B	5
T4	Tool: 2S342-1111-038-PA 1730	6
T5	Insert: CNMG 19 06 24-PR GC4325 Tool: DCLNR 2525M 19	7, 8
T6	Insert: 266RG-22MM01A600M 1125 Tool: 266RFG-2525-22	9
T7	Insert: QD-NK0600-0004-CR 1125 Tool: QD-NN2K60C25A	10

Reference N°.	Other information:	Material: AISI/SAE 1095 Normalized at 900ºC	Scale: 1:10
Responsibility: Dprt. of Mech.	Consulted by:	Document type: Technological process	Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero / Checked by: Andrius Gedvila	Title: Technological process of the motor shaft	Drawing N°: MPfu 11.04.001 / Rev.: A / Date: 2015-06-04 / Lang.: en / Sheet: 9/12

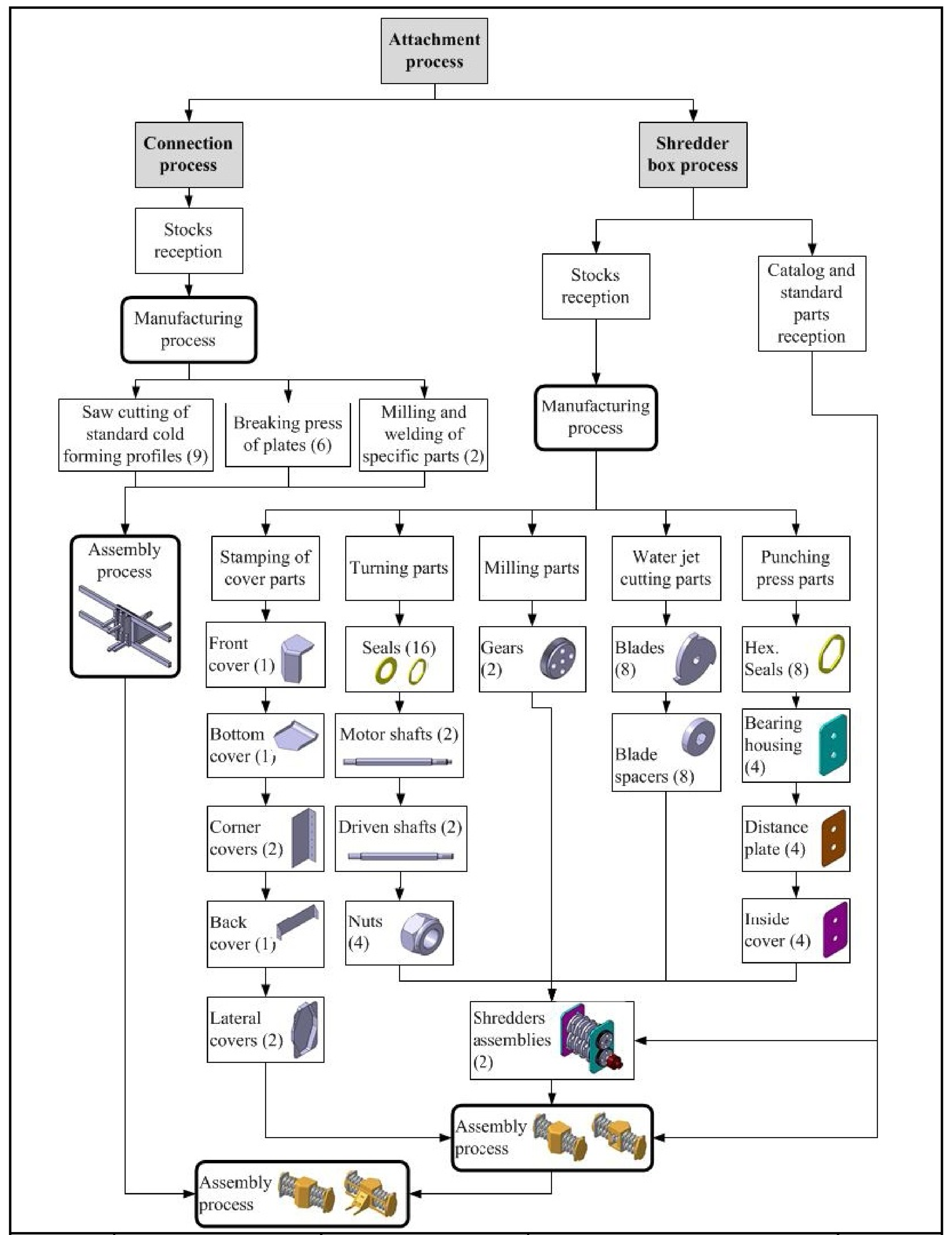




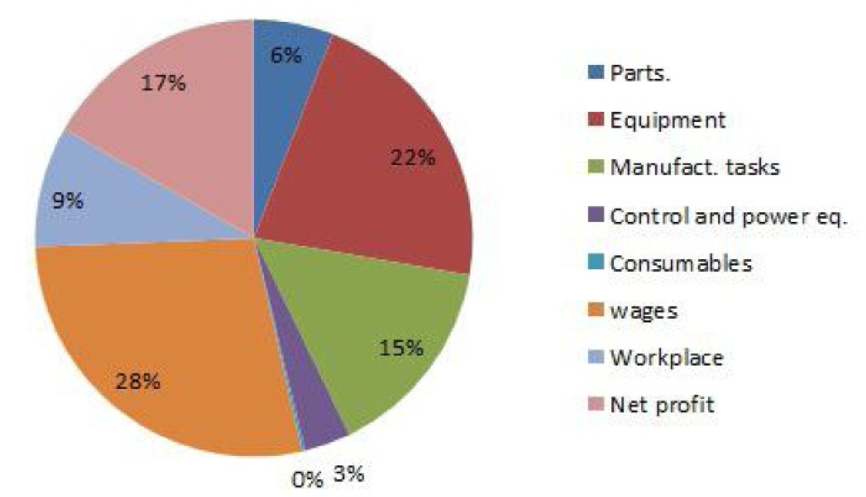
Location of the components of the hydraulic system:

1. Hydraulic oil cooler
2. Electrohydraulic control
3. Implement valve
4. Lift cylinder
5. Tilt cylinder (not exixst)
6. Hydraulic tank and filters
7. Imprement pumps

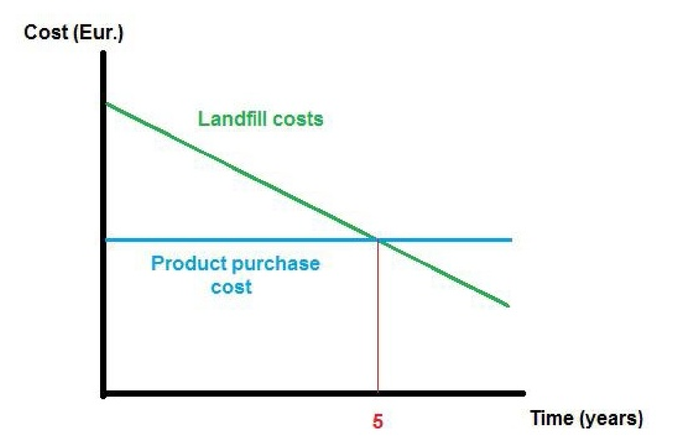
Reference N°.	Other information:	Material:	Scale:
Respponsibility: Dprt. of Mech.	Consulted by:	Document type: Hydraulic scheme	Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero / Checked by: Andrius Gedvila	Title: Compactor hydraulic scheme	Drawing N°: MPfu 11.04.002 / Rev.: A / Date: 2015-06-04 / Lang.: en / Sheet: 10/12



Reference N°.	Other information:	Material:	Scale:
Responsibility: Dprt. of Mech.	Consusted by:	Document type: Block diagram	Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero / Checked by: Andrius Gedvila	Title: Technological process stages	Drawing N°: MPfu 15.04.003 / Rev.: B / Date: 2015.06.04 / Lang.: en / Sheet: 11/12



Manufacturing and construction costs



Break-even point

Reference N°.	Other information:	Material:	Scale:
Responsibility: Dprt. of Mech.	Consusted by:	Document type: Economic characteristics	Document status: Educational
Owner: VGTU MPfu-11	Drawn by: Sara Escudero / Checked by: Andrius Gedvila	Title: Shredder attachment	Drawing N°: MPfu 11.04.004 / Rev.: B / Date: 2015.06.04 / Lang.: en / Sheet: 12/12