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DESIGN OF AN ORGANIC COMPOSTE TURNING MACHINE

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Abstract: This project aims to develop the concept of a mushroom compost turning machine for operation at the Piccoletta Funghi mushroom farm in Goiânia. The scope of the work ranges from the initial sketch, through the selection of mechanical components, estimation of the positioning of each part, and the materials and manufacturing processes for these parts. The project is divided into seven modules: conveyor belt; collection platform; wheel system; vehicle movement; hydraulic system; chassis; and operator cabin. Many of the mechanical components of the final design are inspired by and imported from grain harvesters, such as the collection platform drum and the compost conveyor belt. In addition, the chassis structure is designed to prevent the center of gravity from shifting outside the tractor's stability base, thereby reducing the chances of tipping over, according to a manual from Hazards of Occupations Safety Training in Agriculture. The operator's cab was developed based on anthropometric studies to ensure ergonomic comfort for the operator. This cabin houses the seat and hydraulic controls, which are used to activate the four hydraulic motors that provide the torque and power necessary to operate the conveyor belt, harvesting platform, and rear wheels, which are independent. This project achieves its initial goal, as the machine has the mobility and adaptability necessary for working on small mushroom farms. A step beyond this work would be to build a prototype to analyze the vehicle's strengths and weaknesses and refine the design.
Keywords: belt, hydraulic motor, torque, harvesting platform, hydraulic pump, maintenance.

1. INTRODUCTION

The mushroom industry is booming in Brazil. According to Santos (2018), between 1996 and 2018, there was a 400% increase in the consumption of this fruit, and according to Gomes (2018), a 10% growth is expected by 2021. The more competitive market and more demanding consumers are driving demand for technological innovations in this market.

Among these innovations, the compost turning machine is an example of an asset that has been used to increase production on mushroom farms. Despite this, the market offers few options for this machine, making it difficult to use in some situations.

In this context, it is necessary to design and manufacture a machine that meets the needs of each farm individually, as was the case with the Piccoletta Funghi company in the city of Goiânia.

1.1 MOTIVATION

Currently at Piccoletta Funghi, compost turning is done manually using agricultural forks. In the near future, the owner plans to expand production, making it necessary to mechanize the turning of approximately 11 tons of compost per day.

The machine's workspace is limited and its production needs are relatively small when compared to the capacity of existing machines on the market. This led the owner to seek a designer to develop a customized project.

Some of the requirements for this project are: simplicity of operation and maintenance; and the ability to operate in wet and dirty environments. Because the floor at the site is concrete and flat, and the machine does not require heavy loads, the vehicle does not need to be off-road.

1.2 OBJECTIVE

Create a preliminary design, or concept, for an organic compost turning machine. It must have specific characteristics that allow it to work in small areas, where the operation is not specialized and maintenance costs are low. This will allow for an increase in the turning rate and a consequent increase in production.

1.3 STRUCTURE WORK

The work is divided into five main chapters. The purpose of this chapter is to present the problem and provide an initial introduction to what is developed in the work.

Chapter two presents the concepts and foundations necessary for designing the machine, paving the way for the development of the project and presentation of results in chapter three.

Chapter four concludes and summarizes the results obtained, so that future work can be suggested in chapter five.

2. Y THEORETICAL FOUNDATIONS

2.1 Y MACHINE DESIGN

According to Norton (2011), a machine is a set of interrelated parts that aims to perform useful work by transforming one form of energy into another through the development of forces and movements. The design of an engineering machine is the conception or invention of a machine that is reliable and safe, in addition to performing its function in accordance with the proposed objectives.

Regardless of whether it involves a team or not, machine design has a protocol to be followed so that the consumer receives the product they want. It is necessary to identify the specific needs of the consumer, create an initial visual concept of the product, and identify the engineering characteristics that will ensure correct operation and fulfillment of functions (COLLINS, 2004).

2.2 ENGINEERING MATERIALS

1020 steel is almost always a good candidate for engineering projects due to its many positive characteristics. When this steel is not a good option, all materials that could be selected can be listed based on their strength, rigidity, and cost, weighing these characteristics against the project requirements (BUDYNAS AND NISBETT, 2011).

According to Ferrante (2013), the three main criteria for selecting materials are: mechanical strength, ease of manufacture, and dimensional considerations. Every project must take at least these three factors into account.

2.3 Y CONVEYOR BELT

Conveyor belts are essential in industrial environments where material transport is necessary. To enable transport, belts must have at least the following components: belt, drums (motor and driven), and rollers.

2.3.1 Belt

Belts are flexible machine elements used for power transmission, shock load absorption, and vibration damping and isolation

Vibration damping and isolation. Belts may be subject to some type of slippage or creep, making the angular velocity ratio between the drive and driven shafts neither constant nor proportional to the diameters (BUDYNAS AND NISBETT, 2011).

According to Budynas and Nisbett (2011), V-belts have an advantage over flat belts in that they allow for a small distance between centers. This fact is relevant to the design of the machine proposed for this work, since it is small in size.

In general, V-belts have the geometry shown in Figure 2.1, consisting of the envelope, the cords, and the rubber base.

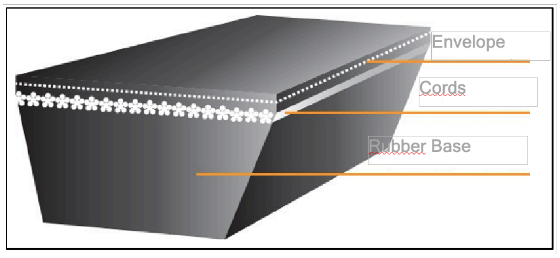

Figure 2.1 – Cross-section of a V-belt.

Source: Multibelt Belt Catalog.

2.3.2 Drums

In a conveyor belt, the drive drum receives power from the motor and transmits it to the belt through friction (FAÇO, 1991). In addition, drums are divided into: flat, for general applications; cambered, for belt alignment purposes; and ribbed.

Faço (1991) also highlights the importance of the drum's external coating in increasing friction between it and the belt, improving power transmission, and eliminating impurities on contact surfaces.

Among the main types of external coating, the following stand out: Smooth, to protect the drum and increase friction; grooved, same as above and also to allow water to flow; diamond-shaped, same as above, but used in reversible conveyors (FAÇO, 1991).

2.3.3 Roller

According to Faço (1991), the roller is a set of cylindrical rollers, set with free rotation, and used to support the weight of the conveyor belt.

2.4 CHAINS

Like belts, chains are also flexible machine elements responsible for transmitting power over relatively long distances. The difference between them is that chains do not slip or creep, enabling long life and the ability to drive multiple shafts from a single power source. Figure 2.2 shows a diagram of a double-row chain and its components.

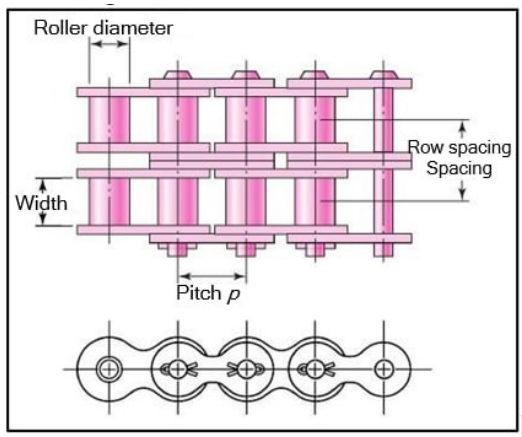

Figure 2.2 – Double-row chain.

Source: Budynas and Nisbett (2011).

According to Collins (2004), attention should be paid to the transmitted power and angular velocity of the pinion to avoid: fatigue of the connecting plate, fatigue of the roller and bushing, and wear due to excessive contact. This relationship is shown in Figure 2.3.

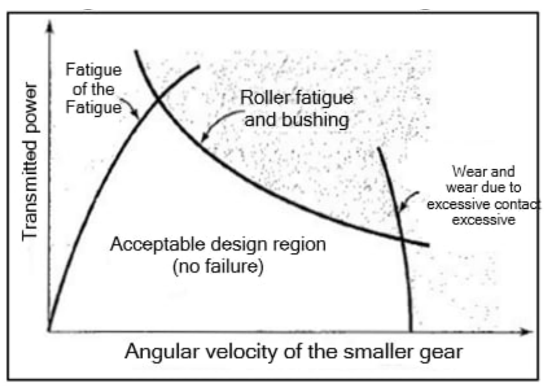

Figure 2.3 – Chain failure diagram.

Source: Collins (2004).

2.5 HYDRAULIC SYSTEMS

Fialho (2014) classifies hydraulic systems in several ways, as follows:

- According to the nominal pressure, according to Table 2.1;

Pressure (bar)	Classification
0 to 14	Low Pressure
14 to 35	Medium Pressure
35 to 84	Medium-High Pressure
84 to 210	High Pressure
Above 210	Extra High Pressure

Table 2.1 – Classification of a hydraulic system according to nominal pressure.

Source: Fialho (2014).

- Depending on the application, these can be continuous or intermittent pressure systems.
- As for the type of pump, it can be constant or variable flow;
- If the direction control is one-way or two-way (reversible pump).

