Design and Fabrication of Test Rig for Acoustic Analysis of Crank Balancer

A Final Year Project Report

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In Partial Fulfillment
of the Requirements for the Degree of
Bachelors of Mechanical Engineering

by

Abdul Ahad Khan

Asjad Ali

Hammad Saleem

Syed Muhammad Mehdi

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EXAMINATION COMMITTEE

We hereby recommend that the final year project report prepared under our supervision by:

Abdul Ahad Khan	249543
Asjad Ali	247693
Hammad Saleem	257678
Syed Muhammad Mehdi	243213

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Supervisor: Dr. Riaz Mufti, Professor	____________________ Dated:
Committee Member: Name, Title (faculty rank) Affiliation	____________________ Dated:
Committee Member: Name, Title (faculty rank) Affiliation	____________________ Dated:

______________________ ___________

(Head of Department) (Date)

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ABSTRACT

This project aims towards the acoustic analysis of the crank balancer of a tractor engine – Massey Ferguson MF 300. Our job is to design and fabricate the Mechanical Test Rig for this analysis. Further analysis using Raw Signals provided by the Test Rig is not in the scope of this project. To avoid extra vibrations or unwanted sound emission by the Test Rig, we have acquired close tolerances in the fabrication process. In our design, we have replicated the original environment of the crank balancer, as being in the engine, to obtain accurate test results and converge to the source of the noise coming out of it. We have provided a direct drive system to the crank balancer via the motor-shaft system. A variable frequency drive system is used to regulate the motor. An open sump tank is designed to directly record the sound signals originating from the crank balancer to avoid any interference.

ACKNOWLEDGMENTS

First of all, we would like to thank Allah Almighty for providing us with the motivation, strength, and persistence to complete this project. We would like to thank our supervisor and mentor Dr. Riaz Ahmed Mufti. He guided us on every step. Without him we might have hit a lot of obstacles in our project. We would also like to thank Dr. Jawad Aslam for helping us with our project. He gave us a valuable insight into the market and with his help we were able to make decisions regarding the market. We would also like to acknowledge the efforts of members of Tribology lab as their experience proved valuable for our project. The impact of MS students in our project can also not be neglected. We would like to thank Mr. Qasim and Mr. Atta for their efforts regarding this project and succoring us on every turn.

This project had a lot of ups and downs. We would like to thank our friends and family for staying beside us and encouraging us to give us motivation to carry this project forward and complete it.

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ABBREVIATIONS

A	Ampere
AC	Alternating Current
AED	Acoustic Event Detection
BS	British Standard
CAD	Computer Aided Design
CNC	Computerized Numerical Control
DC	Direct Current
FEA	Finite Element Analysis
I/O	Input/Output
IP	Ingress Protection
MF	Massey Ferguson
MS	Mild Steel
MW	Medium Wave
OD	Outer Diameter
OEM	Original Equipment Manufacturer
PDP	Positive Displacement Pump
RF	Radio Frequency

RPM	Rounds Per Minute
SAE	Society of Automotive Engineers
SS	Stainless Steel
STC	Sound Transmission Class
V	Voltage
VFD	Variable Frequency Drive
VSD	Variable Speed Drive

CHAPTER 1: INTRODUCTION

1.1 Background and Motivation

Pakistan lacks proper machinery for quality analysis in industrial sector. We generally import the testing machines are very expensive and increase the overall cost of the machine. By taking this project we aim to not only bring proper technology to test the machine being produced in Pakistan but also equip the industry to produce goods of international quality. A defect free machinery will be available in Pakistan for much low cost as compared to the one being imported.

Acoustic analysis is one of the most preferred analyses as you do not have to attach any extra component to the machine. You can just analyze the issue by running the machine and recording its sound. Despite the fact that the process seems easy, the post processing of data is critical and the most gruesome part of the process. Even though we will be designing a test rig to aid in the processing, post processing is out of our scope.

A crank balancer, a part of tractor's engine, is a device that has inertial loads in it, and it helps in balancing the load of a crank shaft. Thus, it reduces the secondary vibrations caused by the engine that can cause severe inconveniences to the driver. If not working properly, it can cause inefficiencies in the engine.

1.2 Problem Statement

The crank balancer of Millat Tractors has started emitting noise which causes inefficiencies in its performance. We have to design the test rig that can be used to identify the defect.

1.2 Objectives and Deliverables

The following are the objectives for this project:

1. To make a proof-of-concept prototype for the industry that helps performing the acoustic analysis of crank balancers.
2. The engine environment of the crank balancer has to be replicated.
3. To offer a variable speed control that can run and test the crank balancer at different speeds.
4. To provide an easy to assemble, handle, and operate design that is economical, and robust at the same time.

Following are the Deliverables of the project:

1. CAD model of the final design considering optimum changeover mechanism.
2. Fabrication of parts according to the design measures.
3. Assembly of parts to run the test rig.
4. Incorporation of a Variable Frequency Drive to regulate the operating speed.

CHAPTER 2: LITERATURE REVIEW

In this chapter, the literature that was reviewed in order to develop this project has been discussed.

2.1 Crank Balancer

Drivers suffer from backache, spinal cord injury, and other health problems as a result of the mechanical vibrations. A tractor engine plays a key role in this regard. Crank balancers (balancer units) are installed on tractors to reduce secondary engine vibrations and reduce engine and tractor vibrations.

Vibration arises in engines for three different reasons:

1. Vibration caused by unbalanced rotatory components such as the crankshaft, flywheel, and clutch.
2. Vibration caused by the impact of the piston while the crankshaft rotates.
3. Vibration caused by the inertia of reciprocating parts, such as pistons.

Weights are employed to neutralize the rotational inertia force, which helps to reduce vibration. The crankshaft gear attached to the balancer's idle gear revolves due to the rotating force of the crankshaft. This gear, on the other hand, is attached to the balancer shaft's gear and rotates the balancer weight. The weights are linked together by spur gears, and rotation of one cause rotation of the other. To provide a net vertical force, these weights rotate in opposite directions to eliminate vibration. When the centrifugal weights reach the low dead center, the balancer synchronizes with the crankshaft in such a way that a pair of pistons are situated at the top dead center. Tractor vibrations are caused by the lack of a damper and the engine's direct connection to the chassis when the balancer is not there.

The oil pump, balancer, and pressure breaker valve are all built into one unit. The roller bearings of the pump actuator shaft and idle gear are lubricated by spraying, while the bush bearings of shafts are lubricated by pressure. Two needle roller bearings are located at the ends of two shafts. The oil pump is controlled by an oil pump gear at the end of the drive shaft.

As the crank balancer is one of the most critical components for engine operation and performance, it's critical to make sure it's in good working order.

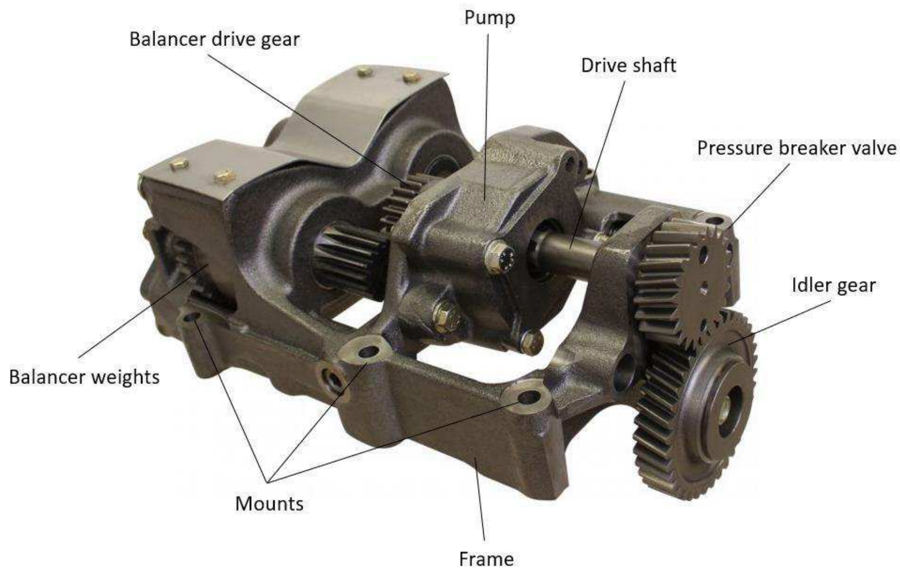

Figure 1: Crank balancer and its main components

2.2 Acoustic Test Rigs

Acoustic testing is very common around the globe. Acoustic test rigs require a proper environment to run a stable test. Depending on the application of the rig the environment is set. Some test rigs use closed environments like acoustic chambers while others can be tested in the open without any special equipment.

2.2.1 Acoustic Chamber

Acoustic chamber is a special room design to separate the noise coming from outside and the noise coming from this system inside the room. Its working can be broken into two parts. First it reflects any noise coming from anywhere, it will reflect the outside noise to the outside environment and the inside noise to the inside environment hence isolating the two environments. Secondly it will try to absorb the internal noise and not reflect it back towards the system. hence the sound waves coming from the system only hit the sensors once providing us with accurate readings.

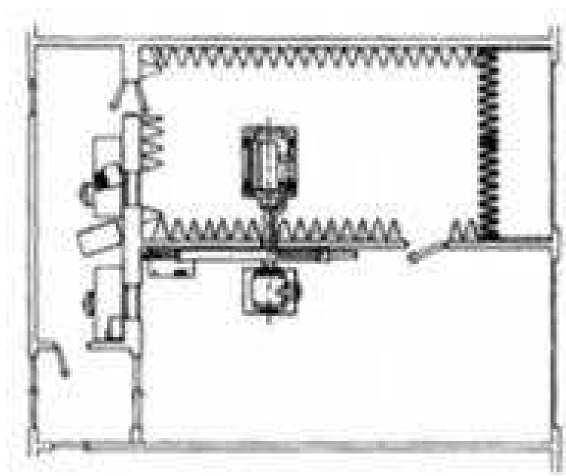

Figure 2: Top view of engine testing facility in anechoic chamber

As you can see in the image above the system being monitored is placed inside and an acoustic chamber end is provided drive externally.

In order to construct an anechoic chamber, we must consider the following parameters:

1. Range of frequencies coming from the system
2. Properties of wedges installed
3. Properties of walls used

4. STC of the chamber
5. Ratio of volume of object to room volume

Figure 3: Anechoic chamber for acoustic testing

2.2.2 Acoustic Analysis Methods

There were several methods used in acoustic testing. They are discussed below:

1. Sound Level Meter

   It is a handheld device that interprets and displays the sound it detects. It consists of a microphone paired with a circuit. Microphone picks up the sound and converts it into an electronic signal and the circuit then displays the required data on the screen. We can use this data to observe the sound coming from the system and detect any irregularities in it.

2. Spectrum Analysis

   It displays and measures amplitude vs frequency of RF and MW signals. It separates and demodulates complex waves into base forms i.e., the sine waves which can be easily interpreted, and irregularity is easy to detect.

3. AED (Acoustic Event Detection)

   Acoustic event detection and classification is a recent discipline that may be included in the broad area of computational auditory scene analysis. It consists of processing acoustic signals and converting them into symbolic descriptions corresponding to a listener's perception of the different sound events present in the signals and their sources.

4. Noise Dosimeter

   A single value of sound does not define its total characteristics, so we prefer special integrated sound-level meters for this method.

2.2.3 Sensor

Mostly the analysis was performed with a microphone paired with an NI card. The characteristics of microphones determine the quality of data collected. For better results, a better microphone is required. With an increase in quality, an increase in price is also seen. Hence, after a lot of research we settled on Bruel & Kjaer model 4188 which has a remarkable sample rate of 10^6. Its specifications are as follows:

Table 1: Specifications of Sensor

Capacitance	12 pF
Diameter	0.5 in
Dynamic range	15.8 – 146 dB
Frequency range	8 – 12500 Hz
Inherent noise	14.2 dB
Lower limiting frequency	5 Hz
Optimization	Free field

Pressure coefficient	-0.021 dB/kPa
Sensitivity	31.6 mV/Pa
Sound field	Free field

Figure 4 gives the visual demonstration:

Figure 4: Bruel & Kjaer Model 4188

2.3 Drive (AC Motor)

An alternating current motor (AC motor) is an electric motor that produces mechanical energy by using magnetism and alternating current. An AC motor's structure includes coils that generate a spinning magnetic field inside a rotor connected to an output shaft, which generates a second magnetic field.

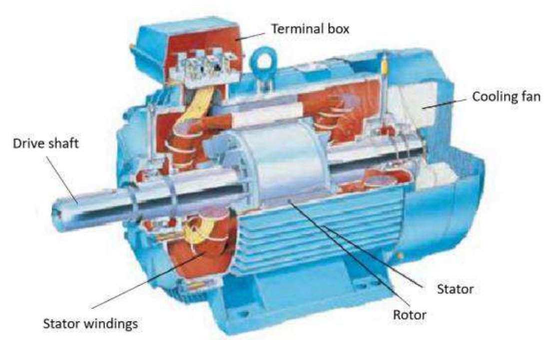

Figure 5: AC motor and its main components

In AC motor, the stator is a gird of electromagnets positioned around the outside which is designed to produce a spinning magnetic field. A solid metal axle, a wire loop, a coil, a squirrel cage constructed of metal bars and links, or some other freely rotating metal object that can conduct electricity can all be found inside the stator. A rotating magnetic field is created when current is delivered to the stator windings of an AC motor. A second revolving magnetic field is formed when this magnetic field produces an electrical current within the electrically conducting rotor. The interaction of the first and second magnetic fields causes the rotor to spin, and the motor to turn as the magnetic field alternates between coil pairs.

Unlike a DC motor, where power is directed to the inner rotor, an AC motor directs power to the outer coils of the stator.

Table 2: Reasons for selecting AC motor

Lower power startup demands protecting the receiving end components as well.
Starting current levels that can be adjusted.
Changes in speed and torque requirements can be adjusted more easily.
Enhanced customizability to meet a variety of configuration needs.
Longer life spans and higher durability.

2.4 Seals

The devices that help in joining systems or mechanisms together by avoiding leakage are termed as mechanical seals. These are usually used to seal fluids at high pressures.

2.4.1 Lip Seals

Lip seals also known as oil seals are generally used to stop the leakage of oil from the system and keeps the dirt and dust from entering the system. It is generally attached to the junction of the Rotary body that exits the system and the boundary of the system.

There are many different types of seals based upon their shape and application, but the general components are similar.

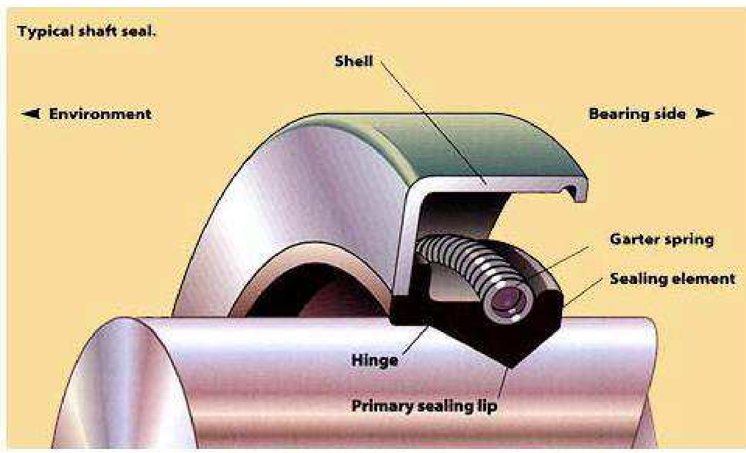

Figure 6: Side cross-section for lip seal installed

As provided in the image above the distant characteristic of a lip seal is the lip pressing against the shaft. The Garter spring in the housing of the seal keeps the lip in contact with the shaft throughout the application. Gardner spring is one of the most sensitive components of the seal. If not installed properly the garden spring can bend and render the seal useless. Housing of the shaft can be made of different materials which depend on the type of application and the operating temperatures.

