Mech 2409

Reduction GearBox

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<u>Brainstorming</u>

In our brainstorming sessions we came up with two gearbox designs. Each gearbox design had the same design specifications, which consists of an output torque of 60 (N.m), a minimum factor of safety of 4 or greater for both yielding and tensile stresses, a distance of 18 - 22 cm between the shafts, an input rpm of 360, and a reduction ratio of ½.

The main difference between these two gearbox designs are their gears; helical and spur gears.

Design 1 Helical Gearbox

This designs helical gears generate large amounts of thrust and use bearings to help support the thrust load. Helical gears can be used for transferring power between non parallel shafts. For the same tooth size (module) and equivalent width, helical gears can handle more load than spur gears because the helical gear tooth is effectively larger since it is diagonally positioned.

Design 2 Spur Gearbox

These are particularly loud, due to the gear tooth engaging and colliding. Each impact makes loud noises and causes vibration, which is why spur gears are not used in machinery like cars. A normal gear ratio range is 1:1 to 6:1. Spur gears are the simplest, hence easiest to design and manufacture. A spur gear is more efficient if you compare it with helical gear of same size spur gear train does not produce axial thrust. So the gear shafts can be mounted easily using ball bearings.

Concept design

We chose the spur gearbox over the helical gearbox. The spur gearbox gave us the maximum number of stress cycles on the shaft, therefore, minimizing the gearbox's maintenance time and increasing its functionality for a longer period of time. It also provides constant and equal gear ratios. Moreover, in spur gears, you don't have to account for axial thrust which is a phenomenon which causes bending in shafts. This is because the teeth in the spur gears are relatively straight and are not inclined at an angle. Furthermore, it is easier to maintain, catch irregularities and damages in the spur gearbox design than in the helical gearbox design. Additionally, this design is the most efficient out of the two because spur gears are 94 to 98% efficient in transmission of power as it has a parallel shaft arrangement and reduction is done in a way that energy is not lost.

Due to its efficiency, there is less heat produced, more shaft life and more lubricant life. This helps reduce the negative environmental impact it has as it does not use up energy and resources unnecessarily.

Finally, this design was the most safest as it has a better damage control and has safer handling as its teeth are not as sharp as helical gears.

Gear and shafts parameters selection

Output Torque = 60 N.m

Input Angular Speed = 37.7 rad/s

Radius in for gear = 0.03m

Radius out for gear = 0.06m

Force due to Output Torque:

$$F = \frac{T}{r}$$

$$F = \frac{60}{0.06}$$

$$F = 1000\text{N}$$

Module for both gears = 4

Number of teeth gear In = 15

Number of teeth gear out = 30

Torque in = 30

Shaft Length (in) = Shaft Length (out) = 0.15 m

Shaft Radius (in) = Shaft Radius (out) = 0.0075 m

Calculations of loading

determine the loading (forces, moment and torsion) on the input and output shafts separately based on the transmission and gear parameters.

Force on the shaft due to tangential load = $Fr = F \times tan(20^\circ) = 1000 \times tan(20^\circ) =$

Fr = 363.895 N ≈ 363.90 N

Moment of Inertia = $I = \frac{\pi}{2} \times r^4 = \frac{\pi}{2} \times (0.0075)^4$

I = 0.000000004970097753 J ≈ $4.9 \times 10^{-9} J$

Bending Moment = $M = Fr \times r = (363.90)(0.0386) = 14.046$

$M = N.m$≈ 14.05 N.m

<u>***To keep it simple we kept both the shaft diameters the same***</u>

Stress Calculations For Shaft IN

Bending Moment stress (IN) = $\sigma = \frac{Mr}{I} = \frac{14.04637256 \times 0.0075}{0.000000004970097753}$

$\sigma = 21196322.37 \frac{N}{m^2} \approx 2.12 \times 10^7 \frac{N}{m^2}$

Shear Stress (IN) = $\tau = \frac{VQ}{IT} = \frac{M \times 4}{3\pi r^2}$

$$\tau = \frac{4(363.89)(0.0075)}{3\pi(0.0075)^2}$$

$$\tau = 24710738.52 \frac{N}{m^2} \approx 2.47 \times 10^7 \frac{N}{m^2}$$

Stress Calculations For Shaft OUT

Bending Moment stress (IN) = $\sigma = \frac{Mr}{I} = \frac{14.04637256 \times 0.0075}{0.000000004970097753}$

$$\sigma = 21196322.37 \frac{N}{m^2} \approx 2.12 \times 10^7 \frac{N}{m^2}$$

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Material selection

select materials for the gears and shafts, consider their mechanical, physical and economical features.