According to Fialho (2014), a hydraulic system is divided into three main systems, as follows:

- Primary Conversion System – Composed of a reservoir, filters, pumps, motors, accumulators, and pressure intensifiers;
- Distribution and Control System – Consisting of flow control valves, pressure control valves, and directional valves;
- Energy Application System — Consisting of actuators, hydraulic motors, and oscillators.

Figure 2.4 refers to the diagram of a hydraulic system and its main components. Figure 2.5 shows the identification of the components presented in Figure 2.4.

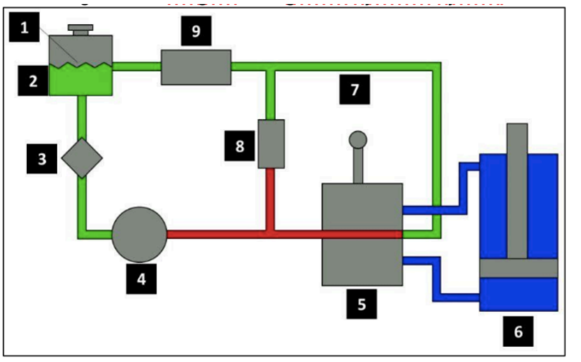

Figure 2.4 – Diagram of a generic hydraulic system.

Source: Lubequip Eximport (2019).

1 - Fluid
2 - Reservoir
3 - Filter
4 - Pump
5 - Control valve
6 - Actuator
7 - Piping
8 - Relief Valve
9 - Cooler

Figure 2.5 – Components of a generic hydraulic system.

Source: Lubequip Eximport (2019).

According to Fialho (2014), the advantages of a hydraulic system are: Easy installation; quick reversal; precise speed adjustments; and better weight x size x power consumption ratio. The disadvantages are: High initial cost; loss due to internal leakage in all components; fire hazard; and loss due to internal and external friction.

The flow can be in series or in parallel. In the case of series flow, the pressures must be added together, while in parallel flow, the pump discharge pressure is equal to the pressure at the valve outlet.

2.5.1 Pumps

According to White (1998), turbomachines are rotating devices that can add or extract energy from a fluid, and are called hydraulic pumps and turbines, respectively. According to Macintyre (1997), pumps are classified as generating machines because they receive mechanical work and transform it into hydraulic energy. Their function is to move liquids by flow, generating useful work, and they are classified as:

- Positive displacement pumps – They have one or more chambers where a propelling member presses the liquid;
- Turbopumps – Also known as rotodynamic pumps, they have a rotating member equipped with blades that exert pressure on the liquid;
- Centrifugal pumps — The centrifugal force derived from the rotation of a rotor presses the liquid.

2.5.2 Hydraulic motors

According to Fialho (2014), a hydraulic motor is a rotary actuator whose function is to convert hydraulic energy into rotary mechanical energy, working in the opposite way to a hydraulic pump.

This type of equipment has the advantage of high torque and power for low rotation. In addition, it has a good weight/power ratio and high efficiency (FIALHO, 2014).

2.6 AGRICULTURAL TRACTOR

An agricultural tractor is a self-propelled machine with stable support on a horizontal, impenetrable surface, used for traction, transport, and providing mechanical power (Massey Ferguson, 2016).

2.7 CHASSIS

According to Tractor Agriculture (2014), the chassis of a tractor is the structure responsible for connecting all the components of the vehicle, such as the engine, transmission, tires, axles, and steering system. The chassis must be strong enough to withstand all the stresses to which the tractor is subjected during operation.

2.8 AGRICULTURAL TRACTOR MAINTENANCE

According to FAESP (2010), maintenance is the set of measures taken to ensure the conservation of the tractor, to keep it in good working condition. In addition, maintenance performed correctly and at the right time increases the useful life of the equipment and reduces operating costs.

Also according to FAESP (2010), each piece of equipment must have an "operation manual," which will enable the monitoring of all necessary maintenance, in addition to the correct way to perform it.

In agricultural tractors, the most common types of maintenance are preventive and corrective maintenance. According to Kardec and Nascif (2009), preventive maintenance is performed with the objective of reducing or avoiding failures, following a previously prepared plan based on defined time intervals. Corrective maintenance, on the other hand, is performed to correct a failure.

3. DEVELOPMENT

3.1 SCOPE

As the objective of this work is to develop the design of a machine for use on small farms, it is necessary to simplify the design as much as possible so that it is reliable and robust. In addition, as this is a design, this work presents a result that is adaptable to different producers.

To this end, parts that can be purchased ready-made will be preferred to avoid manufacturing processes. This work develops the machine up to the sketch stage, which is part of the preliminary stage of a machine design.

3.2 DATA STATEMENT PROBLEM

According to Norton (2011), data must be presented in order for the preliminary design to begin. The concrete floor shed, measuring 13.0 by 7.0 meters, is shown in Figure 3.1.

Figure 3.1 – Photo of the shed.

Source: Author.

Another relevant piece of information is the positioning and size of the organic compost piles. Two piles measuring 8.0 by 1.8 meters, with a height of 1.8 meters, are used, totaling 25.92 m^3 per pile, or 51.84 m^3 in total. It is worth noting that the positions

The piles are interchangeable, allowing for greater flexibility in the design. Figure 3.2 shows one of the two piles located in the shed.

Figure 3.2 – Organic compost pile.

Source: Author.

The producer expects to turn about 11 tons of compost per day, which implies a machine that turns 1,375 kilograms per hour, considering an 8-hour workday. In addition, it is important to note the importance of constant operation, even if it is not so fast, as intermittent operation can reduce the quality of the compost.

3.3 INITIAL SKETCHES

Once the work to be done was defined, the first sketches were made on paper and also in CAD, using SolidWorks software, and are shown in Figs. (3.3) and (3.4). It should be noted that the cabin was not included in the sketches, as it would be adapted to the final structure.

The sketch on paper served as the basis for the SolidWorks design, while the SolidWorks design referenced most of the calculations and definitions that are presented in the following sections of the work.

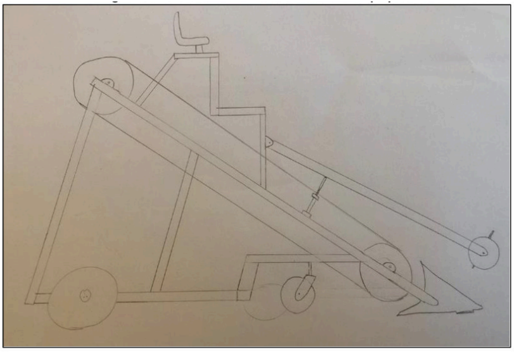

Figure 3.3 – Sketch of the machine on paper.

Source: Author.

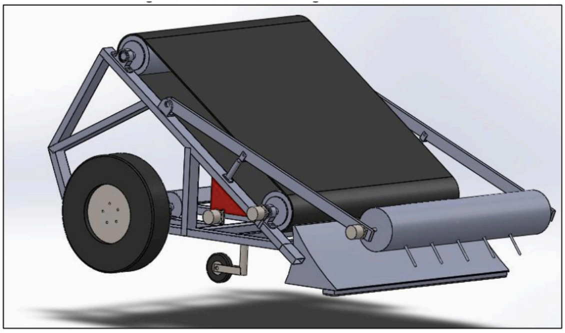

Figure 3.4 – First drawing in SolidWorks.

Source: Author.

3.4 MODULES

As this preliminary design involves several parts with different functions, it has been divided into modules. The modules will be presented and developed between items (3.4.1) and (3.4.7), and their designs detailed.

3.4.1 Conveyor Belt

3.4.1.1General Characteristics

Manufacturers of conveyor belts for grain harvesters unanimously use V-belts, called draper belts, according to a survey of various manufacturers' catalogs. For this project, a conveyor belt with a width as close as possible to 1.8 m should be chosen to meet the customer's demand for assembling compost piles 1.8 m wide.

The Hydraflex Draper 700FD platform from manufacturer John Deere has a central belt with a width of 1778.0 mm, which is an acceptable dimension when compared to the 1800.0 mm required. In addition, this belt's primary function is to transport material at high speed and in very high volumes, harvesting grain on large farms. It can therefore be said that it can safely transport organic compost, making it the belt of choice for this project.

Conveyor belts for central tracks on harvesters are usually installed at an angle of 40 to 45° to the ground. For this project, an angle of 35° will be used, which is less than the belt's design angle, to ensure that it does not encounter any problems during operation.

Another important feature of the selected belt is that it is specified by the manufacturer to operate at 287 m per minute, which is much higher than what will be used in this application, ensuring that it will perform well and have a longer service life.

The drive drum for moving the belt is a hollow cylinder with a diameter of 1778.00 mm and 273.05 mm.

From the sketch shown in Figure 3.4, the length of the belt is calculated to be 6461.42 mm. Comparing the maximum operating speed of the belt with its total length, approximately 44 complete rotations per minute were obtained.