Following are some types of lip seals:

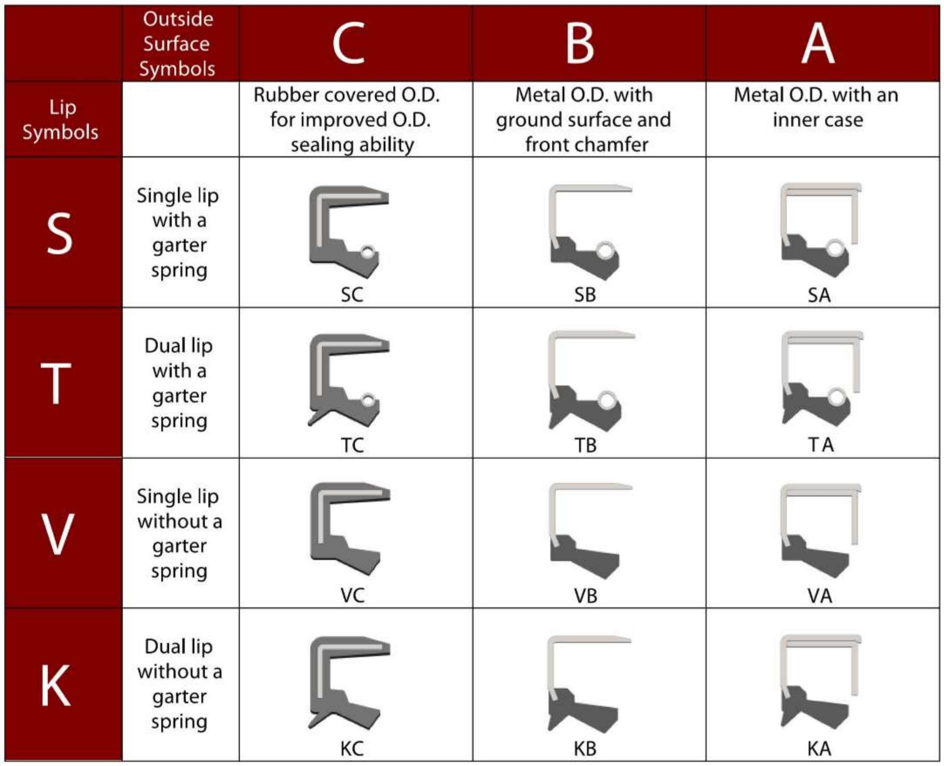

Figure 7: Lip seal types

Several variables must be considered when selecting oil seals. The following are the factors that determine the oil seal.

1. Shaft Speed
2. Temperature
3. Pressure
4. Shaft Hardness
5. Shaft Surface Finish
6. Concentricity
7. Shaft and Bore Tolerances

8. Runout
9. Lubricant

By considering factors there are ready-made manuals that can help to find the right lip seal for your application.

2.5 Bearings

A bearing is a component that provides relative motion of parts with minimal friction. Depending on design they may provide free linear motion or free rotation motion. A bearing is not only designed to reduce friction but also to absorb forces, which varies with bearing type. There are two types of forces:

Thrust Force: thrust load also known as axial load is the load applied in parallel to axis of rotation of bearing.

Radial Force: radial force is the force being applied perpendicular to the axis of rotation of the bearing.

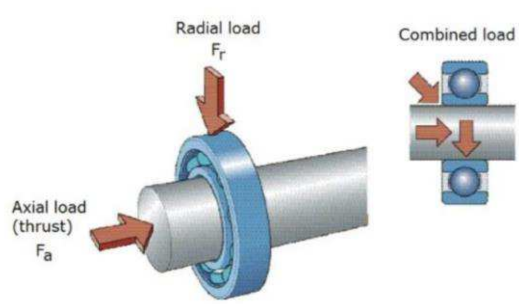

Figure 8: Loads on bearings

Generally, both these loads act simultaneously which if not considered in design can result in failure of bearing.

In terms of motion allowance, there are two types of bearings:

1. Linear bearing: They allow a linear relative motion between components while reducing friction.
2. Rotational bearing: They allow rotational motion in components while reducing friction.

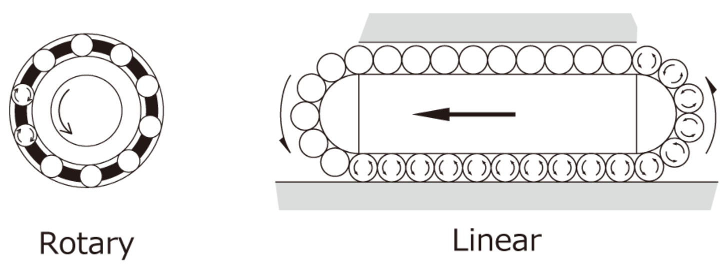

Figure 9: Linear and rotary bearing

We will be discussing rotational/rolling bearings as they are relevant to the project.

2.5.1 Rotary/Rolling Bearings

Rolling bearings have the following elements: Inner ring; outer ring; rolling component (ball/needle/rollers/cones/spheres); cage

As it is easier to roll something rather than slide it as it reduces friction and provides a smooth motion. Rolling elements in bearings carry the load without much friction as the sliding friction is replaced with rolling friction. Rolling element bearings can be divided into two major types:

- Ball bearings
- Roller bearings

2.5.1.1 Ball Bearings

Ball bearings are one of the most common types of bearings. It consists of a row of balls as rolling elements, that are trapped using cages. Ball bearings provide exceptionally low friction during rolling but have limited load-carrying capacity. This is because of the small area of contact between the balls and the races. They can support axial loads in two directions besides radial loads.

Table 3: Advantages, disadvantages, and main types of ball bearings

Advantages	• Good wear resistance • Do not need much lubrication • Low friction, thus less energy loss • Long service life • Easy to replace • Comparatively cheap
Disadvantages	• May break due to shocks

	• Can be quite loud • Cannot handle large weights
Main types	• Deep Groove • Angular Contact • Thrust • Double-row

2.5.1.2 Roller Bearings

Roller bearings contain cylindrical rolling elements instead of balls as load carrying elements between the races. An element is considered a roller if its length is longer than its diameter (even if only slightly). Since they are in line contact with the inner and outer races (instead of point contact as in the case of ball bearings), they can support greater loading.

Table 4: Advantages, disadvantages, and main types of roller bearings

Advantages	• Can take high radial loads • Easy maintenance • Low friction • Used to adjust axial displacement • Greater accuracy • Low vibrations

Disadvantages	• Noisy • Expensive
Main types	• Cylindrical • Spherical • Needle • Tapered rollers

Depending on the application, one can select the bearing type as each one of the above possesses different characteristics.

2.6 Motor Controller

A motor controller is a device or combination of devices that can control the functioning of an electric motor in a predetermined way. To control the speed and direction of a motor, motor controllers can use electromechanical switching or power electronics devices.

A motor controller may have a manual or automatic means for:

1. Starting and stopping the motor
2. Selecting forward or reverse rotation
3. Setting and controlling the speed
4. Limiting or adjusting torque
5. Safeguarding against overloading

6. Protecting against over-current
7. Monitoring of high motor temperatures

2.6.1 VFD

VFD stands for "Variable-frequency Drive." It is also known as VSD, i.e., "Variable-speed Drive." A variable-frequency drive (VFD) is a type of motor drive that is used in electromechanical drive systems to manage AC motor speed and torque by varying the motor input frequency. Electric drive systems, such as VFDs, are preferred because of stricter emission regulations and the need for enhanced reliability and availability.

A variable-frequency drive (VFD) is a component of a drive system that includes three primary sub-systems: an AC motor, a main drive controller assembly, and a operating interface.

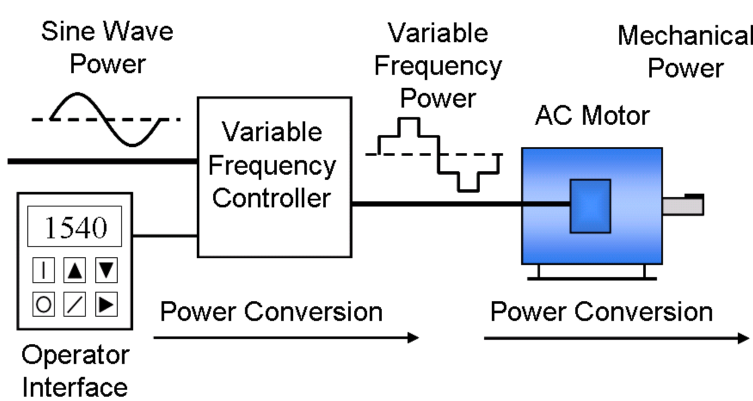

Figure 10: Variable-frequency drive and its sub-systems

Due to economic considerations, the AC electric motor utilized in a VFD system is normally a three-phase motor. A rectifier bridge converter, a direct current (DC) connection, and an inverter make up the VFD controller, which is a solid-state power electronics conversion system. The AC line input is converted to AC inverter output in AC-AC drives. The operating interface allows the operator to start and stop the motor, as well as change the operating speed and other manage parameters.

Table 5: Factors to be considered while selecting VFD

Supply voltage, i.e., 230V AC, 400V AC, etc.
Application, i.e., motor (1-phase or 3-phase)
Parameter to be controlled, i.e., speed
Regulation degree, i.e., 1:70V/1.7A, etc.
Physical dimensions, weight, and mounting place
IP rating, and thermal class

Figure 11: Model VFD System

CHAPTER 3: METHODOLOGY

3.1 Design Approach

We were provided with a crank balancer in the start of our project. With the help of the given parts, manuals, and other parameters, we needed to design a test rig that will operate to run the crank balancer as close to the original conditions as it finds in the actual tractor engine.

The givens and the considerations taken are as follows:

1. Crank balancer
   The engineering drawings, manuals, and other specifications (OEM) of the crank balancer aids in determining the test rig's approximate dimensions and the precise supports required to carry it while operating. Another factor to be considered is the crank balancer's weight.
2. Oil flow sequence (PDP incorporated to crank balancer)
   Crank balancer carries a positive displacement pump (PDP) that pumps the lubricating oil as the balancer operates. Because the pump is rotary PDP and pumps oil in a specific direction based on the inlet and outlet, the oil flow sequence examined helps determine the direction of rotation required, which in turn will determine the rotation of the motor shaft. The specifications of the pump are also provided, which aid in determining the hydraulic power required by the pump, which is to be provided by the motor.
3. Disassembled crank balancer
   Disassembled crank balancer helps in the study of the balancer's basic mechanics and operation. It also aids in the understanding of how the components are put together and function together, with the idler gear rotating the driveshaft, which then rotates the balancer weights via the balancer drive gear. The inertial loading

that the motor will have to carry is to be calculated using the physical dimensions and weights of the rotating parts. Another consideration is the helical gear which is made integral to the drive shaft. It has two holes on the front face that we will use to run the assembly through by inserting two pins.

The entire design is worked out with the constraints that we have. All of the components are designed around and according to what we have been provided, while staying within the constraints, as imposed by the physical dimensioning and working of the crank balancer.

3.2 Variable Parameters

The only thing provided to us was the crank balancer. Hence, the rest:

1. The design of sump
2. A driving mechanism
3. Crank balancer support/mounting
4. Motor support/mounting

was held variable and we had to figure it out to design the test rig.

The sump design was complicated because we needed to ensure that the sound from the crank balancer could be heard without interference from other factors like reflected waves within the sump, and while doing that, the sump must also be in such proportion that it can hold enough oil needed for the lubrication.

A gear system can be driven in a variety of ways. We needed to find a solution that could be implemented without requiring any changes to the original system. Because this project will be used in the industry, crank balancers will be replaced frequently. Furthermore, the addition of an extra part may have an impact on the analysis' quality.

The support of crank balancer is to be considered too. We had to make sure that the crank balancer was held in a way that approached its original position in the engine, and that the support was strong enough to hold it while it was operating, as we were aiming to keep conditions as near to the original as possible.

Other parameters can be configured with the motor mounting that will be put alongside to supply power to run the test equipment.

3.3 Power Calculations for Drive (AC Motor)

To drive the crank balancer using a motor, here are some calculations to find the factors affecting the drive and resulting requirements:

3.3.1 Pump Load

The hydraulic power demand (in kilowatt) is given by:

$$P_{hydraulic} = \frac{Q * H * \rho}{367}$$

Where,

Q is flowrate (in $m^3.h^{-1}$)

H is head (in meters)

ρ is density at the inlet (in $kg.l^{-1}$)

For the flowrate, taking the maximum value (as specified by the manufacturer) that the pump could deliver oil at:

$$Q = 0.9\, l.s^{-1} = 3.24\, m^3.h^{-1}$$

For the head, as the pressure difference to be generated is known, that can be converted to the head required from the pump employed.

$$\Delta P = 345\ kPa = 3.45\ bar = 42.705\ m$$

For the density, it depends on the type of oil used.

$$\rho = 872.5\ kg.m^{-3} = 0.8725\ kg.l^{-1}$$

Then, by putting all the values, we get:

$$P_{hydraulic} = 0.3353\ kW$$

Considering the pump is only 60% efficient, the shaft power required comes as:

$$P_{shaft} = \frac{P_{hydraulic}}{0.6}$$

$$P_{shaft} = 0.5589\ kW$$

As per the advice of suppliers to overdesign (typically 10-50% depending on the power and the technology of the pump) for the actual installed power of the pump, the installed power then:

$$P_{installed} = P_{shaft} * 1.5$$

$$P_{installed} = 0.8383\ kW$$

For the torque associated to this power:

$$T_{pump} = \frac{P_{isntalled}}{\omega}$$

Taking the maximum speed that the pump shaft is going to rotate:

$$\omega = 2933\ rpm = 307.143\ rad.s^{-1}$$

Then the torque required comes as:

$$\boldsymbol{T_{pump} = 2.729\ N.m}$$

3.3.2 Inertial Load

The inertial torque to be accommodated can be calculated as:

$$T_{inertia} = J * \alpha$$

Where,

J is mass moment of inertia (in $kg.m^2$)

α is angular acceleration for the drive shaft (in $rad.s^{-2}$)

For the mass moment of inertias of all the rotating parts, the figure is shown below giving the details of transmission and helping in interpretation of how the inertias are then shifted to the drive shaft:

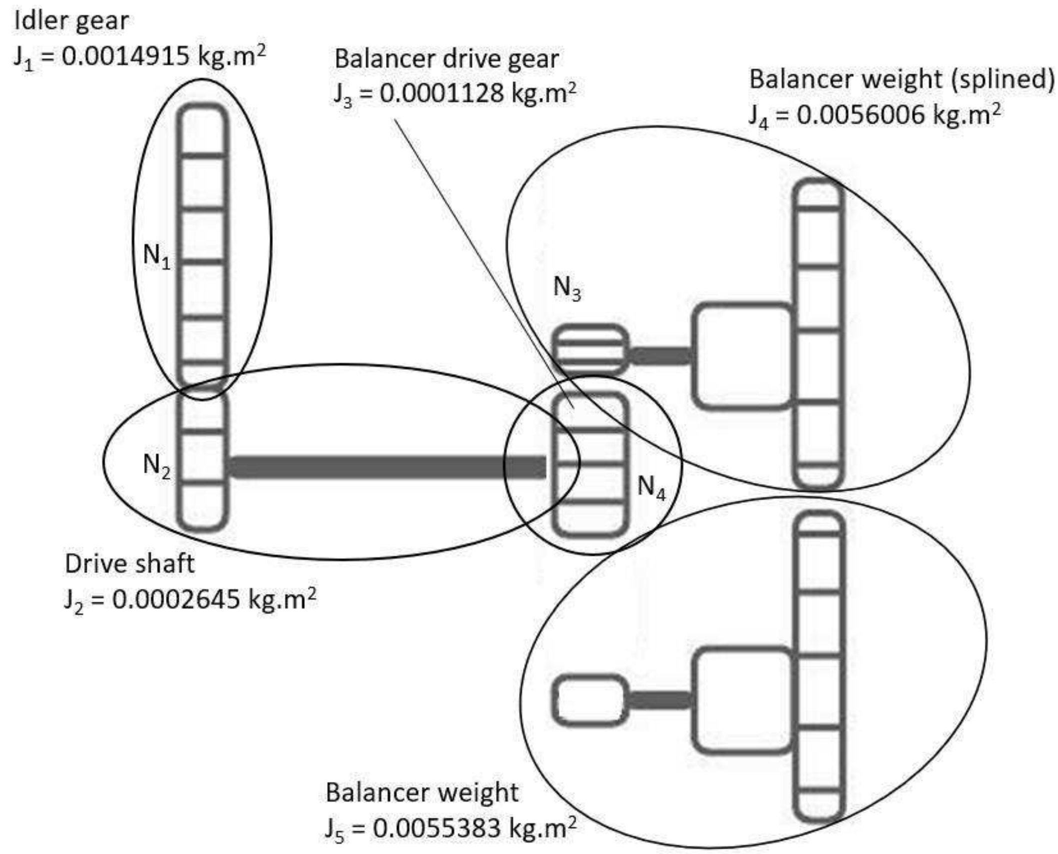

Figure 12: Schematic for transmission and inertias representation

Shifting all the inertias to the drive shaft, we have:

$$J_{eq} = J_1 * (\frac{N_2}{N_1})^2 + (J_2 + J_3) + (J_4 + J_5) * (\frac{N_4}{N_3})^2$$

$$J_{eq} = 0.02592\ kg.m^2$$

For the acceleration, assuming that we accelerate constantly to go from 0 to 900 rpm in 10 seconds, then we have:

$$\alpha = 9.425\ rad.s^{-2}$$

Then the inertial torque,

$$\boldsymbol{T_{inertia} = 0.2443\ N.m}$$

3.3.3 Total Torque and Power Requirement

The motor to be employed has to provide both the torques for the pump and inertial loading. Then the total torque will be:

$$T_{total} = T_{pump} + T_{inertia}$$

$$\boldsymbol{T_{total} = 2.9733\ N.m \approx 3\ N.m}$$

For the power, again considering the maximum speed for the drive shaft, we have:

$$\omega = 2933\ rpm = 307.143\ rad.s^{-1}$$

$$P_{total} = T_{total} * \ \omega$$

$$\boldsymbol{P_{total} = 921.429\ W = 0.921\ kW}$$

Incorporating the auxiliaries to be attached and other mechanical losses as 50 percent of the the P_{total}, we have:

$$P = P_{total} * 1.5$$

$$\boldsymbol{P = 1382.144\ W = 1.382\ kW}$$

3.4 Design Phase

The design phase comprised of several stages with reiterations to reach the final manufacturable, robust, and working design. The design iterations and the final design are discussed below:

3.4.1 First Design

The first design of the test rig assembly was unique because of its mounting mechanism. The idler gear on the crank balancer was used to drive it. The concept assembly is shown in the figure:

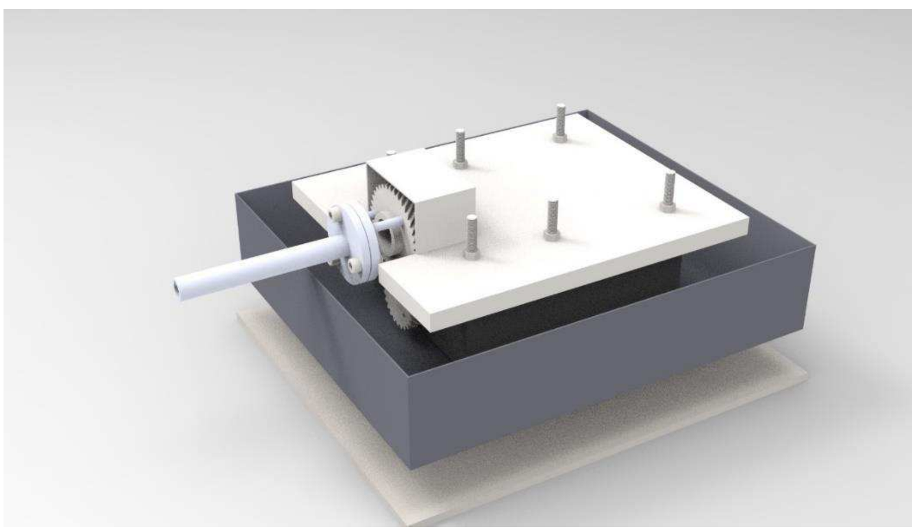

Figure 13: Idler Coupled Assembly

3.4.1.1 Idler Coupling Mechanism

In this mechanism, the idler gear was modified by drilling holes in it. Using studs through those holes, the gear was coupled with the motor shaft using a flange coupling. As anticipated, this would easily drive the crank balancer with lower speed (rpm) provided by the motor.

3.4.1.2 Design

The design of this mounting mechanism had three main components: idler gear with holes drilled, flange coupling, and studs. The studs would help in connecting the coupling with the idler gear by fastening nuts. The flange would further be connected with the shaft and this will complete the drive mechanism.

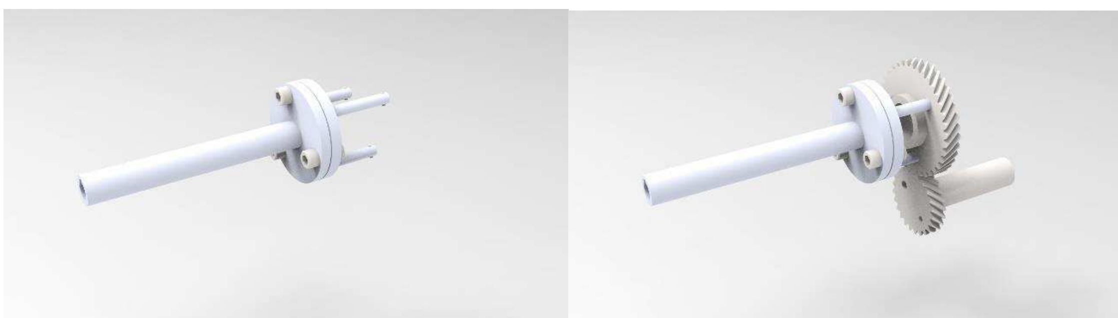

Figure 14: Idler Coupled Mechanism Sub-Assembly

3.4.1.3 Setbacks

This design had the following setbacks due to which it was ruled out:

Feasibility Issues

This design had some feasibility issues. Drilling holes through the idler gear was not an easy thing. For multiple testing of different crank balancers, we had to machine their idler gears, or we had to disassemble their idlers to replace with the one with holes. But there was also not enough space on the gear to properly fasten the studs and tighten them for a proper drive.

Functional Issues

There was a functional issue of alignment. The studs, if not in perfect alignment, could misalign the drive and thus cause considerable vibrations. Those vibrations would ultimately reduce the efficiency of the drive and the rig would not function as expected.

3.4.2 Second Design

The first design of the test rig had motor and crank balancer in fixed position. While, to drive the small helical drive gear, we designed a spring-loaded mechanism. The whole concept is shown in the figure below:

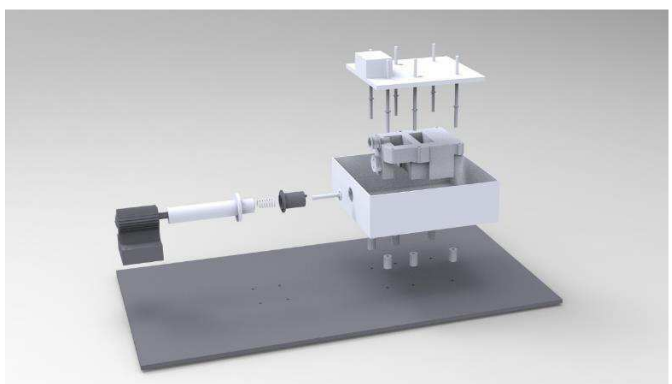

Figure 15: Spring Loaded Drive Assembly

3.4.2.1 Spring-loaded mechanism

The spring-loaded mechanism would have pins at its outer face. Those pins would mate with the existing holes in the drive gear. We could drag the pins out of the holes in lateral direction anytime. This would be very helpful in the crank balancer changeovers for multiple testing.

3.4.2.2 Design

There is a flanged cylinder directly mounted with the shaft from behind. A hollow cylinder is the mating part of the solid flanged cylinder. A spring comes in between these mating parts. To keep the spring at its place and restrict the hollow mating cylinder, a customized bolt would pass through the center of the hollow cylinder, through the spring, and would get screwed in the solid flanged cylinder. The hollow cylinder would have pins at its outer face. The customized bolt would have holes on its head to pass the pins through it. Moreover, the hollow cylinder would be able to move laterally along the solid flanged cylinder. The hollow cylinder would not rotate freely around the solid flanged cylinder.

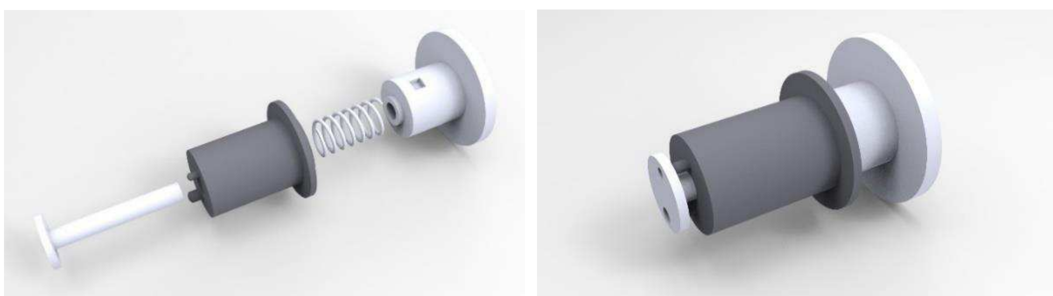

Figure 16: Spring Loaded Mechanism Sub-Assembly

3.4.2.3 Setbacks

This design had the following setbacks due to which we could not manufacture it:

Design Complexities

We came up with multiple concepts for our spring-loaded mechanism and even the most basic one had design complexities in it. For instance, mounting a spring inside a hollow cylinder and then fastening that hollow cylinder with the main shaft. Restricting the rotation to transmit torque and allowing lateral movement to drag the spring – all of these constraints were not easy to achieve. Our goal was to get the job done with a minimal mechanism.

Limited Resources

To achieve an accurate sub-assembly of spring-loaded mechanism, we would need close tolerances. Because to mate the parts with design patterns, if all the entities of a pattern would have some range of tolerances (let's say 0.13 mm, tolerance offered by CNC lathe), the other mating part would also have such tolerances. Thus, achieving the right fit would be difficult.

Functional Issues

We foresaw some functional issues as well. For instance, dragging the drive pins would have been easier using the spring-loaded mechanism but restricting the pins was a challenge. Although we thought about introducing levers to keep restrict the pins at compressed position, but it would clash with the whole assembly.

Overall, we found out that it was not practical to use a spring-loaded mechanism.

3.4.3 Third Design

After ruling out the first design, we came up with the idea of sliding the crank balancer into the pins. The crank balancer would have six pillars of various sizes under it, to keep it horizontal, depending on its hole-references. The assembly along with the exploded view is shown:

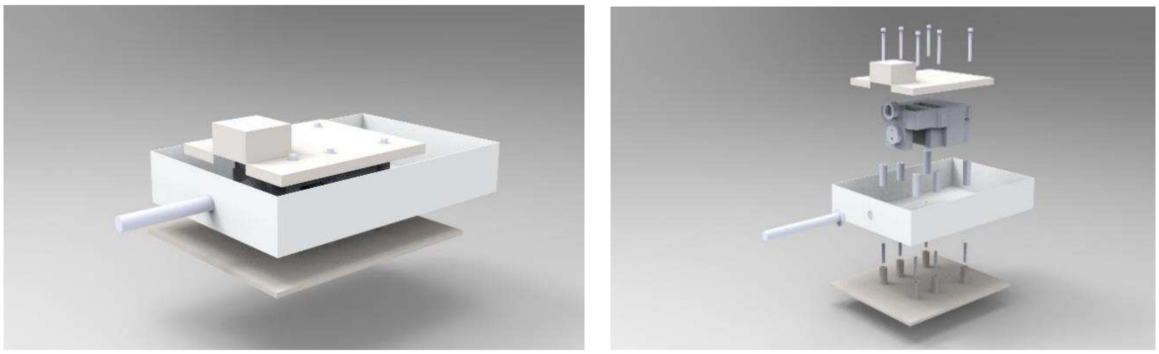

Figure 17: Top Mount Concept Assembly + Exploded

On the base plate, there were six spacers supporting the sump tank. The sump tank had six pillars of various sizes, depending on the bottom-hole references of the crank balancer, inside. The shaft with pins at its tip-face would stay stationary. We would lift the crank balancer and slide it over the pillars into the pins to mate it with the drive. Then, using bolts, we would place the top cover and fasten the assembly from the top.

3.4.3.1 Setbacks

This design had the following setbacks that became the reason of ruling it out:

Functional Issues

The bottom-hole references of the crank balancer frames were not flat for the front two holes. Thus, sliding linearly was not possible. Moreover, after random sampling of multiple crank balancer frames, we found out that the flat surface area at the bottom references had varying tolerances. Thus, if we had improvised for the non-flat areas for a specific crank balancer, it would not have been possible for us to run multiple crank balancers on this design.

3.4.4 Final Concept

After carefully ruling out the unmanufacturable parts/designs, we sifted down to our final design. The general schematic diagram of the design is shown below:

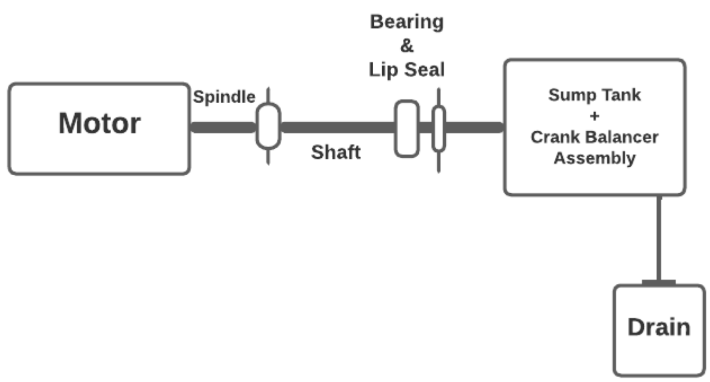

Figure 18: Final Design Schematic

The motor has a sliding mechanism at the base. The male spindle of the motor has an extruded key. The shaft has a corresponding female pattern to mate the spindle – thus it couples there. There is a lip seal mounted in a flange and the shaft goes through the lip seal. The flange and lip seal always stay at a fixed position on the shaft, while the shaft itself can laterally move with the sliding mechanism of motor. There is an open section on the side of the sump facing the shaft. So, the flange is fastened to that side with a silicon gasket inside. The shaft has internal threads on the tip side. Two customized fabricated pins are screwed in those threads. These pins are used to drive the small helical gear through the holes on its front side. This gear, on the other end, is mechanized with the pump. The sump tank has metal spacers under it. Six custom designed studs pass through the base plate, spacers, sump base, crank balancer, and the top cover respectively. This overall assembly remains static.

3.4.4.1 Assembly

The CAD model assembly of the design with actual assembly is shown as follows:

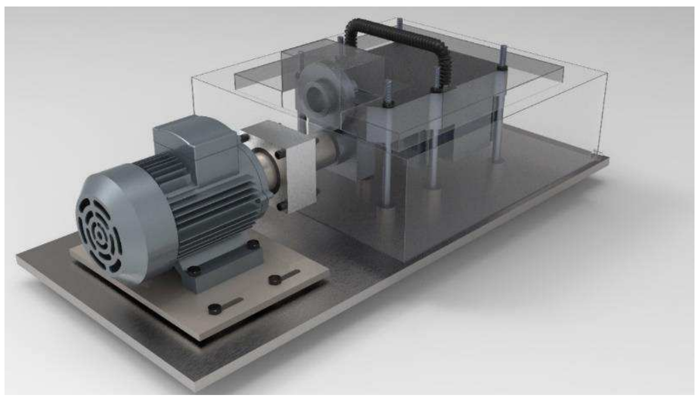

Figure 19: Final Design Assembly (CAD)

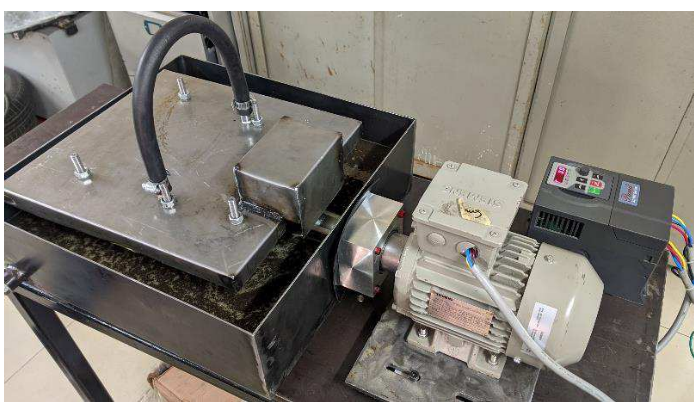

Figure 20: Final Design Assembly (Actual)

3.4.4.2 Exploded View

The Exploded view below shows the details representation of the whole assembly.

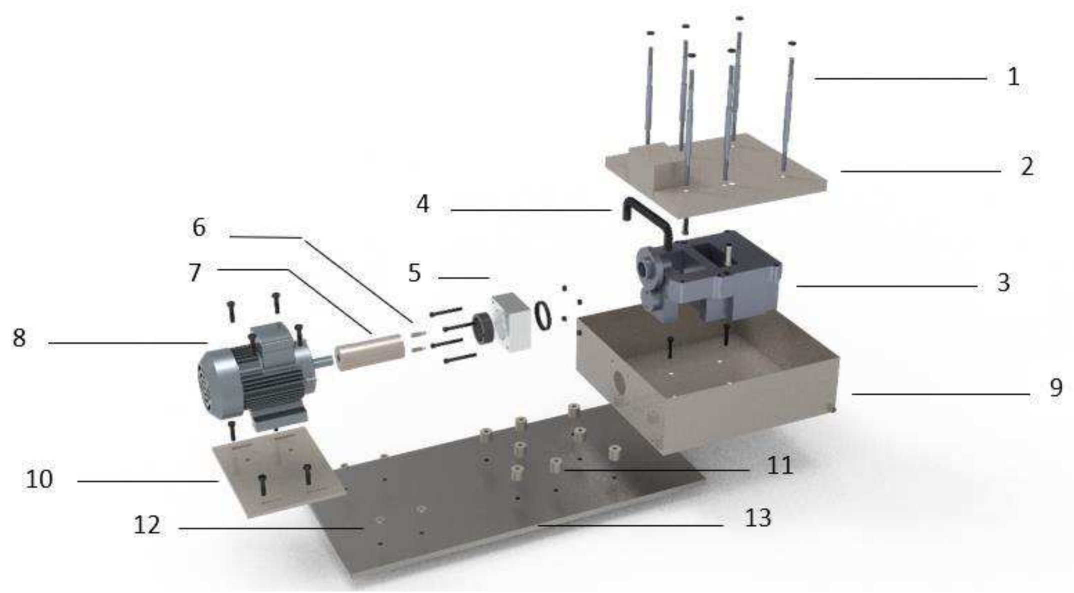

Figure 21: Final Design Assembly (Exploded)

3.4.4.3 Components

The components to be manufactured are labeled as follows:

1. Studs
2. Top Cover
3. Crank Balancer
4. Oil Bypass
5. Bracket for Lip Seal
6. Drive Pins
7. Shaft

8. Motor
9. Sump Tank
10. Motor Base
11. Sump Spacers
12. Motor Spacers
13. Base Plate

The details of the labeled parts is given in the following section:

1. **Studs**

Material: Mild Steel

Dimensions: Length: 215.6 mm, Maximum Diameter: 14 mm

Six studs (supporting columns) pass through the baseplate, spacers, sump, crank balancer, and the top cover to hold them all. To sit the crank balancer, a thick rod is taken and machined for bolts, crank balancer holes, and spacers, while the middle part is left thick to let the crank balancer sit on it. The studs are threaded at both ends for fastening.