Material Chosen for Shaft : AISI 1040 HR

Yield Strength: σ_{max} = 290000000 $\frac{N}{m^2}$

Tensile Strength = 525000000 $\frac{N}{m^2}$

Maximum Combined Stress: $\tau_{max} = \sqrt{\left(\frac{\sigma_x + \sigma_y}{2}\right)^2 + (\tau_{xy})^2}$

$$= \sqrt{\left(\frac{21196322.37 + 0}{2}\right)^2 + (24710738.52)^2}$$

τ_{max} = 26887573.69 $\frac{N}{m^2} \approx 2.69 \times 10^7 \frac{N}{m^2}$

Combined Factor of Safety: F.S = $\frac{\sigma_{max}}{2 \times \tau_{max}}$

$$= \frac{290000000}{2\times(26887573.69)}$$

$$F.S = 5.39282576 \approx 5.40$$

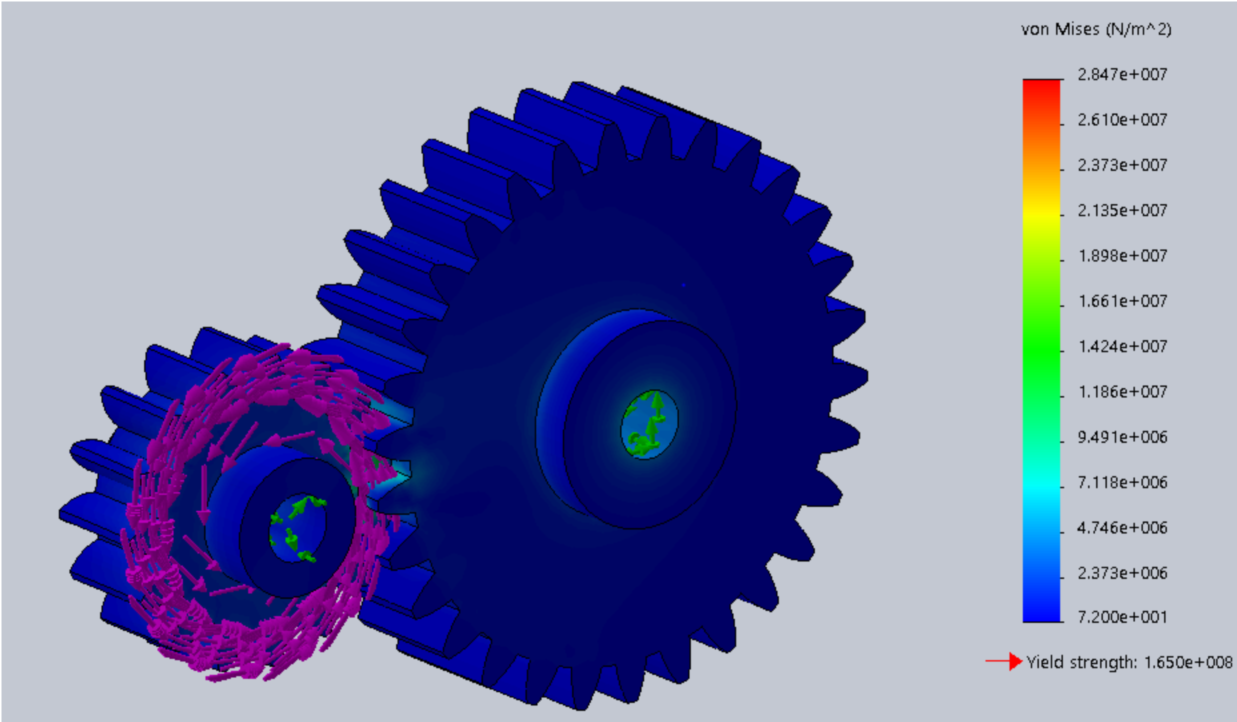
Physical and economical benefits

We are utilizing the same material for all our parts; therefore, in the case of material purchase, we have a high chance of receiving bulk buy benefits which include cheaper and economical prices, and better relationship with the supplier. Additionally, our material AISI 1040 HR is 500$ per ton (without shipping and tax costs); which is relatively cheap.