To assist in transporting the compound, 80 mm square base blades with a length of 800 mm are installed. The material will remain in area 1, shown in Figure 3.5, which has a value of 0.005 m² (calculated by AutoCAD). The diagonal line represents the conveyor belt and the square represents the shovel. To calculate the volume transported per shovel, simply multiply the area by the length of the wood, which is equal to 0.004 m³. As one shovel will be placed on the right side and another on the left side, the total volume per line is 0.008 m³.

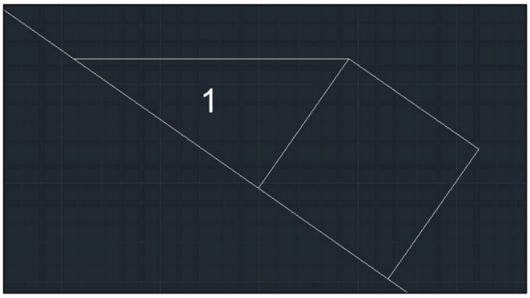

Figure 3.5 – Shovel transport diagram.

Source: Author.

From this point on, it is necessary to know the density of the material. To do this, five random mass measurements were taken using a container with a volumetric capacity of 1.0

liter, or 0.001 m^3. The density in this case is only to provide a basis for calculations and does not require high dimensional accuracy, so the average of the measured values serves as the measurement result. The measured values are shown in Table 3.1 below.

Measurement	Value (kg)
1	0.179
2	0.155
3	0.141
4	0.180
5	0.213
Average	0.174

Table 3.1 – Measured density values for 1.0-liter containers.

Source: Author

Dividing the average mass by the volume in cubic meters gives a density of 174 kg/m^3. Multiplying the density by the total volume per blade row, we conclude that each row transports approximately 1.392 kg.

Two other considerations to be made concern the belt rotation speed and the number of blades installed. It can be considered that each paddle performs 10 rotations per minute at a speed of 1.077 m/s, thus carrying 13.92 kg/min. If 20 rows of paddles are installed, the total transport capacity of this module will be 278.4 kg/min, or 16,704 kg/h.

Comparing the estimated hourly capacity with the capacity required by the producer, 1375 kg/h (presented in section 3.2), it can be seen that there is a large safety margin between the two values, since the estimated value is approximately 1000% greater than that required.

For this project, no load rollers or return rollers will be used, as it is a short conveyor belt. Instead, a wooden board will be used, positioned with the fibers parallel to the direction of the belt's roll, to serve as a support in case of overload.

3.4.1.2 *Definition of the required motor power*

According to Kulinowski (2009), the effective tension to move a belt is presented in Equation 3.1, which is a good way to calculate the power to move the belt, as it takes into account friction, temperature, belt tensions, among others. All of the following calculations are presented according to the English system of units.

$$F= L.Kt\ (Kx + Ky.\ Wb+ 0.015.Wb\ )+ Wm\ (L.Ky \pm H\ )+ Tp + Tam + Tac \quad (3.1)$$

- Belt length (L) — The value extracted from the sketch in SolidWorks is shown in Equation 3.2.

$$L= 6461.42\ mm= 21.19ft \qquad (3.2)$$

- Temperature correction factor (Kt) — The value of Kt is taken from the graph shown in Figure 3.7. For this work, the operating temperature is approximately 77°F (25°C), therefore $Kt = 1.0$.

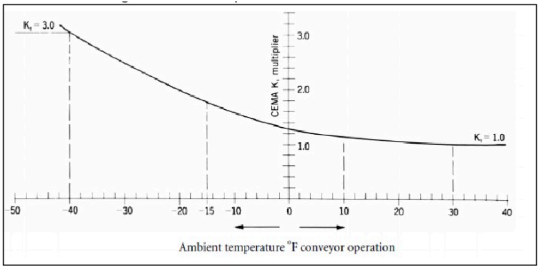

Figure 3.6 – Temperature correction factor.

Source: Kulinowski (2009).

- Idler belt correction factor (Kx) — As this project does not have an idler belt, $Kx = 0$.
- Factor for calculating belt force and bending load on idle belts (Ky) — As this design does not have an idle belt,$Ky = 0$.
- Material weight (Wm) — Equation 3.3

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is used for material weight, where the amount of material transported (Q) is given in tons per hour and the speed of the conveyor (V) in feet per minute.

$$Wm = \frac{2000*Q}{60*V} \tag{3.3}$$

It is worth noting that the mass of the installed blades must be considered. As there will be 20 rows of blades, each with 2 columns of wood, the total volume of wood is 0.16 m^3. Considering a wood density of 1000 kg/m^3, the mass of wood is 160 kg, or 0.16 tons per revolution. With 16.704 tons/h of compost and 96.000 tons/h of wood being transported, totaling 112.704 tons/h of material at 1.077 m/s, or 212 fpm, we obtainWm = 17.7 lbs/ft, after substitution in Equation 3.3.

- Vertical displacement of material (H) — As the material is transported uphill, this value is positive and equal to 1610.48 mm, or 5.28 ft.
- Total tension required to rotate each pulley of the belt conveyor (Tp) — Each pulley is responsible for an increase in tension. As the system in this work has two pulleys, one on the tight side and the other on the slack side, based on the data presented in the table in Figure 3.7, it can be concluded that the tension is Tp = 300 lb.

Location of Pulleys	Degrees Wrap of Belt	Pounds of Tension at Belt Line
Tight side	150° to 240°	200 lbs/pulley
Slack side	150° to 240°	150 lbs/pulley
All other pulleys	less than 150°	100 lbs/pulley
Note: Double the above values for pulley shafts that are not operating in antifriction bearings.		

Figure 3.7 – Tension for rotating pulleys.

Source: Kulinowski (2009).

- Force to continuously accelerate the material on the belt (Tam) — This force is calculated using Equation 3.4, obtaining $Tam = 6.87 lbs$, with the variables already defined above.

$$Tam = \frac{2000*Q}{3600*32.2} * \frac{V}{60} \tag{3.4}$$

- Resistance generated by accessories (Tac) — As a wooden board will be used to help support the belt, preventing it from bulging, an overestimated value of 100 lbs will be considered here.

Substituting all values presented in Equation 3.1, we obtain the effective voltage$F = 896.6lb$. From Equation 3.5, we find that the power is equal to 4.7 kW. Considering a reduced motor efficiency equal to 0.9, the power required by the hydraulic motor is 5.2 kW.

$$P = \frac{212*F}{30000} * \frac{745.7}{1000}$$

3.4.1.3 Definition of the required motor power

A hydraulic motor is used to drive the conveyor belt, due to the ease of varying the angular speed and good torque-to-size ratio. In other words, a small motor provides high torque.

Since the power required to drive the belt has already been defined, a hydraulic motor that provides this power can be found in catalogs. From the Parker manufacturer's catalog, the TC0100 model was chosen, whose technical specifications are presented in Table 3.2, and photo and approximate dimensions in Figures 3.8 and 3.9, respectively.

Torque	**140 Nm**
Power	**6.1 kW**
Pressure Maximum	**134 bar**
Speed Angular	**460 rpm**
Oil Oil	**45 l/min**
Weight	**5.88 kg**

Table 3.2 – Parker TC0100 technical specifications.

Source: https://www.parker.com

Figure 3.8 – Parker TC series.

Source: https://www.parker.com

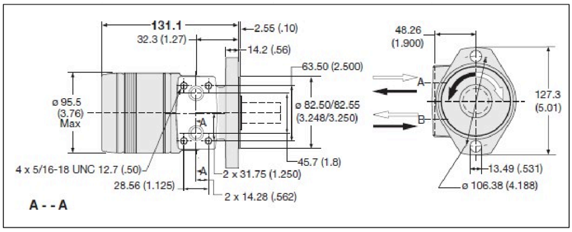

Figure 3.9 – Dimensions of hydraulic motor for conveyor belt.

Source: https://www.parker.com

3.4.1.4 Materials and Manufacturing

The drive drum and driven drum must have their material selected, and in the case of the latter, their manufacturing process defined.

The drum will not be subject to high loads or impacts, so it is important to ensure that they do not suffer from oxidation, as the environment may be humid.

For this purpose, SAE 1020 steel drums should be used. In addition, it may be useful to machine grooves in the area of contact with the belt to allow water to be expelled.

3.4.2 HARVESTING PLATFORM

3.4.2.1 General Characteristics

The harvesting platform is responsible for collecting the compost from the pile and throwing it onto the conveyor belt. In this work, a platform similar to that used by harvesters is used. These harvesters use a hollow, freely rotating drum with a fixed shaft inside that is not concentric with the drum. Claws are attached to the fixed shaft and positioned outside the drum through holes, but with enough clearance to rotate freely around the shaft, allowing the claws to be retractable.

This system is best visualized in Figs. (3.10) and (3.11).

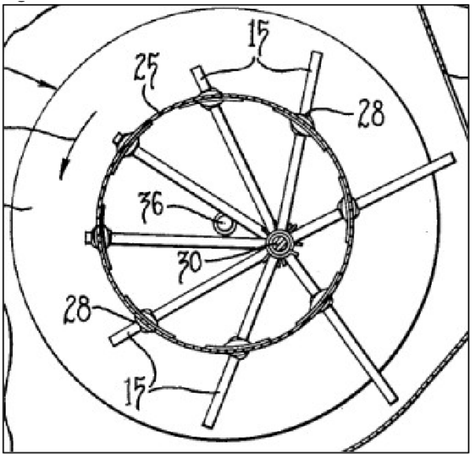

Figure 3.10 – Side view of the drum with claws.