Figure 22: Customized Supporting Stud

2. **Top Cover**

Material: Sheet Metal

Dimensions: 395 mm x 350 mm x 16 mm

The top cover is also a sheet metal part that restricts oil splashing inside the sump. It has six holes to pass the studs while two holes for the oil channel of the crank balancer's pump. There is an additional elevated shed with dimensions 94 x 128 x 64 mm that cases the idler gear and restricts its splashing.

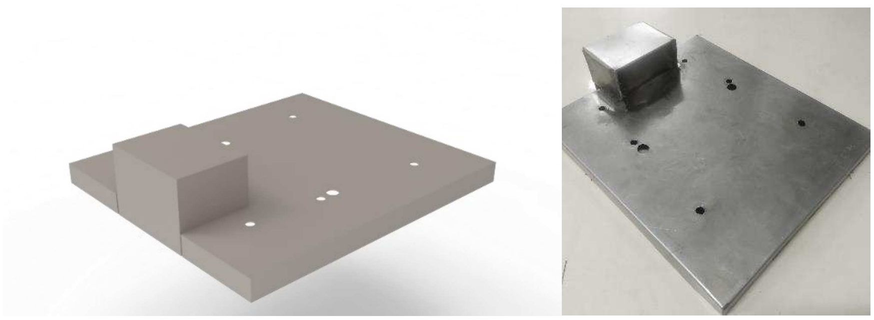

Figure 23: Top Cover

3. **Crank Balancer**

Material: Grey Cast Iron GG 25 (BS Grade 260)

Dimensions: 290 mm x 220.70 mm x 125.60 mm (maximum)

The crank balancer of the tractor model Massey Ferguson MF 300 is mounted on studs and is driven by the motor to understand the noise coming out of it.

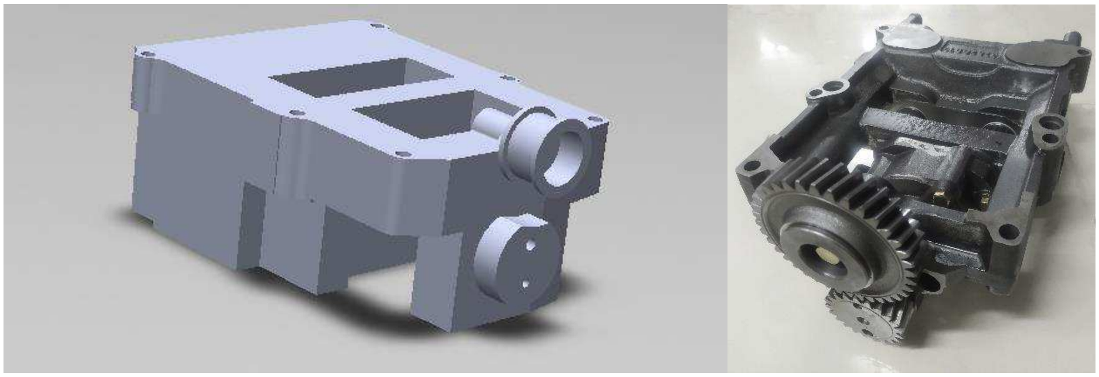

Figure 24: Crank Balancer

4. **Oil Bypass**

Material: Rubber Pipe with Metal Clamps, MS Sockets

Dimensions: Socket Diameter: 15 mm

The positive displacement pump supplies oil to the upper part of the engine. Thus, there is a supply point and a return point of oil on the crank balancer. To bypass that circuit, a pipe with sockets is used that is mounted on the holes and supported by the top cover. Thus, the oil goes through the supply line at a maximum pressure of around 3.45 bar and returns through the second hole via same pipe.

Figure 25: Bypass Socket and Pipe

5. Bracket for Lip Seal

Material: Aluminum

Dimensions: 140 mm x 140 mm x 55 mm

A bracket is holding the lip-seal to seal the oil from the front wall. There is a roller bearing installed in the bracket to keep the shaft aligned. This portion always stays on the shaft at a fixed position. The whole sub-assembly can move laterally with the motor sliding mechanism. This portion further is fastened with the sump tank with a gasket in-between when the Test Rig is functional.

Figure 26: Aluminum Bracket with Seal and Bearing

6. Drive Pins

Material: Mild Steel

Dimensions:

- Small Pin; Length: 33 mm, Front Diameter: 6 mm
- Large Pin; Length: 35.5 mm, Front Diameter: 8.5 mm

The pins are designed and fabricated to drive the crank balancer. These pins are inserted into the holes of drive gear of the crank balancer. These are the most critical parts of the whole assembly.

Figure 27: Drive Pins

7. Shaft

Material: Mild Steel

Dimensions: Length: 145 mm, Diameter: 55 mm

The shaft has a hollow female section to mate the motor's spindle (Diameter: 18.50 mm, Depth 43.2 mm). There is a cut-extruded slot for the keyed part of the motor spindle. On the opposite face, it has threaded holes (Hole 1: Diameter: 8.85 mm, Hole 2: 6.05 mm) to screw the pins that are driving the crank balancer.

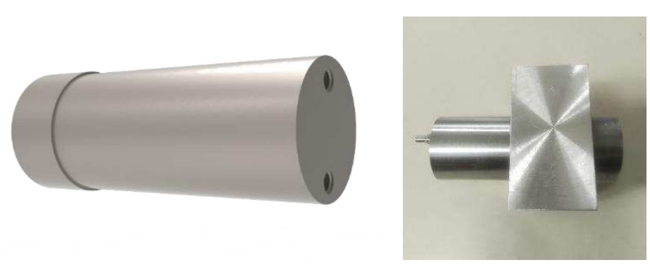

Figure 28: Drive Shaft

8. Motor

The motor is driving the crank balancer to help us analyze its acoustics. It is a Siemens series 1LE0102, 4 pole having IP55 for degree of protection. The specification are given:

Table 6: Motor Specifications

Rated Output	0.75 kW
Rated Speed	1405 rpm
Rated Torque	5.1 Nm
Rated Current	1.87 A
Rated Power Factor	0.72
Starting Current	4.8 A
Moment of Inertia	0.0019 kgm^2

According to our calculations, the motor with the above-mentioned specifications can easily drive the crank balancer at our desired rpm.

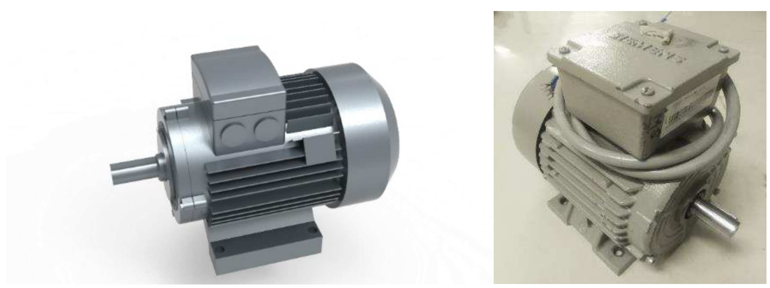

Figure 29: Motor

9. Sump Tank

Material: Sheet Metal

Dimensions: 483 mm x 458 mm x 154 mm, Sheet Gauge: 12

The sump tank is a sheet metal part containing the oil with the crank balancer submerged inside the oil up to in-between the dedendum and center of the idler gear. The base of the sump has eight holes for six studs and two bolts. The side facing the shaft has an opening of 60 mm to let the shaft pass.

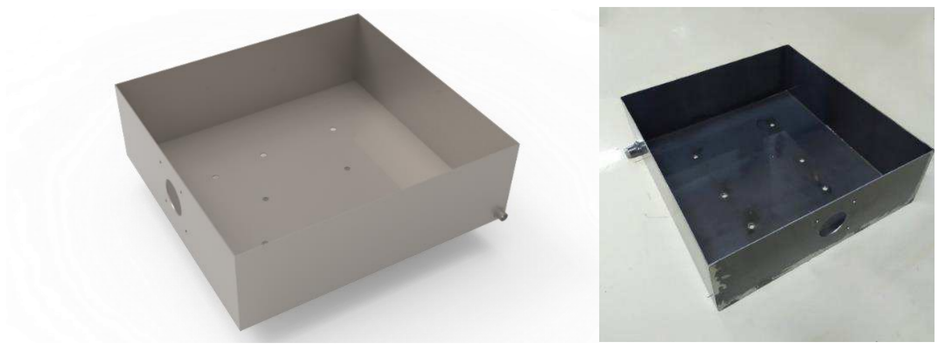

Figure 30: Sump Tank

10. Motor Base

Material: Mild Steel

Dimensions: 290 mm x 176 mm x 10 mm

The motor base mechanism comprises of a plate having 4 slots and 4 holes. The holes are to mount the motor on the plate while the slots are bolted with the main base plate to let the motor slide for adjustment and fitting.

Figure 31: Motor Base Plate

11. Sump Spacers

Material: Mild Steel

Dimensions: Length: 30 mm; Diameter: 30 mm

The spacers with through-holes are placed and mounted on the very top of the base plate. These spacers are to adjust the elevation of the sump. Six spacers are parts of the studs' assembly supporting the crank balancer, while two spacers are solely supporting the sump tank to keep at its place.

Figure 32: Sump Spacers

12. Motor Spacers

Material: Mild Steel

Dimensions: Length: 23 mm; Diameter: 35 mm

These spacers are designed and fabricated to adjust the elevation of motor according to the crank balancer. They come right under the motor base slots and help in sliding the motor as well. They also provide a space to the motor bolts under the motor base plate.

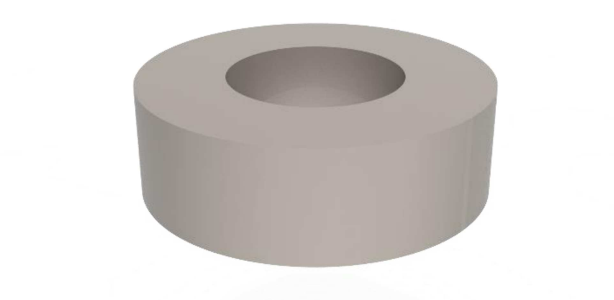

Figure 33: Motor Spacers

13. Baseplate

Material: Mild Steel

Dimensions: 1483.42 mm x 783.26 mm x 15 mm

The base plate is a support for the whole rig and is mounted on the table frame. It has 10 holes as per the design. The sump & crank balancer assembly is mounted on the base plate using studs while the motor base is mounted using bolts over the base plate.

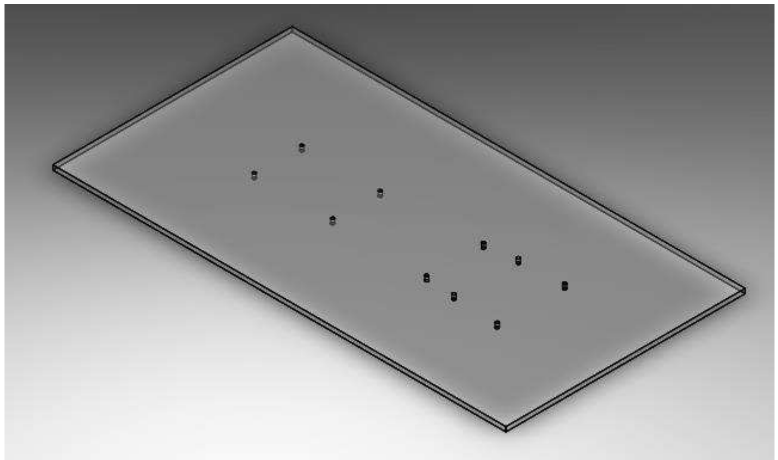

Figure 34: Base Plate

3.4.4.4 Additional Parts

1. **Drain**

Material: Mild Steel

Dimensions: Length: 12.7 mm, OD: 25.4 mm

The drain is important while changeovers. The oil is drained out of the sump while assembling and disassembling the crank balancer off the rig. A mild steel bushing is welded with the sump near the base position. There is a stopper to restrain the oil. The drained oil goes to another container.

Figure 35: Socket and Stopper

2. **Gasket**

Material: Rubber

Dimensions: As per the requirements.

Rubber sheet is used to make customized gaskets to seal the oil at the base as well as on the wall opening.

Figure 36: Gasket Rubber

3. **Nuts**

Material: Galvanized Iron

Size: M10 x 1.5, M8 x 1.25

Twelve basic nuts with internal diameter of 10 mm and pitch 1.5 mm are for the balancer assembly. Six of them are fastened under the baseplate while the remaining six are fastened over the top cover. Eight additional nuts: 4 with internal diameter 10 mm and pitch 1.5 mm while remaining 4 with internal diameter of 8 mm and pitch 1.25 mm are used to fasten the motor slider mechanism.

Figure 37: Nuts

4. **Lip Seal**

Material: Silicone

Size: Inner Diameter: 55 mm, Outer Diameter: 70 mm, Width: 10 mm

The lip seal used in the bracket to seal oil from the front wall is *single lip seal* with a garter spring. Referring to the table mentioned in the Literature Review, this is the SC type seal.

Figure 38: Lip Seal

5. Bearing

The test rig design demands for a bearing with close tolerances, negligible vibrations, high radial and axial load, and minimum noise. In this scenario, a roller bearing is the most suited option. After a bit of research, needle roller bearing is finalized. The bearing used is NKI 55/25 TN from SKF bearings catalogue.

Table 7: Bearing Specifications

Internal Diameter	55 mm
External Diameter	72 mm
Width	25 mm
Dynamic Load	46.8 kN
Static Load	110 kN
Fatigue Load	13.5 kN
Reference Speed	6700 r/min
Limiting Speed	7500 r/min
Weight	0.26 kg

This bearing fits all the design parameters and fits perfectly. The figure shows its appearance and provides more insight to it:

Figure 39: Needle Roller Bearing

6. Variable Frequency Drive

The AC inverter drive is used to regulate and control the motor speed. The series of this device is SD100. Following table shows the specifications of the device:

Table 8: Motor Controller Specifications

Series	SD100
Model	01D5-43 (1.5kW – 3Phase AC 380V)
Power In	3 x 380V-480V AC at 50/60Hz
Power Out	3x 0V- Vin AC at 0-400 Hz
I/O Board	5 digital, 2 analog inputs 2 digital,2 relay,2 analog outputs -10 to 10V
IP Rating	20
Power dissipation	49 W
Net Weight	1.3 kg

The figure shows the VFD used to control and regulate the motor:

Figure 40: VFD (SD100)

3.5 Materials

Material selection is one of the most important aspects of any project. A material must adhere to all design specifications while still being cost effective. Market research is one of the most significant aspects of reducing down the selection of materials. You cannot use material that is not accessible on the market in a project because it will require you to make unfavorable alterations afterwards. Even if you have a great design, using the wrong material might lead to project failure.

Table 9: Factors to be considered while choosing material

Physical properties
Design requirements
Machinability
Cost
Availability

We start with design specifications and market research. The design requirements include strength, resilience, and can bear significant fatigue.

The materials potentially we can use for the project scrutinized after the consideration of the required properties and market availability are:

1. **Aluminum**

In terms of design parameters, aluminum is a good option. It offers sufficient strength, can be easily machined, and is lightweight, i.e., aluminum parts may be handled without difficulty. The major downside of adopting this material is the cost.

2. **Stainless Steel (SS)**

   Stainless steel is one of the best materials to consider when working on such projects since it not only has a longer life under load, but it is also corrosion free, meaning it is not affected by the environment, and it is more resilient. The market provides material that is simple to use and comes at a reasonable price. The sole disadvantage of adopting this material is the difficulty in machining and processing.

3. **Mild Steel (MS)**

   Mild steel can be considered as a tamer form of stainless steel because it can only provide a fraction of the attributes that stainless steel can, such as strength and resilience. MS is known for its low cost and machineability. MS and its products are the most often used materials on the market.

