UNIVERSIDAD SAN FRANCISCO DE QUITO USFQ

Colegio de Ciencias e Ingeniería

Design and Construction of a Chassis Dynamometer for a Formula SAE Vehicle Test Bench.

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Trabajo de fin de carrera presentado como requisito

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DE TRABAJO DE FIN DE CARRERA

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RESUMEN

La propuesta del problema de este proyecto fue diseñar y construir un dinamómetro de chasis para el capítulo FSAE de la Universidad. Los requisitos del proyecto eran que el dinamómetro debería dar: la velocidad, la aceleración y el par de las ruedas del vehículo, y debería estar diseñado para futuras mejoras. Después del análisis de fabricación, el diseño del prototipo fue de un conjunto de dos rodillos y un eje para una rueda del vehículo, utilizando un sensor capacitivo para medir las RPM de los rodillos y luego calcular la velocidad del vehículo. Se realizó un proceso de diseño para determinar las propiedades geométricas de los rodillos y del eje, como el ancho de los rodillos y el tipo de rodamiento lo requiere nuestro problema. Los componentes: eje, rodillos y cojinetes, fueron diseñados para vida infinita, y los cálculos de soldadura se realizaron considerando que estaba bajo fatiga. Se realizaron diferentes iteraciones de simulación del prototipo para verificar los cálculos teóricos y el factor de seguridad de la construcción. Se realizó una prueba de velocidad para determinar que los valores obtenidos con el prototipo eran correctos. La prueba de velocidad consiste en la comparación del valor de velocidad del cuentarrevoluciones de la moto y el valor de velocidad obtenido con el prototipo. En conclusión, el prototipo se diseñó correctamente cumpliendo con los requerimientos y expectativas del cliente, entregando gráficos correctos de velocidad, aceleración y par de rueda. Algunos desarrollos futuros para el prototipo son el uso de pantallas para mostrar el valor de los gráficos sin el uso de una computadora. Además, la implementación de la dinamo, por lo que el par de la rueda se puede calcular con el cambio de voltaje, en lugar de calcularse solo de forma teórica.

Palabras clave: dinamómetro de chasis, análisis de fabricación, prueba de velocidad, simulación, torque de las ruedas, rodillos, ejes, rodamientos, cálculos de soldadura.

ABSTRACT

The problem proposal of this project was to design and build a chassis dynamometer for the FSAE chapter of the University. The requirements of the project were that the dynamometer should give the speed, acceleration, and wheel torque of the vehicle, and it should be designed for future improvements. After the manufacturing analysis, the design for the prototype was of a two rollers and shaft assembly for one wheel of the vehicle, using a capacitive sensor to measure the RPM of the rollers, and then calculate the speed of the vehicle. A design process was made to determine the geometric properties of the rollers and shaft, as the width of the rollers and the type of bearing require to our problem. The components: shaft, rollers, and bearings, were designed for infinite life, and the welding calculations were made taking the consideration that it was under fatigue. Different simulation iterations of the prototype were made to verify the theorical calculations and the construction safety factor. A speed test was made to determine that the values obtained with the prototype were correct. The speed test consists of the comparation of the speed value of the tachometer of the motorcycle and the speed value obtained with the protype. In conclusion, the prototype was design correctly meeting the requirements and expectations of the customer, giving a correct speed, acceleration, and wheel torque graphics. Some future developments for the prototype are the use of displays to show the value of the graphs without the use of a computer. Also, the implementation of dynamo, so the wheel torque can be calculated with the change of voltage, instead of being calculated just theorical.

Key words: chassis dynamometer, manufacturing analysis, speed test, simulation, wheel torque, rollers, shafts, bearings, welding calculations

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INTRODUCTION

Annual deaths by car accidents are 1.2 million worldwide and are considered the second death cause for people between 5 and 29 years, and the third one between 30 and 44 years. Many accidents could be prevented by having a great control of the mechanic systems of cars (CEPAL, 2021). Measuring each parameter of the car's systems is not an easy task, it is important to simulate the exact same conditions as the ones on the roads. When it comes to measurements, there are several devices that can help; one of them is the chassis dynamometer.

Chassis dynamometers are devices that use rollers to obtain information from the car wheel's rotation. The use of this test cells has been one of the most important parts of vehicle development and validation for decades, it helps developers to design clean and efficient vehicles (SAE, 2017).

Modern dynamometers can help to measure speed, acceleration, power, fuel consumption and stability in many conditions such as flat terrain or climbs. This device can be combined with others to obtain gas emissions, slip rates, driving force and many other parameters. However, there are still some designs related problems that difficult having exact measurements; these problems are related to calibration, large system deviation, variations in time and uncertain mathematical models, as most of the models are experimental (Zhang & Zhou, 2020).

Problem definition

The SAE (SAE International) chapter of Universidad San Francisco de Quito (USFQ) is currently developing a formula SAE vehicle, which is intended to participate in future competitions. Despite all the theoretical calculations that are needed for the design process, it is important to measure the basic parameters of the vehicle. Currently, the Automotive Workshop at USFQ does not have testing devices that can be used for the formula SAE development and so it is needed to design a device that can give the minimum necessary measurements of the formula SAE vehicle.

The scope of this project is to obtain measurements of at least speed, acceleration, and torque of the formula SAE vehicle. Due to funding limits, it is difficult to achieve other measurements. However, the project will be designed with the possibility of implementing upgrades in future work. The main objective of the project is to design and build a modular chassis dynamometer for the Formula SAE vehicle developed by the SAE student chapter at USFQ. It is expected to obtain an initial fully functional prototype that can measure the basic parameters of the vehicle.

SUBSYSTEM SELECTION

An important aspect to consider for the construction of a dynamometer is the engineering criteria. For this, the entire system must be divided into small subsystems to analyze and define which process is the most efficient for the to analyze and define which process is the most efficient for constructing construction of a dynamometer. To analyze the most optimal process, it is needed to define a matrix with all the possible options, and then grade them between one to five, where one is the least desired option, and five is the most desired option. The process with the highest number is optimal for construction.

In this case, four subsystems were defined: lock, refrigeration, structure, and electronics. To define the most viable process for the construction of the dynamometer, an analysis will be tested between two to four possible options of different construction processes with their due engineering criteria. Based on different research reports, most of the dynamometers are cooled by an air system, therefore, the refrigeration subsystem will not be analyzed like the others, since there is only one viable option in the market that is economically affordable.

Lock subsystem

The locking system of a dynamometer is used to completely block the rollers, so the car can leave the test bench after taking the measurements. For this, there are three options which will be analyzed.

1. **Automatic brake with disk:** The automatic brake with disk process uses a disk system to stop the rollers by means of friction that is activated by an automatic system. This process is very efficient and useful, since it does not need a person to activate it, but due to the complexity of building, maintenance, it is not a viable process.
2. **Manual Brake:** The Manual Brake is a system that uses friction to stop the rollers. Using a lever, a friction force is applied to the rollers. This system principle is similar to the brake disk, but it has to be activated manually by the operator.
3. **Automatic Pin Brake:** The automatic Pin Brake is a process that uses a pin that is inserted through an automatic system between the rollers, preventing their movement. This process is efficient and useful since a person is not needed to do this work, but due to its high cost, difficult maintenance, and construction, it is not a viable process.

Engineering criteria.

To make the decision matrix, the engineering criteria used for the lock subsystem is the following:

1. **Utility:** Ease with which the braking system blocks the rollers so that the vehicle can leave the test bench. (1 = the rollers are not locked, 5 = the rollers are totally locked).
2. **Costs:** How much savings does the option present for the assigned budget (1 = very expensive, 5 = very cheap).
3. **Effectivity:** Accuracy with which the system fulfills the blocking function (1 = not at all effective, 5 = very effective).
4. **Maintenance:** Frequency and difficulty of maintenance (1 = frequent maintenance, 5 = little or no maintenance).
5. **Assembly:** Ease of incorporating the system into the structure of the dynamometer (1 = very complicated to incorporate, 5 = very simple to incorporate).

Lock Subsystem				
Engineering Criteria		Automatic Disk brake	Manual Brake	Automatic Pin Brake
Utility	15%	5	1	4
Cost	45%	1	4	2
Efectivity	10%	5	3	4
Maintenance	20%	2	4	3
Assembly	10%	2	3	2
Total		2,3	3,35	2,7

Table 1. Engineering Criteria Matrix: Lock Subsystem

With this analysis, it is concluded that for the locking system, the Manual Brake process is the most optimal. It is a low-cost system due to its low construction complexity, which makes it easier to build and maintain.

Despite, the low rate in the utility criteria, the manual brake is still the best option. Comparing with the other criteria, it is the cheapest and de most easy to assembly. As well, this option it as good as the other options for the effectiveness of the dynamometer. So, as seen in the table above, the average is much better than the other options. In conclusion, the manual brake is the best option for the case in question.

Structure subsystem

The structure of the dynamometer refers to two aspects. The first is the layout of the roller support and how its configuration will allow to store the rest of the components while being able to hold the weight of the vehicle. The second is the configuration of the rollers where the wheels of the vehicle are going to be laid for the required measurements.

Two structures are proposed for the support of the components:

1. **Semi-fixed structure with ramp:** It consists of a support that needs to be kept in a fixed place, not so easy to move to other places, which has a ramp to raise the vehicle's traction tires.
2. **Detachable Structure:** It refers to a compact "box" type structure that can be easily moved from one area to another so that with the help of hydraulic jacks, the vehicle is suspended, and the drive wheels can be seated on the rollers.

For the configuration of the rollers, there are two options:

1. **One roller configuration:** The tire used for the measurements is located over a single roller which holds the entire weight.
2. **Two rollers configuration:** The tire used for the measurements is held between two rollers so that the weight and balance is distributed between both.

Engineering Criteria.

To make the decision matrix, the engineering criteria for the structure subsystem is the following:

1. **Mobility:** Ease of moving the dynamometer with all its components from one place to another (1 = very difficult to move, 5 = very easy to move).
2. **Costs:** How much savings does the option present for the assigned budget (1 = very expensive, 5 = very cheap).
3. **Materials:** Amount of material that has to be obtained and adapted to what is needed (1 = a lot of material and a lot of work, 5 = little material and a little work).
4. **Maintenance:** Frequency with which maintenance will be required for the system (1 = a lot of maintenance, 5 = little or no maintenance).
5. **Assembly:** Ease of installation and calibration of the rest of the components required to make the dynamometer functional (1=difficult installation, 5=easy installation).

Structure						
Engineering Criteria		Ramp	Detachable	One Roller	Two Roller
Movility	15%	2	4	3	3
Costs	40%	1	3	2	3
Materials	25%	2	4	2	3
Maintenance	5%	4	2	3	2
Assembly	15%	2	4	1	5
Total		1,7	3,5	2,05	3,25

Table 2. Engineering Criteria Matrix: Structure Subsystem

In this case, the decision matrix was not made the same as the first one. For this case, we will consider two options for the type of structure it will deal with. First, we compare doing a ramp or detachable system. It is a clear win for the detachable structure, this type of structure win in mobility, materials, and assembly. The scores are twice with respect to the scores given to the same criteria for the ramp structure. The score is three times bigger for the detachable system in the cost. The only category in which the ramp wins is maintenance, however, this is not as critical as the other criteria. As well, we can see that the average is greater for the second option, so it was decided to do this option.

Also, it was decided between doing one or two roller structures. From the analysis made, it is seen that the assembly is a lot much easier to do. We can see that despite there is not such a great difference between the two and one roller options, the two rollers have better scores in all engineering criteria except in maintenance. As seen from the table above, the average for the two-roller option is the best choice, so that is why this option has been chosen.

The analysis of the decision matrix shows that the best combination option of the two proposed configurations is of a semi-fixed structure with two rollers. In the first instance, the objective could be to build a large structure to cover the entire drive train, however, due to costs and time, it is proposed that the dynamometer work for a single drive train tire using a hydraulic jack to support the vehicle in the dynamometer.

Electronic subsystem

The electronic subsystem will be used to collect and translate data of interest, rotational speed, torque, and power. The rotational speed and torque are obtained through sensors and the power will be obtained by doing the product between torque and angular speed. Two sensor options have been considered for the rotational speed: optical and inductive. A DC motor and a resistance will be used to calculate the torque. The DC motor will be connected to the shaft with a pulley and a transmission band.

Sensor options for speed are:

1. **Optical Sensor:** Sensor widely used in the automotive sector that takes advantage of the rotation of gears to measure speed.
2. **Inductive Sensor:** Detects metallic objects that pass through it with the principle of electromagnetic induction. Its small range of detection is optimal for this application.

Engineering Criteria.

To make the decision matrix, the engineering criteria for the electronics subsystem are the following:

1. **Availability in the market:** How easy is it to find the sensor in the Ecuadorian market, particularly in Quito (1 = very difficult to find, 5 = easily available).
2. **Adaptability:** How complicated is it to program the sensor to the need we have (1 = requires adjustments to use it, 5 = the sensor is designed for what is needed)
3. **Calibration:** How easy it is to install and calibrate the sensor for the purpose (1 = difficult installation and requires several adjustments, 5 = easy installation and does not require further adjustment).
4. **Costs:** How much savings does the option present for the assigned budget (1 = very expensive, 5 = very cheap).

Electronics			
Engineering Criteria		**Hall Effect**	**Inductive**
Availability in the market	30%	2	4
Adaptability	25%	3	2
Calibration	20%	2	4
Costs:	25%	2	4
Total		2,25	3,5

Table 3. Engineering Criteria Matrix: Electronic Subsystem

The decision matrix showed that the best option to measure the speed in the dynamometer is the infrared sensor.

Standard Documentation

The standard documentation for the construction and design of the chassis dynamometer is based on two SAE Standards:

1. SAE J2784: FMVSS Inertia Dynamometer Test Procedure for Vehicles Below 4540 kg GVWR. See Annexes.
2. SAE - Chassis Dynamometer Testing. See Annexes.

DESIGN FOR MANUFACTURING

Components for the manufacture and assemble

For the chassis dynamometer consist of the following components: an absorption unit or driver, the rollers that are ones that develop the torque, a device that measures the torque and rotational speed, an absorber that is a rotor that is connected to the equipment that is meant to test. The list of components is listed below:

1. 4 bearings to hold the shaft and the rollers.
2. 2 shaft and rollers assembly. One of these shafts is the transmission shaft that will be connected to the AC motor. And the other is the support shaft.
3. 1 shaft pulley, that will be used to transmit the rotational speed from the rollers and the shaft to the AC motor.
4. The base that will support the entire system. For the base, we are going to use square steel profiles.
5. 1 inductive sensor to measure the rotational speed. To place at the correct height the sensor, a holder will be used.
6. 1 band to transmit the rotational speed from the rollers to the AC motor.
7. 2 M4 bolts to join the inductive sensor holder with the base.
8. 8 M8 bolts to join the bearing with the base.

Manufacturing technologies

Almost all the manufacturing processes will be done in a job shop. For the manufacturing technologies, the first thing to do is the design, cut, and assembled of the steel profiles of the base. For this, a power saw will be needed with the proper fixtures. To join the different steel parts, a welder will be used with the proper electrode. For the base, no heat treatment will be needed.

For the shaft and the rollers, also a process of welding will be needed, to join the inner shaft with the still lid and the roller. In this case, as the process demands a high geometric tolerance and quality welding, it will be sent to be manufactured in a specialized mechanical workshop. Also, the rollers will be under some heat treatment and chemical attack to generate the necessary hardness and friction surfaces. At last, just some tiny process of hole drilling on the steel profiles will be done, so the bearings can be placed on the base.

To better understand the process, a manufacturing flow diagram is shown below:

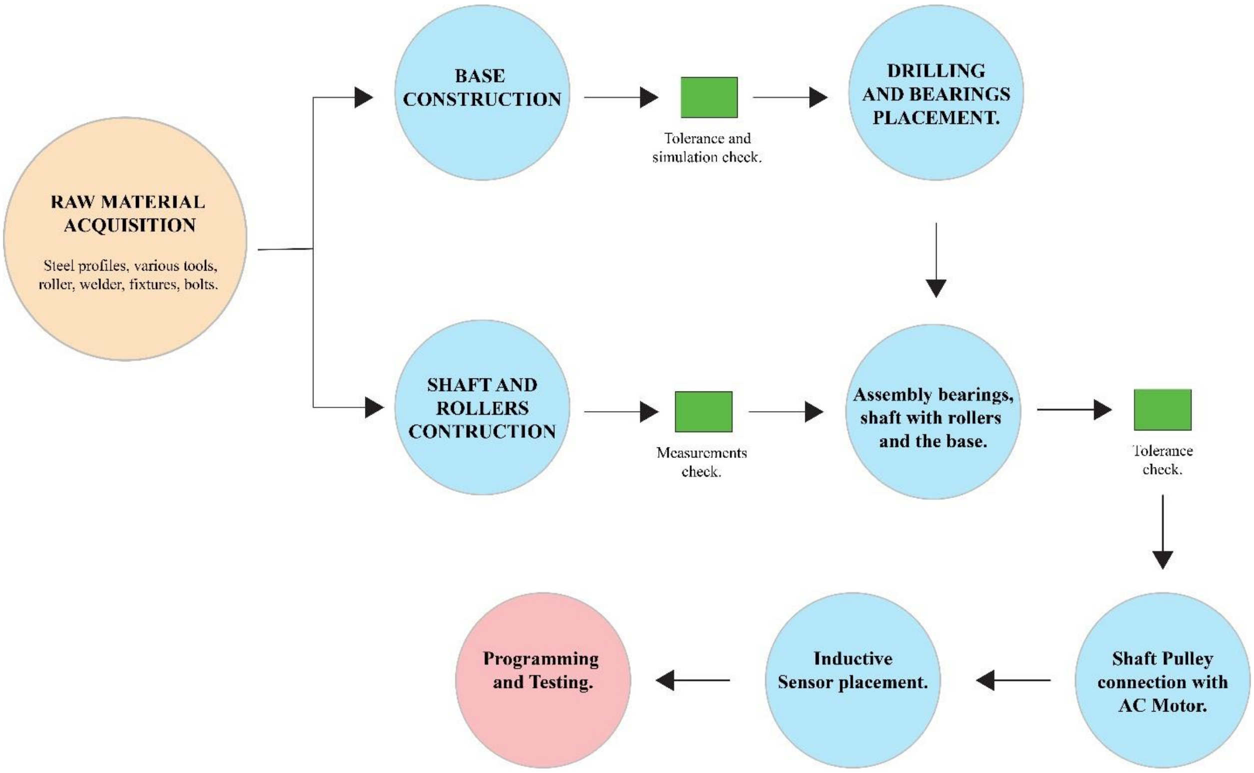


Figure 1. Manufacturing flow diagram

For the manufacturing schedule, the raw material acquisition will be done on the last week of October, and it will last approximately 5 days, so all material can be found before Friday 28th. Meanwhile, the shaft and rollers construction will be sent to a specialized job shop so the geometric tolerance can be followed, also because this is critical part of the system, so the welding must be done by a professional. The base construction will be done on Week 2, first days of November, and it will last approximated 3 days. The drilling and bearings placement can be done in one day. The assembly of the shaft with the bearing will be done between Week 3 and Week 3. While doing this, the connection with the shaft motor also will be taking place, so if any problem appears, it can be solved quickly. The programming and testing part will be a priority since Week 3, because a lot of investigation and iterations will be done.

The next figure summarizes the entire schedule.

Manufacturing schedule									
No	**Manufacturing Steps**	**Week 1** October 24 to October 30	**Week 2** October 31 to November 6	**Week 3** November 7 to November 13	**Week 4** November 14 to November 20	**Responsible for tasks**	**Input Materials**	**Cost**	**Performance Indicator**
1	Raw Material Acquisition					All Team	Steel profiles, various tools, roller, welder, fixtures and bolts	$250,00	Labor Materials
2	Base Construction					Juan Jose Plaza	Steel profiles, welder and steel saw	$0,00	Labor Materials
3	Shaft and Rollers Construction					Luis Felipe Sanchez	Specialized job shop	$150,00	Delivery In Full On Time Rate
4	Drilling and bearings Placement					Juan Jose Plaza and Martin Cardenas	Drill, bearings and bolts	$0,00	Processes and Procedures Developed
5	Assembly bearings, shaft with rollers and the base					Martin Cardenas	Bearings, base, shaft with rollers	$0,00	Processes and Procedures Developed
6	Shaft Pulley Connection with AC Motor					Mateo Montenegro and Luis Felipe Sanchez	Pulley, DC Motor, base, band	$0,00	Operating Margins
7	Inductive Sensor Placement					Mateo Montenegro	Inductive sensor, holder, drill, bolts	$0,00	Operating Margins
8	Programming and Testing					All Team	Computer and cables	$0,00	Rework Rate
							Total	**$400,00**	

Figure 2. Manufacturing Schedule

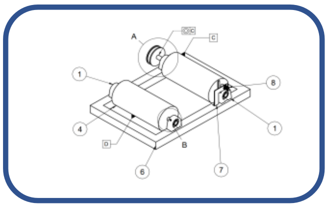
Process Sheet and verification plan

In the following figure, the different process for each part is describe with the corresponding manufacturing description and verification control. As the process has a variety of different tiny processes, only the most critical ones were taken in account. The critical manufacturing parts are the base construction, transmission shaft, support shaft, Inductive holder, and rollers construction.

Process Sheet

USFQ

Project	Chassis Dynamometer
Filled by	Mateo Montenegro
Date	28/10/2022
Approved by	Patricio Chiriboga
Date	**Fall 2022**



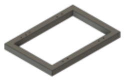
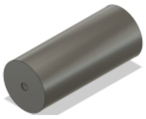
Stage	Piece	Scheme	Processes	Description	Control
1	Base		1	Cut	Caliper
			2	Welding	Visual
			3	Drilling	Caliper
2	Transmission Shaft		1	Cut	Meter
			2	Turning	Caliper
3	Support Shaft		1	Cut	Meter
			2	Turning	Caliper
4	Inductive sensor holder		1	Cutting	Meter
			2	Drilling	Meter
			3	Folding	Meter
5	Rollers		1	Cut	Meter
			2	Milling	NA
			3	Drilling	Caliper
			4	Welding	Visual

Figure 3. Process Sheet

Processing Flow Diagram

In this section, the following figure shows in more depth the process and tolerance that each subsystem will follow for the construction. The most important assemblies are the base structure, shaft and rollers, DC motor – shaft torque transmission, and inductive sensor holder.