Source: Massey Ferguson Limited (1964).

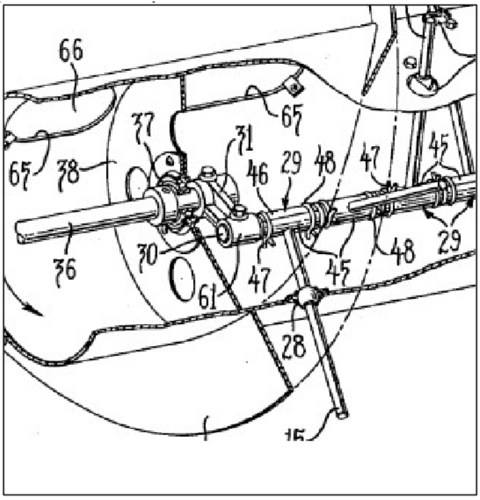

Figure 3.11 – Perspective view of the drum with claws.

Source: Massey Ferguson Limited (1964).

The advantage of retractable claws stems from their ability to be visible while the hole is in position to reach the pile, and then to be retracted as the hole approaches the belt.

For this project, the parts that must be purchased are the supports and guides for the claws, shown in Figs. (3.12) and (3.13) respectively, which are used to support the claws.

Because these are dynamic parts that are constantly subject to friction and impact, manufacturers use nylon and Teflon in the manufacture of the bracket and guide, respectively.

Figure 3.12 – Claw support.

Source: https://www.alibaba.com

Figure 3.13 – Claw guide.

Source: https://agrodoctor.eu/header/74620-auger-finger-guide-777199-claas-603754-claas.html

To select the drum size, the belt width is used as a reference, which, according to section 3.4.1, is 1778.00 mm. The drum length is then selected as 1700.00 mm, while the diameter is selected as 300.00 mm.

To make good use of the drum length, 10 equidistant and alternately positioned claws are used, i.e., 180° apart from each other. The alternating positioning reduces the load on the motor that rotates the drum.

The length of the claw should be approximately 210.00 mm, allowing it to be extended when grabbing the compost, but also retracted sufficiently when releasing it. In addition, its diameter should be approximately 14.00 mm to be compatible with commercially available guides and supports.

The drum section is shown in Figure 3.14.

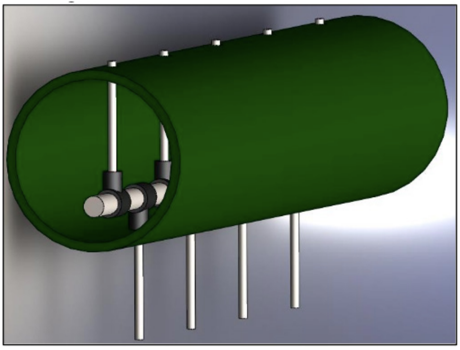

Figure 3.14 – View of the drum section.

Source: Author.

In addition to the drum, a system with two arms is required to support the collection platform. To increase the contact area of the claws with the compost pile, the arms are movable, performing an up and down movement. Two hydraulic actuators are used to enable their movement.

When positioning the actuators and choosing the length of the arm, it must be taken into account that the greater the working radius of the platform, the better the collection will be. However, if the value is too high, the lever arm can make the pressure to lift the arm too high.

It will be considered that the arm is 2.0 m long and that it is subjected to a force of 0.5 kN at its end, referring to half the weight of the drum and components. The drum will be positioned 1.5 m from the point of application of the force. This diagram is shown in Figure 3.15.

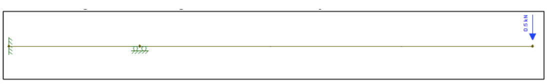

Figure 3.15 – Diagram of the beam subjected to a force of 0.5 kN.

Source: Author.

Using Ftool software, it is calculated that the force required by the actuator to lift the arm, as shown in Figure 3.16, is 0.7 kN.

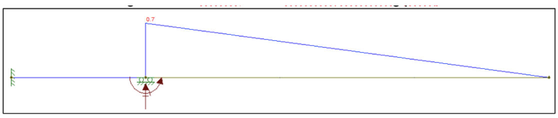

Figure 3.16 – Stress at the actuator mounting point.

Source: Author's own.

To apply the lifting force to the arms, the actuator sizing method proposed by Fialho (2014) is used. A pressure of 2.5 bar from the pump is considered to lift each actuator, totaling 5.0 bar extracted from the pump. It should be noted that the downward movement of the pistons is performed due to the pressure exerted by the weight of the assembly itself.

The first step in selecting the actuators is to calculate the piston diameter according to Equation 3.6. Knowing that the pressure is $Ptb = 2.5 bar = 250000 Pa$, the forward force is $Fa = 700 N$, and that the actuator efficiency is considered $n = 0.9$.

$$Dp = 2\sqrt{\frac{n*Fa}{\pi*Ptb}} \quad (3.6)$$

Substituting the values, we obtain that the piston diameter must be at least 56.6 mm. Searching the Euro Hidraulics manufacturer's catalog, we find that there is a compatible actuator with a sleeve diameter of 63.0 mm, whose other dimensions are shown in Figure 3.17. In addition, Figure 3.18 shows the dimensions of the base of the selected actuator.

SHIRT	HASTE	KK	A	H	E	Y	PJ	EE	COURS
25	12	M10 x 1,25	14	5	40	50	53	G 1/4	600
	18	M10 x 1,25	14						
		M14 x 1,5	18						
32	14	M12 x 1,25	16	5	45	60	56	G ¼	800
	22	M12 x 1,25	16						
		M16 x 1,5	22						
40	18	M14 x 1,5	18	-	63	62	73	G 3/8	1000
	28	M14 x 1,5	18						
		M20 x 1,5	28						
50	22	M16 x 1,5	22	-	75	67	74	G ½	1200
	36	M16 x 1,5	22						
		M27 x 2	36						
63	28	M20 x 1,5	28	-	90	71	80	G ½	1400
	45	M20 x 1,5	28						
		M33 x 2	45						

Figure 3.17 – Dimensions of the selected actuator.

Source: http://www.eurohidraulics.com.br/

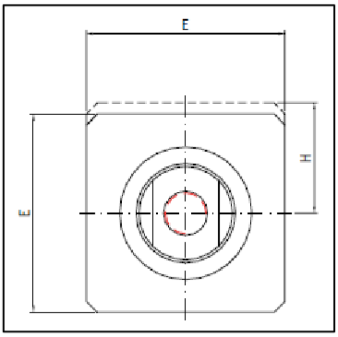

Figure 3.18 – Actuator base dimensions.

Source: http://www.eurohidraulics.com.br/

As shown in Figure 3.17, the maximum stroke is 1400.0 mm, which is much greater than expected for the project. Therefore, the exact stroke can be defined after the chassis specifications have been defined.

3.4.2.2 Materials and Manufacturing

The drum, claws, and inner shaft still need to be selected, since the claw support and guide are purchased from manufacturers.

The drum can be made of SAE 1020 steel using a calendering process. The holes in its outer radius can be made using a steel drill bit.

The claws will be subject to bending moments when they come into contact with the compost pile. This requires that the material be flexible, yet resistant to breakage when subjected to this type of stress. Therefore, SAE 4130 steel tubes can be purchased SAE 4130 steel tubes, which, despite their higher price, have high mechanical strength.

Finally, the material for the inner shaft must be selected. The shaft will not be subjected to significant bending stresses, since the stresses exerted by the claws cancel each other out, as there are 5 forces in one direction and 5 forces in the opposite direction, as shown in Figure 3.19 and explained in section 3.4.2.1. Since the surface of the shaft will be under constant dynamic friction with the internal nylon surfaces of the claw support, it is advisable to apply an anti-abrasion coating to the shaft.

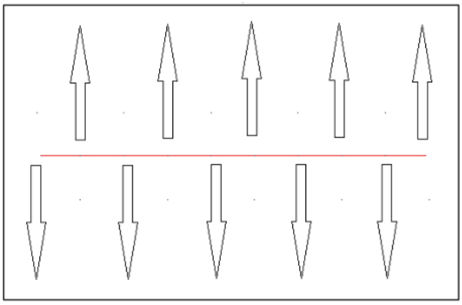

Figure 3.19 – Direction of requests on the internal axis.

Source: Author's own.

3.4.2.3 Motorization

A hydraulic motor can be used to drive the worm gear. To select the motor, the torque must be calculated using Equation 3.7, which uses angular acceleration (α) and mass moment of inertia (I).

$$T = \alpha I \qquad (3.7)$$

Before applying Equation 3.7, it is necessary to define some parameters. One of them is the approximate mass of the drum. Table 3.3 shows the volume of the drum, which multiplied by the density of the material results in the mass.