Table 10: Material Selection Criteria

Properties	Materials		
	Aluminum	SS	MS
Strength		✓	✓

Weight	✓		
Corrosion	✓	✓	
Workability	✓		✓
Welding		✓	✓
Cost			✓
Availability	✓		✓

(The assessment of the properties in the table is based on the project's requirements, as well as on a comparative basis) .

3.6 Finite Element Analysis

Every design has some critical parts that are likely to fail and considered as the weakest links of a chain. In this test rig, the drive pins are very critical. To ensure that they are in safe limits, here is a finite element analysis for these pins:

3.6.1 FEA with Full Assembly

The analysis is performed on the full assembly while focusing on the pins. Static analysis was chosen assuming that pins are on the verge of moving. The Von Mises plot of the stresses displays no considerable stresses when compared to the yield strength of mild steel that is 250 MPa. Moreover, the supporting columns (studs) also did not show any failure.

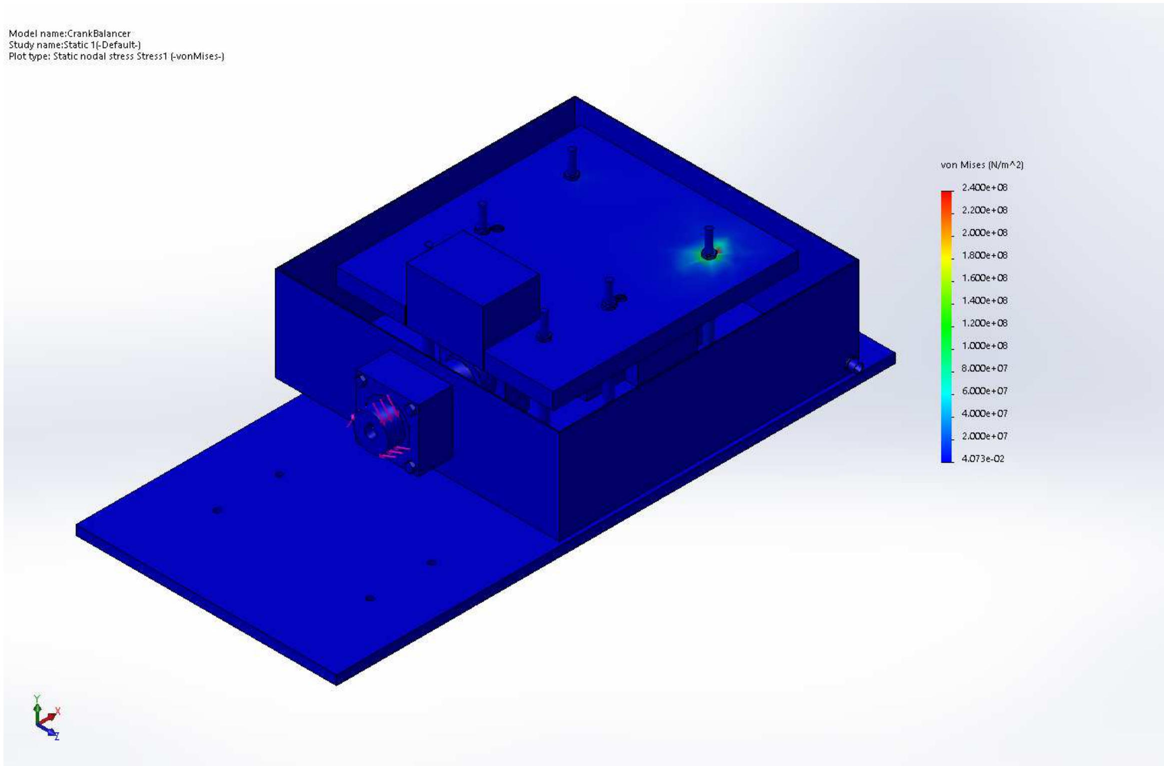

Figure 41: Von Mises (with Assembly)

The displacements caused under stresses are shown in the figure using a sectioned view. It can be seen quite clearly that the whole rig is in place with negligible displacement under the stresses. The pins also do not show any considerable displacements rather they stay within the holes of their drive point.

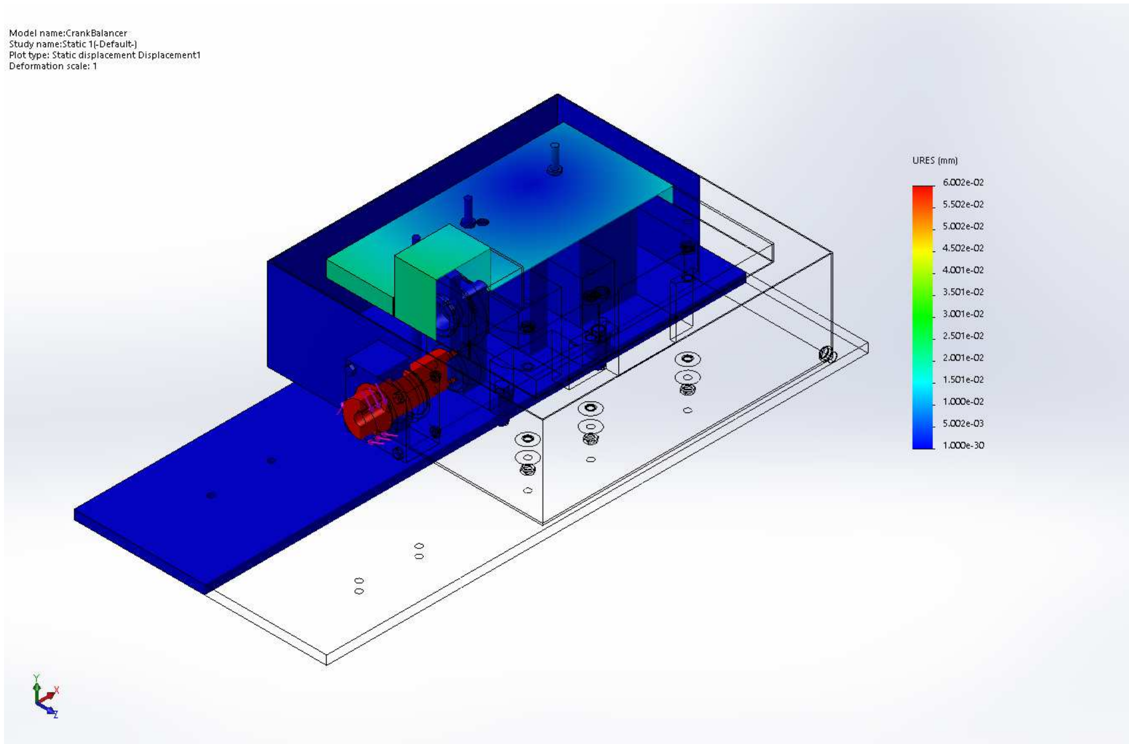

Figure 42: Displacement (with Assembly)

3.6.2 Drive Pins Closeup

A closer look on the drive pins is discussed in this section. As the pins are being used to drive the crank balancer, our concern is with two types of stresses: shear stress and bearing stress. Shear stress is due to the torsion while bearing stress is due to the contact that the pins make with the inner surface of the holes.

Considering the rated torque of 5.1 Nm, the forces on the small and large pins are 1700 N and 1200 N respectively.

The theoretical shear stresses for small and large pins are 60.13 MPa and 21.15 MPa respectively. Moreover, the theoretical bearing stresses on small and large pins are 141.66 MPa and 47.05 MPa respectively.

Comparing these to the shear strength of mild steel that is 345-525 MPa, the design is quite safe. Moreover, the yield strength of mild steel (250 MPa) also depicts that the design is in a safe range.

The Von Mises plot of the FEA analysis in the figure also verifies that the stresses generated in the pins are in a safe range.

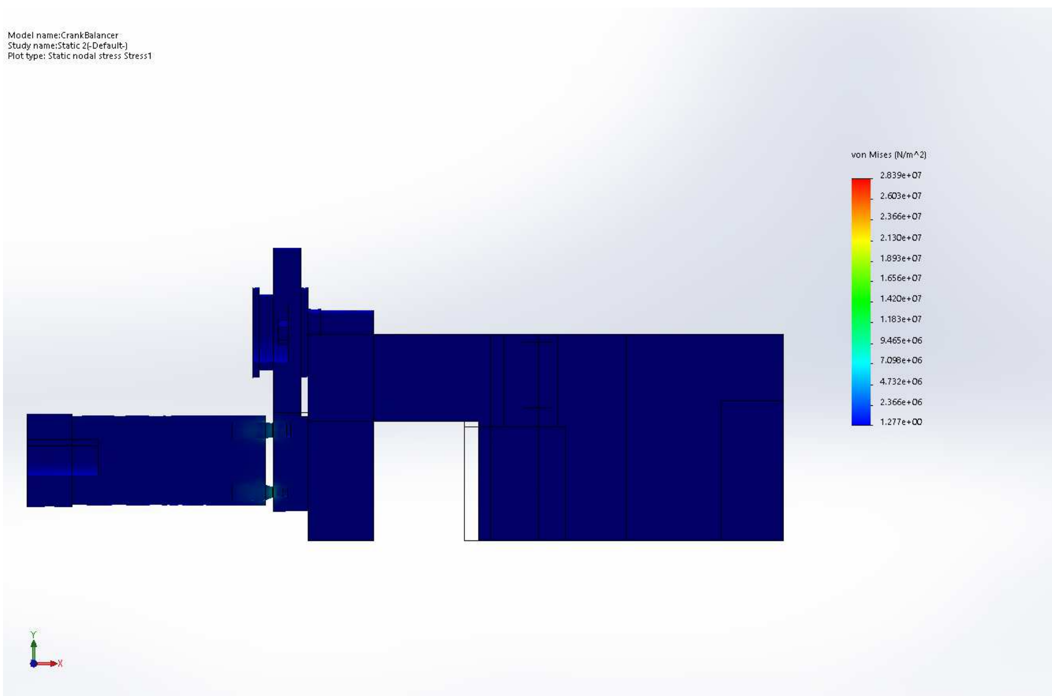

Figure 43: Von Mises (Close-up)

The displacement plot shows negligible displacement in the targeted areas. The pins stay within the holes under the stresses generated. That means the displacements are also in a safe range.

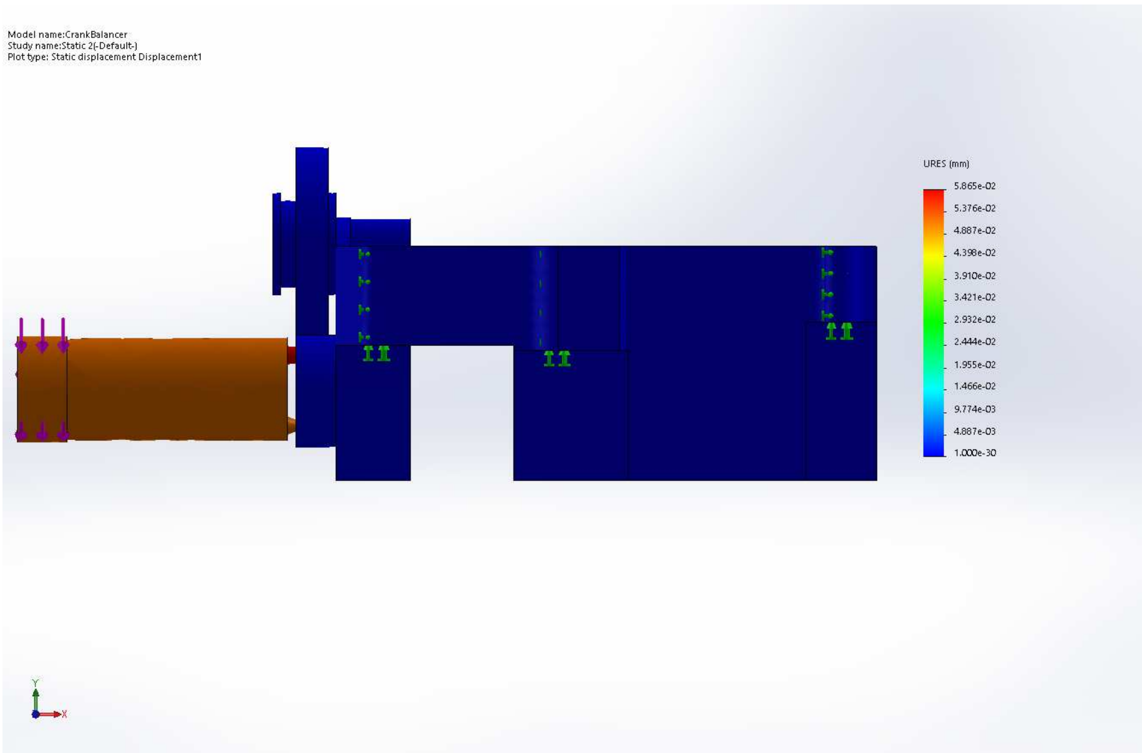

Figure 44: Displacement (Close-up)

3.7 Manufacturing Processes

Many of our components will be produced in job shops is they require a certain amount of accuracy. Following is a list of components along with their manufacturing processes.

Table 11: Manufacturing Processes for the Components

Sr No.	Component	Material	Process	Drawing
1	Sump	Sheet metal (Galvanized Iron)	Bending Welding	Appendix
2	Studs	Mild steel	Threading Welding Machining (lathe)	Appendix
3	Top cover	Sheet metal (Galvanized Iron)	Welding	Appendix
4	Base plate	Mild steel	Milling	Appendix
5	Shaft	Mild steel	Drilling	Appendix

			Shaping	
6	Spacers	Mild steel	Drilling	Appendix
7	Lip seal Bracket	Aluminum	Milling	Appendix
8	Motor Base	Mild steel	Milling	Appendix

CHAPTER 4: RESULTS AND DISCUSSIONS

This chapter covers the observations from our test runs. Moreover, the performance and result of the test rig are discussed.

4.1 Testing

The test results are dependent on many of the design parameters. The test output is expected to be in close correspondence to the design. Following steps are important in testing:

4.1.1 Design Verification

The test rig should perform as envisioned by the designer. Thus, design verification is important. This can only be verified by running the machine. The test rig is designed to run the crank balancer at a speed of 3000 rpm with a motor power of 0.75kW. The part should run smoothly without any external vibrations to ensure a reasonable testing environment for the crank balancer. The sump tank should not leak to spill any oil outside the test rig. The drive pins should work fine without yielding and the supporting columns (studs) of the balancer should not buckle.

The design process is reviewed and reiterated in case of any deviations from the anticipated results.

4.1.2 Design Validation

When the design is verified successfully, it should be validated. Verified results are evaluated further in this phase. The goal is to reach as close as possible to the requirements of the user. Readjustments are made until the design is validated. But if it does not work, the design process should be reiterated to successfully validate the design.

4.2 Test Outcomes

The test rig for acoustic analysis of the crank balancer showed the following outcomes in the test runs:

4.2.1 Running Test Rig

The test rig is run using the motor controller at variable speeds. The response of the rig is observed. The shaft has a slight misalignment that causes problem at higher rpms. The load from the shaft gets transmitted into the sump and the crank balancer undesirably. The test rig vibrates, and the speed of the crank balancer slows down at higher rpms provided by the motor.

4.2.2 Test Rig Integrity

The test rig shows great integrity when we run it. The weakest points are the drive pins and the supporting columns (studs). The pins do not yield and perfectly drive the crank balancer. The supporting columns of the crank balancer perfectly hold it. Thus, the test rig is statically in correspondence with the design anticipations.

4.2.3 Oil Control and Leakages

There is around 16 liters of oil in the sump tank. The sump tank, on the other hand, has 6 holes in the base while 5 on the front wall. The test rig when assembled is sealed perfectly to not let even a single drop of oil out while static or running. The drain of the oil is controlled through a socket welded on the side of the sump tank.

The oil channel of the pump is bypassed using a pipe, clamps, and a customized socket. The oil is thus perfectly bypassed without any leakage and spillage.

The gears of the crank balancer cause oil splashing when run in oil. But the top cover designed to control splashing do not let any of the oil spill out of the tank.

CHAPTER 5: CONCLUSION AND RECOMMENDATION

This chapter covers a comprehensive conclusion obtained from the test results and experiences. Several areas of improvement are noticed and thus, provided in the recommendations section.

5.1 Conclusion

The design process consisted of countless iterations to reach a manufacturable, robust, and working design. The confidence in the final design derived to fabricate the test rig accordingly. The assembly was very challenging with a lot of fastenings, adjustments, and alignments. Once assembled and run for testing, the test rig showed exceptional results in a lot of areas. Although there are areas of improvement that can rule out the small issues and this test rig will become fully able to perform the acoustic analysis of crank balancers at any speed but considering it as a novel approach of thinking out of the box and designing something new, this project is a success.