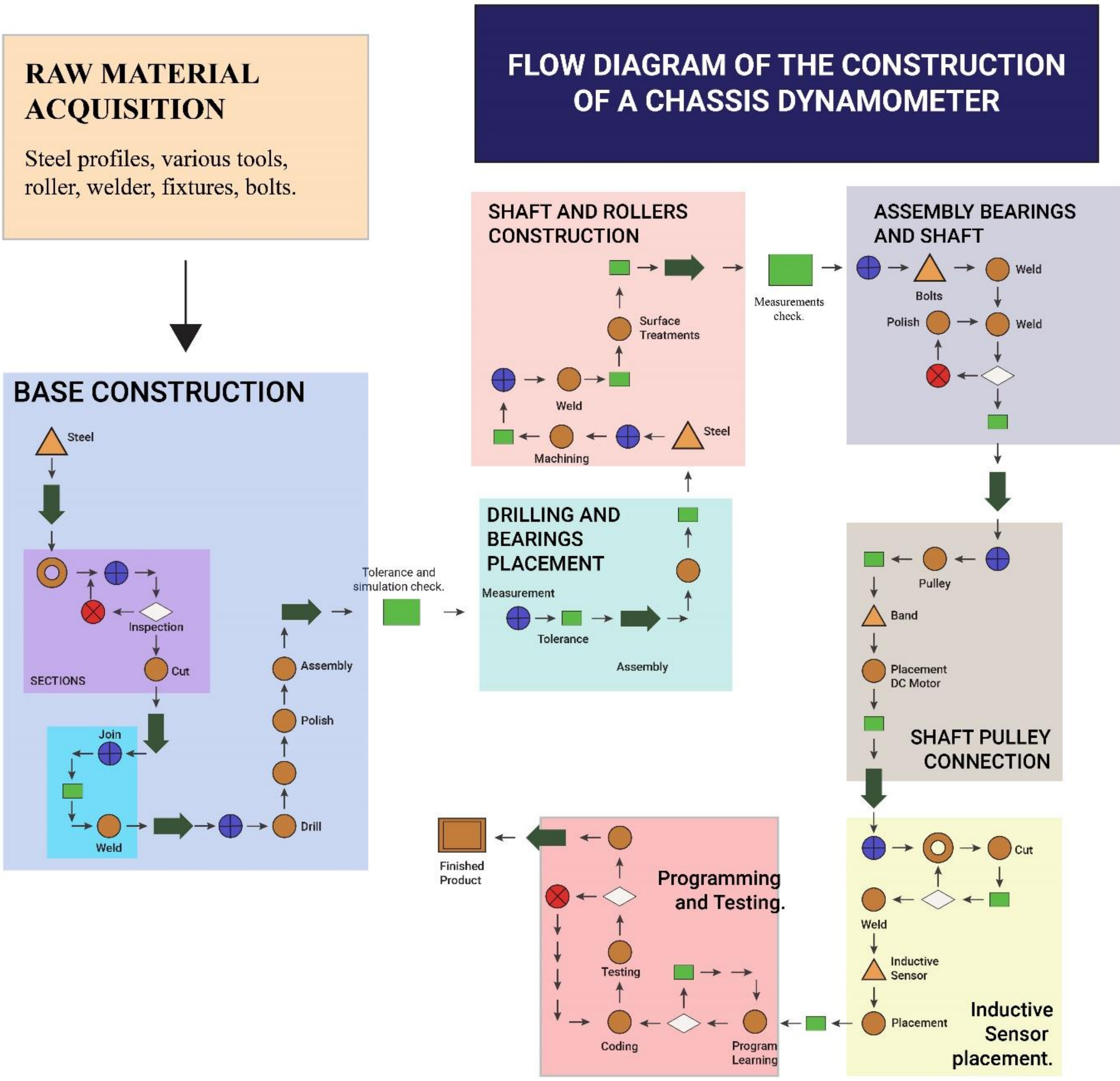


Figure 4. Flow diagram of the construction of a chassis dynamometer

STATIC CALCULATIONS

Calculations and Simulations on Static Assumptions

For the calculations, were defined three states for the dynamometer: before, during and after the test. The following calculations are for the first and third states, where the shaft and rollers will be blocked and the FSAE will be located over the dynamometer. During these stages, the FSAE exert two loads: Momentum and torque. The momentum is generated by the weight of the SAE and the torque is generated when the FSAE gets out of the dynamometer, accelerating and loading a torque on the blocked shaft.

Data and Suppositions

The objective of these first calculations is to find the dimensions of all the pieces that will be used to assemble the chassis dynamometer. Based on the literature, the rollers are to be designed for a maximum torque of 71.5 Nm. In an SAE Vehicle, the dynamometer normally supports a torque of 55 Nm (SAE, 2004). With a safety factor of 1.3, the dynamometer will be designed for a torque of 71.5 Nm.

The wheels of the FSAE at USFQ were measured. Dimensions are as follows:

Figure 5. Formula SAE wheel dimensions

Static Calculations

Based on our design parameter, it is assumed that the maximum weight that the FSAE will have been 400 kg, and that each wheel supports 100 kg (symmetric division) of the weight, so that divided by two rollers, gives 50 kg for each roller. That multiplied by 9.81m/s2 of gravity, gives us a force of 490.5 N.

$$\text{SAE vehicule: } 400\ \text{kg}$$

$$\frac{400\ \text{kg}}{4} = 100\ \text{kg per wheel}$$

$$\frac{100}{2} = 50\ \text{kg per roller}$$

$$\mathrm{F} = \mathrm{m} * \mathrm{a} = 50\ \text{kg} * 9.81\text{ms}^{-2} = \mathbf{490.5\ N}$$

Roller and shaft data

The largest size of pipe available in Dipac is 6” diameter (SAE J403 1008) which is the pipe that is going to be used for the rollers, in addition, according to the SAJ standard for test plants, the angle that must be formed between the center of the wheel and the centers of the roles must be between 60 and 70°.

Considering that the wheels of the formula SAE are 55cm diameter, that the pipe is 6” diameter, and assuming that an angle of 65º is optimal. The following distance between rollers is obtained.

$$\mathrm{R} = 275\ \text{mm} \rightarrow \text{Wheel radius}$$

$$\mathrm{r} = 76.2\ \text{mm} \rightarrow \text{Roller radius}$$

Using sine law and a 65º proposed angle, it is possible to calculate the distance between the center of the rollers. The triangle formed between the center of the rollers and the center of the wheel is isosceles. A sketch of the proposed triangle is shown below.

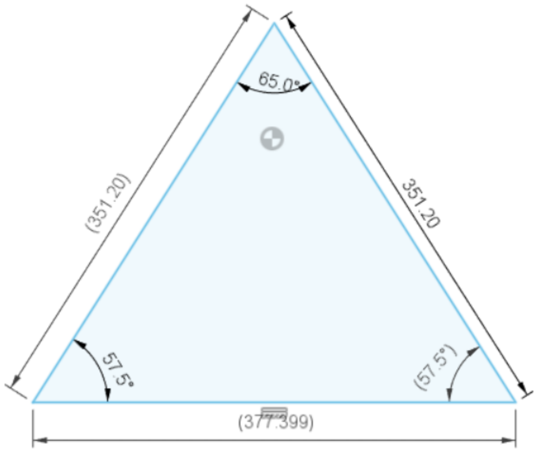


Figure 6. Angles

$$\frac{\sin 65}{\mathrm{d}} = \frac{\sin 57.5}{351.2}$$

$$\mathrm{d} = 377.4 \text{ mm}$$

As the angle range for the wheel is between 60º and 70º, it is possible to round the distance d to 380mm without leaving the suggested range. The final design for the roller's position is shown below.

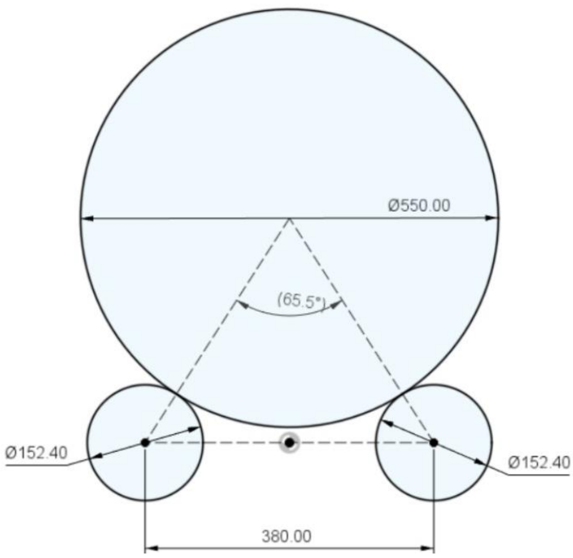


Figure 7. Roller and Wheel sketch design

As the joint between the wheel and the chassis of the FSAE has a clearance of 70mm over the ground, it is necessary to check that the current design will not have any interference with the joint. From CAD design it is possible to obtain the distance between the lowest point of the wheel and the highest part of the rollers.

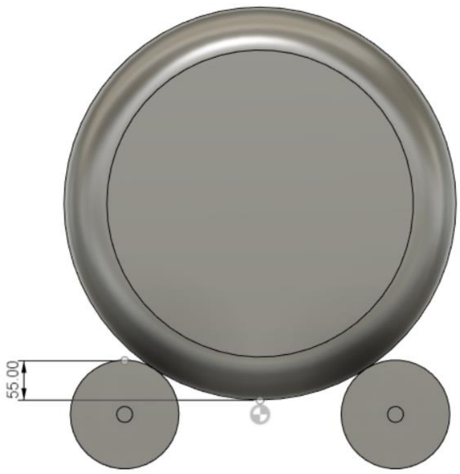


Figure 8. CAD Rollers and Wheel

The distance obtained is 55mm, which is enough to avoid any interference with other parts of the FSAE.

Shaft design for one force

For the roller design it is going to be consider the first and third stage of the dynamometer usage. Considering that the roller will be attached to the shaft by means of caps, these will act as a fixed support at both ends while the load is applied in the middle of it. The free body diagram shows the loads to which the roller will be subjected.

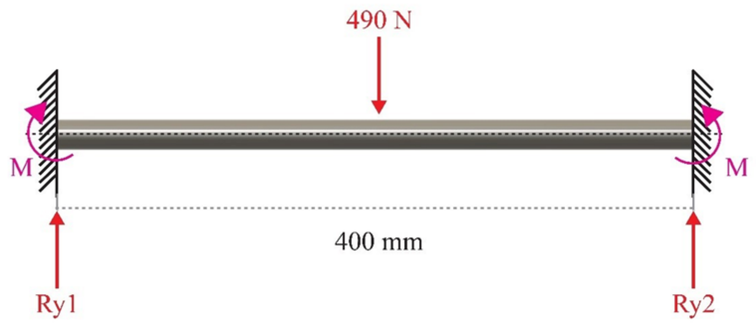


Figure 9. Shaft Body Diagram

The force and moment reactions must be obtained to perform the stress analysis. Considering that the roller covers are proposed as fixed supports (since they join the roller with the axis), this problem becomes statically indeterminate. As stated by Beer et al. (2015) to solve this type of problem, it must be broken down into some instances and obtaining its deflection and slope equations, it is possible to find the missing reactions (p.636). For this case, the roller is broken down into three already known cases whose equations are tabulated in Appendix D of the book.

Case 1.

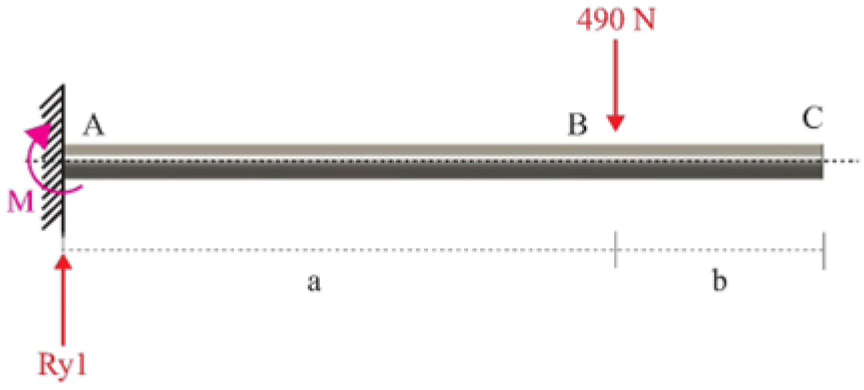


Figure 10. Case 1 Shaft Diagram One Force

$$(\theta_B)_P = (\theta_C)_P = -\frac{Pa^2}{2E\,I}$$

$$(y_B)_P = (y_C)_P + (\theta_B)_P b$$

$$(y_B)_P = -\frac{Pa^2}{3E\,I} - \frac{P\,a^2}{2EI} b$$

$$(y_B)_P = -\frac{Pa^2}{3E\,I}(2a + 3b)$$

Case 2.

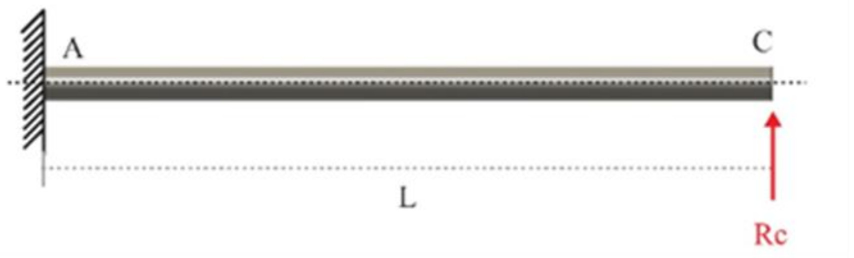


Figure 11. Case 2 Shaft Diagram One Force

$$(\theta_B)_{R_B} = \frac{R_B L^2}{2EI}$$

$$(y_B)_{RB} = \frac{R_B L^3}{3EI}$$

Case 3.

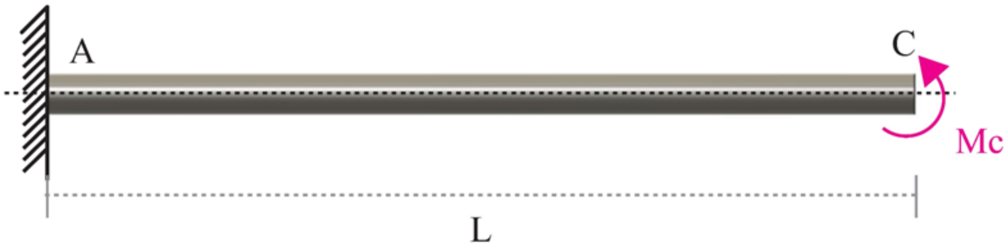


Figure 12. Case 3 Shaft Diagram One Force

$$(\theta_B)_{M_B} = \frac{M_B L^2}{EI}$$

$$(y_B)_{M_B} = \frac{M_B L^2}{2EI}$$

For this case:

With the first border condition, that at point B the deflection and the slope must be cero, the following equations are obtained:

$$a = b = \frac{L}{2}$$

$$M_B = M_A = -\frac{PL}{8}$$

$$R_B = R_A = \frac{P}{2}$$

$$\text{Border Conditions: } X = L \qquad \theta_B = 0$$

$$\theta_B = (\theta_B)_P + (\theta_B)_{R_B} + (\theta_B)_{M_B}$$

$$0 = -\frac{Pa^2}{SEI} + \frac{R_B L^2}{2EI} + \frac{M_B L^2}{EI}$$

$$\text{Border Conditions: } x = L \qquad y_B = 0$$

$$y_B = (y_B)_P + (y_B)_{R_B} + (y_B)_{M_B}$$

$$0 = -\frac{Pa^2}{3EI}(2a + 3b) + \frac{R_B L^3}{3EI} + \frac{M_B L^2}{2EI}$$

To get the maximum moment, the following values are replaced with the previous data exposed:

$$M_A = -\frac{PL}{8}\frac{= (490.5 * 0.2)}{8} = 12.26\ \text{Nm}$$

For the maximum stress:

$$\sigma = \frac{MC}{I}$$

$$C = 3\ \text{in} = 76.20\ \text{mm}$$

$$I = \frac{1}{2} * \pi * (R^4 - r^4)$$

$$\sigma = \frac{12260 * 76.20}{\frac{1}{2} * \pi * (76.20^4 - 73.20^4)} = \frac{1868424000}{125764477.8} = 0.1189\ \text{MPa}$$

With this result, it is possible to understand that stresses are not critical for the roller design. Due to this, the roller selection will be based on the geometry needed.

This section will analyze the forces to which the shaft will be subjected at two critical moments of operation. To obtain the measurements, the vehicle must be mounted on the dynamometer (first stage), then it will be accelerated so that the transmission rotates the rollers (second stage, data collection) and finally the vehicle must be braked so that it stops rotating. The rollers can be disassembled (third stage). Note that for the first and second stages the shaft will be static and subjected to the weight of the vehicle, so reaction forces and moments will be experienced in the bearings. On the other hand, in the second stage, the rotation will generate a torque on the shaft. The following figure shows the scheme of all the moments, forces, and torques to which the axes would be subjected.

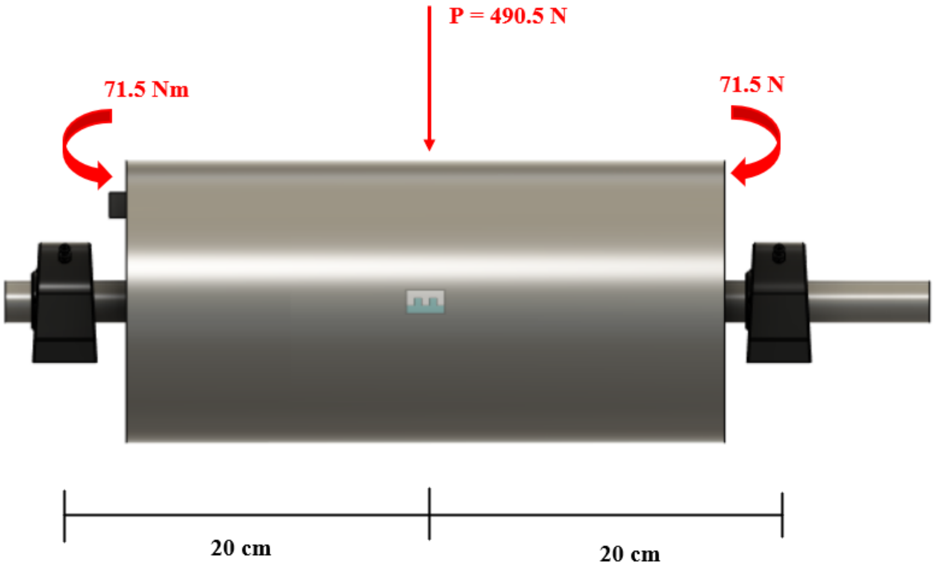


Figure 13. Moment and Shaft Diagram

Torque in B was supposed to be 71.5 Nm based on previous measurements of Formula SAE vehicles. For the reactions, due to symmetry we are going to suppose that they are as follows:

$$R_A = R_C = 245.25 \text{ N}$$

The distance between the support point of the vehicle to the bearings is the same since the moment they will experience will be the same. With a distance from the center to the end of 0.20 m, the moments will be:

$$M_A = M_C = 490{,}5 \text{ N} \times 0{,}2 \text{ m}$$

$$M_A = M_C = 98{,}1 \text{ Nm}$$

Effect of the loads to which the shaft was shown to be subjected (bending and torsion), the analysis of bending stress and shear stress was carried out. Going back to the diagram, we have:

$$\tau_{xy} = \frac{16 \cdot T_B}{\pi \cdot d^3}$$

$$\tau_{xy} = \frac{16 \cdot 71500}{\pi \cdot d^3}$$

$$\tau_{x\,y} = \frac{364150}{d^3}\ \text{MPa}$$

$$\sigma_x = \frac{32\ \cdot M}{\pi \cdot d^3}$$

$$\sigma_x = \frac{32\ \cdot (98\ 100)}{\pi \cdot d^3}$$

$$\sigma_x = \frac{999\ 240}{d^3}\ \text{MPa}$$

Using the maximum shear criterion, first analytically:

$$\tau_{max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{x\,y}{}^2}$$

$$\tau_{max} = \sqrt{\left(\frac{99\ 9240}{2 \cdot d^3}\right)^2 + \left(\frac{364150}{d^3}\right)^2}$$

$$\tau_{max} = \frac{618243{,}78}{d^3}\ \text{MPa}$$

And comparing the formula provided by Shigley where n is the safety factor of 1.5 for design.

$$\tau_{max} = \frac{S_y}{2 \cdot n}$$

$$\tau_{max} = \frac{235}{3}$$

Combining both formulas:

$$\tau_{max} = \tau_{max}$$

$$\frac{618243{,}78}{d^3} = \frac{235}{3}$$

$$d = 20\ \text{mm} = 0{,}79\ \text{in}$$

Choosing a 7/8-inch (22.225) shaft corresponding to the Dipac catalog, the safety factor of this design will be:

$$\tau_{max} = \frac{S_y}{2 \cdot n}$$

$$\frac{618243{,}78}{d^3} = \frac{235}{2 \cdot n}$$

$$\frac{618243{,}78}{(22{,}225)^3} = \frac{235}{2 \cdot n}$$

$$n = 2.09$$

Shaft calculations for two forces

We are going to analyze the shaft now considering a better distribution of the force. Instead of considering only one force on the center of the shaft, we are going to assumed that the force is distributed on. The joints between the rollers and the shaft. The body diagram will be as follows:

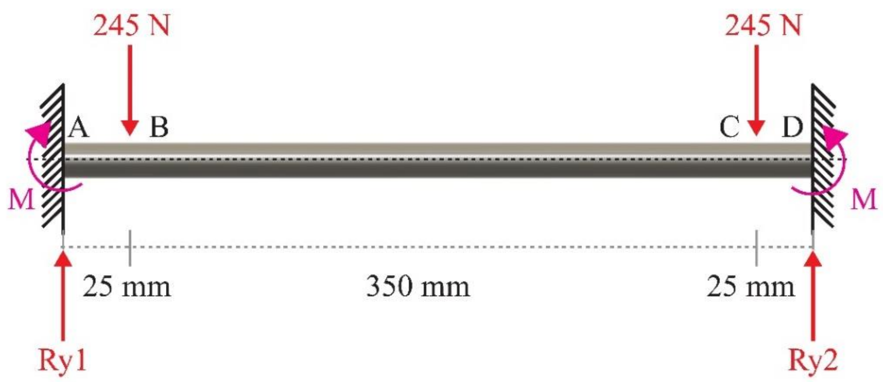


Figure 14. Shaft Body Diagram for two forces

For the force calculations:

$$\sum F_x = 0$$

$$\sum F_y = 0$$

$$Ry_1 - 245\ \text{N} - 245\ \text{N} + Ry_2 = 0$$

$$\mathrm{Ry_1} = \mathrm{Ry_2} \text{ for symmetry}$$

$$\mathrm{Ry} - 490 = 0$$

$$\mathrm{Ry} = 490 = \mathrm{Ry_1} = \mathrm{Ry_2} = \frac{\mathrm{Ry}}{2} = \frac{490}{2} = 245\ \mathrm{N}$$

For the moment calculations, we have an indeterminate to the second degree on the beam. But, as the forces are symmetric, we know the value of the reaction forces. With this, we can easily calculate the moment at one of the ends of the shaft, and as they are symmetric, the moment is going to be the same in the other side.

The deformation at the right-end (point D) of the shaft is going to be as follows:

$$(\theta_D)_p = (\theta_B)_{P_1} + (\theta_C)_{P_2}$$

$$(\theta_D)_p = -\frac{P_1 a_1^2}{2EI} - \frac{P_2 a_2^2}{2EI}$$

We know the Boundary conditions at the end of the shaft (Point D). The slope and the deflection must be zero:

$$[x = L; \theta_D = 0]$$

So, for the slope equation, using the book Mechanics of Materials (Beer, 2015, p. 460) we have:

$$\theta_D = (\theta_D)_p + (\theta_D)_R + (\theta_D)_M$$

$$(\theta_D)_R = +\frac{R_C L^2}{2EI}$$

$$(\theta_D)_M = \frac{M_C L}{EI}$$

$$\theta_D = -\frac{P_1 a_1^2}{2EI} - \frac{P_2 a_2^2}{2EI} + \frac{R_C L^2}{2EI} + \frac{M_C L}{EI}$$

Doing some algebra, and putting values we have the following equation:

$$M_D(L) = \frac{P_1 a_1^2}{2} + \frac{P_2 a_2^2}{2} - \frac{R_c L^2}{2}$$

$$M_D(400) = \frac{(245)(25^2)}{2} + \frac{(245)(375^2)}{2} - \frac{(245)(400^2)}{2}$$

$$M_D = -5742.18 \text{ Nmm}$$

This means, that the momentum is positive, but clockwise direction. Therefore,

$$M_D = 5742.18 \rightarrow \text{Clockwise}$$

$$M_A = -5742.18 \rightarrow \text{Counterclockwise}$$

Effect of the loads to which the shaft was shown to be subjected (bending and torsion), the analysis of bending stress and shear stress was carried out.