	External	Internal
Height	1,778 mm	1.758 m
Radius	0.300 m	0.290 m
Volume	0.1257 m^3	0.1161 m
Volume of Steel	0.009 m^3	

Table 3.3 – Drum information.

Source: Author

Since the density of SAE 1020 steel is approximately 7800 kg/m^3, the mass of steel is 75 kg.

For this project, an acceleration of 2π radians per second squared will be considered. The moment of inertia of mass is calculated using Equation 3.8, knowing that the mass is 75 kg and the radius is 0.150 m.

$$I = mr^2 \qquad (3.8)$$

Substituting the values in Equation 3.8, we obtain that the moment of inertia of mass is equal to 1.688 kg.m^2. Therefore, using Equation 3.7, we arrive at a value of 10.6 Nm.

Therefore, the Parker TC0036 hydraulic motor from the Parker manufacturer's catalog is chosen. Its technical specifications are presented in Table 3.4, its housing is the same as the Parker TC0100 in Figure 3.8, and its dimensions are shown in Figure 3.20.

Torque	44 Nm
Power	4.2 kW
Maximum pressure	134 bar
Angular speed	902 rpm
Oil consumption	34 l/min
Weight	5.38 kg

Table 3.4 – Parker TC0036 technical specifications.

Source: https://www.parker.com

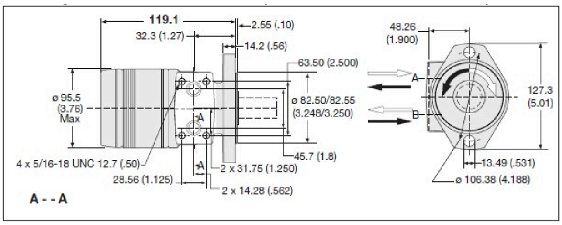

Figure 3.20 – Dimensions of the hydraulic motor for the screw conveyor.

Source: https://www.parker.com

3.4.3 Wheeled system

As the vehicle will operate in a location with reduced mobility, it is important that it has the ability to make tighter turns. One way to achieve this is through the use of rear-wheel drive with free-turning front wheels, commonly known as crazy wheels.

This steering system is characterized by operating with independent rear wheels, meaning that if you want to turn left, only the right wheel will turn, while the left wheel remains stationary, and vice versa. The front wheels only serve to keep the vehicle on four wheels, as they rotate freely around their axis.

Since the ground in the tractor's operating area is concrete and smooth, smooth tires for asphalt could be used. However, it is better to opt for agricultural tires, due to the small price difference and greater robustness. In addition, according to Macmillan (2002), agricultural tires, as shown in Figure 3.21, are recommended for common farm operations. Through internet searches, it was observed that the 7.50-16 off-road tractor tire, with a diameter of 800 mm, from Chevrolet C-10 pickup trucks, has the desired characteristics, as well as a size consistent with that used in this work.

Figure 3.21 – Tire for common agricultural work.

Source: Macmillan (2002).

The other parts needed for installation, such as the axle end, wheel, and hub from the C-10 pickup truck, are easily found on the domestic market, reducing the cost of the steering system. Figs. (3.22), (3.23), and (3.24) show the tire to be used, the axle end, and the wheel hub, respectively.

Figure 3.22 – Selected agricultural tire (Chevrolet C-10).

Source: https://produto.mercadolivre.com.br/MLB-1111346726-pneu-750-16-off-road-f75pick- upruraltratorgaiola-recap-_JM?quantity=1

Figure 3.23 – Axle tip of the C-10.

Source: https://produto.mercadolivre.com.br

Figure 3.24 – Wheel hub of the C-10.

Source: https://produto.mercadolivre.com.br

3.4.4 Vehicle Movement

To drive the rear wheels, the initial idea was to use a combustion engine combined with a set of gears, or to use two hydraulic motors, one on each wheel. Since a combustion engine and a hydraulic pump would already have to be installed to drive the hydraulic motors on both the conveyor belt and the collection platform, it was decided to also use hydraulic motors on the wheels.

In addition to the advantages of hydraulic motors mentioned in section 3.4.1.3, the lack of need for complex gear sets reduces the total weight of the machine, as well as the complexity of maintenance. It should be noted that regardless of the rotation speed of the chosen motor, the hydraulic motor offers it also has the advantage of adjustable angular speed and the possibility of rotating in the opposite direction (reverse).

The system is simple to operate, requiring basically a hydraulic control with two levers, one for each hydraulic motor. Each lever has three positions: +1, to move forward; 0, motor with zero angular velocity; and -1, where the motor rotates in the opposite direction, enabling reverse movement.

Therefore, the instructions presented in Table 3.5 should be used to guide the vehicle. It is worth noting that, although three positions are used in the control, between 0 and +1 and

0 and -1, there are infinite positions for the lever between -1 and +1. Each one allows a different oil flow to the motor, and the higher the flow, the higher the rotation.

	Lever	
	Left Right	Right
Forward	1	1
Reverse	-1	-1
Left turn	0	1
Right turn right	1	0

Table 3.5 – Instructions for moving the vehicle.

Source: Author.

The connection between the motor shaft and the wheel is made using a crown-pinion assembly with chain transmission, with the crown positioned at the end of the shaft and the pinion on the motor shaft, as shown in Figure 3.25. The coupling between the driven wheels and the structure is achieved through bearings, positioned upside down, for better weight distribution over its housing.

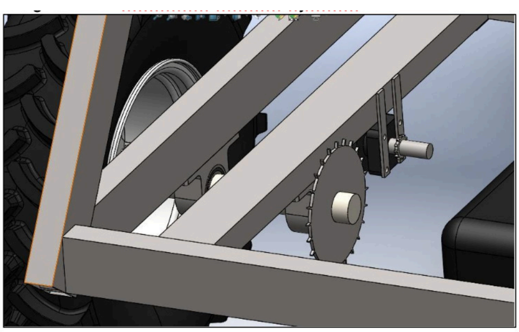

Figure 3.25 – Connection between hydraulic motor and wheel.

Source: Author.

3.4.4.1 Definition of torque to rotate the wheel

To select the hydraulic motor, it is necessary to know the torque required to rotate the wheel and, consequently, the weight of the vehicle. For calculation purposes, it is assumed that the weight of the vehicle is equally distributed among the four wheels and that its total mass is 1,500 kg, equivalent to 14,715 N (3,678.75 N/wheel).

The torque is given by Equation 3.9, where FT is the total force, Rr is the wheel radius, and f a safety factor.

$$\tau = FT * Rr * f \qquad (3.9)$$

For the torque equation, it is necessary to calculate the total force (FT), according to Equation 3.8. The total force is the sum of: Rolling force (FR), climbing force (FS), and acceleration force (FA).

$$FT = FR + FS + FA \qquad (3.10)$$

- Rolling force — This is the force required to overcome rolling resistance. According to Canale (1989), the rolling force is given by Equation 3.11, where fr is the rolling resistance coefficient and W is the weight supported by the wheel. The weight supported by the wheel is 3678.75 N. The coefficient fr is obtained from Figure 3.26, and for this case, it will be considered 0.02 (tractor traveling on concrete).

$$FR = fr * W \qquad (3.11)$$

VEHICLE	CONCRETE	AVERAGE HARDNESS	SAND
PASSENGER CARS	0,015	0,10	0,30
HEAVY TRUCKS	0,012	0,08	0,25
TRACTORS	0,02	0,04	0,20

Figure 3.26 – Rolling resistance coefficients.

Source: Canale (1989).

Substituting the values in Equation 3.11, we obtain $FR = 73.58N$.

- Climbing Force — This is the force that the wheels must overcome to travel uphill. The maximum angle (α) of 10 degrees will be considered. From Equation 3.12, provided by Canale (1989), we arrive at the value of FS.

$$FS = W * sen\alpha \qquad (3.12)$$

Substituting the values in Equation 3.10, we obtain $FS = 638.81N$.

- Acceleration Force — This is the force required to bring the vehicle to its maximum speed in the shortest time possible. Equation 3.13 is used to calculate this, with the maximum speed ($vmáx$) equal to 2.2 m/s and the acceleration time (ta) equal to 4.0 s.

$$FA = \frac{W * vmáx}{9.81 * ta} \qquad (3.13)$$

Substituting the values in Equation 3.13, we obtain $FA = 206.25N$.

There is sufficient data to obtain the total torque value, according to Equation 3.9. Knowing that the wheel radius is 0.4 m and the safety coefficient is 1.1, we have a total torque equal to $TT = 404.20\ Nm$.

If the motor were connected directly to the wheel, a motor with a torque equal to 404.20 Nm would be required. However, as the transmission will be by chain and a crown-pinion pair, the torque required by the motor will be lower.

3.4.4.2 Definition of the crown-pinion pair

Initially, a transmission ratio (i) must be adopted to define the crown-pinion pair, whose chosen value is 6.0. As it is ideal to purchase commercial gears, all definitions were made according to the Kamart manufacturer's catalog.

The chain to be used is single, not double. Thus, to calculate the pitch diameter of the pinion, two choices must be made based on the catalog: the number of teeth on the pinion and the gear pair pitch (p).