5.2 Recommendations

Considering the areas of improvement and future prospects of this test rig, following recommendations are given:

5.2.1 Design Improvements

Following are the design improvements that can be implemented in further design iterations:

1. There should be a flexible coupling incorporated on the shaft to rule out the minor alignment issues.

2. The pipe used for oil bypass should be replaced by connecting pipe with internal threads at the terminals.
3. The front wall of the sump should be made thicker to handle the shaft-bracket assembly well.
4. The bracket holding the shaft should be supported from the base on the base plate rather than the sump wall to avoid vibrations transmission.
5. There should be undercut groves in the supporting columns to incorporate O-rings and ensure better oil sealing.

5.2.2 Functional Improvements

For future prospects and iterations of this prototype, following functional improvements are suggested:

1. The motor specifications can be improved to provide more power and run the part at its maximum rpm range.
2. The changeover time can be improved by improving the assembly and fastening processes.
3. The test rig supporting bench should be fixed and the motor should have dampers at the base to damp vibrations.
4. Tandem lip seals can fully eliminate the oil leakage caused by the shaft hole.

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APPENDIX I: ENGINEERING DRAWINGS OF PARTS

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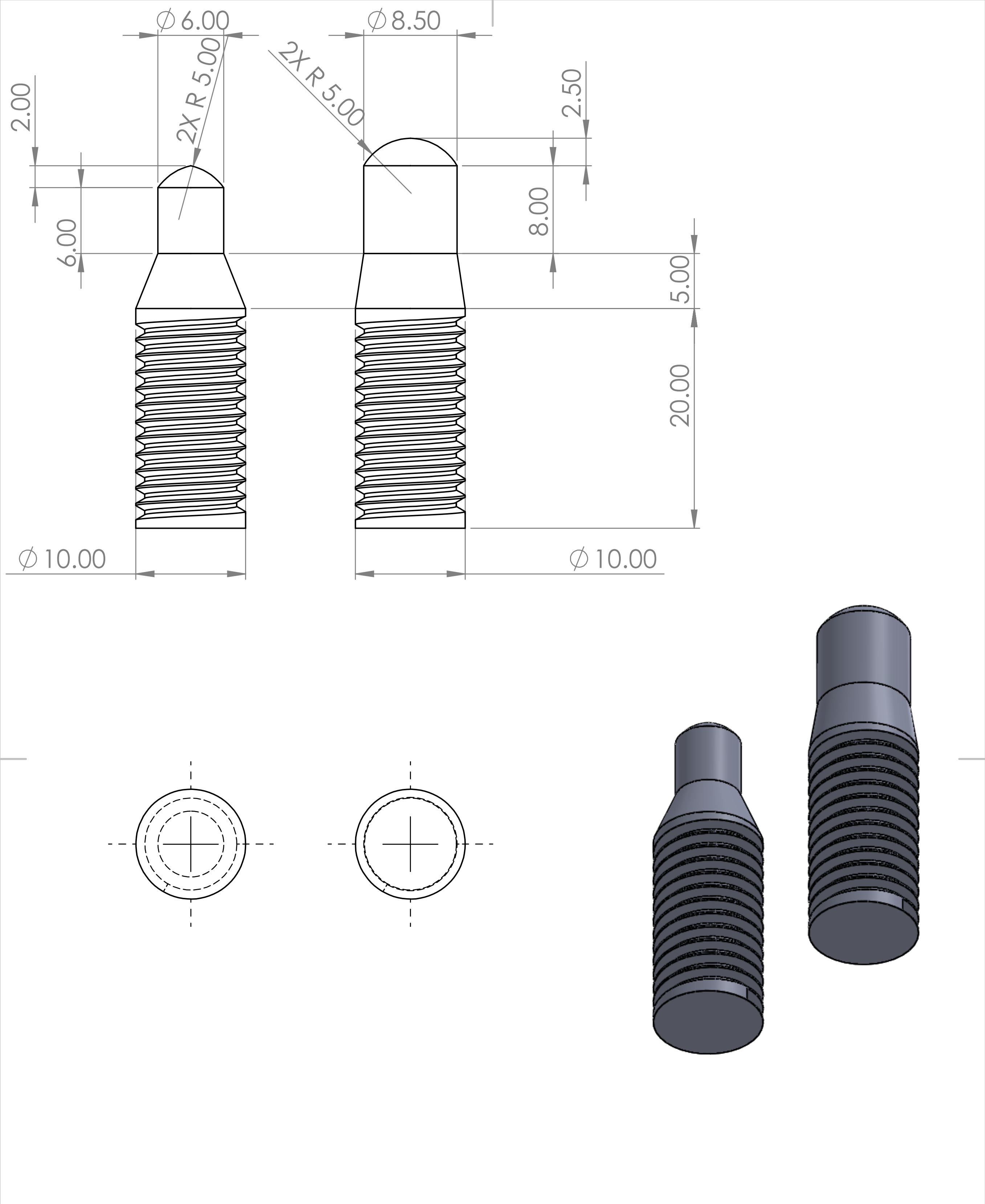


UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES: ± 0.05 mm

FINISH: Machined

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: Pins

MATERIAL: Mild Steel

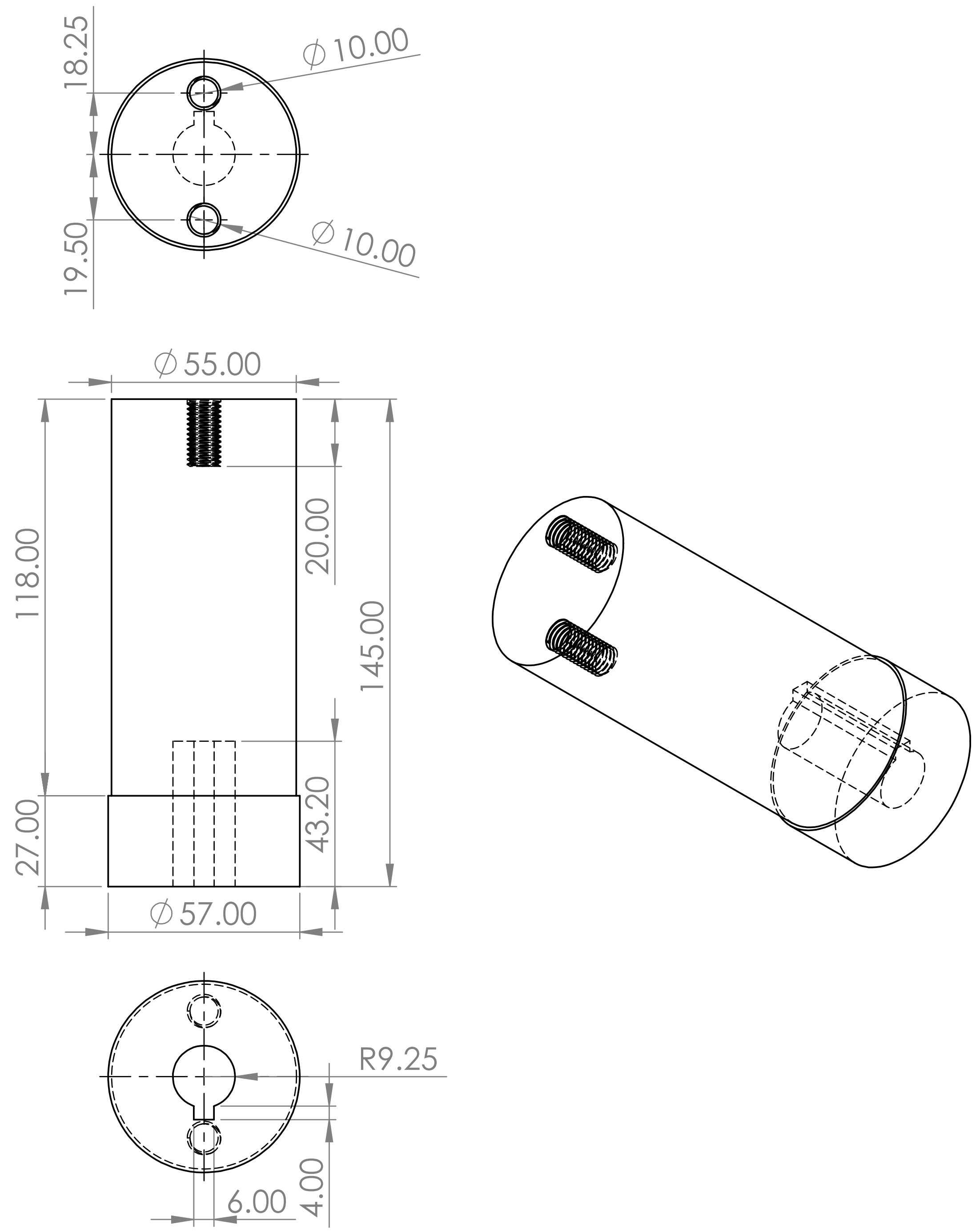
DWG NO.

A4

WEIGHT:

SCALE:2:1

SHEET 1 OF 1



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DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

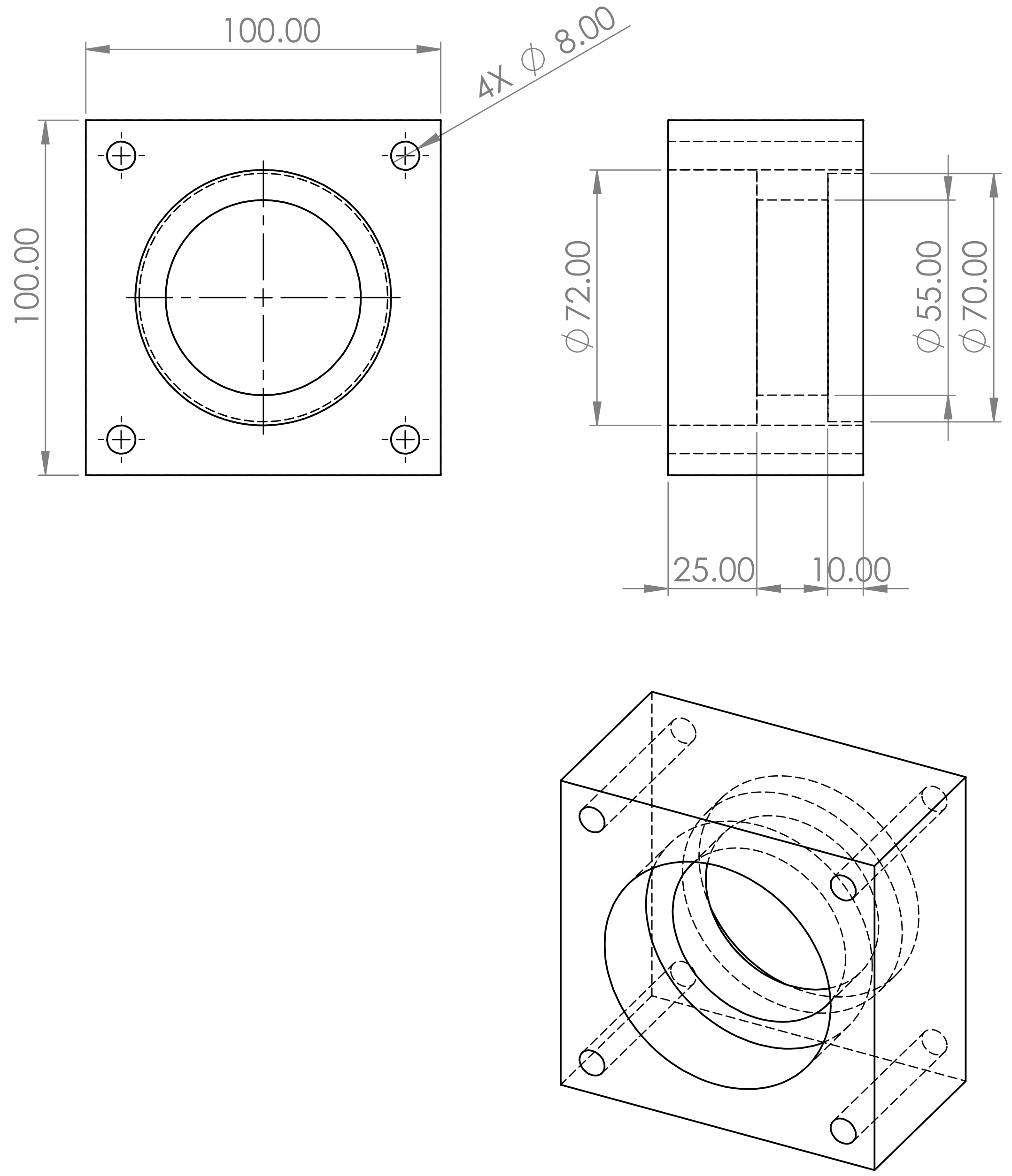
Shaft

A4

WEIGHT:

SCALE:1:2

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: 0.1 mm LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A					

TITLE:

Bracket

MATERIAL: Aluminum

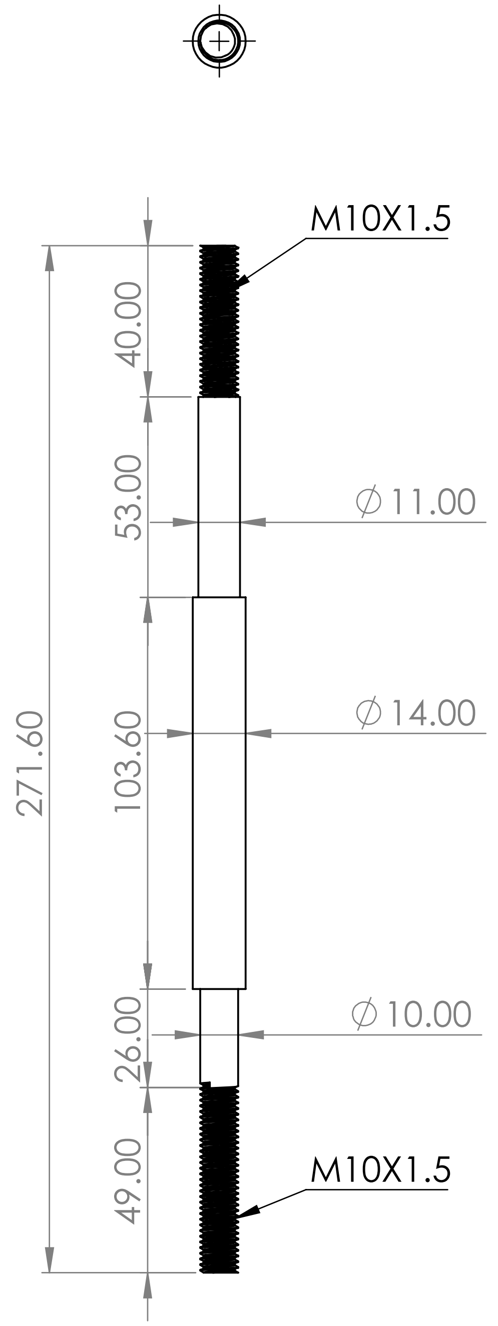
DWG NO.

A4

WEIGHT:

SCALE:1:2

SHEET 1 OF 1



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DRAWN						
CHK'D						
APPV'D						
MFG						
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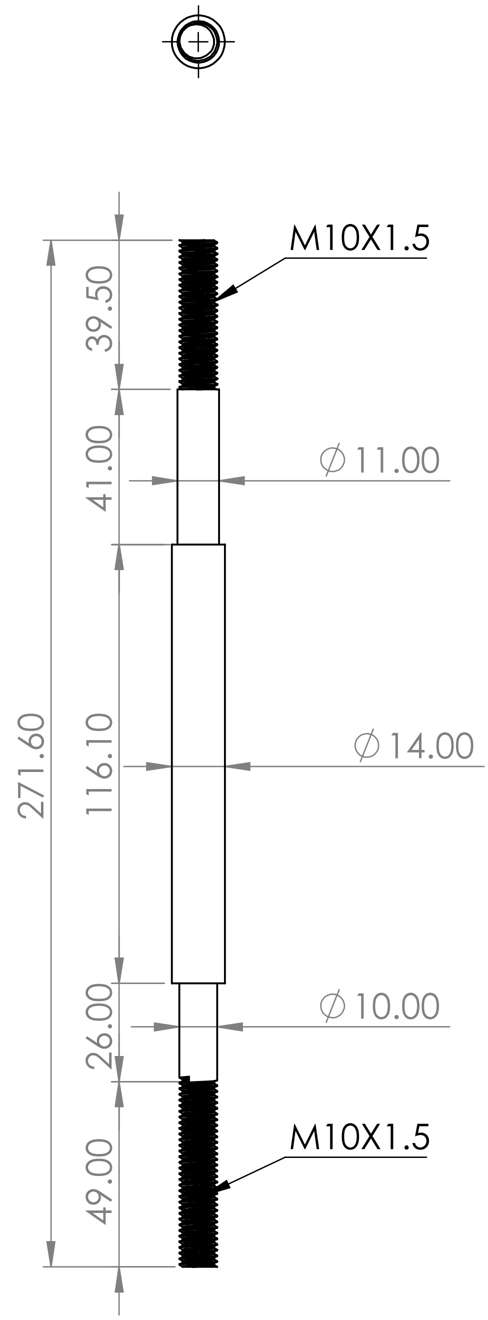

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DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Stud Back

MATERIAL:

DWG NO.