For the Shear Stress:

$$\tau_{xy} = \frac{16 \cdot T_B}{\pi \cdot d^3}$$

$$\tau_{xy} = \frac{16 \cdot 71500}{\pi \cdot d^3}$$

$$\tau_{xy} = \frac{364150}{d^3} \text{ MPa}$$

For the normal Stress:

$$\sigma_x = \frac{32 \cdot M}{\pi \cdot d^3}$$

$$\sigma_x = \frac{32 \cdot (5.742 \text{ N} * \text{m})}{\pi \cdot d^3}$$

$$\sigma_x = \frac{183.744}{d^3} \text{ MPa}$$

Using the maximum shear criterion:

$$\tau_{max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{x\,y}{}^2}$$

$$\tau_{max} = \sqrt{\left(\frac{183.744}{2 \cdot d^3}\right)^2 + \left(\frac{364150}{d^3}\right)^2}$$

$$\tau_{max} = \frac{364150.023}{d^3}\ \mathrm{MPa}$$

Comparing with the text formula, we obtain the safety factor:

$$\tau_{max} = \frac{S_y}{2 \cdot n}$$

$$\tau_{max} = \frac{235}{3}$$

$$\tau_{max} = \tau_{max}$$

$$\frac{364150.02}{d^3} = \frac{235}{3}$$

$$d = 16.689\ \mathrm{mm} = 0{,}656\ \mathrm{in}$$

The obtained diameter for the shaft is 16.689mm, which corresponds to 0.656in. According to catalogs obtained, the best option is a 7/8” shaft. The final security factor for this shaft is:

$$n = \frac{S_y}{2\tau_{max}}$$

$$n = 3.54$$

FATIGUE CALCULATIONS

Shaft Fatigue calculations

For the shaft fatigue calculations, it was assumed the same load distribution as the static case. The loads for this case are as follows. Where P=490.5N, T=71.5Nm and L=0.2m.

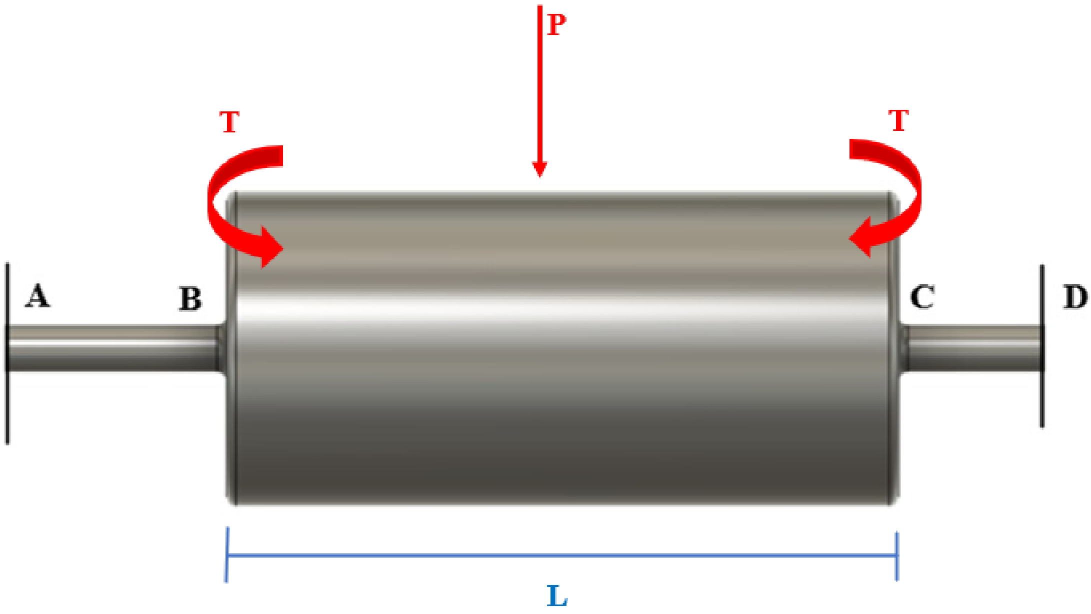


Figure 15. Critical Points Shafts

The critical point for this shaft is located at the sides of the shaft, just after the bearings. The loads and material properties are listed below.

Material: AISI 1018

$$S_y = 235 \text{ MPa}$$

$$S_{ut} = 410 \text{ MPa}$$

Loads:

$$R_y = 245 \text{ N}$$

$$M_{max} = -5742.18 \text{ Nmm}$$

$$T = 71500 \text{ Nmm}$$

This shaft will be designed for infinite life with the DE-Goodman theory (Budynas & Nisbett, 2015).

Endurance Limit

The endurance limit for this shaft is calculated below.

$$S_e = k_a k_b k_c k_d k_e k_f S'_e$$

As the S_{ut} for AISI 1018 is 410 MPa, the following estimation can be made.

$$S'_e = 0.5 S_{ut}$$

$$S'_e = 0.5(410) = 205 \text{ MPa}$$

The Surface modifying factor corresponds to

$$k_a = a S_{ut}^b$$

We select the Machined or cold-drawn factors a=4.51 and b=-0.265

$$k_a = 4.51(410)^{-0.265}$$

$$k_a = 0.916$$

The size factor will be assumed with the static load case, where the diameter of the shaft is 7/8" (22.225 mm).

$$k_b = 1.24 d^{-0.107}$$

$$k_b = 1.24(22.225)^{-0.107}$$

$$k_b = 0.890$$

As there is a combined load case, the loading factor is assumed 1.

$$k_c = 1$$

For this design, the shaft will not reach any extreme temperatures, the temperature factor will be assumed to be 1.

$$k_d = 1$$

For the reliability factor, it will be assumed a reliability of 95%, which corresponds to a factor of 0.868 (Budynas & Nisbett, 2015)

$$k_e = 0.868$$

The miscellaneous-effects factor will also be assumed to be 1 as there are no special conditions for the shaft.

$$k_f = 1$$

With all the modifying factors, the endurance limit can be calculated.

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

$$S_e = (0.916)(0.890)(1)(1)(0.868)(1)(205)$$

$$S_e = 145.06 \text{ MPa}$$

Stress Concentrators

For this design, the shaft will not have stress concentrator as there are no area changes or any other kind of stress concentrators.

$$K_f = K_{fs} = 1$$

Fluctuating loads

For this case, the bending moment can be considered completely reversible as the shaft will be rotating. And the torque will be held constant.

$$M_m = 0 \text{ Nmm}$$

$$M_a = 5742 \text{ Nmm}$$

$$T_m = 71500 \text{ Nmm}$$

$$T_a = 0 \text{ Nmm}$$

Using the DE-Goodman theory and a safety factor of 1.5, it is possible to obtain the needed diameter for the shaft (Budynas & Nisbett, 2015).

$$d = \left(\frac{16n}{\pi}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(K_{fs}T_a)^2\right]^{\frac{1}{2}} + \frac{1}{S_{ut}}\left[4(K_f M_m)^2 + 3(K_{fs}T_m)^2\right]^{\frac{1}{2}}\right\}\right)^{\frac{1}{3}}$$

$$d = \left(\frac{16(1.5)}{\pi}\left\{\frac{1}{145.06}\left[4\big((1)5742\big)^2 + 3\big((1)0\big)^2\right]^{\frac{1}{2}}\right.\right.$$

$$\left.\left. + \frac{1}{410}\left[4\big((1)0\big)^2 + 3\big((1)71500\big)^2\right]^{\frac{1}{2}}\right\}\right)^{\frac{1}{3}}$$

$$d = 14.28\text{mm}$$

As the 7/8" shaft needed for the static loading is greater than the shaft needed for fatigue, it will be selected for the design.

Fatigued rollers Calculations

The rollers being the part that supports the distributed weight of the vehicle and will rotate due to its traction. For this reason, at the time of operation, they will be subject to a point load plus the torque produced by the tires. The following free body diagram shows the loads to which each roller will be subjected.

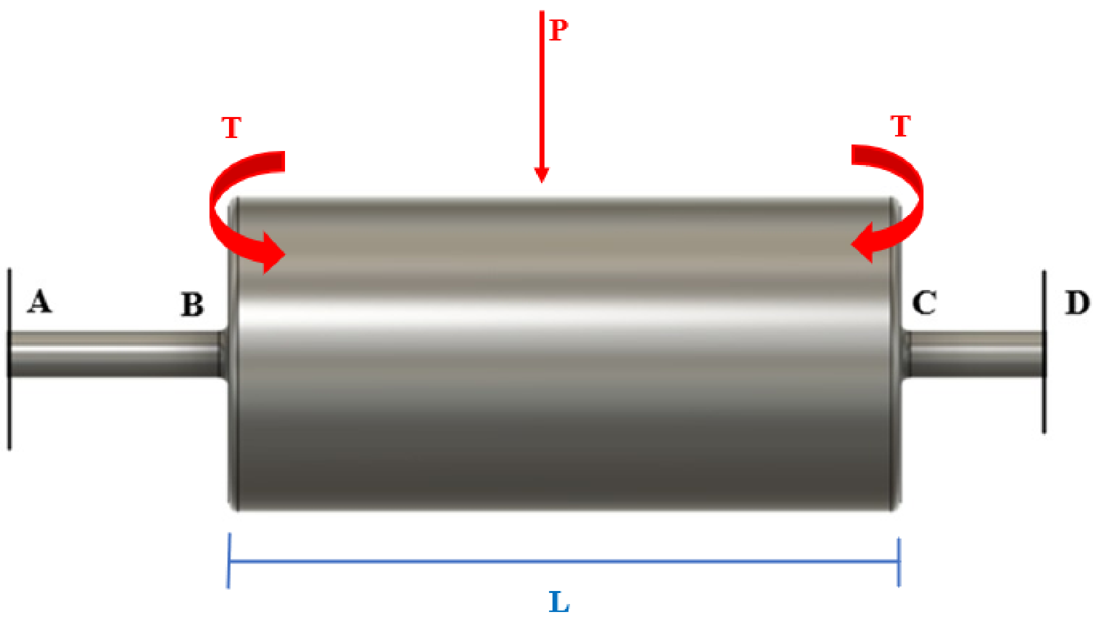


Figure 16. Force on Rollers

Being the load P = 490.5 N determined according to the design criteria, T = 71.5 Nm, which is the torque that the researched bibliography establishes as the torque exerted by the traction of a vehicle of formula SAE and L = 350 mm, which is the length that It was determined that it is required to support the tires. The roller has an external diameter of 6 in (152.4 mm) and a thickness of 3 mm.

With these data, it is necessary to recognize where the most critical points are and for this, the shear force and moment diagrams are required. With the equations obtained in the static calculations, the value of the maximum moment can be obtained, but it is necessary to locate it. For this and to check the calculations, the free online software Optimal Beam was used, which provided the following graph.

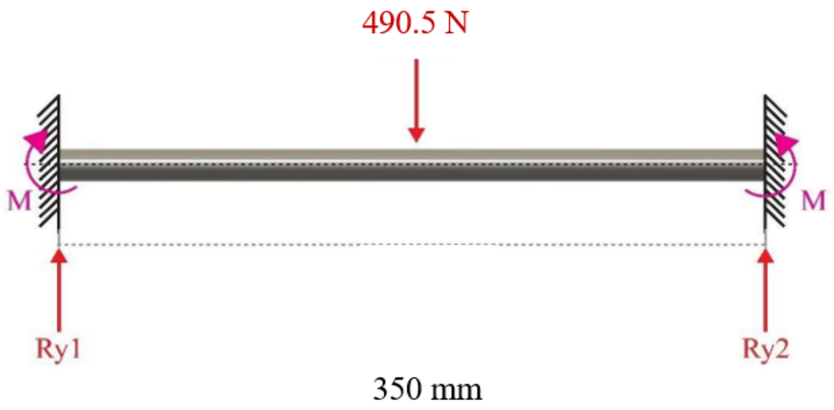


Figure 17. Body Diagram with One force

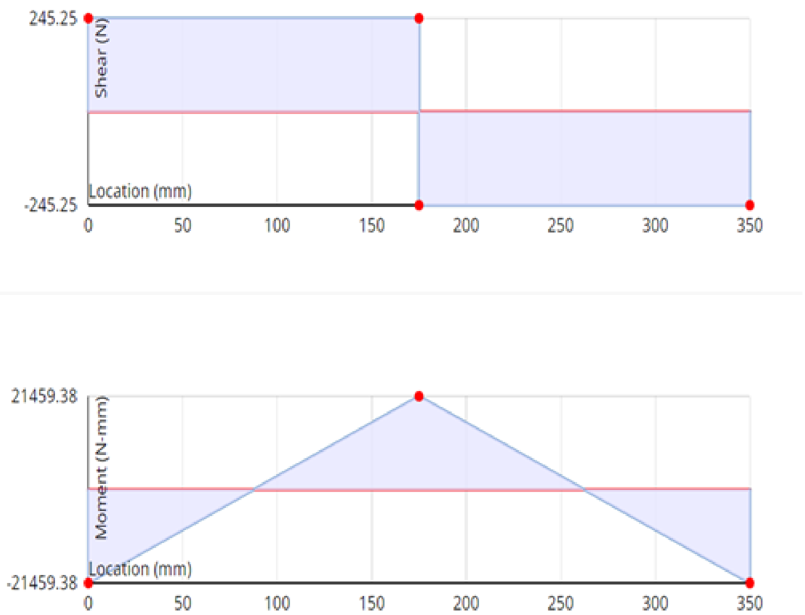


Figure 18. Force and Momentum graphs

It can clearly be seen that the maximum moments will occur at the ends of the roller (where it is attached to the axis) and in the middle zone. Being the middle area where the load of the vehicle's weight is based, it will be taken as the most critical area.

It must be considered that, as indicated by the free body diagram, the rollers are hollow and have a thickness of 3 mm. The following figures indicate the most critical zone of the hollow roller and the infinitesimal analysis of the most critical zone.

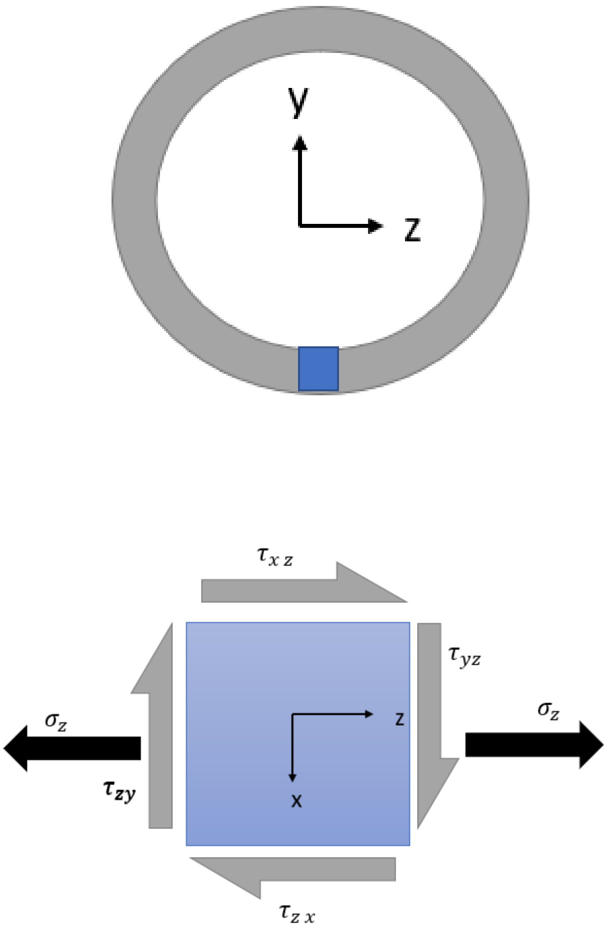


Figure 19. Most critical Zones on the hollow roller

With the critical zone identified, the type of stress that will be subjected must be recognized. First, this is the shear stress effect of the torque which, being a fixed value due to the traction of the tires, will be a constant value. On the other hand, the bending stress effect of the moments in the roller will alternate between tension and compression, this can be corroborated by the moment diagram and by the fact that the roller will be in rotation, alternating this value from maximum to minimum, being this way the fully reversible bending stress. The following figures show both efforts as a function of time.

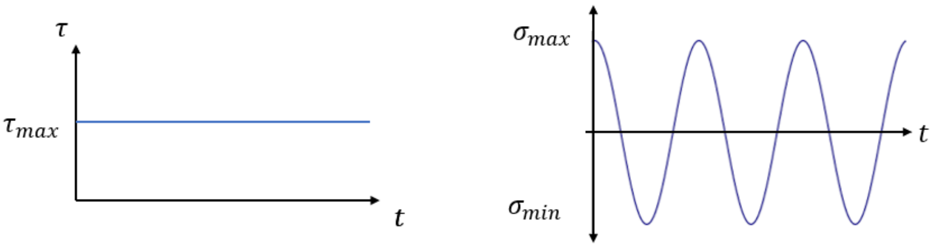


Figure 20. Fully reversible bending stress diagrams

Note that due to the type of stress when fatigue stresses are analyzed, the shear amplitude stress will be zero, while for the bending stress its average stress will be zero. To obtain the missing values for the Von Misses analysis, as the literature dictates, the rotational inertia, and the moment of inertia for shear and bending stress, respectively, are required. As the cylinder is hollow, the formula will have a variation since the difference between the external diameter and the internal diameter is required, as shown below.

$$J = \frac{\pi}{32}\left({d_{ext}}^4 - {d_{in}}^4\right)$$

$$J = \frac{\pi}{32}(152.4^4 - (152.4 - 6)^4)$$

$$J = 7860779.86 \text{ mm}^4$$

$$I = \frac{\pi}{64}\left({d_{ext}}^4 - {d_{in}}^4\right)$$

$$I = \frac{\pi}{64}(152.4^4 - (152.4 - 6)^4)$$

$$I = 3930139.93 \text{ mm}^4$$

Now with these values we only need the maximum moment of bending. The formula obtained with the static analysis is the following.

$$M_{max} = \frac{P\,L}{8}$$

$$M_{max} = \frac{490.5 \cdot 0.35}{8}$$

$$M_{max} = 21450.45\ \mathrm{Nm}$$

With all the required values and knowing that c is equal to $d_{ext}/2$ the stresses are:

$$\sigma = \frac{M\,c}{I}$$

$$\sigma = \frac{21450.45\ \cdot \frac{152.4}{2}}{3930139.93}$$

$$\sigma = 0.42\ \mathrm{MPa}$$

$$\tau = \frac{T\,c}{J}$$

$$\tau = \frac{71.5\ \cdot \frac{152.4}{2}}{7860779.86}$$

$$\tau = 0.6931\ \mathrm{MPa}$$

Note as the bending stress is completely reversible the value obtained is the same in tension as in compression. In this way, the Von Misses stress equations can be used. Since there are no stress concentrators in the rollers, all the values of K will be equal to 1.dsd.

$$\sigma_a' = \left\{\left[(K_f)_{bending}\,(\sigma_a)_{bending} + (K_f)_{axial}\frac{(\sigma_a)_{axial}}{0.85}\right]^2 + 3[(K_{fs})_{torsion}(\tau_a)_{torsion}]^2\right\}^{1/2}$$

$$\sigma_a' = \{[1\ \cdot\ 0.42 + 0]^2 +\ 3[1\ \cdot 0]^2\}^{1/2}$$

$$\sigma_a' = 0.42\ \mathrm{MPa}$$

$$\sigma'_m = \left\{\left[(K_f)_{bending}\,(\sigma_m)_{bending} + (K_f)_{axial}(\sigma_m)_{axial}\right]^2 + 3[(K_{fs})_{torsion}(\tau_a)_{torsion}]^2\right\}^{\frac{1}{2}}$$

$$\sigma'_m = \{[0+0]^2 + 3[1 \cdot 0.6931]^2\}^{\frac{1}{2}}$$

$$\sigma'_m = 1.20\ \mathrm{MPa}$$

With the efforts of Von Misses and using a sufficiently conservative theory of safety (such as Goodman's), the safety factor of the designed components can be obtained. To use Goodman's theory, the ultimate effort is required, which will be calculated as dictated by Budynas and Nisbett (p. 292). The investigated properties of the type of steel to be used are S_{ut} = 340 MPa, S_y = 285 MPa. The following table summarize the values k for the endurance limit.

k Factor	Description	Coments	Calculations (If it requires)	Value
ka	Surface condition modification factor	Cold Drawn	$ka = aS_{ut}{}^b$ $ka = 4.51 \cdot 340^{-0.265}$	0.96
kb	Size modification factor	Diameter 6 in	$kb = 0.91 \cdot d^{-0.157}$ $kb = 0.91 \cdot 6^{-0.157}$	0.687
kc	Load modification factor	Bending		1
kd	Temperature modification factor	Ambient temperature		1
ke	Reliability factor	Reliability of 95 %		0.687
kf	Miscellaneous-effects modification facto	None effect		1

Table 4. Values k for the endurance limit

Using the previous values, we can use the following equation.

$$\mathrm{Se} = \mathrm{ka} \cdot \mathrm{kb} \cdot \mathrm{kc} \cdot \mathrm{kd} \cdot \mathrm{ke} \cdot \mathrm{kf} \cdot \mathrm{Se}'$$

$$\mathrm{Se} = 0.96 \cdot 0.687 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1 \cdot (0.5 \cdot S_{ut})$$

$$\mathrm{Se} = 0.96 \cdot 0.687 \cdot 1 \cdot 1 \cdot 0.868 \cdot 1 \cdot (0.5 \cdot 340)$$

$$\mathrm{Se} = 97.32\ \mathrm{MPa}$$

Using the Goodman criteria

$$\frac{1}{n} = \frac{\sigma'_a}{\mathrm{Se}} + \frac{\sigma'_m}{S_{ut}}$$

$$\frac{1}{n} = \frac{0.42}{97.32} + \frac{1.20}{340}$$

$$n = 127.5$$

With this it is can be concluded that the design is safe and has infinite life.

WELDING CALCULATIONS ON SHAFT AND ROLLERS

The objective of the welding calculations is to find the theorical size of the leg of the filet, so welding can support easily the apply force if the electrode is type E60XX. Then, measure the actual size of the leg of the filet, and calculate the safety factor.

For the welding calculations we are going to assume that the two joints are uder bending moment. The wheel of the car will exert a force downwards, as show in the next figure.

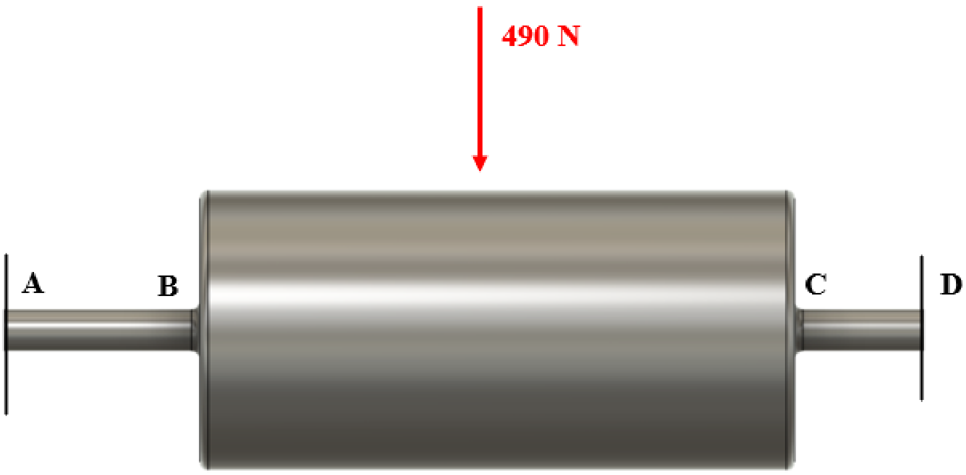


Figure 21. Welding force diagram 1

This force will flow to the edges of the shaft, creating a bending moment as shown in the next figure.