The pitch adopted is $p = 9.525mm$ and the number of teeth on the pinion is 12. From Equation 3.14, the pitch diameter of the pinion can be calculated, knowing that the factor X is given by the manufacturer, according to Figure 3.27.

Number of Teeth	X Factor
09	2,924
10	3,236
11	3,549
12	3,864
13	4,179
14	4,494
15	4,810

Figure 3.27 – X Factor.

Source: http://kamart.com.br/catalogo.html

$$DPp = p * fx \qquad (3.14)$$

Therefore, the pitch diameter of the pinion is equal to 36.805 mm. This gear is the one with KMT reference 1.35.12, as shown in Figure 3.28. The pitch diameter of the crown wheel must be selected based on the chosen transmission ratio. The transmission ratio is given by Equation 3.15.

$$i = \frac{DPc}{DPp} \qquad (3.15)$$

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Isolating the transmission ratio, we conclude that the pitch diameter of the ring gear is equal to the product of the pitch diameter of the pinion and the transmission ratio, that is, $DPp = 220.83\ mm$.

Looking at Figure 3.28 for a gear whose pitch diameter is equal to 220.83 mm, we see that it does not exist. Therefore, we adopt a gear whose pitch diameter is as close as possible. Therefore, the gear whose KMT reference in Figure 3.28 is equal to 1.35.76, whose diameter is 230.49 mm, will be chosen for this work.

Substituting the tabulated pitch diameter values in Equation 3.15, we arrive at a transmission ratio of $i = 6.26$.

Reference	Diameter Positive	Number of of Teeth	Bore Guide	Hub Diameter	Total Width
KMT	DP	Z	F	D	L
1.35.09	27,85	09	-	17	16
1.35.10	30,82	10	-	20	16
1.35.11	33,81	11	-	23	16
1.35.12	36,80	12	12	26	16
1.35.13	39,80	13	12	29	16
1.35.14	42,81	14	12	32	16
1.35.15	45,81	15	12	35	20
1.35.16	48,82	16	12	38	20
1.35.17	51,84	17	12	41	20
1.35.18	54,85	18	12	44	20
1.35.19	57,87	19	12	47	22
1.35.20	60,89	20	12	50	22
1.35.21	63,91	21	12	53	22
1.35.22	66,93	22	12	56	22
1.35.23	69,95	23	12	59	22
1.35.24	72,97	24	15	62	22
1.35.25	76,00	25	15	65	22
1.35.26	79,02	26	15	68	22
1.35.27	82,05	27	15	70	22
1.35.28	85,07	28	15	72	22
1.35.30	91,12	30	15	78	25
1.35.32	97,18	32	15	84	25
1.35.35	106,26	35	15	93	25
1.35.38	115,34	38	15	103	25
1.35.40	121,40	40	15	110	25
1.35.45	136,55	45	20	126	25
1.35.48	145,64	48	20	130	25
1.35.54	163,82	54	20	72	30
1.35.57	172,91	57	20	72	30
1.35.60	182,00	60	20	80	30
1.35.76	230,49	76	20	80	30
1.35.95	288,08	95	20	80	30
1.35.114	345,68	114	20	98	30

Figure 3.28 – Kamart Gears

Source: http://kamart.com.br/catalogo.html

From Equation 3.16, it is possible to calculate the torque required from the motor, given the transmission ratio obtained previously.

$$TM = \frac{TT}{i} \tag{3.16}$$

Substituting the values, we obtain a total motor torque of $TM = 64.57 Nm$.

3.4.4.3 Hydraulic motor selection

With the torque required to turn each wheel already calculated in section 3.4.4.2, the Parker catalog can be consulted to find the motor required for this application. The Parker TC0050 motor was chosen, whose technical specifications are presented in Table 3.6, and its dimensions in Figure 3.29.

Torque	67 Nm
Power	4.7 kW
Maximum pressure	134 bar
Angular Speed	688 rpm
Oil consumption	34 l/min
Weight	5.67 kg

Table 3.6 – Parker TC0050 technical specifications

Source: https://www.parker.com

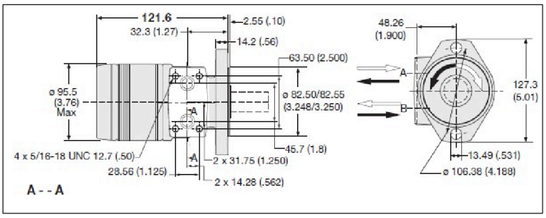

Figure 3.29 – Dimensions of hydraulic motors for movement.

Source: https://www.parker.com

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3.4.4.4 Materials and Manufacturing

The motors must be positioned on the chassis structure using two SAE 1020 steel plates welded to the chassis, which function as flanges.

3.4.5 Hydraulic System

In order for the previously selected hydraulic motors to function, some components are still required, such as: Hydraulic Pump; hydraulic pump drive motor; oil reservoir; fuel reservoir; hydraulic control; and connections.

3.4.5.1 Hydraulic Pump

When selecting the pump, look for equipment that meets the oil flow required by the hydraulic motors. As these are variable speed and flow, look for a pump that meets the maximum flow condition. Table 3.7 shows the oil flow of the motors selected so far, as well as the total flow required.

	Flow rate (l/min)
01 x Parker TC0100	45
01 x Parker TC0036	34
02 x Parker TC0050	68
Total	147

Table 3.7 – Oil flow rate of hydraulic motors.

Source: Parker (2009)

A pump with a flow rate of 147 l/min should be sought in catalogs. According to Macintyre (1997), it is interesting to choose a gear rotary pump due to its high pressure capacity and its ability to provide medium pressures. In the Danfoss manufacturer's catalog, it can be seen that the Danfoss JS75C equipment meets the flow and pressure requirements of hydraulic motors.

From Figure 3.30, it can be seen that for a flow rate of 150 l/min, the pump must operate at approximately 2000 rpm. Seeking 2000 rpm of rotation and the maximum working pressure of the pump equal to 140 bar, in Figure 3.31, it can be seen that the power required by the motor is approximately 40 kW, or 54 hp. Finally, from Figure 3.32, it can be seen that the overall efficiency of the pump is approximately 87%.

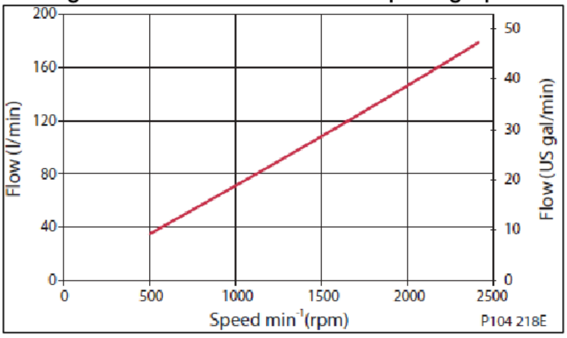

Figure 3.30 – Flow rate versus speed graph.

Source: https://www.danfoss.com/pt-br/

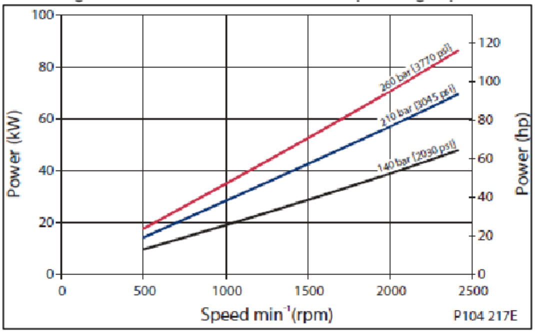

Figure 3.31 – Power versus speed graph.

Source: https://www.danfoss.com/pt-br/

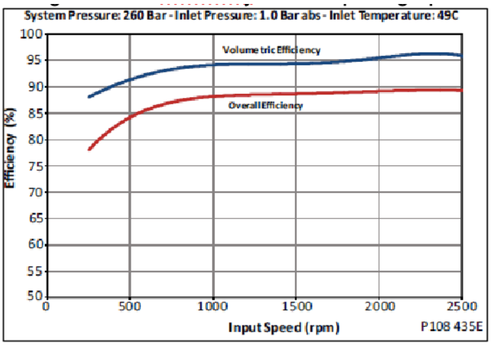

Figure 3.32 – Efficiency versus speed graph.

Source: https://www.danfoss.com/pt-br/

3.4.5.2 Hydraulic pump drive motor

As presented in section 3.4.5.1, a motor that provides at least 54 hp of power is required to drive the pump. There is a choice between electric or combustion engines. Electric motors have the disadvantage of requiring batteries to operate, so combustion engines were chosen.

To meet the power requirements, a partial engine from a passenger car can be purchased. The advantages of this type of engine are:

- The less powerful ones easily reach 50 hp;
- Excellent cost-benefit ratio, since partial engines are new and the price is about three times higher than that of a stationary engine, which provides six times less power;
- Easy to adapt, as they come equipped with a block, crankshaft, rings, pistons, connecting rods, bushings, and oil pump.

The main disadvantage is that partial engines are sold without the cylinder head, so it is necessary to purchase and install one.