A4

WEIGHT:

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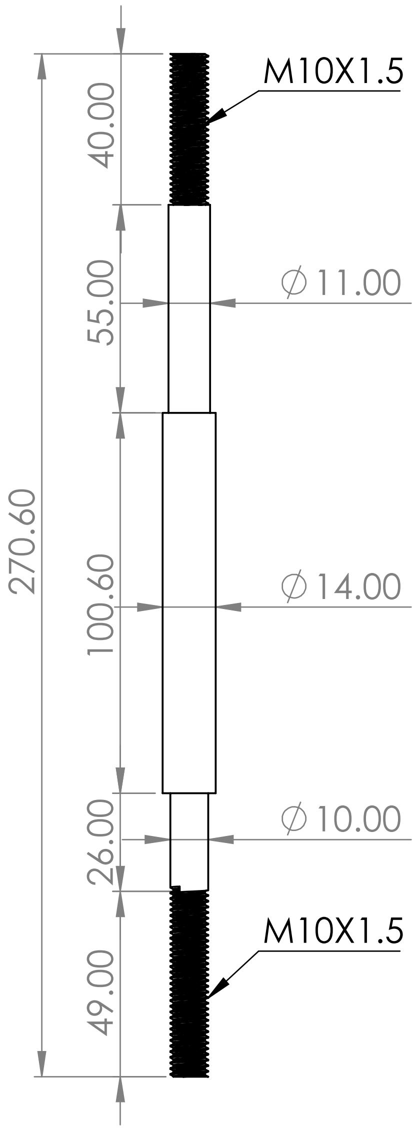
SHEET 1 OF 1



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DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Stud Mid Left

MATERIAL:

DWG NO.

A4

WEIGHT:

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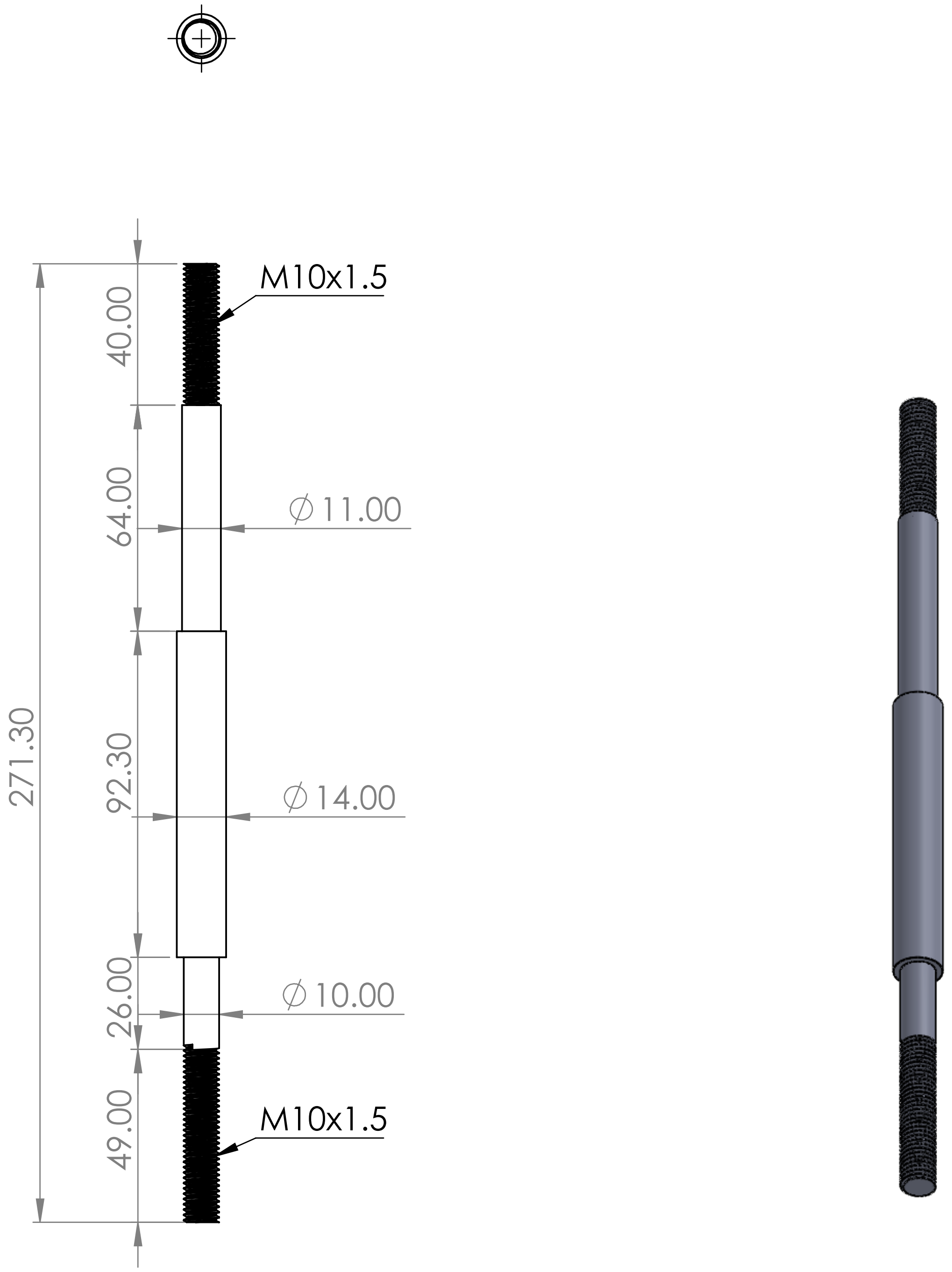
SHEET 1 OF 1



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DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

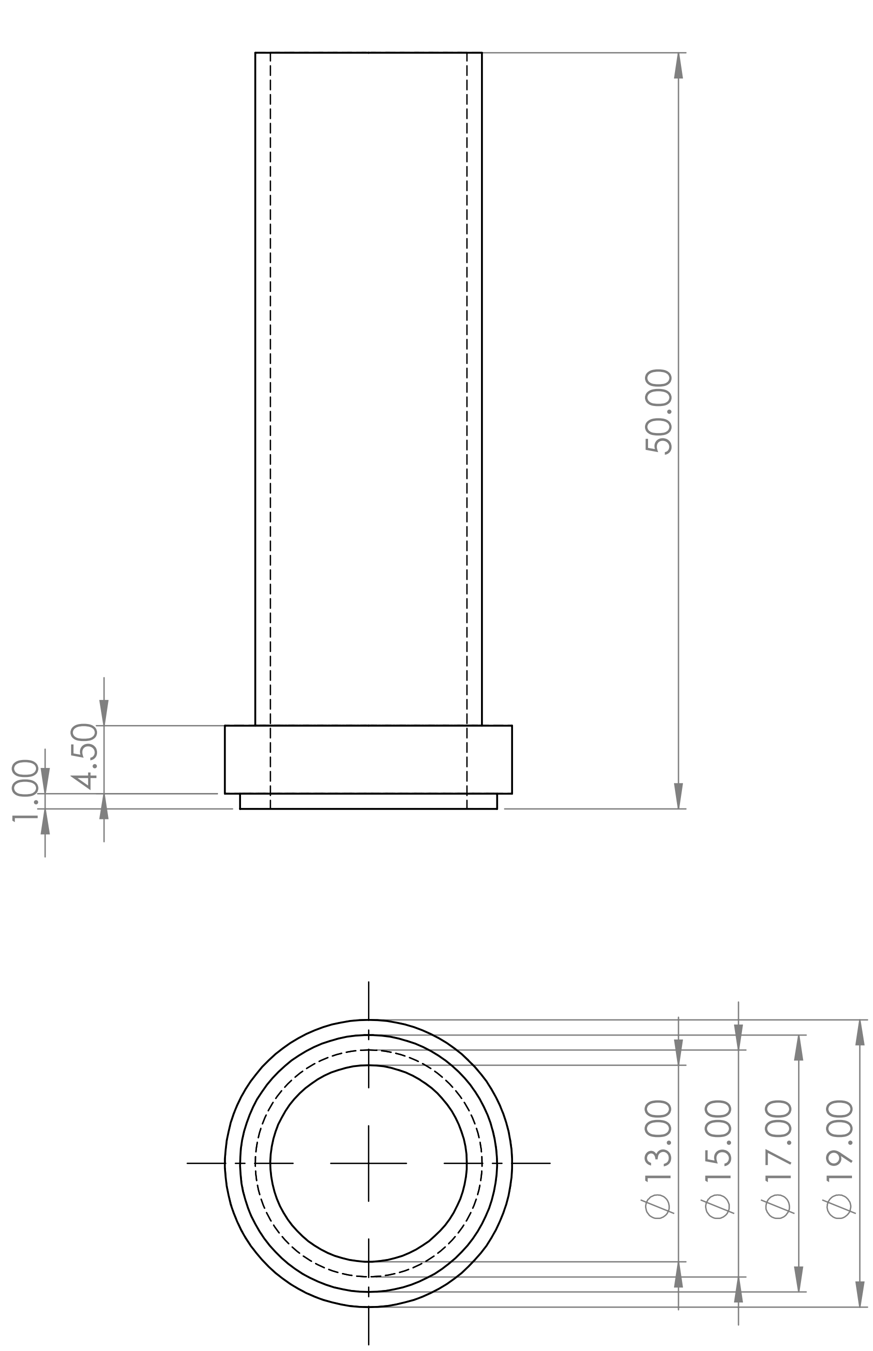
Stud Mid Right

DWG NO.

A4

SCALE:1:2

SHEET 1 OF 1



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LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

Bypass

MATERIAL:

DWG NO.

A4

WEIGHT:

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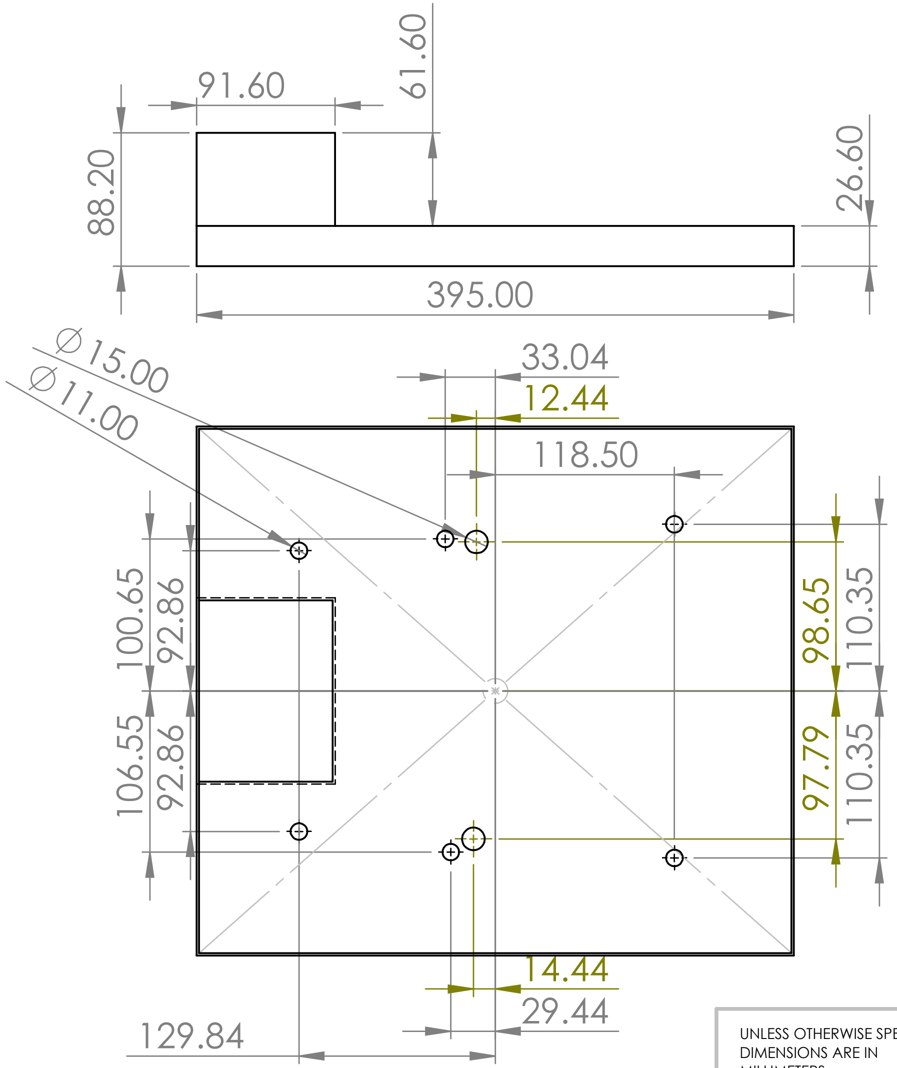
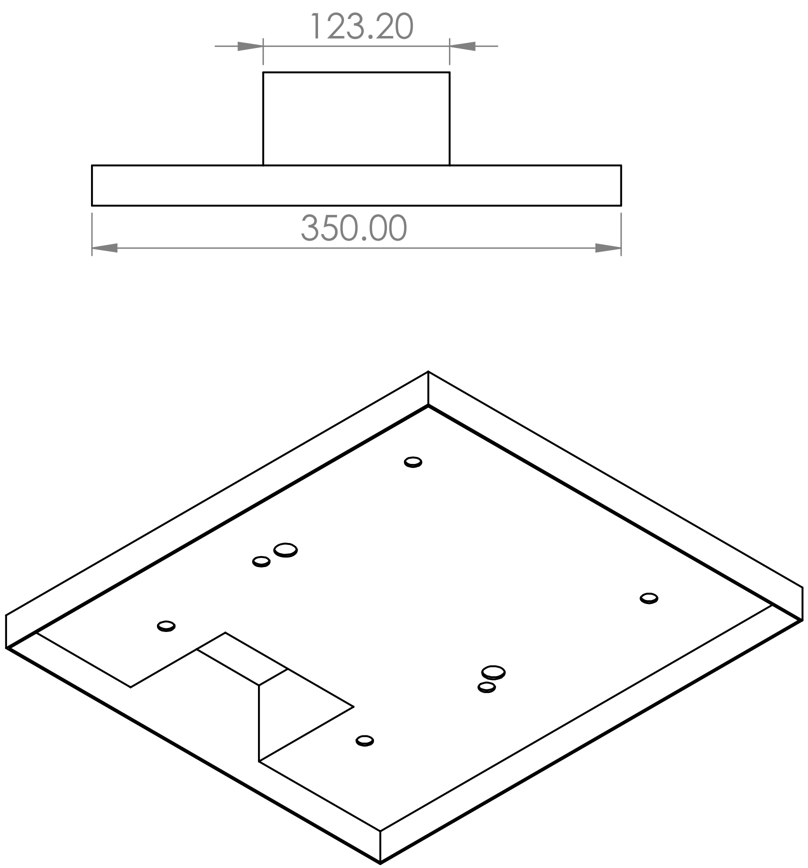
SHEET 1 OF 1





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FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

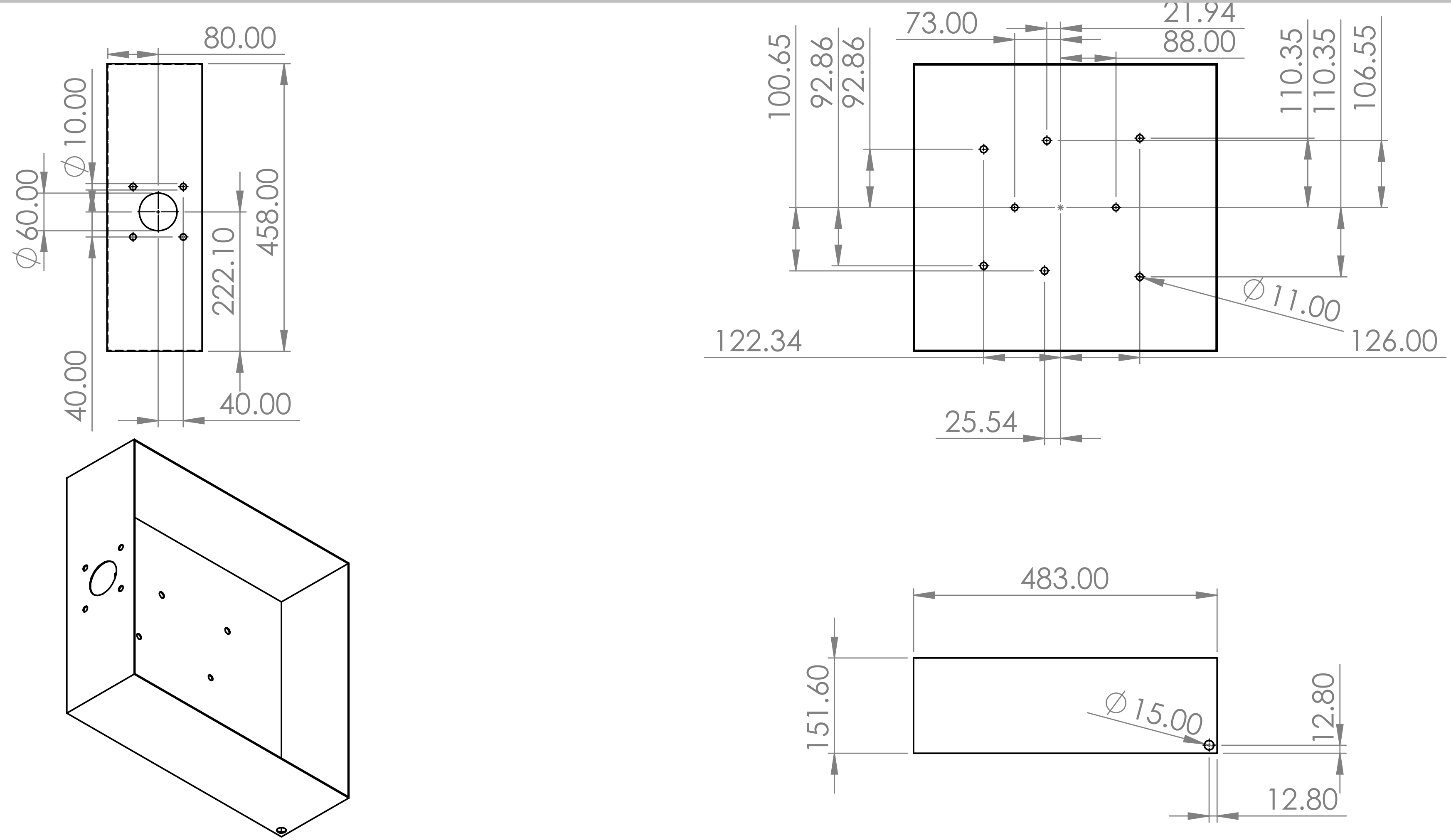
Top plate

DWG NO.