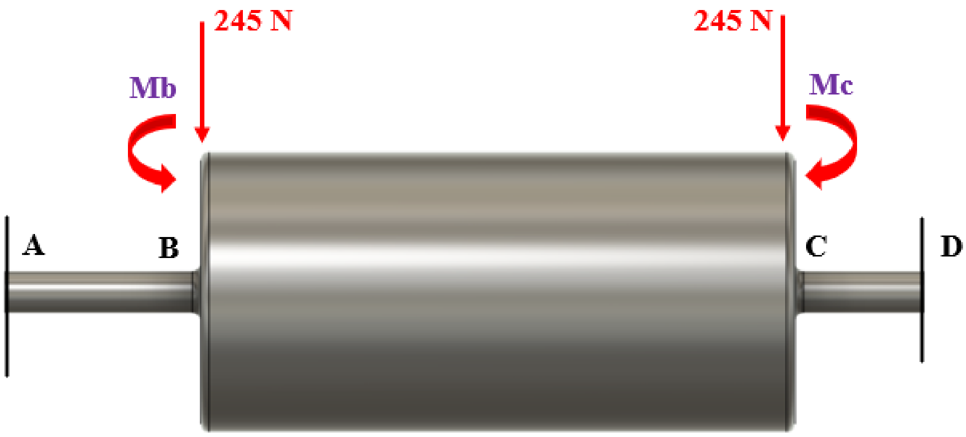


Figure 22. Bending Moment welding calculation

Then, the force will flow to the shaft, and the principal shear stress and secondary bending stress will be as follows:

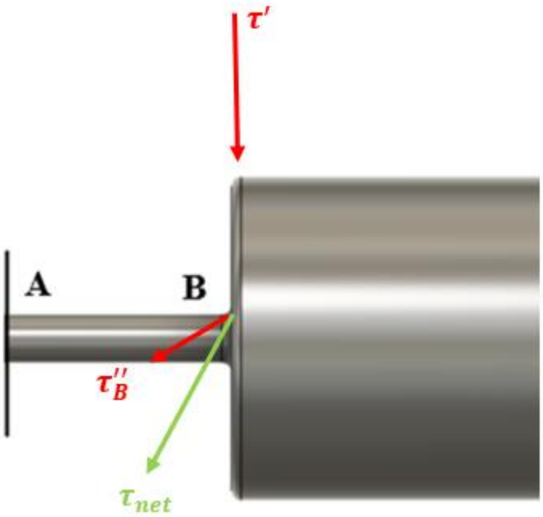


Figure 23. Primary and Secondary Stress

The force of 245 N will create a primary shear stress, and the moment Mb will create a secondary bending stress, that in this case will be critical. For the primary shear stress, it will be calculated with the following equation.

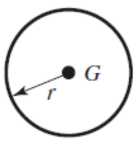
$$\tau' = \frac{P}{A_t}$$

Where τ', is the primary shear stress, P is the shear force, and A_t, is the cross-sectional area.

To continue the calculation, the following data is needed:

$$d = 168.3 \text{ mm}$$

And the table 9.2 of the book “Shigley's mechanical engineering design 9th edition”. (Shigley, 2011, p. 462).



$$A = 1.414\pi hr \qquad I_u = \pi r^3$$

Figure 24. Bending Properties Filet Welding

The cross-sectional area is:

$$A_t = 1.414\pi * h * r$$

Where, h is the leg of the filet, and r is the radius of the tube.

$$A_t = 1.414\pi * \left(\frac{168.3}{2}\right) * h$$

$$A_t = 118.988h \text{ [mm]}$$

Using table 9.2, the bending second moment of inertia is:

Bending second moment of inertia:

$$I_v = \pi r^3 = \pi * \left(\frac{168.3}{2}\right)^3$$

$$I_v = 1872027.541 \text{ [mm}^3\text{]}$$

The moment Mb is calculated as follows:

$$M_{b=c} = P * l$$

Where P is the force and l are the distance between the center of the roller to the fillet welding:

$$M_{b=c} = 245\ \mathrm{N} * 346\ \mathrm{mm}$$

$$M_{b=c} = 84770\ \mathrm{N} * \mathrm{mm}$$

For the primary shear stress and secondary bending stress the calculations are as follows:

$$\tau' = \frac{P}{A_t} = \frac{245\ [\mathrm{N}]}{118.998 * h\ [\mathrm{mm}]}$$

$$\tau' = \frac{2.0588}{h}$$

$$\tau_B'' = \frac{M * c}{I} = \frac{84770}{0.707h * I_v}$$

$$\tau'' = \frac{84770[\ \mathrm{N} * \mathrm{mm}] * \frac{168.3}{2}[\mathrm{mm}]}{0.707h * 1872027.541\ [\mathrm{mm}^3]}$$

$$\tau'' = \frac{7133395.5\ [\mathrm{N} * \mathrm{mm}^2]}{1323523.471\ [\mathrm{mm}^3]}$$

$$\tau'' = \frac{5.3897}{h}\left[\frac{\mathrm{N}}{\mathrm{mm}}\right]$$

Now, the net stress is going to be calculated with the value of the primary shear stress and secondary bending stress, as follows:

$$\tau_{net} = \sqrt{\tau'^2 + \tau_B''^2}$$

$$\tau_{net} = \sqrt{\left(\frac{2.0588}{h}\right)^2 \left[\frac{N}{mm}\right] + \left(\frac{5.3897}{h}\right)^2 \left[\frac{N}{mm}\right]}$$

For the leg of the fillet not to fail, the net stress should be less than the admissible stress of the electrode. For this, the table 9.3 of "Shigley's mechanical engineering design 9th edition" will be used.

Tabla 9-3

Propiedades mínimas del metal de aporte

Número de electrodo AWS*	Resistencia a la tensión, kpsi (MPa)	Resistencia a la fluencia, kpsi (MPa)	Elongación porcentual
E60xx	62 (427)	50 (345)	17-25
E70xx	70 (482)	57 (393)	22
E80xx	80 (551)	67 (462)	19
E90xx	90 (620)	77 (531)	14-17
E100xx	100 (689)	87 (600)	13-16
E120xx	120 (827)	107 (737)	14

* Sistema de numeración del código de especificaciones de la American Welding Society (AWS) para electrodos. En este sistema se usa como prefijo la letra E, en un sistema de numeración de cuatro o cinco dígitos en el cual los primeros dos o tres números designan la resistencia aproximada a la tensión. El último dígito incluye variables en la técnica de soldadura, como la fuente de corriente. El penúltimo dígito indica la posición de la soldadura, por ejemplo, plana, vertical o sobre la cabeza. El conjunto completo de especificaciones se puede obtener solicitándolo a la AWS.

Figure 25. Electrode Properties (Shigley, 2011)

` So, the ultimate tensile strength of electrode 60xx is 427 MPa, so the admissible stress is calculated as follows, with the AWS norm:

$$\tau_{ADM} = 0.3 * (427) = 128.1 \; [MPa]$$

The rest of the calculations is shown as follows:

$$(128.1)^2 \; [MPa] = \left(\frac{2.0588}{h}\right)^2 \left[\frac{N}{mm}\right] + \left(\frac{5.3897}{h}\right)^2 \left[\frac{N}{mm}\right]$$

$$16409.61 \left[\frac{N}{mm^2}\right] * h^2 = 4.2386 \left[\frac{N}{mm}\right] + 29.048 \left[\frac{N}{mm}\right]$$

$$16409.61 \left[\frac{N}{mm^2}\right] * h^2 = 33.287 \left[\frac{N}{mm}\right]$$

$$h^2 = 0.00202\ mm$$

$$h = 0.045\ mm$$

The size of the leg of the fillet is small, and the actual size of the fillet is of 1mm. So, the weld will not fail, but a safety factor will be calculated. First, the maximum stress will be calculated.

$$\tau' = \frac{P}{A_t} = \frac{245\ N}{118.988\ mm * 1\ [mm]} = 2.05\ MPa$$

$$\tau'' = \frac{5.3897}{1\ mm} \left[\frac{N}{mm}\right] = 5.3897\ MPa$$

The secondary bending moment will be the most critical on the assembly. The net stress is:

$$\tau_{net} = \sqrt{\left(\frac{2.0588}{1}\right)^2 \left[\frac{N}{mm^2}\right] + \left(\frac{5.3897}{1}\right)^2 \left[\frac{N}{mm^2}\right]}$$

$$\tau_{net} = 5.7695 \left[\frac{N}{mm}\right]$$

Using the Von Mises safety factor formula and Table 9.3 to find the Sy of electrode 60XX:

$$\tau_{ADM} = 0.577 * Sy_{electrode}$$

$$\tau_{ADM} = 0.577 * 346\ MPa = 199.642\ MPa$$

The safety factor is:

$$n = \frac{\tau_{ADM}}{\tau_{atm}} = \frac{199.642}{5.7595} = 34.6$$

The fillet weld will not fail and if compare with the calculations of the safety factor of the rollers shown in the following table:

Part	Safety Factor
Shaft	2.05
Roller	127.5
Fillet Welding	34.6

Table 5. Safety Factor of rollers, shaft, and fillet welding

BEARINGS SELECTION

Nowadays, angular contact ball bearings are one of the most used ball bearings worldwide. To do the selection of the bearings, we will use as reference the Mechanical Engineering Design Book by Shigley. The iterative process that was followed to select the ball bearings is as follows:

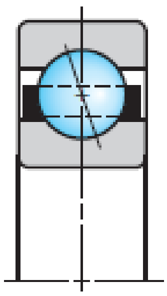


Figure 26. Bearing selection

1. Choose Y_2 from Table 11–1 (Shigley).
2. Find C_{10}.
3. Tentatively identify a suitable bearing from Table 11–2 (Shigley), note C_o.
4. Using F_a/C_o enter Table 11–1 (Shigley) to obtain a new value of Y_2.
5. Find C_{10}.

6. If the same bearing is obtained, stop.
7. If not, take next bearing and go to step 4.

So, the first step is to choose Y_2 from Table 11–1, we will take a first approximation by selecting the middle entry of the table with:

$$X_2 = 0.56 \qquad Y_2 = 1.63$$

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

Table 6. Selection of y2

Assuming a value of $V = 1$ and $a = 3$, and since we do not have any axial load in this case, we obtain the following:

$$F_e = XVF_r + YF_a = 0.56(1)(425.5) + 1.63(0) = 238.28\ N$$

$$C_{10} = a_f F_D \left[\frac{x_D}{x_o + (\theta - x_o)(1 - R_D)^{1/b}}\right]^{1/a}$$

As an unknown parameter we have x_D, which can be calculated as follows assuming a life goal of 20 kh, with a maximum rotational speed is 20000 rpm (Raimondo, 2020).

$$x_D = \frac{60\mathcal{L}_{10}n_D}{L_{10}} = \frac{60(10000)20000}{10^6} = 12000$$

$$C_{10} = 1.2(238.28)\left[\frac{12000}{0.02 + 4.439(1 - 0.99)^{\frac{1}{1.483}}}\right]^{\frac{1}{3}} = 10.86\ kN$$

From table 11-2 (Shigley) we can see that there is an angular contact bearing 02-20 mm has a $C_{10} = 13.3\ kN$ and a C_o is 6.55 kN.

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_S	d_H	Load Ratings, kN Deep Groove C_{10}	C_0	Angular Contact C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Table 7. Angular contact bearing

Proceeding with the step 4, we have:

$$\frac{F_a}{C_o} = \frac{0}{6.55} = 0$$

And so $F_a/C_o < 0.014$, we will use the value of 0.014 as mentioned in Shigley. With that condition, we get a value of $Y_2 = 2.30$. Calculating again F_e, the results are as follows:

$$F_e = XVF_r + YF_a = 0.56(1)(425.5) + 2.30(0) = 238.28\ N$$

With this, we see that the force is the same so the next steps will be omitted since we will obtain the same results. Therefore, the same bearing will be obtained in this iteration since we do not have any value in the axial load. A bearing of 02-20 mm will still be selected.

SPEED, ACCELERATION AND TORQUE CALCULATIONS

Speed

Using the capacitive sensor, it is possible to obtain the time it takes the roller to give one lap. Using this time, it is possible to obtain the speed of the car as follows, where R is the radius of the wheel, r the radius of the roller, P the circumference of the wheel and v_c the linear speed ate the contact point between the wheel and the roller.

$$\omega_{roller} = \frac{2\pi}{\Delta t} \left[\frac{rad}{s}\right]$$

$$v_c = \omega_{roller}(r) = \omega_{wheel}(R) \ \left[\frac{m}{s}\right]$$

Knowing that each revolution of the wheel, the car moves one circumference, it is possible to obtain the linear speed of the FSAE.

$$P_{wheel} = 2\pi R \ \left[\frac{m}{rev}\right]$$

$$v_{FSAE} \left[\frac{m}{s}\right] = P_{wheel} \ \left[\frac{m}{rev}\right] \cdot \omega_{wheel} \ \left[\frac{rad}{s}\right] \cdot \left[\frac{1\,rev}{2\pi\,rad}\right]$$

$$v_{FSAE} = \omega_{wheel}(R) = v_c$$

This expression can be used as there is no slip between the wheel and the roller.

Acceleration

Knowing the angular speed, it is possible to obtain the angular acceleration as follows; as it was demonstrated with speed calculations, the FSAE acceleration will be the same as the linear acceleration at the contact point.

$$\alpha_{roller} = \frac{\Delta\omega}{\Delta t}\left[\frac{rad}{s^2}\right]$$

$$a_{FSAE} = \alpha_{roller}(r)$$

Wheel Torque

As the first prototype does not have a Dyno, a theoretical wheel torque can be calculated using the inertia of the rollers, which is calculated below.

$$I_{shaft} = \frac{1}{2}mr^2$$

$$I_{shaft} = \frac{1}{2}(2.1933)(0.0127)^2$$

$$I_{shaft} = 1.77e - 4$$

$$I_{roller} = \frac{1}{2}M(OR^2 + IR^2)$$

$$I_{roller} = \frac{1}{2}(10.93)(0.08415^2 + 0.07704^2)$$

$$I_{total} = I_{shaft} + I_{roller}$$

$$I_{total} = 0.07113\ [Kg\ \cdot m^2]$$

Once the inertia is calculated, theoretical torque can be found as shown below.

$$T_{wheel} = I_{total}\alpha$$

ENGINEERING EXPERIMENT FOR VERIFICATION

Rollers Simulation

For the simulations, the software used was Fusion 360, for this the axes, the rollers, and the wheel in the center was modeled. It should be remembered that for these simulations, only the loads that were going to be on the rollers were simulated statically, in this case, it was only that of the wheel. Later, the dynamic simulations will be developed, but in this case, to check the resistance of the materials, it was simulated only statically. For the analyses, we worked with Von Mises.

The first of the images is shown below. This image shows the efforts to which the members of the dynamometer are subjected, in this case, the rollers as well as the axles. As can be seen, and as expected, the greatest stresses to which the axle is subjected is just in the contact it has with the wheel, this has a maximum value of 11.76 MPa, which is much lower than the yield stress of the steel, which also tells us that we are going to always work within the elastic range of the material.

The image results are shown below.

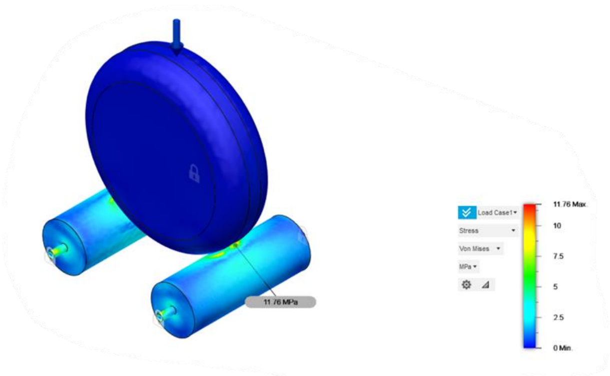


Figure 27. Stress Simulation

The following image shows the results of the displacements in meters in the materials of the dynamometer. As in the previous case, the largest displacements occur in the contact of the rollers with the wheel of the Formula SAE vehicle, the deformations are low, the largest having a value of 0.01894 meters.

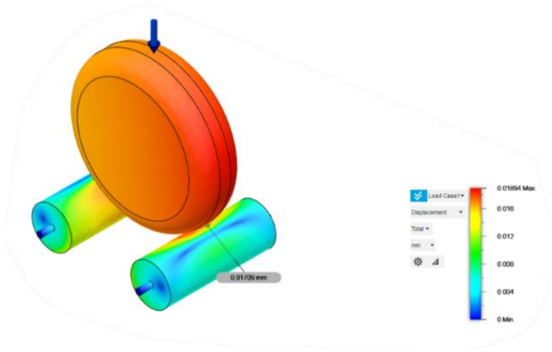


Figure 28. Displacement Simulation

Fusion is one of the most important software for simulations within engineering, in addition to the fact that there is no limit to the number of nodes with which you can work made us opt for this Software, within the simulations, we worked with a total of 122,056 nodes and a total of 76,917 elements.

⊟ **Mesh**

Type	Nodes	Elements
Solids	122056	76917

Figure 29. Fusion simulation analysis

From the analysis, a summary of the results is shown below:

Maximum Stress	11.76 MPa
Maximum Displacement	0.019mm

Table 8. Simulation Summary

In conclusion, we determine that the rollers will not yield, and the deformations will not be significant for further calculations. Furthermore, we conclude that que critical point will be on the join between the shaft and roller. So, in the next section we will analyze the normal and shear stress in the joint and determine whether they will fail or not.

Natural Frequency Simulation of the roller and shaft

The objective of this simulation is to determine the natural frequency of the roller and shaft as a unique body, and to identify if the frequency that the rollers move will ever be the same as their natural frequency, creating resonance that will affect the data.

For the simulation, Fusion 360 program modal frequency was used. The two ends of the shaft were put a fixed constrain and a 6000 RPM motion on the center of the roller to simulate a high-speed test. The results are the following:

Figure 30. Mode 1 Natural Frequency Simulation

The table of values are the following:

Frecuencia	Participación X	Participación Y	Participación Z
Modo 1: 237.1 Hz	0	0.0001	0
Modo 2: 314.2 Hz	0.0007	7.5677	73.8982975
Modo 3: 314.4 Hz	0	73.9104986	7.55589977
Modo 4: 485.9 Hz	70.2682972	0	0.0005
Modo 5: 665.4 Hz	0.0007	0.979899988	0.599100022
Modo 6: 665.9 Hz	0.0003	0.598199992	0.998700038
Modo 7: 982.8 Hz	0	5.01030013	7.31929988
Modo 8: 983 Hz	0	7.32510015	5.01849987

Table 9. Natural frequency data simulation

The primary mode has a value of frequency of 237.1 Hz. The dynamometer will move at maximum 6000 RPM, that is 100 Hz, that is below the primary mode of the assembly of the shaft and roller, so resonance will be no resonance.

Rollers Friction coefficient calculation

To determine the friction coefficient of the rollers, an experiment was made. The experiment consists of putting a rubber object that simulates the wheel of the FSAE, on the surface of the roller, and the rotate it a certain angle until it falls. For the calculations it is needed the weight of the rubber and the angle at which the object falls.

$$Rubber\ object\ weight: 21\ g$$

$$N = w * 9.8 = 0.021kg * 9.8 \left[\frac{m}{s^2}\right] = 0.2058\ N$$

To determine the angle, the photo of the initial position and the photo of the last position before slipping were used and overlay. As shows in the following figure:

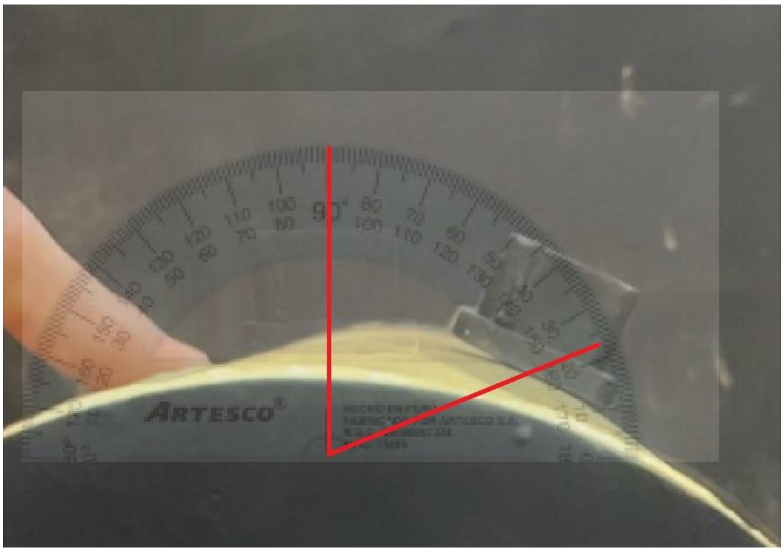


Figure 31. Angle of slipping Rollers

The angle that the piece slip is approximately 55°. With this, the calculation can be done as follows.

$$F_r = N * \mu$$

Then equal the x component of the weight of the rubber object because this is the force that must surpass to start the motion.

$$W_x = mg * \cos\theta = 0.021\ \mathrm{kg} * 9.8 \left[\frac{m}{s^2}\right] \cos 55 = 0.118\ \mathrm{N}$$

$$0.118\ \mathrm{N} = 0.2058\ \mathrm{N} * \mu$$

$$\pi = \ 0.58$$

Jones and Childers reports friction coefficients of about 0.7 for dry roadway and 0.4 for wet roadway (Novikov, 2018). So, the protype has a good friction coefficient, but to increase the friction of the rollers to simulate better the street surface, a friction tape was put to avoid glide and to have a better grip.

TEST AND RESULTS

Our test aims to demonstrate the correct functionality of the Chassis dynamometer prototype. As a chassis dynamometer tries to simulate street conditions, the test will be done by a motorcycle on the street.

The test also aims to find the time it takes for the data to stabilize, so it is possible to measure the data correctly with the dynamometer.

Speed Test

The speed test consists of comparing the values of the speed data of the dynamometer and the data acquired by the motorcycle in street conditions. The conditions of the two tests will be the same: 3000 RPM, 4000 RPM, and 5000 RPM in the first gear, because the motorcycle with the engine on has a normal of 2000 RPM.

For the speed of the motorcycle, it will be done four tests on the same RPM. Then, an average velocity value will be calculated and collected. Then, the motorcycle will be put over the rollers of the dynamometer, and it will begin to run on first gear at a RPM until the graph stabilizes. Then, the time it took the graph to stabilize will be determined, and the speed value of the dynamometer at this time will be collected. Both values of speed will be compared to obtain a relative error value to determine if the dynamometer is working correctly or not. The same test will be done for the 4000 RPM and 5000 RPM.

First, the personal protection equipment was used by the test subject.