3.4.5.3 Oil reservoir

According to Fialho (2014), oil storage requires a reservoir with at least three times the oil flow rate. Therefore, the reservoir for this application must have a capacity of at least 441 liters, or 0.441 m^3. This tank must be made of metal to withstand the high temperatures of the oil, so it can be manufactured according to the desired measurements.

The position of the reservoir can be defined according to the characteristics of the chassis.

3.4.5.4 Fuel tank

For fuel storage, you can choose a fuel tank from any passenger car. The Willys Jeep tank has a capacity of 30 liters and is a good candidate. Its dimensions are shown in Figure 3.33.

Figure 3.33 – Willys Jeep fuel tank.

Source: https://produto.mercadolivre.com.br

3.4.5.5 Hydraulic control

The hydraulic control is the component responsible for allowing the operator to control the oil flow to the hydraulic motors. As four hydraulic motors and two actuators (acting together) are used, five hydraulic control modules are required, connected in series, with double-acting valves.

Figure 3.34 shows the selected hydraulic control, whose technical specifications are compared with those required for the job in Table 3.8.

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Figure 3.34 – Hydraulic control.

Source: https://produto.mercadolivre.com.br

	Technical Specifications	Maximum Project Requirement
Pressure	250 bar	134 bar
Flow	50 l/min	45 l/min

Table 3.8 – Comparison between control specifications and project requirements.

Source: Author

In addition, this control has the advantage of allowing the selector lever to be locked, enabling a constant oil flow. The manufacturer recommends that equipment connected in series, such as in this project, use pressure passage nipples.

It is worth noting that as the controls will be connected in series, in a situation of maximum project demand, the pump would have to supply a pressure of 541 bar, which would require an unfeasibly powerful pump. Therefore, the operating manual must specify how the operator should monitor the pump pressure. Physical restrictions can also be used on the hydraulic control levers to prevent the flow from exceeding a certain value.

Once assembled, the complete hydraulic control system will look similar to Figure 3.35.

Figure 3.35 – 4-way hydraulic control.

Source: https://produto.mercadolivre.com.br

3.4.5.6 Connections

The connections will not be detailed in this work, but there must be hoses for suction, pressure, return, and the appropriate adapters.

3.4.5.7 Hydraulic system diagram

Using FluidSIM Hydraulics software, the circuit required to activate the two pistons that lift the collection platform, the hydraulic pump, and the hydraulic motors for the wheels, drum, and conveyor belt was diagrammed. The diagram is shown in Figure 3.36.

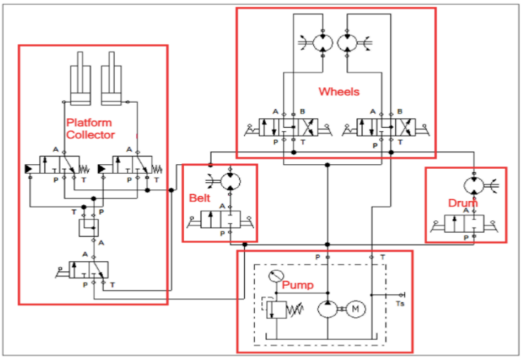

Figure 3.36 – Components identified in the hydraulic system diagram.

Source: Author's own.

3.4.6 Chassis

For this project, it was determined that the chassis would be a lattice structure composed of rectangular metalon structural tubular profiles measuring 100 x 50 x 3 mm, as shown in Figure 3.37. Metalon was chosen because it is cost-effective, durable, resistant, and easy to acquire.

The welds for fixing the profiles can be of the coated electrode type, as they are cost-effective and applicable to metalon. Welding flat joints in metalon is relatively easy to perform, but angle joints can be challenging, mainly due to the possibility of perforations due to the thin walls.

Attention must be paid to the welding current, as if it is too high, perforation will occur. If it is too low, there may not be sufficient penetration, or slag inclusion may occur. In addition, it must be ensured that the welder maintains good temperature control temperature through segment welding and precision in the position and height of the arc through steady, well-supported hands and the use of a short electrode.

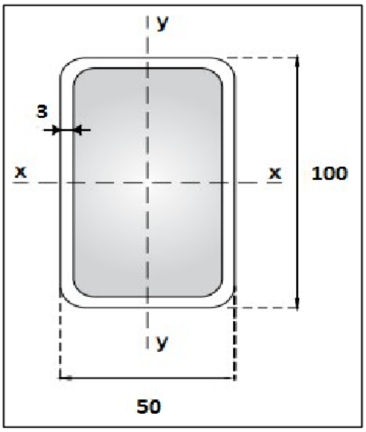

Figure 3.37 – Dimensions of the chosen profile.

Source: Vallourec (2016).

The method for defining the chassis design is basically to ensure that it joins all the vehicle modules harmoniously, obeying some pre-established positions due to their criticality. These are the position of the front wheels and the position of the tractor's heaviest components, which will be presented below.

According to HOSTA (2004), almost 50% of fatalities on farms involving tractors occur due to steering above the vehicle's limit, leading to overturning. The center of gravity (CG), which is dynamic during operation, must remain within the limits of the tractor's base, as shown in Figure 3.38, to prevent overturning.

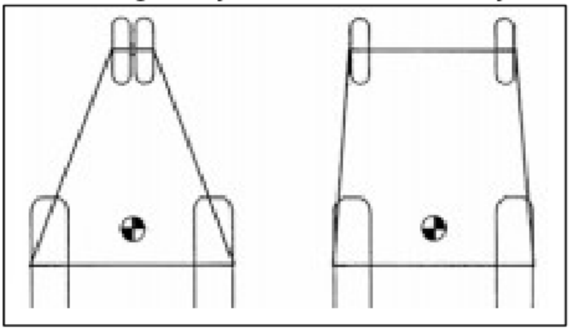

Figure 3.38 – Center of gravity within the stability limits of the base.

Source: HOSTA (2004).

One way to prevent the CG from escaping the base is by increasing the distance between the front wheels. In this work, to reduce the probability of tipping, even if the vehicle operates at very low speeds, the front wheels will be installed on two profiles welded to the tractor chassis.

The oil reservoir, fuel tank, and combustion engine are the heaviest components of the vehicle and should therefore be installed as close to the ground as possible. According to HOSTA (2004), the greater the distance between the CG and the ground, the greater the chance of tipping. For this reason, in this project, we chose to make the section between the chassis axles closer to the ground.

3.4.7 Operator Cab

The purpose of the cab is to allow the operator to operate all the controls necessary for the use of the vehicle from a comfortable position. A comfortable position is understood to be an anthropometrically correct position. According to Barbosa, Santos, and Deganutti (2006), vibration is the main factor that leads to pathologies in tractor drivers, so certain parameters must be followed to reduce vibration.

The ISO 5353 standard specifies that the operator's seat should be arranged as shown in Figure 3.39. Note that the vehicle in this study will not have a steering wheel, so the dimensions relating to this component have been ignored. Figure 3.40 shows the same seat as in Figure 3.39, but from a top view.

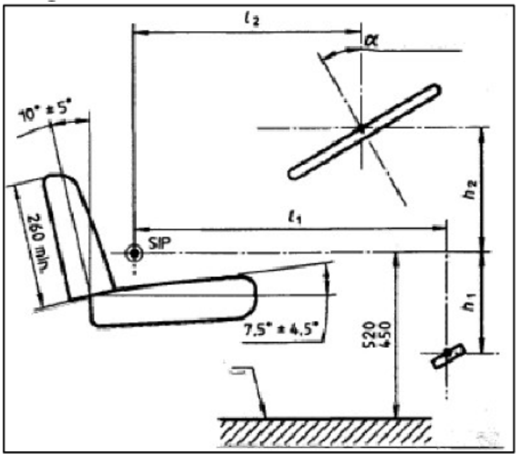

Figure 3.39 – Side view of the seat.

Source: ISO 5353.

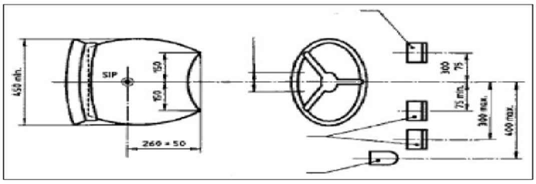

Figure 3.40 – Top view of the seat.

Source: ISO 5353.

The dimensions used in this study were based on the anthropometric dummy developed by Barbosa, Santos, and Deganutti (2006), which is shown in Figure 3.41.

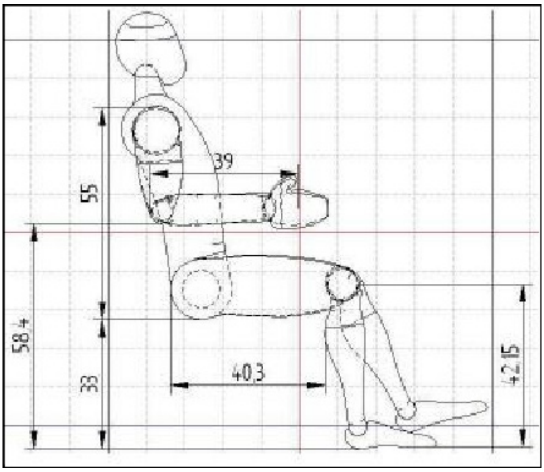

Figure 3.41 – Anthropometric dummy of the tractor driver.