A3

SCALE:1:5

SHEET 1 OF 1



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	NAME	SIGNATURE	DATE		TITLE:	
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CHK'D						
APPV'D						
MFG						
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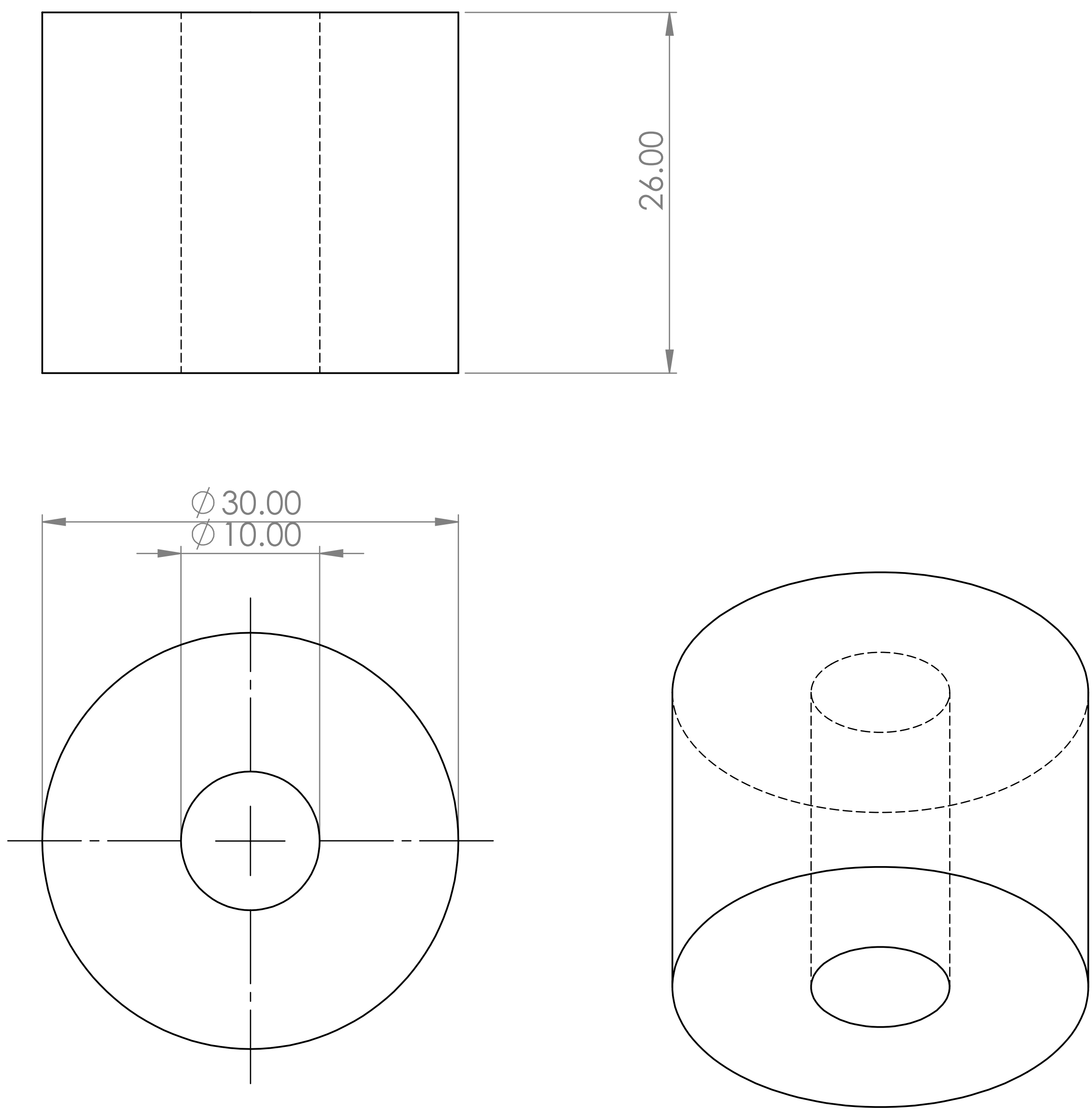

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	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

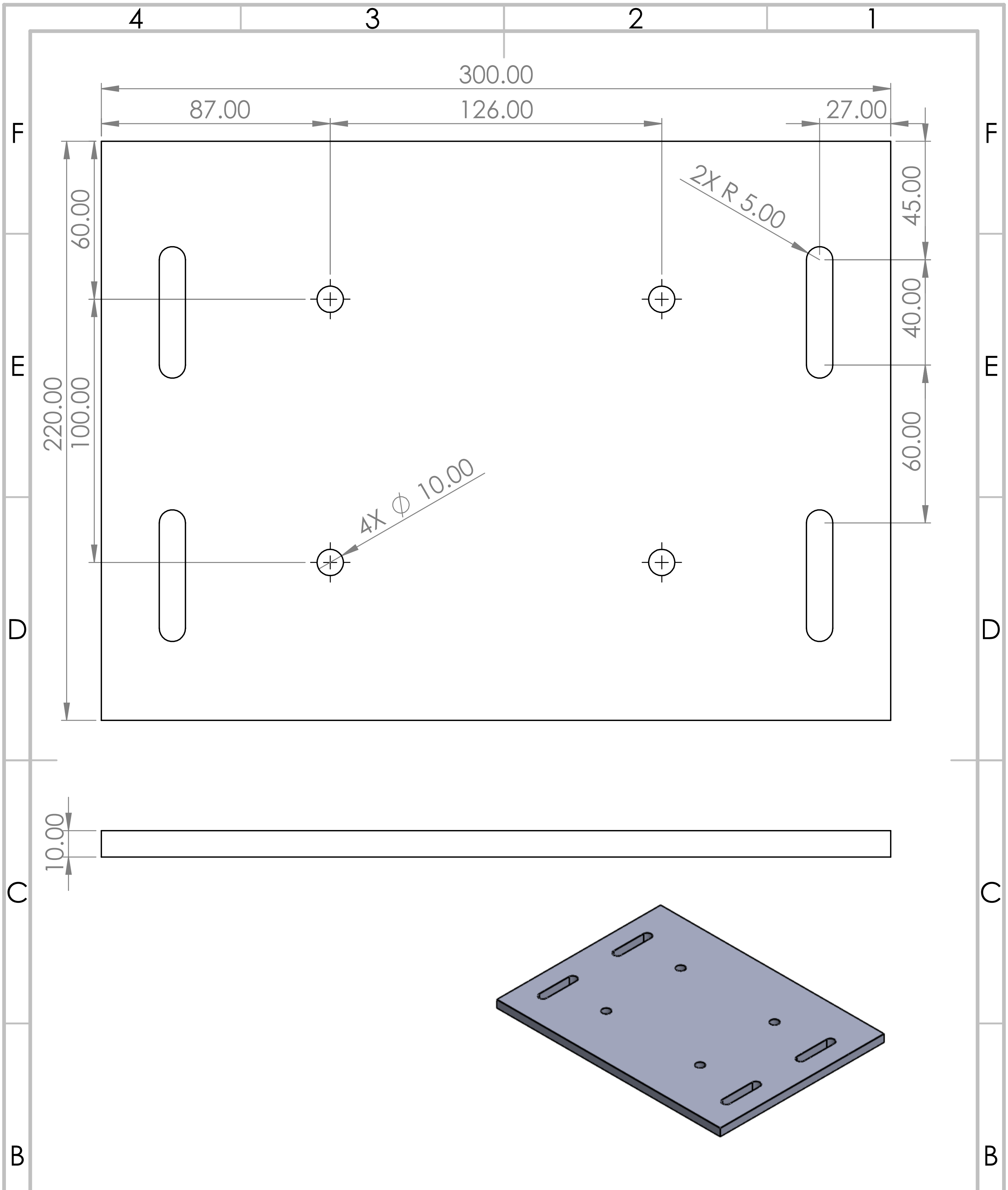
Sump Spacer

DWG NO.

A4

SCALE:1:1

SHEET 1 OF 1



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CHK'D					
APPV'D					
MFG					
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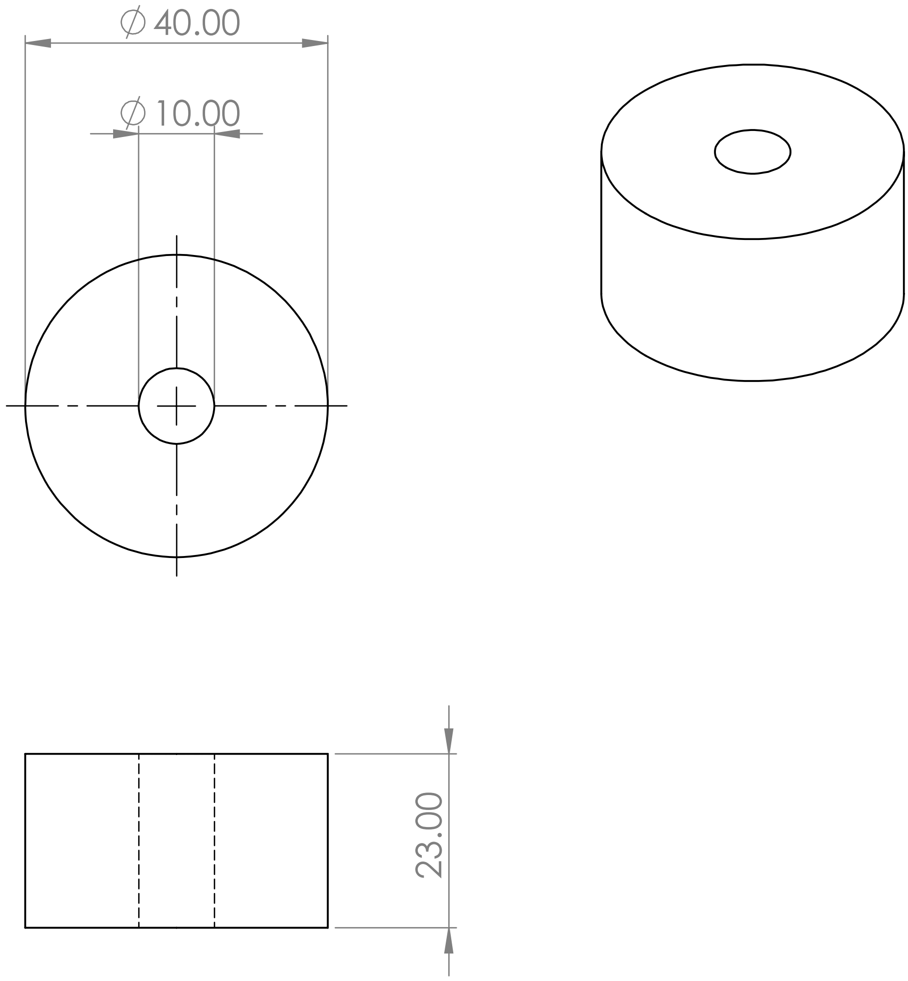

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TOLERANCES: 0.1 mm
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

DWG NO.

Motor Spacers

A4

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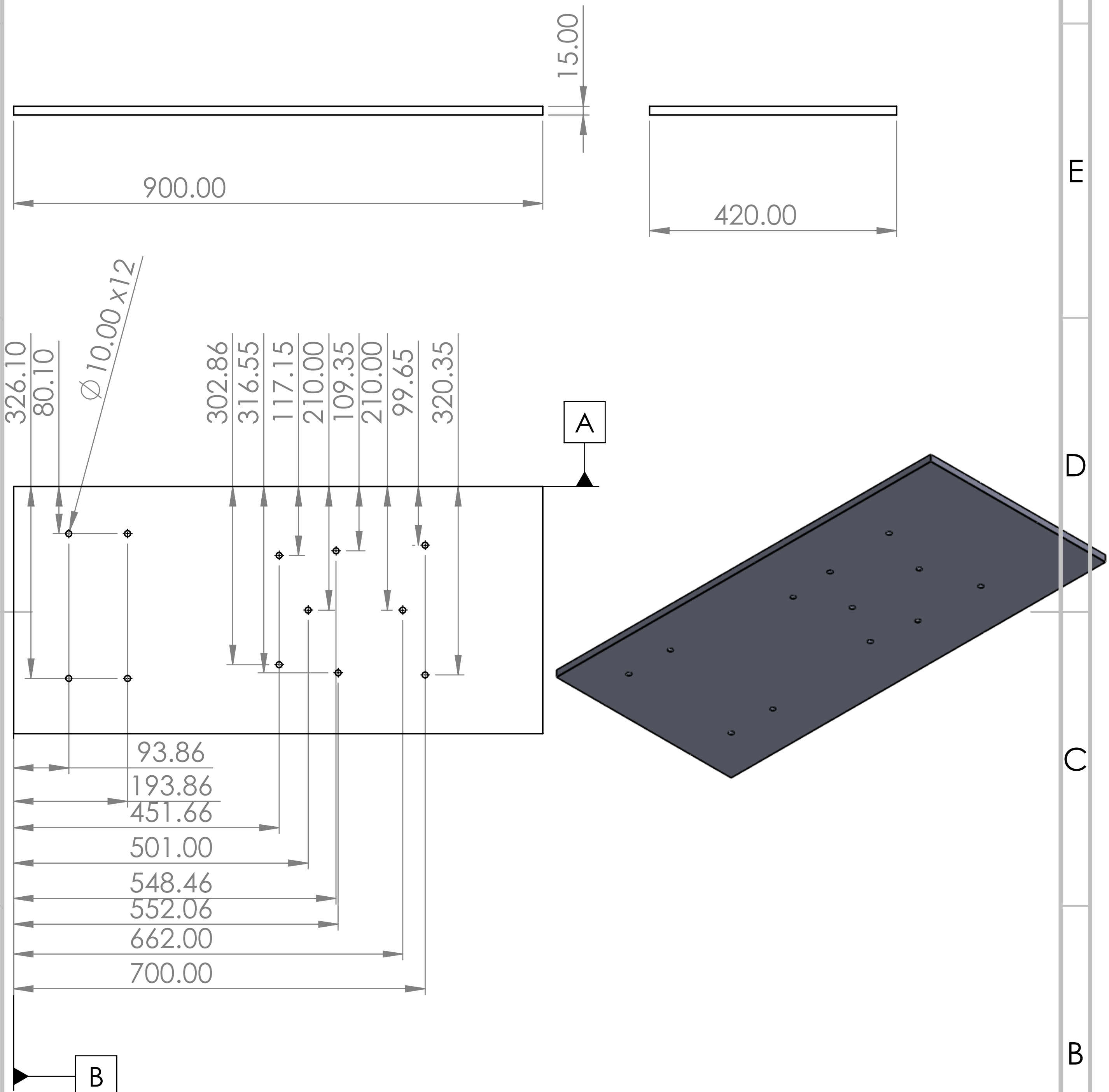

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DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
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CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

BasePlate

MATERIAL:

DWG NO.

A4

WEIGHT:

SCALE:1:20

SHEET 1 OF 1