Figure 32. Personal Protection Equipment

Then, the test began at a 3000 RPM:

Figure 33. Speed of the motorcycle at 3000 RPM

# Test	Speed Value [km/h]
1.	11
2.	12
3.	12
4.	13
Average	**12**

Table 10. Speed values 3000 RPM

Then, at 4000 RPM:

Figure 34. Speed of the motorcycle at 4000 RPM

# Test	Speed Value [km/h]
1.	15
2.	16
3.	17
4.	15
Average	**15.75**

Table 11. Speed values 4000 RPM

And finally at 5000 RPM:

Figure 35. Speed of motorcycle at 5000 RPM

# Test	Speed Value [km/h]

1.	20
2.	22
3.	21
4.	20
Average	**20.75**

Table 12. Speed values 5000 RPM

To perceive better the speed of the motorcycle at the different RPM, a table with the results is shown below:

RPM	Speed [Km/h]
3000	12
4000	16
5000	20

Table 13. RPM and Speed values

Then, the same test was done on the dynamometer:

Figure 36. Dynamometer normal functionality

The graph of the speed values at 300 RPM of the dynamometer is shown below:

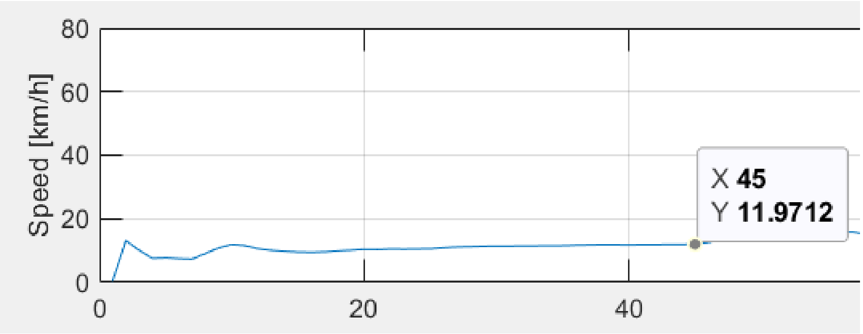


Figure 37. Speed value Dynamometer 3000 RPM

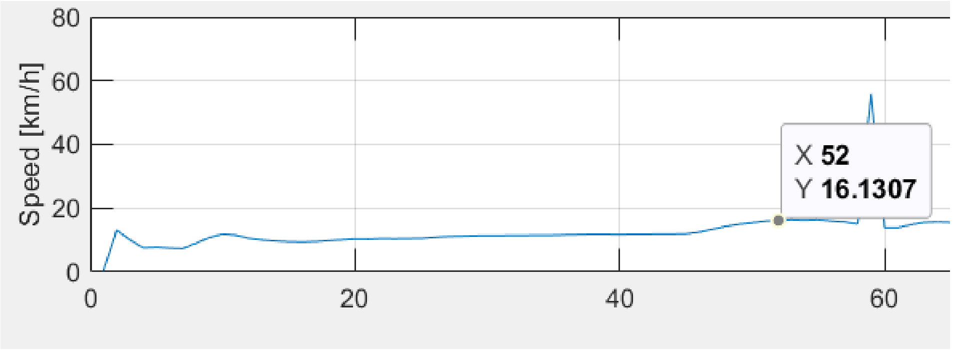


Figure 38. Speed Value Dynamometer 4000 RPM

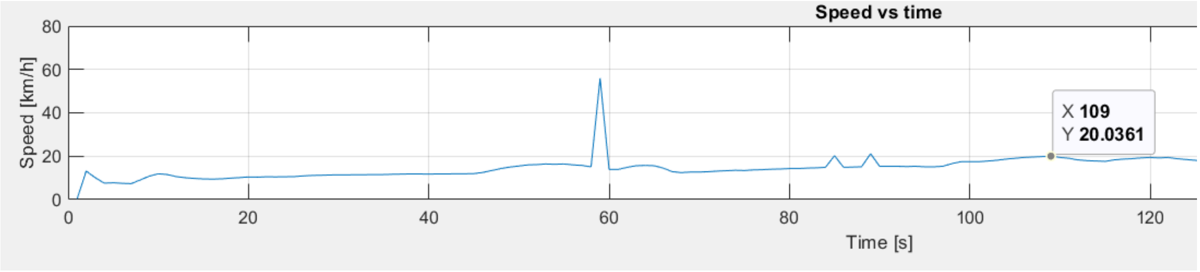


Figure 39. Speed Value Dynamometer 5000 RPM

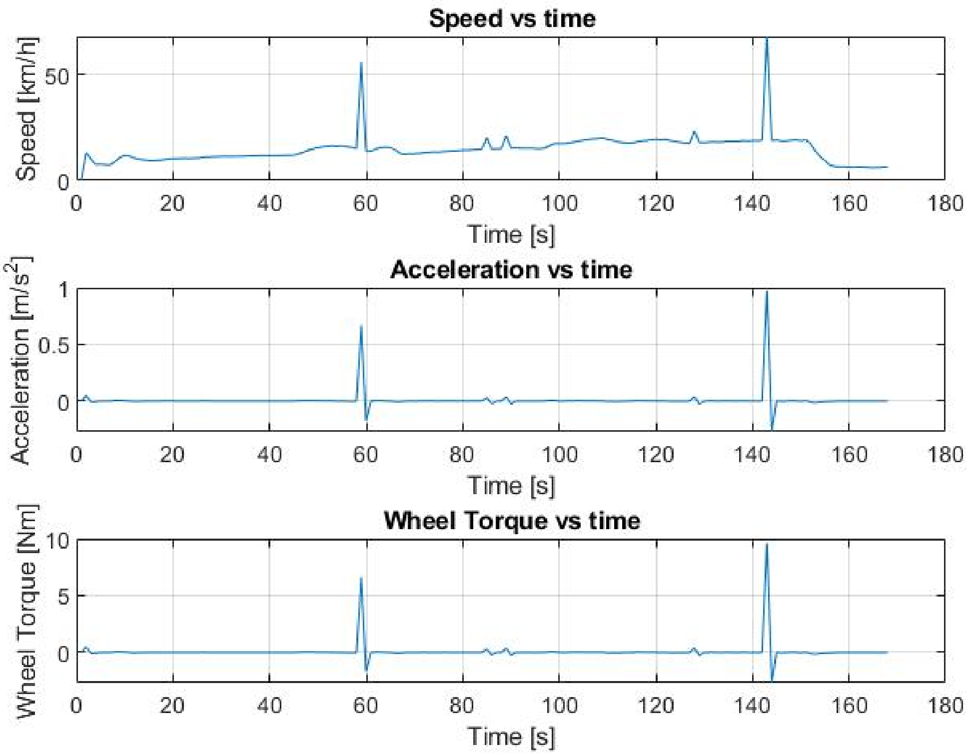


Figure 40. Speed, Acceleration, and wheel torque graphs

The value of time of the graph to stay constant was approximated between 20 to 30 seconds after changing the RPM, so the time that must pass to determine that the graph will stay constant is 25 seconds.

SAFETY THROUGH DESIGN

This section aims to identify and evaluate the possible risks that we may encounter during the fabrication of the chassis dynamometer. For this, it was followed four main steps: identification, evaluation, action, and monitoring. Identification of the risks will cover the examination of the requirements, objective, scope, specifications, schedule, and budget. For a better approach, brainstorming was made among our colleagues with the guidance of a manufacturing expert. In the evaluation, it was detected 6 types of risks: Technical, operational, economical, commercials, resource management, and security.

- **Technical risk:** These risks are related to the inappropriate evaluation of technical machinery.
- **Operation Risk:** These are the risks caused by failed processes, systems, policies, and operations malfunctions.
- **Economic Risk:** These are the risks caused by a lack of economic resources (budget), that may lead to the acquisition of inappropriate resources.
- **Commercial Risk:** These are the risks that encompass the changes in input prices, variations in demand, and access to raw resources, especially in the national market.
- **Resource Management Risk:** These are the risks caused by the misuse of resources while crafting and assembling the dynamometer.
- **Safety Risk:** These are the risks related to the physical health of the workers.

For a better understanding, it was chosen a different color for each type of risk. As shown in the following figure.

	Risk	Color
1	Technical	
2	Operation	
3	Economic	
4	Commercial	
5	Resource Management	
6	Security	

Table 14. Risk definition

To analyze the relevance of each risk, it was made a monitoring table. This table compares the probability and impact of each risk on a 1 to 5 scale. Where 5 corresponds to the highest probability of impact. The risks were distributed as shown in the following figure.

Impact	**5**		Burn during the welding process	Equipment calibration error		
	4		Slip between wheel and roller		Lack of budget for optimal equipment	
	3	Entrapment when lifting the vehicle	Vehicle leveling error	Lack of equipment in the national market	Geometric Tolerances Error	
	2		Workshop accidents	Inhalation of toxic gases in the welding process		
	1	Errors in the calculation of material to use		waste of resources		
		1	**2**	**3**	**4**	**5**
		Probability				

Table 15. Impact VS Probability table

Once the risk register table was made, it was able to create a risk register table. This table is used to identify the priority of each risk and to delegate a responsible member to each one of them. Also, a decision to reduce or eliminate the risk was made and explained in the table. To end, a status of each risk was placed, where "Pending" means that the risks have not yet been presented so far, and "Active" means that the risks are been presented right now. This table is shown in the following table.

Code	Description	Priority =Impact x Probability	Responsable	Decision	Status	Comments
001	Geometric Tolerances Error	3x4=12	Juan José Plaza	Share. Cross-check between team members	Pending	Get equipment to measure tolerances
002	Equipment calibration error	5x3=15	Juan José Plaza	Reduce. Do calibration tests	Pending	Consult with a professional
003	Slip between wheel and roller	4x2=8	Juan José Plaza	Avoid. Use slippery materials	Pending	Research available options
004	Vehicle leveling error	3x2=6	Mateo Montenegro		Pending	Get the leveling equipment
005	Lack of budget for optimal equipment	4x4=16	Martín Cardenas	Get prepared. Reserve funds	Active	Make multiple market investigation
006	Lack of equipment in the national market	3x3=9	Martín Cardenas	Get prepared. Look for local alternatives	Active	
007	Errors in the calculation of material to use	1x1=1	Mateo Montenegro	Get prepared. Training of team members	Active	
008	waste of resources	1x3=3	Martín Cardenas	Reduce. Optimize the use of resources	Active	Find places where the wasted material can be used
009	Workshop accidents	2x2=4	Luis Felipe Sánchez	Get prepared. Get trained in first aid	Active	Risks evaluation
010	Entrapment when lifting the vehicle	3x1=3	Luis Felipe Sánchez	Avoid. Restrict access in danger zones	Pending	
011	Burn during the welding process	5x2=10	Luis Felipe Sánchez	Avoid. Use of proper equipment	Pending	Check the status of the equipment
012	Inhalation of toxic gases in the welding process	2x3=6	Mateo Montenegro	Avoid. Use of proper equipment	Pending	Ventilate the work area

Table 16. Responsible Risk description

MAINTENANCE AND OPERATING MANUAL

General Description

The FSAE Chassis Dynamometer is designed to measure speed, acceleration, and theoretical wheel torque. The following figure shows device.

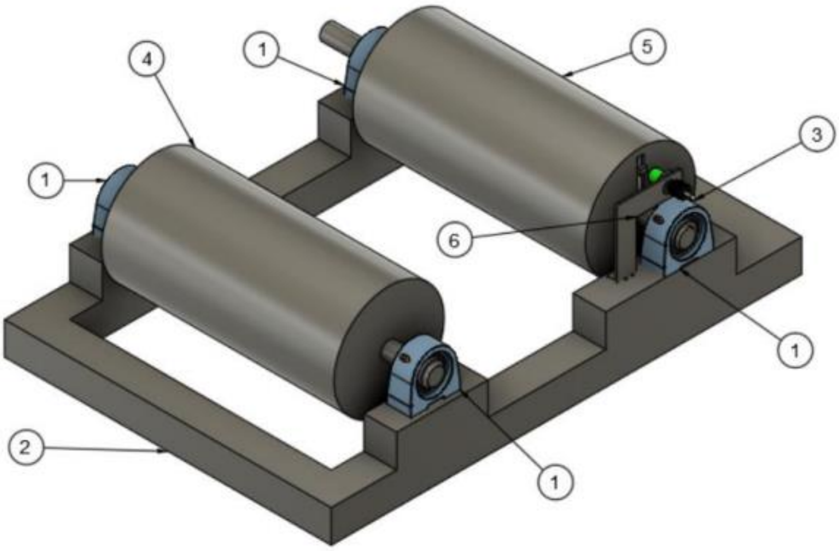


Figure 41. Parts of Chassis Dynamometer

Safety

This symbol shows a safety indication and must not be ignored. When you see this symbol, be alert to the possibility of personal injury and read the indications carefully. Here are some general indications for the operation of the FSAE Chassis Dynamometer.

The FSAE Chassis Dynamometer contains mobile parts which can cause accidents. When functioning, keep away from the moving parts.

- Use adequate personal protection equipment such as googles, gloves, ear protection and adequate clothing for the activity.
- Keep the area clear during the whole operation of the FSAE Chassis Dynamometer.
- The FSAE Chassis Dynamometer and vehicle alignment must be inspected prior each use. In case there is any problem, the operation must be stopped until it is fixed.

- Parts may be hot after use, be careful when manipulating the equipment after a test has been done.
- Wait until the FSAE Chassis Dynamometer has completely stopped to manipulate any part of the equipment.

Installation and Operation

Space requirements.

The FSAE Chassis Dynamometer must be located on a flat horizontal surface. The minimum space depends on the vehicle to test. It is recommended to leave enough space to walk and keep a safe distance from the vehicle.

System Requirements and Set up.

To run the program, a computer with MatLab is needed.

If the computer has not been used with the FSAE Chassis Dynamometer before, follow these steps.

1. Turn on the computer and open MatLab.
2. Once MatLab is open, connect the FSAE Chassis Dynamometer via USB Cable.
3. On the Command Window must appear a message asking to install an Add-on for Arduino.
4. Install the Add-on for MatLab.

To Set up the FSAE Chassis Dynamometer follow these steps.

1. Open MatLab
2. Connect the Arduino UNO via USB.
3. In line 5 of the code, type the serial port that is been used by the Arduino (To check the serial port used by Arduino UNO refer to Section 3.3.)

```
s=serialport('COM9', 9600);
```

4. If necessary, update the wheel diameter in the variable "wheel"

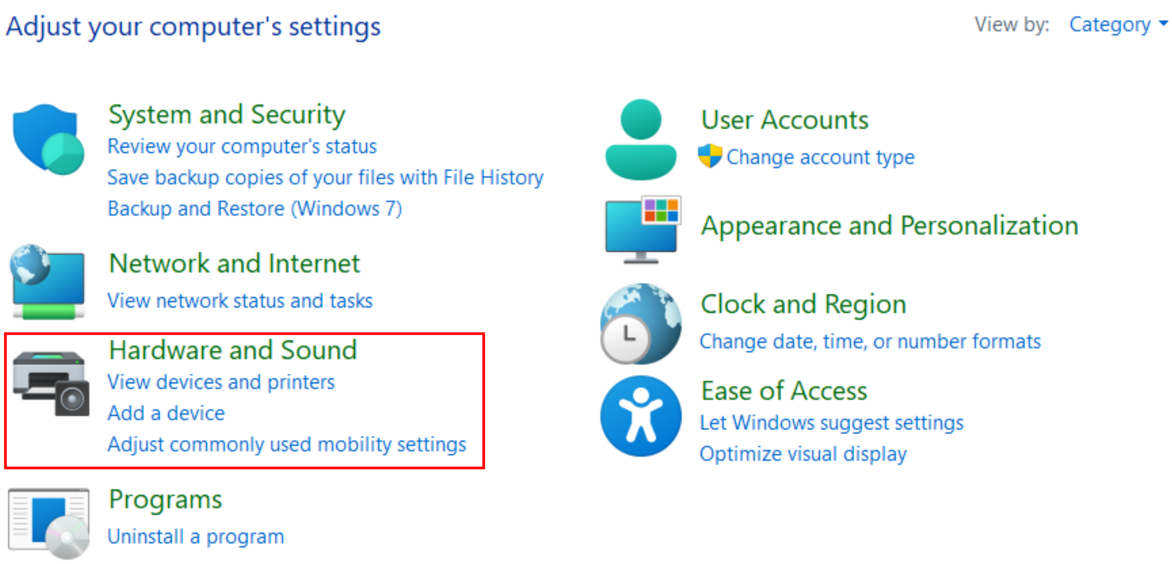
```
R=0.1683; %meters
wheel=0.55; %meters
```

After these steps, the FSAE Chassis Dynamometer is ready to be used.

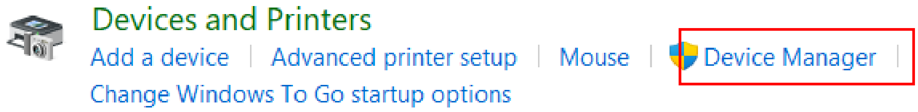
Checking the Serial Port used.

If you don't know which Serial Port is been used by the Arduino UNO Processor follow these steps.

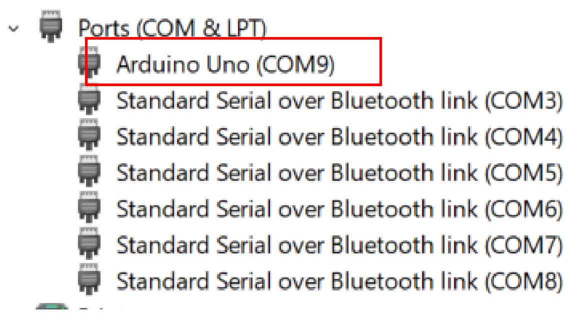
1. Connect the Arduino UNO to the computer.
2. Open de Control Panel and go to hardware and sound option.



3. Open the Device Manager

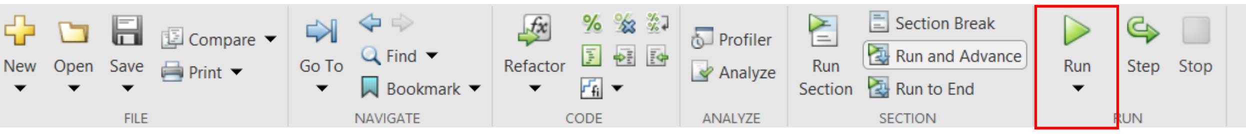


4. Open the option "Ports" and find the Arduino UNO device. Between parenthesis you will find the Serial Port used by the device.

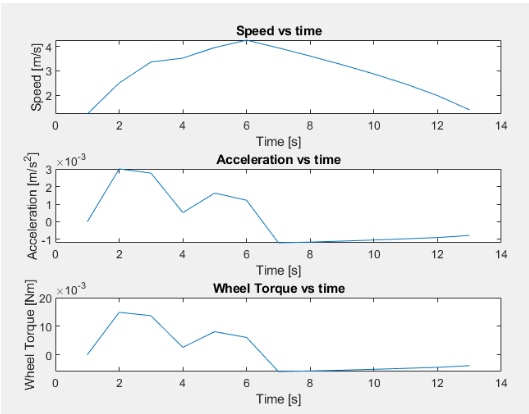


Running a Test.

1. Place the FSAE Chassis Dynamometer in the place where the test is going to be conducted.
2. Mount the vehicle with the help of a jack.
3. Balance the vehicle with other jacks and align the vehicle with the FSAE Chassis Dynamometer. Make sure that the vehicle is stable before starting the test.
4. Turn on the computer that is going to be used for the measurements.
5. Do the initial set up (Section 3.2.)
6. Run the program.



7. Start the vehicle and verify alignment (Vehicle should not move to the sides of the rollers)
8. MatLab will automatically display real time graphs.



- When running a test, keep away from the FSAE Chassis Dynamometer, as the moving parts can cause an accident.

Troubleshooting

This section contains the most common problems that can appear and possible solutions.

Problem	Solution
`Warning: The specified amount of data was not returned within the Timeout period for 'readline'. 'serialport' unable to read any data. For more information on possible reasons, see serialport Read Warnings.`	The program is running but is not receiving information from the sensor. Make sure that the test is being conducted and the sensor is detecting the roller.
`Error using serialport (line 116) Unable to connect to the serialport device at port 'COM9'. Verify that a device is connected to the port, the port is not in use, and all serialport input arguments and parameter values are supported by the device. See related documentation for troubleshooting steps.`  `Error in Data_Logger (line 5) s=serialport('COM9', 9600);`	1. Make sure the Arduino UNO module is connected to the computer. If it is, try reconnecting it. 2. Make sure that the Serial Port is not being used by another program (for example the Arduino IDE). If this is the case, close any other program and restart MatLab. 3. Check that the Serial port selected is the correct one (Section 3.3)

Maintenance

This section contains the most important parts that will need to be considered when the chassis dynamometer is in use.

Bearings.

To extend the useful life of a bearing, the following steps must be followed.

1. Assembly and Lubrication

To ensure that a bearing optimizes its work and can initially reach its expected useful life cycle, this element must be carefully assembled, in order to avoid any type of damage to the bearing.

2. Bearing Alignment

The passage of time causes the deterioration of all equipment and bearings are no exception. You have to verify the correct alignment of the bearings, in order to be sure that these components are well aligned, it is recommended to take them to a specialized workshop.

3. Continuous lubrication

It is important to keep track of the bearing while it is running. We have to carry out a series of relubrications so that its work continues to be carried out at full capacity.

4. Monitoring of Basic Conditions

Knowing that the function of the bearings is to serve as support elements for the shaft, and due to the action of their periodic state of operation, it is advisable to inspect the state of these elements at least once a month.

Steel Base.

Since the steel base is not a critical element, its maintenance does not have to be done as often. The most common thing to do is paint the base in case there are sections that lose paint. The paint that was used can be purchased at any local store through the following link:

https://kywitiendaenlinea.com/product/pint-spray-amarilla-no-53-abro/

To know when to maintain the base, this must be done when through a visual way imperfections or areas are seen, especially in the welds, maintenance should be given. For this, all the bolts and nuts must be removed, in addition to removing the sensor support, as well as the rollers.

Capacitive Sensor.

The steps that must be followed to maintain the capacitive sensor are the following:

- ✓ Check the connections that the capacitive sensor has.
- ✓ Remove the sensor from the bracket.
- ✓ Verify that the sensitivity of the sensor.
- ✓ If this has changed, adjust it with the help of a screwdriver.
- ✓ In case it is damaged, it is best to buy another because its cost is low.
- ✓ Reinstall the sensor in its support.

In Ecuador, the capacitive sensor can be purchased in a simple way through the following link:

https://articulo.mercadolibre.com.ec/MEC-517719638-mgsystem-sensor-capacitivo-npn-ljc18a3-b-zb-6-36v-arduino-_JM#position=6&search_layout=stack&type=item&tracking_id=9a42e3b8-58ed-45fe-8ac6-3bb5a39b92b3

Capacitive Sensor

Rollers & Shafts

In the same way, the rollers and shafts must be maintained. For this, the best way to provide maintenance is through visual inspection, in the event that it is seen that these components are not giving the same performance or have deformations (very difficult to be the case), they will have to be give immediate maintenance.

The best way to provide maintenance is through visual inspection, in each use of the equipment it should be seen that the components comply with their function, in addition to not seeing any damage to the structure.

In the case of shaft alignment, in general, rigid and flexible couplings can be used. Although flexible couplings are preferable because they compensate for some of the misalignment, there are generally accepted standards for shaft misalignment with various types of couplings.

To verify the alignment of a shaft, and because its procedure is quite complex, and to avoid possible confusion when describing how this maintenance should be carried out in words. Because it is considered the best way to understand these concepts visually, a tutorial is attached in which you can follow a step-by-step to correctly align shafts. The link is as follows:

https://www.youtube.com/watch?v=IhsRmm7uTE4

Alignment of a Shaft - Tutorial

COST ANALYSIS

The budget that was given by the client was 400 dollars. After all the cats, the total expense was 407.69 dollars.

Description	Quantity	Cost ($)
Square steel profile	1	26.62
Floor pillow x 2	2	10.10
Floor pillow x 2	2	10.10
Set rollers	2	300.00
Electronic Kit	Various	43.50
Bolts	8	3.52
Square steel profile 2	1	13.85
Use of USFQ equipment	Various	0.00
Total		407.69

Table 17. Budget Summary

From the table above it can be seen that there is a detailed analysis of all the expenses incurred. There are 7 expenses. Of these, the set of rollers were the pieces that had a higher cost due to the precision that was needed, in addition to the fact that for this it was prioritized to have a good weld made by professionals, to have an adequate alignment and balance.

There was a lower cost with the purchase of the bolts, this because not so many of these pieces were needed. In the same way, the use of the USFQ equipment was free of charge, since all the equipment that was used is inside the materials workshop belonging to mechanical engineering.

It should be noted that it was sought to minimize the cost of all components, for this, from the beginning of the construction of the dynamometer it was possible to have cheap components that ensured proper functioning.

In the same way, deterministic methods were applied which helped us to see how the dynamometer could be mass-produced. The deterministic methods that were applied were:

In the same way, deterministic methods were applied which helped us to see how the dynamometer could be mass-produced. The deterministic methods that were applied were:

1. Lot to Lot (LxL): The batch-to-batch technique is the simplest of all, it consists of placing orders or production runs equal to the net needs of each period, thus minimizing inventory maintenance costs.

2. Constant Period Technique: This method sets an interval between requests arbitrarily. This allows the economic quantity to order and produce to be adjusted for each order.
3. Economic Order Quantity (EOQ): It is a method that, taking into account the deterministic demand for a product, the cost of holding the inventory, and the cost of requesting an order, produces as output the optimal number of units to order to minimize product maintenance costs.
4. Periodic Order Quantity (POQ): This method is a combination of the EOQ and the constant period method.
5. Minimum Total Cost: This method is based on the Economic Order Quantity foundation, in which the more similar maintenance costs and setup costs are, the closer one will be to determining the optimal order quantity.

The results that were obtained for these methods where it was assumed that the cost of storage was 0.00 $ because it was assumed that the university was not going to charge any extra cost, the cost of ordering was 450 dollars since it was considered the difference in the value of materials ($407.69) and labor. The results are the following.