Source: Barbosa, Santos, and Deganutti (2006).

3.5 CT IN SOLIDWORKS

Finally, the concept of the machine in SolidWorks, as developed in this study, is presented in Figs. (3.42) and (3.43).

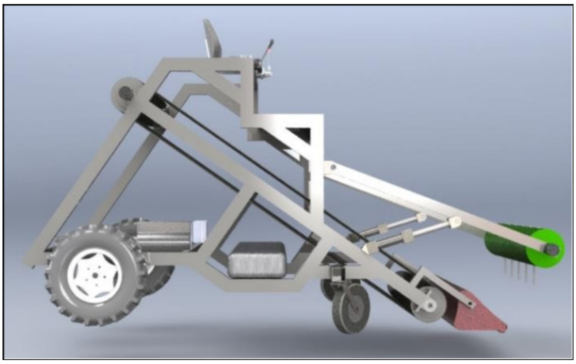

Figure 3.42 – Side view of the tractor.

Source: Author.

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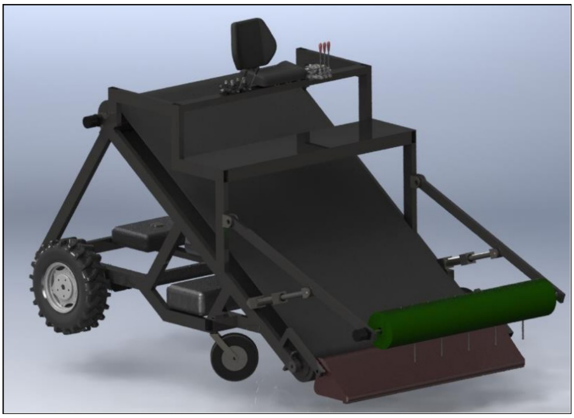

Figure 3.43 – General view of the tractor.

Source: Author.

3.6 INSTRUCTIONS MAINTENANCE

The operator must always have the vehicle maintenance instructions on hand so that it can be operated and maintained correctly. This item addresses the preventive interventions to be performed on the most critical modules.

3.6.1 Conveyor Belt Maintenance

The conveyor belt is a complex component, but the alignment of the belt and the condition of the drums should be checked periodically. The check is intended to look for wear, tears, misalignments, and loose screws.

3.6.2 Harvesting Platform Maintenance

The maintenance to be performed on the collection platform is basically lubrication, cleaning, and retightening. However, the drum with the claws is a component that must always be checked after use, as it is constantly subjected to forces that can break or weaken the claws.

3.6.3 Wheel System Maintenance

Maintenance of the wheel system is simple and consists of calibrating the air pressure in the tires to maintain their grip and stability. Consult the manufacturer's manual to check the pressure and frequency required for this maintenance.

In addition, the system's fastening screws can be retightened on a weekly basis, as this is a simple and easy task.

With regard to the chains used for power transmission, the following items should be noted:

- Alignment of the sprockets;
- Sprocket wear;
- Following a frequency of stops, remove the chains, clean them with degreaser, dip them in oil, and remove the excess by draining;
- Lubricate them with oil.

3.6.4 Hydraulic System Maintenance

The most immediate maintenance tasks to be performed on the hydraulic system are basically checking the oil level, changing the oil, and replacing the filters.

4. CONCLUSION

This work fulfills the objective of developing the concept of an organic compost turning machine with high mobility and simple operation and maintenance.

The chosen conveyor belt, John Deere Hydraflex Draper 700FD, is robust enough for the operation, and its installation angle and working speed allow for a margin of safety. The blades screwed onto the belt can be replaced with others of various sizes, making the design adaptable.

The drum of the collection platform is a simple piece of equipment, but it is ideal for this type of work. Regarding the volume of the compost pile, it has height limitations due to the maximum movement of the drum arm, which is approximately 1.40 m. In addition, there is a limitation on the width of the pile due to the length of approximately 1.80 m of the drum. Therefore, once the prototype is operational, the piles can be organized in different ways, maintaining the total volume of 51.84 m^3.

The wheel system is entirely selected, not designed. This way, those interested in assembling such a tractor can purchase wheels, tires, and axle ends according to their needs.

Choosing a hydraulic system has several advantages, but it is an expensive system that requires careful installation. The suggested manufacturer, Parker, is very active in the domestic market due to its quality, so parts are easily accessible. Those interested in building this equipment can choose other manufacturers, making sure that the power/torque demands are met. The same applies to the Danfoss pump.

Regarding the combustion engine, the ideal would be to purchase an engine specifically designed for tractors, as its technical specifications are far superior to those adapted from a passenger vehicle. However, its price can be a limiting factor.

The chassis structure and operator's seat are important because they are the main factors responsible for operator safety. It is therefore necessary to pay special attention to the technical standards mentioned above, such as ISO 5353 and the HOSTA Task Sheets, which provide sufficient information for vehicle operator safety.

5. SUGGESTIONS FOR FUTURE WORK

This work provides the basis for the design of an agricultural tractor for turning organic compost, but there are still a multitude of actions that can be taken to refine the design. These are:

- Study of the ideal angle and length of the support arm of the harvesting platform;
- Study of the forces acting on and finite element analysis of the chassis structure and operator cab;
- Dimensioning of hydraulic oil transport pipes;
- Detailed analysis of seat anthropometry for greater operator ergonomics;
- Preparation of a complete vehicle maintenance plan, with frequency and steps for performing maintenance services;
- Cost analysis, comparing with other equipment on the market that performs similar services;
- Construction of a prototype.

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APPENDIX A – WORK PLAN

MINISTÉRIO DA EDUCAÇÃO
UNIVERSIDADE FEDERAL DE GOIÁS
ESCOLA DE ENGENHARIA ELÉTRICA, MECÂNICA E DE COMPUTAÇÃO

EMC UFG

PLANO DE TRABALHO DO PROJETO FINAL

Título: Concepção de uma Máquina de Viragem de Composto Orgânico
Desenvolvimento de projetos de equipamento

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Certif. Estudos	

Resumo

O projeto consiste da elaboração e concepção de uma máquina de viragem de composto orgânico para uma unidade produtora de cogumelos da região de Senador Canedo. As máquinas existentes no mercado não atendem às necessidades do produtor, por terem o custo elevado, dimensões não compatíveis com o espaço de trabalho do cliente, e difícil acesso no Brasil, visto que a maioria é de fabricação chinesa e indiana. Esse motivo levou o produtor a solicitar o projeto de um equipamento que tenha a capacidade de produção, a mobilidade em pequenas áreas, o custo e a confiabilidade necessários para ser implementado.

Como se trata do projeto da concepção de um equipamento, não será feita a estratificação completa, como dimensionamento de eixos, engrenagens, mancais, dentre outros elementos de máquinas. Além disso, também não serão feitas análises de cargas e tensões, deflexão e rigidez, e nem simulações computacionais. Na concepção serão definidos como devem ser dispostos e organizados os diferentes sistemas mecânicos presentes na máquina, bem como os materiais recomendados para cada elemento, processos de fabricação e o deve ser fabricado e comprado.

I. Objetivos.

Este trabalho tem como objetivo, elaborar uma máquina de viragem de composto orgânico, conforme as necessidades do cliente de capacidade de produção, tempo de funcionamento diário e semanal, e facilidade de manutenção e operação. É também escopo do projeto fazer uma análise de viabilidade econômica do equipamento, comparando-o com outros existentes no mercado, bem como fazer a seleção de peças, de materiais, processos de fabricação e layouts de todos os sistemas mecânicos.

II. Metodologia (atividades a serem desenvolvidas).

As atividades a serem desenvolvidas, são:

- Definição das características necessárias para operação da máquina;
- Desenvolvimento da concepção da máquina, conforme critérios e restrições definidas anteriormente;
- Desenho 3D, em SolidWorks do conceito;
- Seleção de peças, materiais e processos de fabricação;
- Fazer estudo de viabilidade econômica, comparando o conceito com as máquinas semelhantes já existentes no mercado.

Para cumprir estas atividades, será feito o estudo de máquinas que possuam sistemas semelhantes, para compreensão de como as mesmas foram concebidas, além do estudo de bibliografias que sejam relevantes ao projeto, buscando adaptar os sistemas a este projeto. Como este é uma máquina não existente no mercado, a experiência e orientação serão essenciais para a finalização.

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III. Cronograma.

O cronograma das atividades a serem realizadas é apresentado na Tabela 1.

Tabela 1 – 1° semestre letivo de 2019.

Etapas do Projeto	MAR	ABR	MAI	JUN	JUL
1. Pesquisa Bibliográfica	X				
2. Desenvolvimento	X	X	X	X	
3. Análise dos resultados	X	X	X	X	
4. Elaboração da Monografia			X	X	
5. Defesa do Projeto Final				X	X

Goiânia, 07 de março de 2019.

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