Model	Cost of holding	Cost to order	Total Cost
POQ	$ -	$ 1,800.00	$ 1,800.00
Constante Period	$ -	$ 1,350.00	$ 1,350.00
BPF	$ -	$ 1,350.00	$ 1,350.00
EOQ	$ -	$ 1,800.00	$ 1,800.00
L x L	$ -	$ 3,600.00	$ 3,600.00

Table 18. Deterministic Methods

Material Planning: Lot to Lot											
Item	Available Inventory	Concepts	Time period (weeks)								
			1	2	3	4	5	6	7	8	Total
Dynamometer	0	Gross requirements	1	1	1	1	1	1	1	1	
		Available Inventory	0	0	0	0	0	0	0	0	
		Net requirements	1	1	1	1	1	1	1	1	
		Order reception	1	1	1	1	1	1	1	1	
		Release order	1	1	1	1	1	1	1	1	
		Cost of holding	$ -	$ -	$ -	$ -	$ -	$ -	$ -	$ -	$ -
		Preparation cost	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	$ 3,600
		Total cost	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	$ 450	**$ 3,600**

Table 19. Lot to Lot

Material Planning: POQ											
Item	Available Inventory	Concepts	Time period (weeks)								
			1	2	3	4	5	6	7	8	Total
Espejo Tipo B	0	Gross requirements	1	1	1	1	1	1	1	1	
		Available Inventory	0	1	0	1	0	1	0	1	
		Net requirements	1	0	1	0	1	0	1	0	
		Order reception	2		2		2		2		
		Release order	2		2		2		2		
		Cost of holding	$ -	$ 0	$ -	$ 0	$ -	$ 0	$ -	$ 0	$ 0
		Preparation cost	$ 450	$ -	$ 450	$ -	$ 450	$ -	$ 450	$ -	$ 1,800
		Total cost	$ 450	$ 0	$ 450	$ 0	$ 450	$ 0	$ 450	$ 0	**$ 1,800**

Table 20. POQ

Material Planning: Constant Period											
Item	Available Inventory	Concepts	Time period (weeks)								
			1	2	3	4	5	6	7	8	Total
Dynamometer	0	Gross requirements	1	1	1	1	1	1	1	1	
		Available Inventory	0	2	1	0	2	1	0	1	
		Net requirements	1			1			1		
		Order reception	3			3			2		
		Release order	3			3			2		
		Cost of holding	$ -	$ -	$ -		$ -	$ -		$ -	$ -
		Preparation cost	$ 450	$ -	$ -	$ 450	$ -	$ -	$ 450	$ -	$ 1,350
		Total cost	$ 450	$ -	$ -	$ 450	$ -	$ -	$ 450	$ -	$ 1,350.00

Table 21. Constant Period

Material Planning: EOQ											
Item	Available Inventory	Concepts	Time period (weeks)								
			1	2	3	4	5	6	7	8	Total
Dynamometer	0	Gross requirements	1	1	1	1	1	1	1	1	
		Available Inventory	0	439	438	877	876	1315	1314	1753	
		Net requirements	1	0	-437	0	-875	0	-1313	0	
		Order reception	440		440		440		440		
		Release order	440		440		440		440		
		Cost of holding	$ -	$ 0	$ 0	$ 0	$ 0	$ 0	$ 0	$ 0	$ 0
		Preparation cost	$ 450	$ -	$ 450	$ -	$ 450	$ -	$ 450	$ -	$ 1,800
		Total cost	$ 450	$ 0	$ 450	$ 0	$ 450	$ 0	$ 450	$ 0	$ 1,800

Table 22. EOQ

Material Planning: BPF											
Item	Available Inventory	Concepts	Time period (weeks)								
			1	2	3	4	5	6	7	8	Total
Dynamometer	0	Gross requirements	1	1	1	1	1	1	1	1	
		Available Inventory	0	545	544	543	545	544	543	544	
		Net requirements	1	0	-543	-542	0	0	-542	0	
		Order reception	546			3			2		
		Release order	546			3			2		
		Cost of holding	$ -	$ -	$ -	$ -	$ -	$ -	$ -	$ -	$ -
		Preparation cost	$ 450	$ -	$ -	$ 450	$ -	$ -	$ 450		$ 1,350
		Total cost	$ 450	$ -	$ -	$ 450	$ -	$ -	$ 450	$ -	$ 1,350.00

Table 23. BPF

RESULTS, ANALYSIS AND DISCUSSION

Results

The most important results obtained are the following.

Static Calculations

- Safety Factor=2.09 à For one force
- Safety Factor=3.54 à For two forces

Fatigue

- Se=145.06 MPa
- d=14.28mm

Welding Calculations

- h=0.045 mm
- Safety Factor = 34.6

Bearings

- A bearing of 02-20 mm was selected.

Inertia of the rollers

- $I_{total} = 0.071 \text{ kgm}^2$

Simulations

- Maximum stress: 11.76 MPa
- Maximum Displacement: 0.019 mm

Roller's friction Coefficient

- $\mu = 0.58$

The results of the graphs that were obtained once the chassis dynamometer was tested for different revolutions per minute (rpm), are the following:

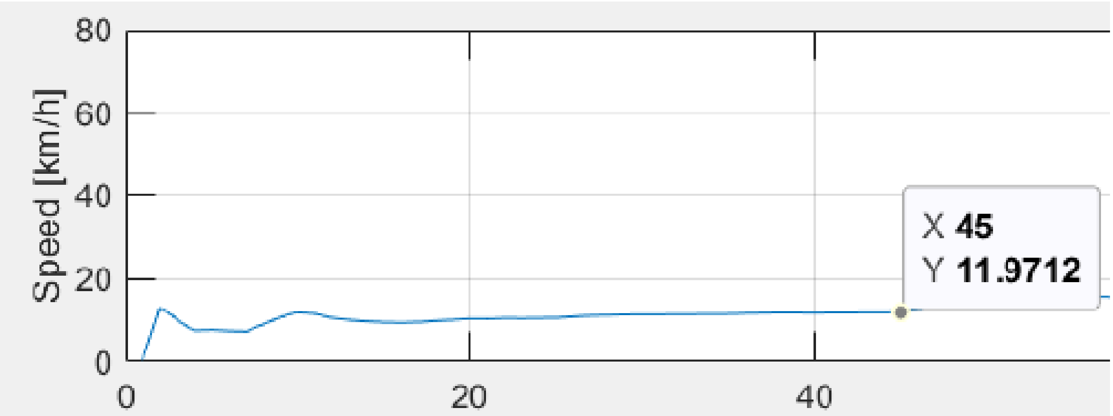


Figure 42. Speed value Dynamometer 3000 RPM

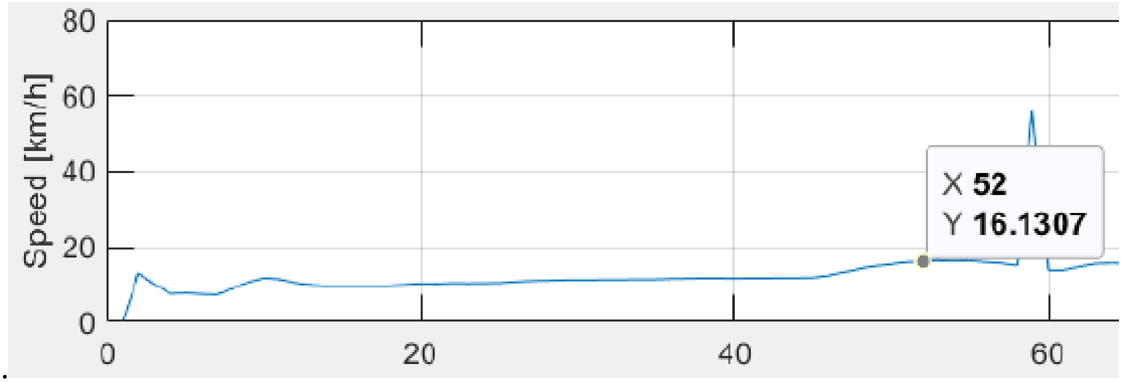


Figure 43. Speed Value Dynamometer 4000 RPM

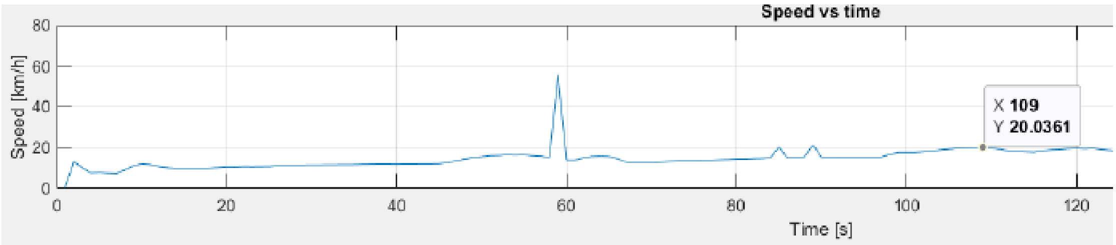


Figure 44. Speed Value Dynamometer 5000 RPM

Analysis and Discussion

The main results were presented in this section, it can be seen briefly that all the calculated safety factors are greater than 1, which indicates that our components will not fail, in the same way there are several calculated diameters for both for fatigue as well as for the static case, these diameters are the ones that in theory should be used for the construction of the parts of the dynamometer.

In the same way, it is possible to select the most suitable bearing so that it can work under the load conditions imposed for the problem through an analysis of bearings. The simulations have helped us a lot, with this you can see that the maximum effort that is made is not that significant, and the maximum displacement is not great at all, it is only 0.019 mm.

Compared with street conditions, the friction coefficient of the rollers is very similar to real conditions, which means that the results obtained with the calculations, as well as when the dynamometer is tested, are very similar to the real ones.

In the part of graphs that there are several results, specifically there are results at 3000, 4000 and 5000 rpm. From the graphs it can be seen how after there is a period of acceleration, the graph proceeds to stabilize, for different times it can be clearly evidenced, when it has more revolutions, the speed, in kilometers per hour, is greater.

Within the tests that were carried out with the chassis dynamometer, they could be compared. It was seen how the results obtained, with the real data are very similar.

FUTURE WORK

- In the future, a dynamometer cover can be implemented, that is, the structure can be strengthened so that, if this is the case, it does not fail.
- Due to the great weight of the vehicles with which one works, and to make less effort when raising the vehicle to the dynamometer, a ramp can be implemented.
- Currently, the use of a computer is a requirement so that the graphs can be visualized. In the future, the implementation of a display is recommended to better be able to obtain the data obtained.
- As mentioned at the beginning, the manual brake subsystem was not implemented, this is a good recommendation to be implemented in the future to prevent the wheel of the vehicle from skidding with the roller.

CONCLUSIONS

- The prototype was correctly designed to satisfy the needs of the client and for its use. It was verified that the customer was satisfied with the product, in addition to verifying the proper functioning of this dynamometer, comparing the readings obtained with the actual measurements.
- The best way to build the prototype was analyzed, for this the most economical way to build the prototype was seen, in addition to the fact that all the operation was verified with real data.
- A chassis dynamometer was built which serves as a basis for future improvements. That is to say, the design is not limited only to the current operation that it has, in addition to this, in the future other accessories can be implemented or improvements to the structure can be made to be able to make other measurements or give more stability to the entire structure.
- It was possible to build a functional prototype with a reduced budget, for this, the way to obtain the cheapest possible parts that, in addition to this, give a good performance was seen.
- The importance of a good analysis from the beginning was understood, since doing everything from the beginning, checking all the calculations as well as all the simulations, gave as a result that the error obtained in the dynamometer measurements was very small.
- It is important to emphasize that in addition to this, the importance of the selection of materials was understood, since within the national market (Ecuador) the supply of materials such as steel profiles, or bearings is very limited, so it had to be adjusted to local materials so that in this way the dynamometer can be built.

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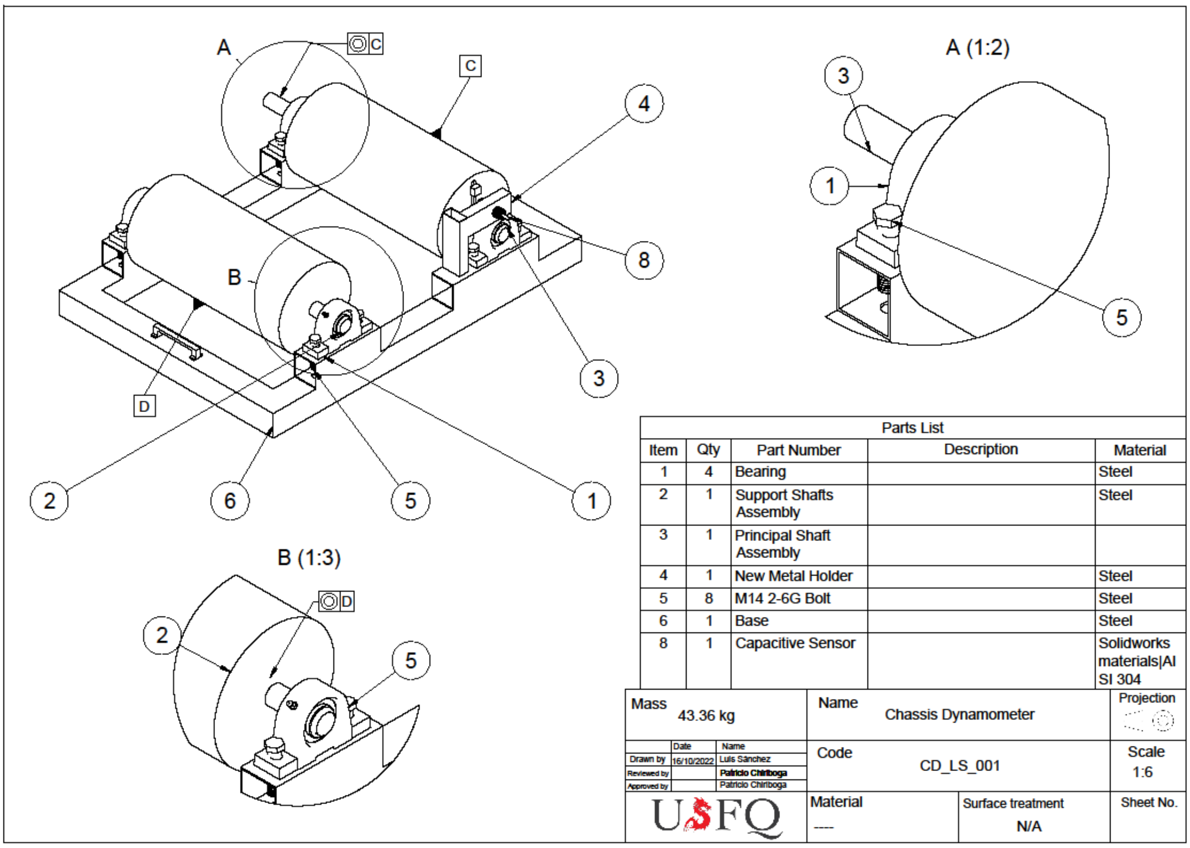
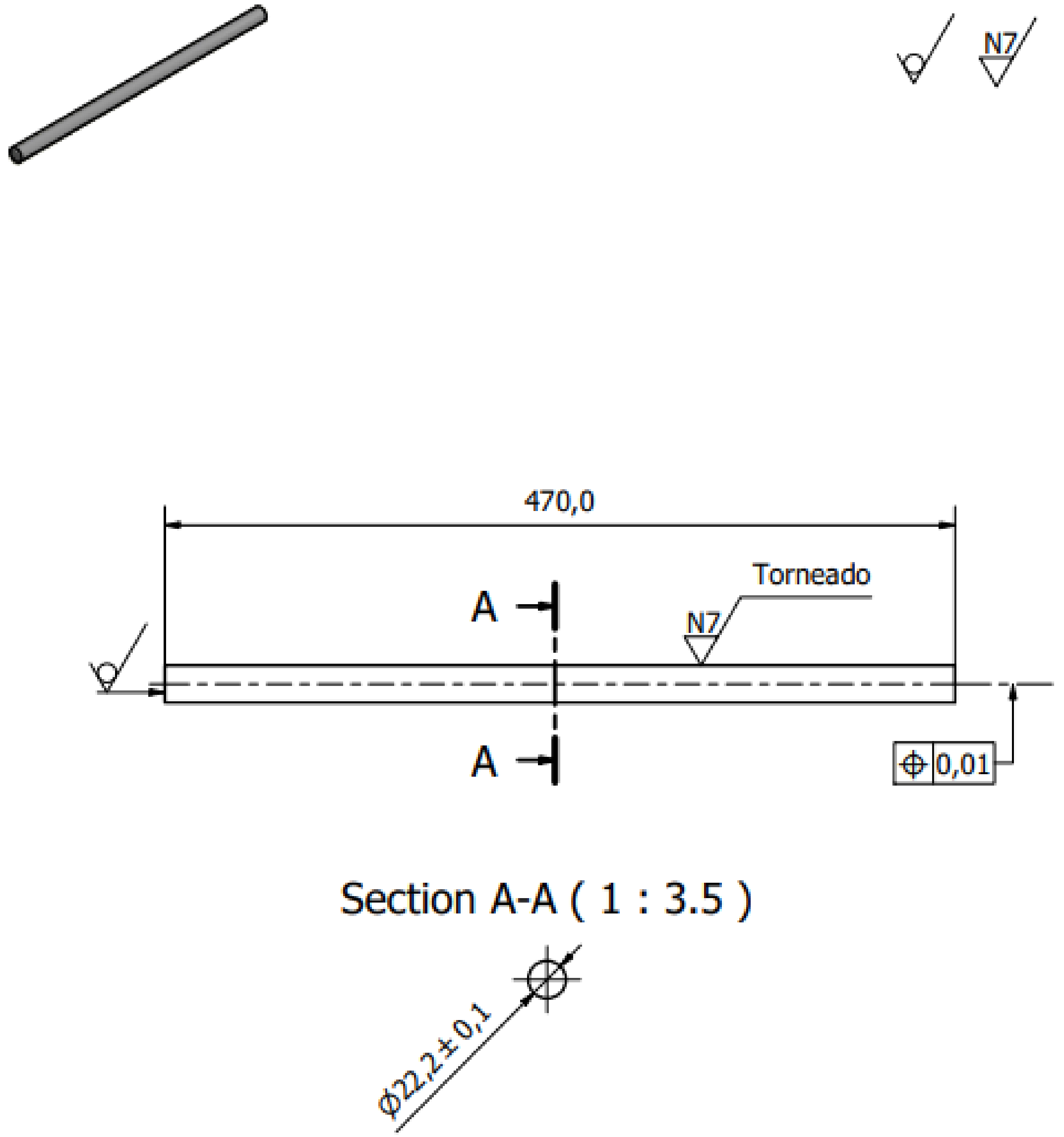
APPENDIX A (MECHANICAL DRAWINGS AND CODES)

Mechanical Drawings



Parts List

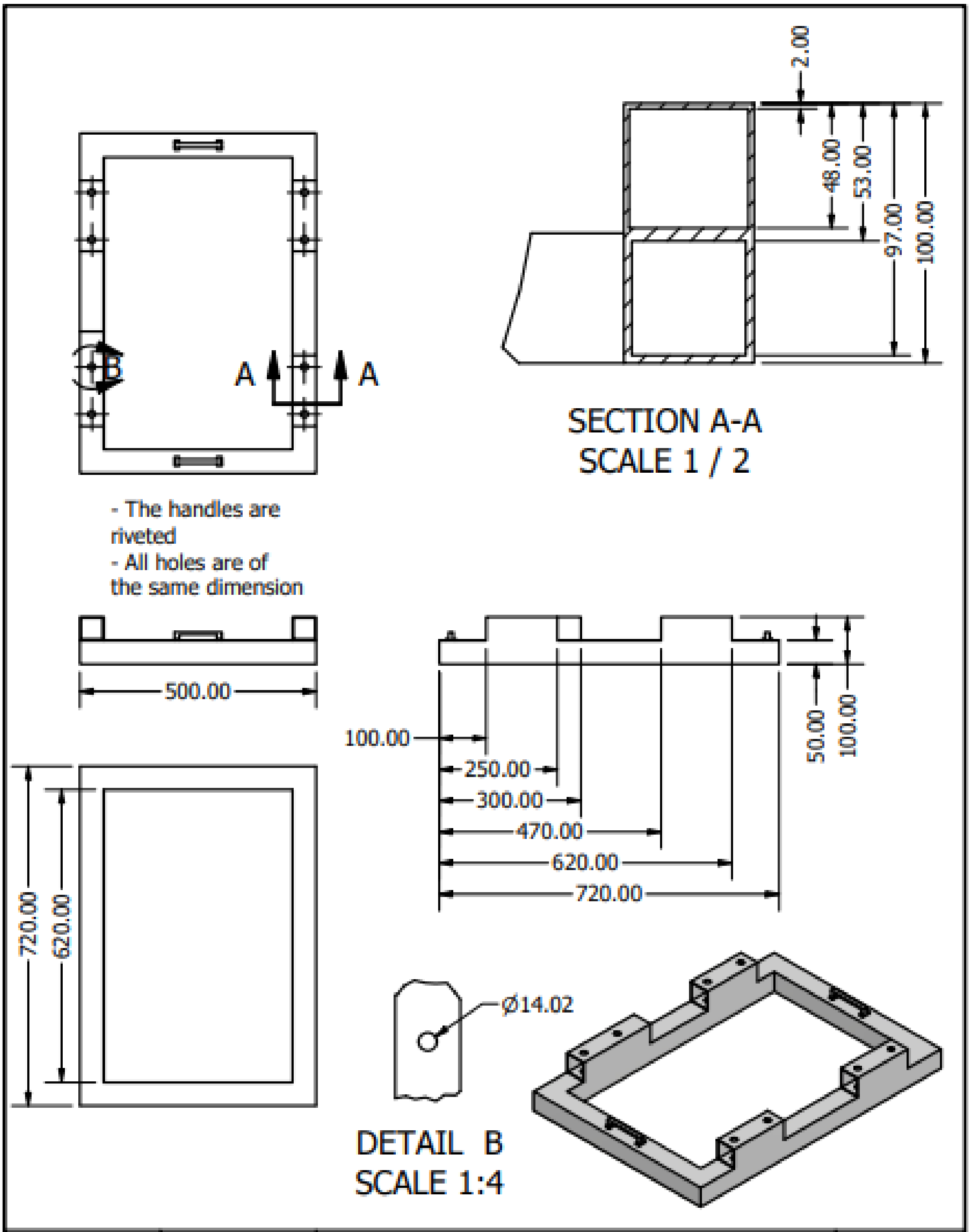
Item	Qty	Part Number	Description	Material
1	4	Bearing		Steel
2	1	Support Shafts Assembly		Steel
3	1	Principal Shaft Assembly		
4	1	New Metal Holder		Steel
5	8	M14 2-6G Bolt		Steel
6	1	Base		Steel
8	1	Capacitive Sensor		Solidworks materials|AISI 304



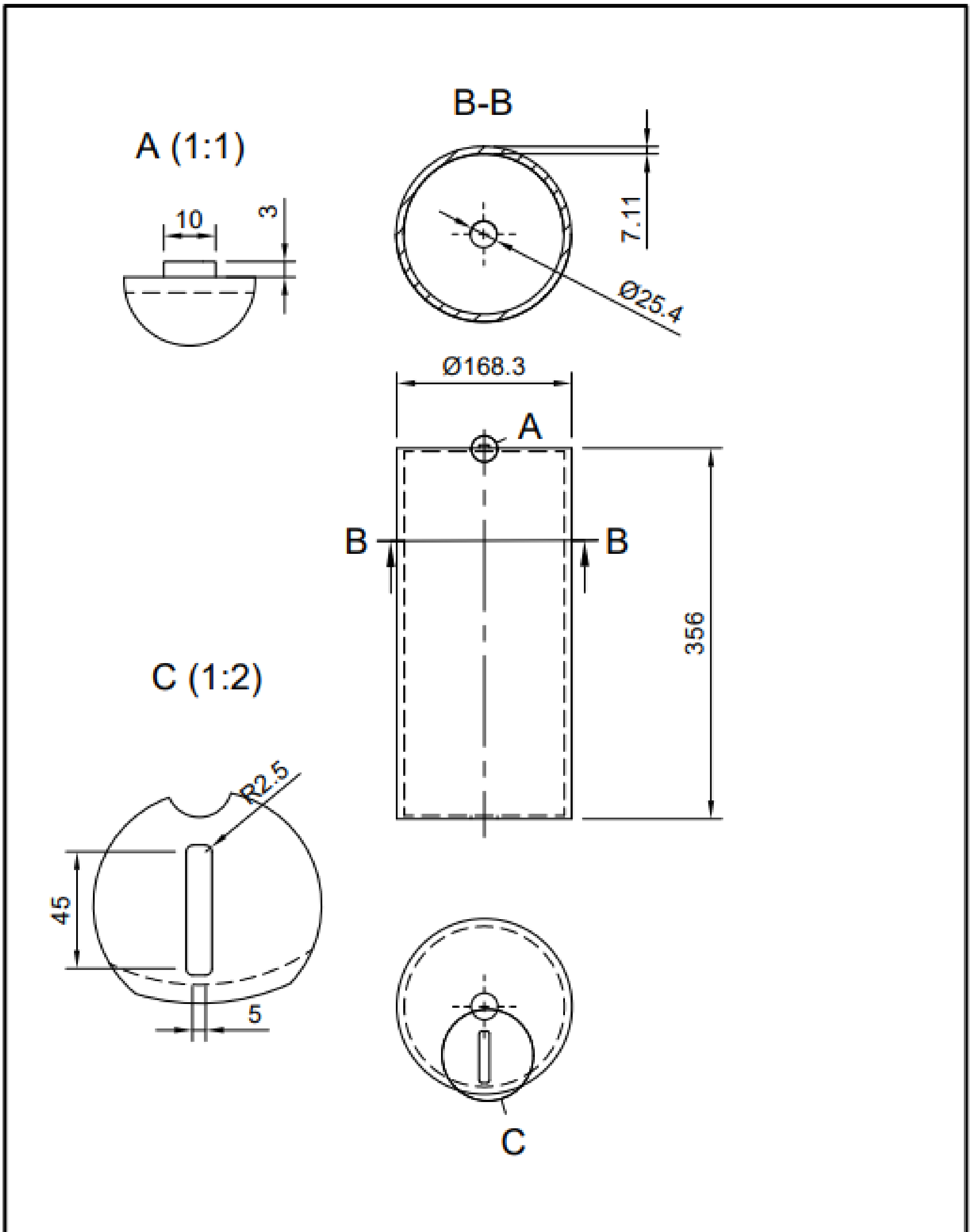
TOLERANCE	MASS	MATERIAL	
±0.1	1.43 kg	Steel	

	DATE	NAME	DENOMINATION	SCALE
DRAWN ON	10/12/2022	Mateo Montenegro	Accompanying Shaft	1 : 3.5
REVIEWED BY		Patricio Chiriboga		
APPROVED BY		Patricio Chiriboga		

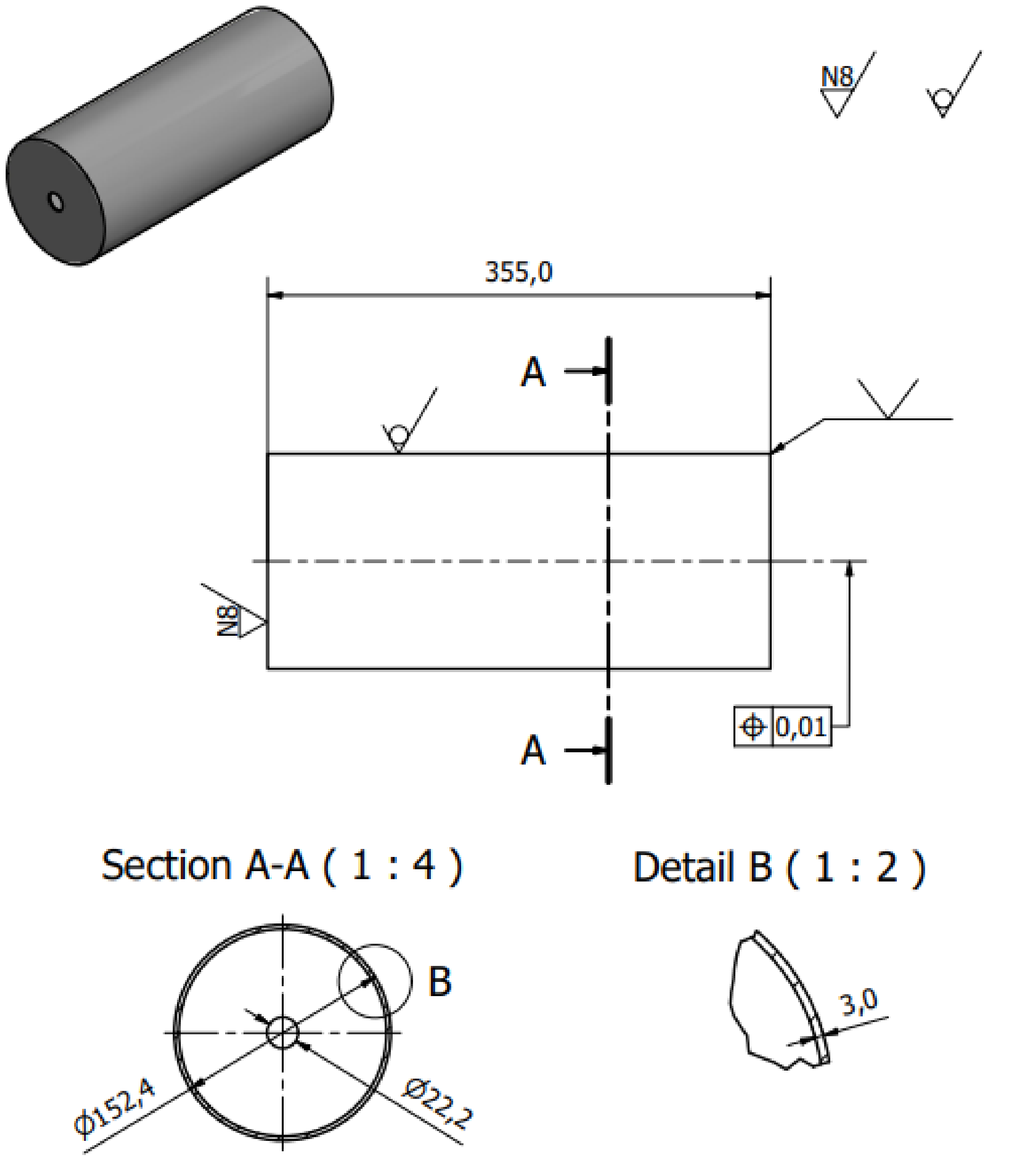
	COVERING	HEAT TREATMENT	SHEET No
USFQ	No Covering	No HT	1



TOLERANCE		MASS	MATERIAL		
±0.1		12,195 kg	Steel		
	DATE	NAME	DENOMINATION		SCALE
DRAWN ON	21/12/2022	Martín Cárdenas	Base		1 / 11
REVIEWED BY		Patricio Chiriboga			
APPROVED BY		Patricio Chiriboga			
USFQ			COVERING	HEAT TREATMENT	SHEET No
			No Covering	No HT	1



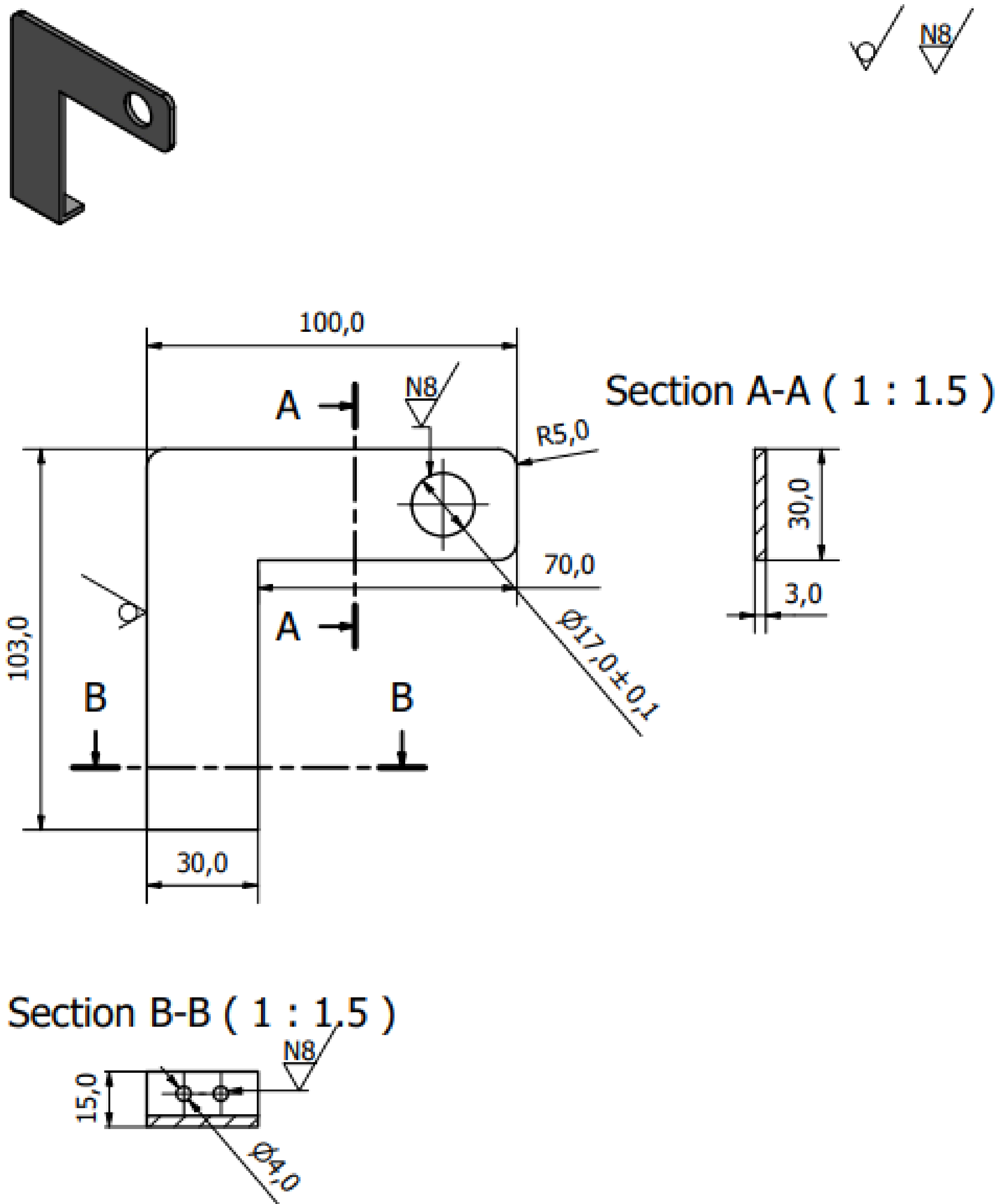
Mass 10927.838 g		Name Principal Roller		Projection
	Date	Name	Code CD_LS_002	Scale 1:2
Drawn by	01/11/2022	Luis Sánchez		
Reviewed by		**Patricio Chiriboga**		
Approved by		Patricio Chiriboga		
USFQ		Material Steel	Surface treatment N/A	Sheet No.



Note:
- Welding shall be carried out by a qualified welder

TOLERANCE	MASS	MATERIAL	
±0.1	4.57 kg	Steel	
		DENOMINATION	SCALE
		Roller	1 : 4
		COVERING: No Covering / HEAT TREATMENT: No HT	SHEET No: 3

	DATE	NAME
DRAWN ON	10/12/2022	Mateo Montenegro
REVIEWED BY		Patricio Chiriboga
APPROVED BY		Patricio Chiriboga



Note:
- The width of the whole bracket is 3.0 mm

TOLERANCE	MASS	MATERIAL		
±0.1	0.12 kg	Steel		

	DATE	NAME	DENOMINATION	SCALE
DRAWN ON	10/12/2022	Mateo Montenegro	Inductive Sensor Bracket	1 : 1.5
REVIEWED BY		Patricio Chiriboga		
APPROVED BY		Patricio Chiriboga		

	COVERING	HEAT TREATMENT	SHEET No
USFQ	No Covering	No HT	2

Codes

Arduino Code

```
//time variables
  int previous;
  int time;
  float delta;

  //sensor and other variables
  int signal;
  int presignal=900;
  int sensor=A1;
  int stop=3;
  int i=0;
  int state;
  float test;
  int light=12;

void setup() {
  Serial.begin(9600);
  pinMode(sensor, INPUT);
  pinMode(stop,INPUT);
  pinMode(light,OUTPUT);
  previous=millis();
  }

void loop() {
  signal=analogRead(sensor);

  if((signal>1000) && (presignal<1000)){
    time=millis();
    delta=(time-previous);

    Serial.println(delta);

    previous=millis();


    delay(10);
  }
  presignal=signal;

  state=digitalRead(stop);
  if (state==0){
    test=0;
    Serial.println(test);
    digitalWrite(light,HIGH);
```

```
  }
  else{
    digitalWrite(light,LOW);
  }
}
```

Matlab Code

```
close all;
clear variables;
clc;

s=serialport('COM9', 9600);
i=1;
j=0;
k=1;
R=0.08415; %meters
wheel=0.55; %meters
inertia=0.07131; %kg-m2;

while (k==1)
    %Speed
    delta(i) = str2double(readline(s));
    roller_w(i)=((2*pi)./delta(i))*1000; %rad/s
    speed(i)=(roller_w(i)*R*3.6); %km/h
    wheel_w(i)=((speed(i)./3.6)/wheel);


    if(j==0)
        prew(i)=roller_w(i);
    end

    %Acceleration
    angular_a(i)=(roller_w(i)-prew(i))/delta(i);
    acceleration(i)=angular_a(i)*R;

    %Torque
    roller_torque(i)=inertia*angular_a(i);
    wheel_torque(i)=(roller_w(i)./wheel_w(i))*roller_torque(i);

    %Variable update
    prew(i+1)=roller_w(i);
    i = i+1;
    j = j+1;
    pause(0.01);

    %Graphs
    subplot(3,1,1);
    plot(speed);
    xlabel("Time [s]");
```

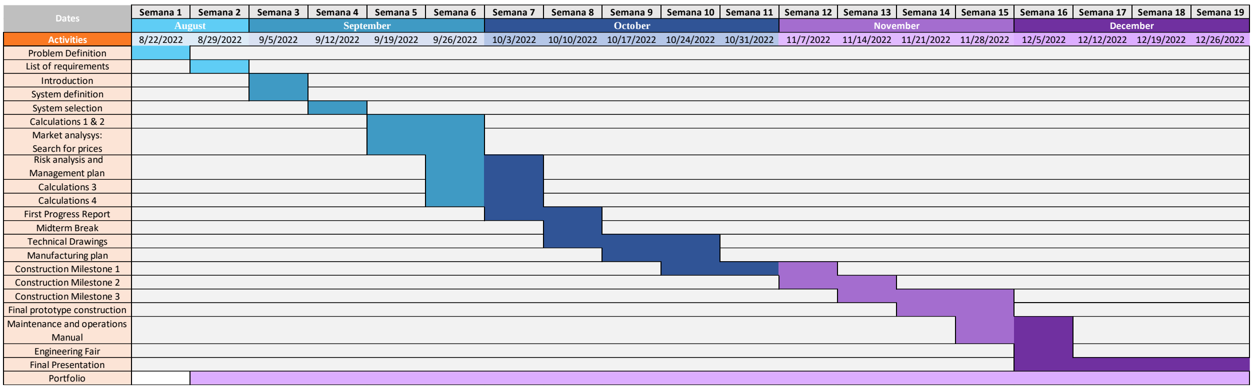
```
    ylabel("Speed [km/h]");
    title("Speed vs time");
    grid on;

    subplot(3,1,2);
    plot(acceleration);
    xlabel("Time [s]");
    ylabel("Acceleration [m/s^2]");
    title("Acceleration vs time");
    grid on;

    subplot(3,1,3);
    plot(wheel_torque);
    xlabel("Time [s]");
    ylabel("Wheel Torque [Nm]");
    title("Wheel Torque vs time");
    grid on;

    %End Process
    state=str2double(readline(s));
    if (state==0)
        k=0;
        break;
    end
end
```

APPENDIX B (PROJECT MANAGEMENT)

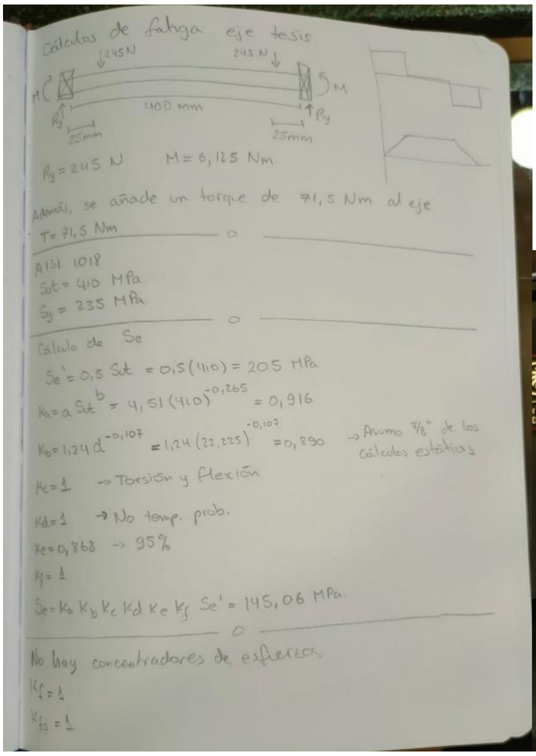


To see the full version of this diagram, follow the link:

https://estudusfqedu-my.sharepoint.com/:x:/g/personal/lfsanchez_estud_usfq_edu_ec/Eb46RbkCe_ZHqJJ4XygDPugBKKlPFUo6OeFfsNNypPNDxQ?e=c7ZQPc

APPENDIX C (EVIDENCE)

Design Process



Cálculos de fatiga eje tesis

245 N　245 N

1100 mm

25 mm　25 mm

R_y　M

$R_y = 245$ N　$M = 6,125$ Nm

Además, se añade un torque de 71,5 Nm al eje

$T = 71,5$ Nm

AISI 1018

$S_{ut} = 410$ MPa

$S_y = 235$ MPa

Cálculo de Se

$S_e' = 0,5\,S_{ut} = 0,5(410) = 205$ MPa

$K_a = a\,S_{ut}^{\,b} = 4,51(410)^{-0,265} = 0,916$

$K_b = 1,24\,d^{-0,107} = 1,24(22,225)^{-0,107} = 0,890$ → Asumo 7/8" de los cálculos estáticos

$K_c = 1$ → Torsión y flexión

$K_d = 1$ → No temp. prob.

$K_e = 0,868$ → 95%

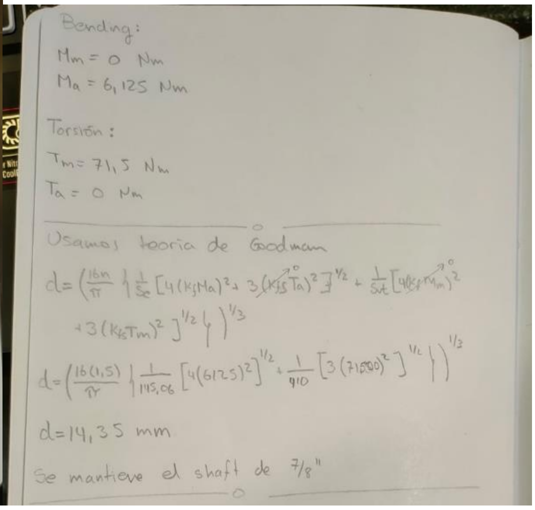
$K_f = 1$

$S_e = K_a K_b K_c K_d K_e K_f\, S_e' = 145,06$ MPa

No hay concentradores de esfuerzos

$K_f = 1$

$K_{fs} = 1$



Bending:

$M_m = 0$ Nm

$M_a = 6,125$ Nm

Torsión:

$T_m = 71,5$ Nm

$T_a = 0$ Nm

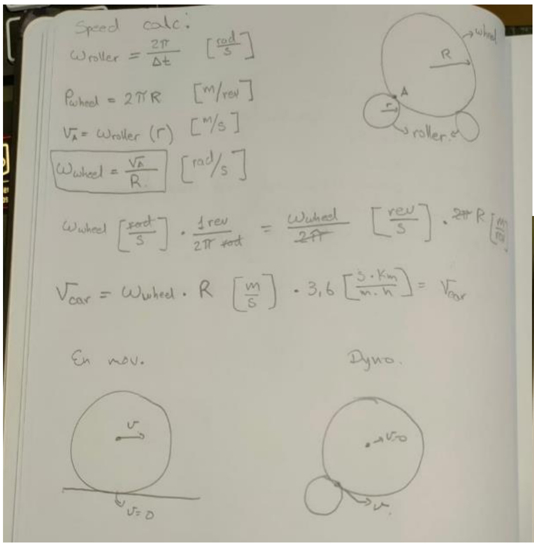
Usamos teoria de Goodman

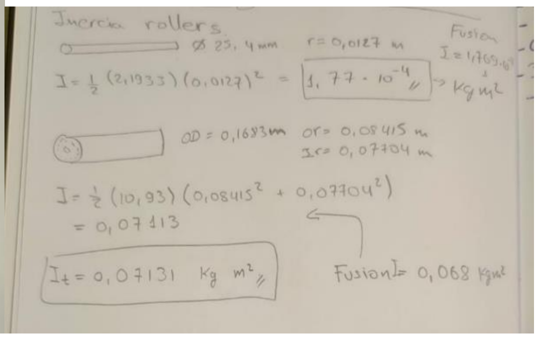
$$d = \left(\frac{16n}{\pi}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(\cancel{K_{fs} T_a})^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4(\cancel{K_f M_m})^2 + 3(K_{fs} T_m)^2\right]^{1/2}\right\}\right)^{1/3}$$

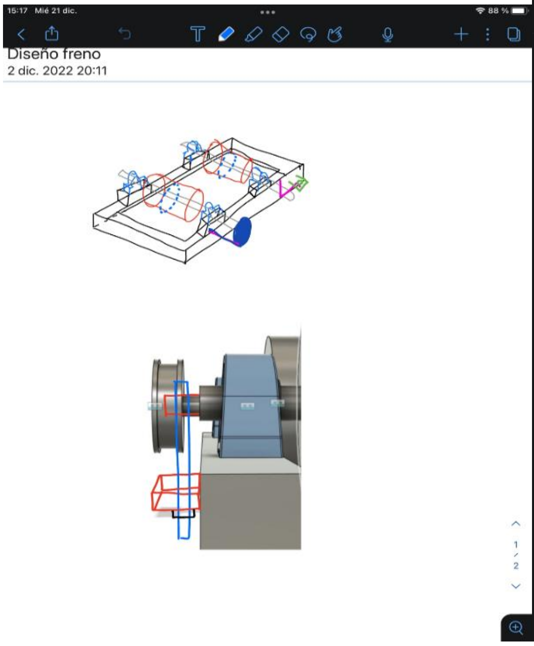
$$d = \left(\frac{16(1,5)}{\pi}\left\{\frac{1}{145,06}\left[4(6125)^2\right]^{1/2} + \frac{1}{410}\left[3(71500)^2\right]^{1/2}\right\}\right)^{1/3}$$

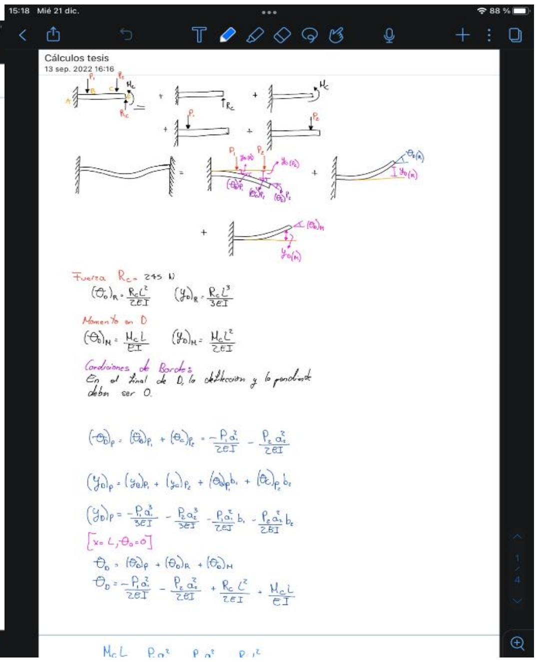
$d = 14,35$ mm

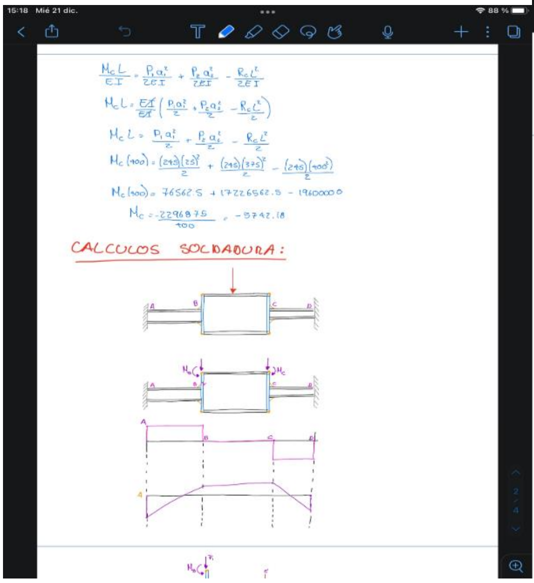
Se mantiene el shaft de 7/8"

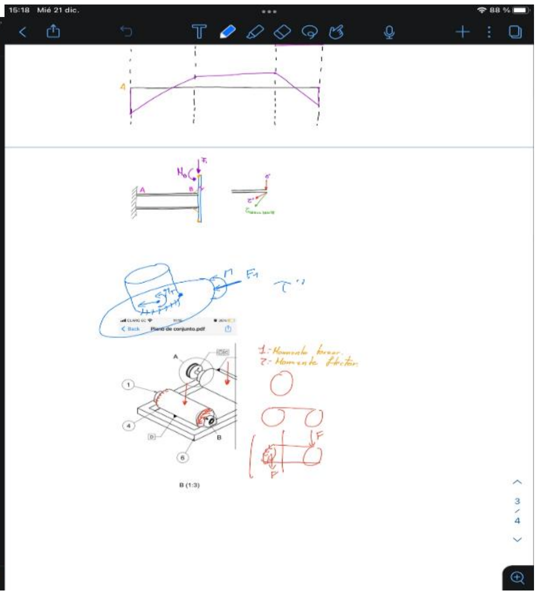

Speed calc:
$\omega_{roller} = \frac{2\pi}{\Delta t} \left[\frac{rad}{s}\right]$
$P_{wheel} = 2\pi R$ [m/rev]
$V_A = \omega_{roller} (r)$ [m/s]
$\boxed{\omega_{wheel} = \frac{V_A}{R}}$ [rad/s]
$\omega_{wheel} \left[\frac{\cancel{rad}}{s}\right] \cdot \frac{1\,rev}{2\pi\,\cancel{rad}} = \frac{\omega_{wheel}}{\cancel{2\pi}} \left[\frac{rev}{s}\right] \cdot \cancel{2\pi} R$
$V_{car} = \omega_{wheel} \cdot R \left[\frac{m}{s}\right] \cdot 3,6 \left[\frac{s \cdot Km}{m \cdot h}\right] = V_{car}$
wheel
R
A
r
roller
En mov.
Dyno.
v
v=0
v=0
v

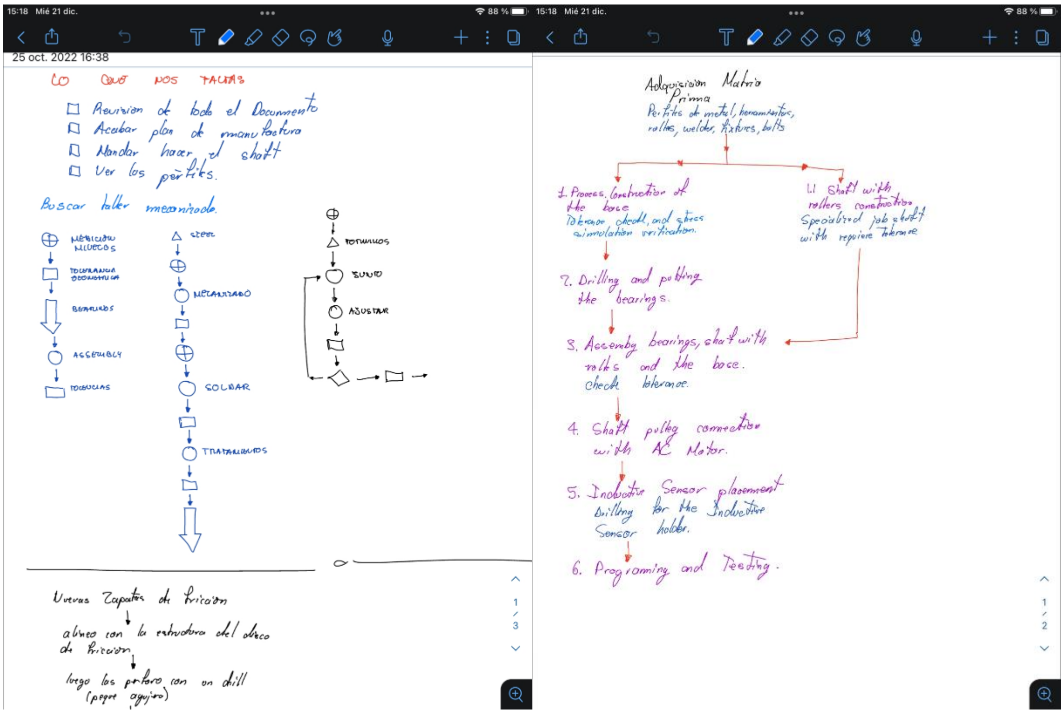

Inercia rollers
Ø 25,4 mm
r = 0,0127 m
$I = \frac{1}{2}(2,1933)(0,0127)^2 = \boxed{1,77 \cdot 10^{-4}}$ → Kg m²
Fusion
I = 1,769·10
OD = 0,1683 m
or = 0,08415 m
Ir = 0,07704 m
$I = \frac{1}{2}(10,93)(0,08415^2 + 0,07704^2)$
$= 0,07113$
$\boxed{I_t = 0,07131 \text{ Kg m}^2}$
Fusion I = 0,068 Kgm²

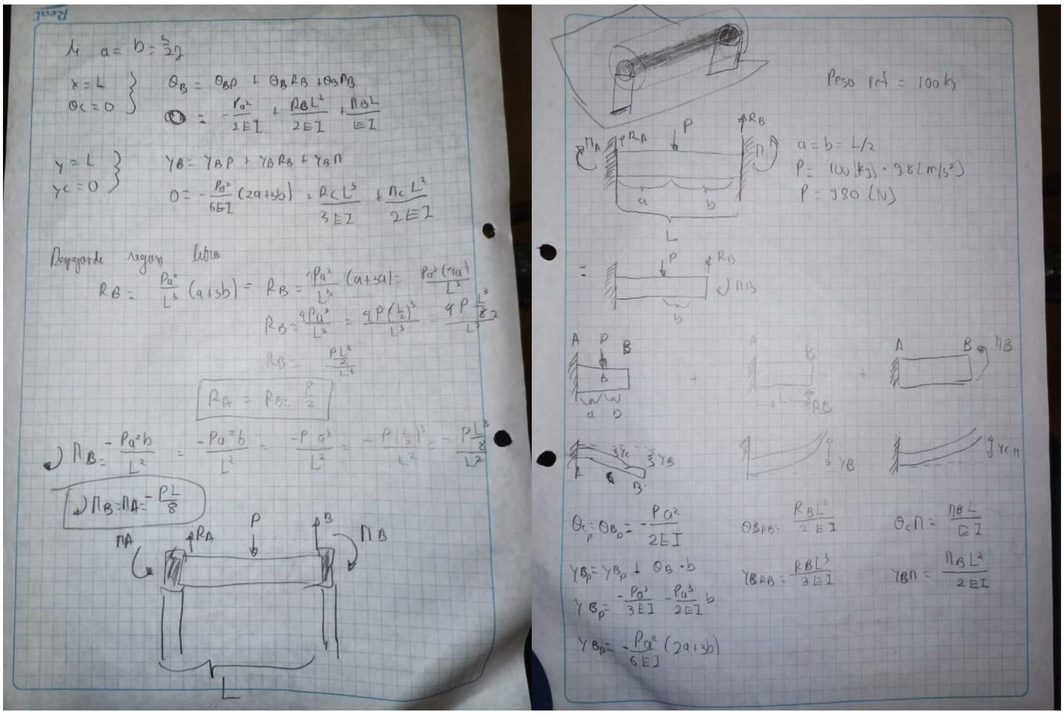

Diseño freno
2 dic. 2022 20:11

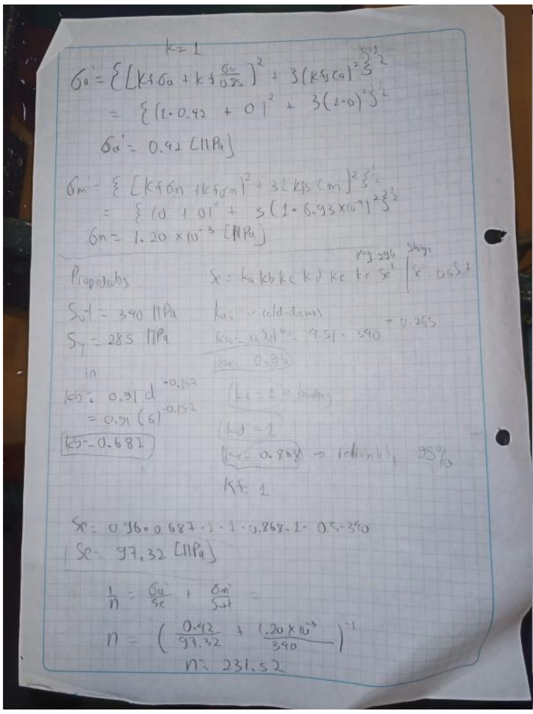

Cálculos tesis
13 sep. 2022 16:16
Fuerza R_c = 245 N
Momento en D
Condiciones de Bordes
En el final de D, la deflección y la pendiente deben ser 0.

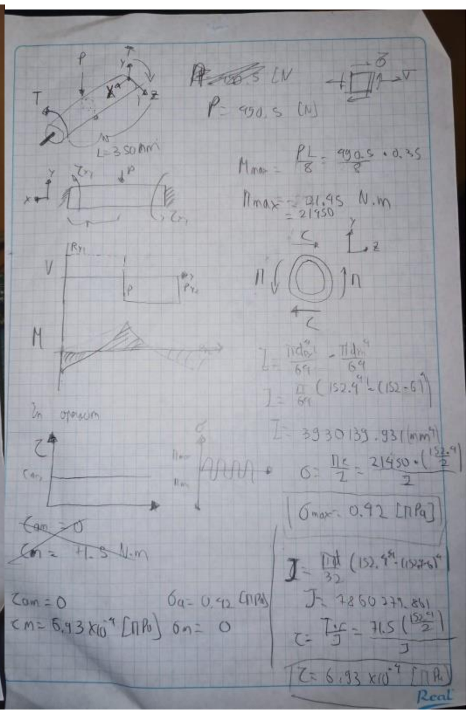

CALCULOS SOLDADURA:


1.- Momento torsor
2.- Momento flector


25 oct. 2022 16:38
Revisión de todo el Documento
Acabar plan de manufactura
Mandar hacer el shaft
Ver los perfiles.
Buscar taller mecanizado.
Adquisición Materia Prima
2. Drilling and polishing the bearings.
4. Shaft pulley connection with AC Motor.
6. Programming and Testing.
Nuevas Zapatas de fricción


Peso ref = 100 kg
a = b = L/2


k = 1
σa' = {[Kf σa + Kf σa/0.85]² + 3(Kfs τa)²}^½
= {(1·0.42 + 0)² + 3(1·0)²}^½
σa' = 0.42 [MPa]
σm' = {[Kf σm + Kf σm]² + 3[Kfs τm]²}^½
= {(0 + 0)² + 3(1·6.93×10⁻⁴)²}^½
σm = 1.20×10⁻³ [MPa]
Propiedades
Se = ka kb kc kd ke kf Se'
pag 296
Shigley
Se' = 0.5 Sut
Sut = 390 MPa
Sy = 285 MPa
ka → cold-drawn
ka = a Sut^b = 4.51 · 390^-0.265
ka = 0.96
in
kb = 0.91 d^-0.157
= 0.91 (6)^-0.157
kb = 0.687
kc = 1 → bending
kd = 1
ke = 0.868 → reliability 95%
Kf = 1
Se = 0.96 · 0.687 · 1 · 1 · 0.868 · 1 · 0.5 · 390
Se = 97.32 [MPa]
1/n = σa'/Se + σm'/Sut
n = (0.42/97.32 + 1.20×10⁻³/390)⁻¹
n = 231.52


P
T
y
x
z
L = 350 mm
P = 990.5 [N]
σ
V
Mmax = PL/8 = 990.5 · 0.35 / 8
Mmax = 21.45 N.m
= 21450
V
M
Ry₁
Ry₂
M
τ
I = π do⁴/64 − π di⁴/64
I = π/64 (152.4⁴ − (152 − 6)⁴)
I = 3930139.931 (mm⁴)
σ = Mc/I = 21450 · (152.4/2) / I
σmax = 0.92 [MPa]
τ
τam = 0
τam = 0
τm = 71.5 N.m
σa = 0.92 [MPa]
σm = 0
τm = 6.93×10⁻⁴ [MPa]
J = π/32 (152.4⁴ − (152.4 − 6)⁴)
J = 7860279.861
τ = T·c/J = 71.5 (152.4/2) / J
τ = 6.93×10⁻⁴ [MPa]
Real

Construction

Purchase Receipts

Ferretería Cano
José Cano e Hijos S.C.C.
Ruc: 1792153867001
Cumbayá: Av. Maria Angelica Idrovo Oo3-37 y General Eloy Alfaro
Telf: (02) 2890305 / 099 6603 927 / 099 6603 928 / 099 6603 924
E-mail: ferrcano@yahoo.com

FACTURA
N° 001002 000164204
DOCUMENTO SIN VALIDEZ TRIBUTARIA

CLIENTE: CHIRIBOGA ARROYO PATRICIO GABRIEL
CED/RUC: 1710634641
DIRECCIÓN: EL BATAN
TELÉFONO: 987892785
CORREO: pchiriboga@gmail.com
OBSERVACIÓN: Refer. Factura N° 000164204
REGIMEN GENERAL

FECHA: 01/11/2022 16:37
VENDEDOR: STEFANO QUIÑONEZ
CAJERO: CECIBEL
FORMA DE PAGO CREDITO

CÓDIGO	DESCRIPCIÓN	CANT.	VALOR UNIT.	DESCTO.	VALOR TOTAL
26820-187	PERFIL TUBO CUADRADO 2 50X50X2.0MM 18.20KG	1.00	29.58	10%	26.62

FERRETERIA CANO
RUC: 1792153867001
TRANSFERENCIA

ELABORADO POR:

RECIBI CONFORME

Subtotal 12%:	29.58
Subtotal 0%:	0.00
Descuento:	-2.96
Valor Neto 12%:	26.62
Valor Neto 0%:	0.00
IVA 12%:	3.19
Total:	29.81

1)No se aceptan reclamos despúes de 5 dias entregada la mercaderia.

2)Se debe presentar el comprobante original junto con el articulo en las mismas condiciones adquiridas, no debe haber sido utilizado ni debe presentar deterioro o daños adicionales que no sean de fábrica.

3)Se realizan cambios de mercaderia solo por algún otro articulo del mismo tema o tipo, pero con diferentes caracteristicas como tamaño, color, accesorios, etc

4)No aceptamos cambios ni devoluciones en material eléctrico.

Art. 50 LORTI. Los agentes retención están obligados a entregar el respectivo comprobante de retención dentro del término no mayor a 5 dias de recibido elcomprobante de venta

Retenciones electrónicas remitidas a ferrcano@yahoo.com

DETALLE DE FACTURA ELECTRONICA
Fact.#008-003-000053100
(Documento sin validez tributaria)
Clave Acceso: 1611202201170799125100120080
0300005310012345678l2
Ambiente: Produccion
Emision: Emision Normal

Ruc: 1707991251001
JARAMILLO ORDONEZ EDWIN RODRIGO
Dir.Matriz: CALLE SAN CARLOS OE4-42 Y PASA
JE B
Dir.Sucursal: VIA TUMBACO KM. 14.5 Y SEGUN
DARIA CENTRO COMERCIAL MALL VENTURA LOCAL
15-1-PA
Contribuyente Especial Nro.: E489
Obligado a Llevar Contabilidad: SI
Fecha: 16/11/2022 15:33:18

Cliente: 1710634641-PATRICIO CHIRIBIGA
Dir.Cliente: CUMBAYA
Telefono: 0987892785
Email:

Cod.	Descri.	Cant.	P.Unit	Total
101880	CHUMACERA DE PISO 205-16 X 1 pulg.	2.00	5.30	10.61

Subtotal 12%:	10.61
Subtotal 0%:	0.00
Tot. Descto.:	1.59
Iva 12%:	1.08
Total	10.10

Su factura electronica la puede descargar en www.comercialjaramillo.com

Si ingresa por primera vez a consultar el comprobante el usuario y password es su numero de identificacion.

DETALLE DE FACTURA ELECTRONICA
Fact.#008-003-000053183
(Documento sin validez tributaria)
Clave Acceso: 2111202201170799125100120080
0300005318312345678l6
Ambiente: Produccion
Emision: Emision Normal

Ruc: 1707991251001
JARAMILLO ORDONEZ EDWIN RODRIGO
Dir.Matriz: CALLE SAN CARLOS OE4-42 Y PASA
JE B
Dir.Sucursal: VIA TUMBACO KM. 14.5 Y SEGUN
DARIA CENTRO COMERCIAL MALL VENTURA LOCAL
15-1-PA
Contribuyente Especial Nro.: E489
Obligado a Llevar Contabilidad: SI
Fecha: 21/11/2022 12:29:33

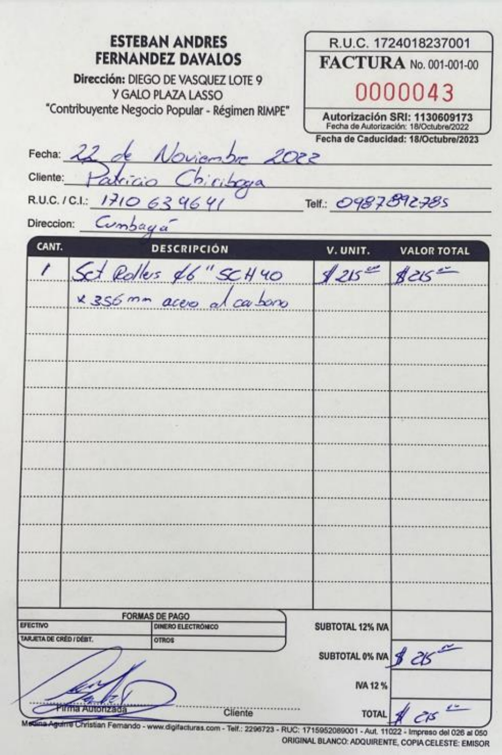
Cliente: 1710634641-PATRICIO CHIRIBIGA
Dir.Cliente: CUMBAYA
Telefono: 0987892785
Email:

Cod.	Descri.	Cant.	P.Unit	Total
101880	CHUMACERA DE PISO 205-16 X 1 pulg.	2.00	5.30	10.61

Subtotal 12%:	10.61
Subtotal 0%:	0.00
Tot. Descto.:	1.59
Iva 12%:	1.08
Total	10.10

Su factura electronica la puede descargar en www.comercialjaramillo.com

Si ingresa por primera vez a consultar el comprobante el usuario y password es su numero de identificacion.



ESTEBAN ANDRES FERNANDEZ DAVALOS
Dirección: DIEGO DE VASQUEZ LOTE 9 Y GALO PLAZA LASSO
"Contribuyente Negocio Popular - Régimen RIMPE"

R.U.C. 1724018237001
FACTURA No. 001-001-00 0000043
Autorización SRI: 1130609173
Fecha de Autorización: 18/Octubre/2022
Fecha de Caducidad: 18/Octubre/2023

Fecha: 22 de Noviembre 2022
Cliente: Patricio Chiriboga
R.U.C. / C.I.: 1710634641 Telf.: 0987892785
Direccion: Cumbayá

CANT.	DESCRIPCIÓN	V. UNIT.	VALOR TOTAL
1	Set Rollers ⌀6" SCH 40 x 356 mm acero al carbono	$215.00	$215.00

FORMAS DE PAGO	
EFECTIVO	DINERO ELECTRÓNICO
TARJETA DE CRÉD / DÉBT.	OTROS

SUBTOTAL 12% IVA	
SUBTOTAL 0% IVA	$215.00
IVA 12 %	
TOTAL	$215.00

Firma Autorizada Cliente

Medina Aguirre Christian Fernando - www.digifacturas.com - Telf.: 2296723 - RUC: 1715952089001 - Aut. 11022 - Impreso del 026 al 050
ORIGINAL BLANCO: ADQUIRENTE, COPIA CELESTE: EMISOR

Electronics
ECUADOR
Robótica y Electrónica

RUC: 1715227466001

FACTURA

No. 001-010-000000749

NÚMERO DE AUTORIZACIÓN

2511202201171522746600120010100000007498201995910

FECHA Y HORA DE AUTORIZACIÓN: 25/11/2022 10:00 AM

AMBIENTE: PRODUCCIÓN

EMISIÓN: NORMAL

CLAVE DE ACCESO

2511202201171522746600120010100000007498201995910

Electronics

Edwin Stalin Nasimba Villarreal

Dirección matriz: Versalles N25-28 y Av Cristóbal Colon

Teléfono: 0994624335

OBLIGADO A LLEVAR CONTABILIDAD: NO
CONTRIBUYENTE RÉGIMEN RIMPE

Razón social / Nombres y apellidos:	Patricio Chiriboga
Identificación:	1710634641
Fecha emisión:	25/11/2022
Dirección:	Cumbaya
Teléfono:	0987892785

Cod. Principal	Cod. Auxiliar	Cantidad	Descripción	Precio unitario	Descuento	Precio total
COMPON12	COMPONEN	1,00	Componentes Electronicos	38,839286	0,00	38,84

SUBTOTAL 12%	38,84
SUBTOTAL 0%	0,00
SUBTOTAL NO OBJETO DE IVA	0,00
SUBTOTAL EXENTO DE IVA	0,00
SUBTOTAL SIN IMPUESTOS	38,84
TOTAL DE DESCUENTO	0,00
ICE	0,00
IVA 12%	4,66
IRBPNR	0,00
PROPINA	0,00
VALOR TOTAL	43,50

Forma de pago	Valor
01 - Sin utilización del sistema financiero	43,50

Información adicional

Correo de cliente: pchiriboga@usfq.edu.ec

Enviar los comprobantes de retenciones electrónicas al correo: n3dwin@hotmail.com

Contribuyente Régimen RIMPE: .

Meeting Schedule

Day	Meeting
Monday	14:30 – 16:00
Tuesday	11:20 – 13:00; 16:00 – 17:30
Wednesday	14:30 – 16:00
Thursday	11:20 – 13:00; 16:00 – 17:30
Friday	9:00 – 13:00 (If necessary)
Saturday	If necessary
Sunday	If necessary