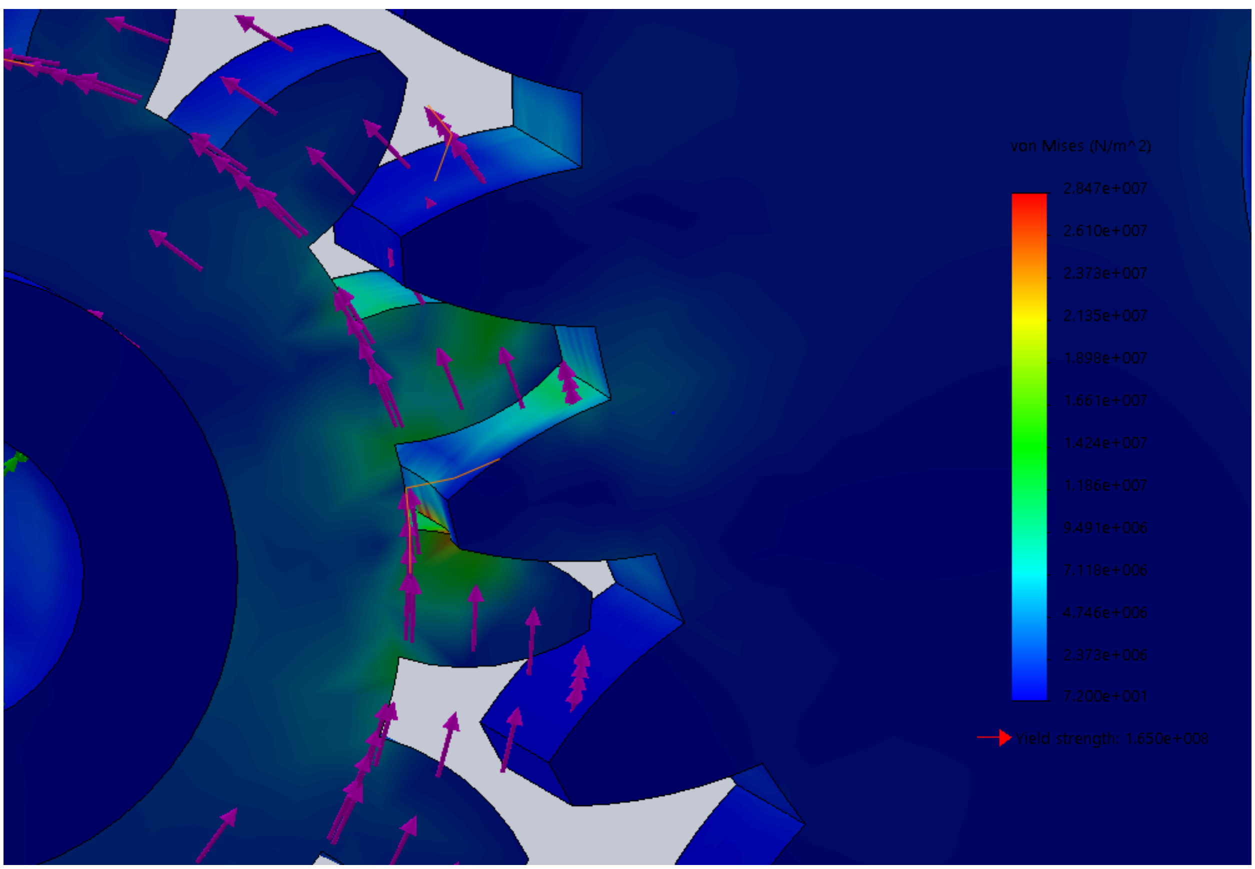
Furthermore, AISI 1095 NM offers maximum surface hardness with high strength and wear resistance. This greatly benefits our gearbox’s lifetime and efficiency.

FEM analysis

use SolidWorks to do stress/strain and deformation analysis of gears at their contact teeth. External loading should be calculated manually and boundary conditions should be prescribed and properly applied to the model. Based on static failure theories, the safety factor of gears under static loading must be calculated.


von Mises (N/m^2)
2.847e+007
2.610e+007
2.373e+007
2.135e+007
1.898e+007
1.661e+007
1.424e+007
1.186e+007
9.491e+006
7.118e+006
4.746e+006
2.373e+006
7.200e+001
Yield strength: 1.650e+008

Gear Stress


von Mises (N/m^2)
2.847e+007
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2.135e+007
1.898e+007
1.661e+007
1.424e+007
1.186e+007
9.491e+006
7.118e+006
4.746e+006
2.373e+006
7.200e+001
Yield strength: 1.650e+008

Gear Stress

Force due to external Loading = $Fr = F \times tan(20\circ) = 1000 \times tan(20\circ)$ =

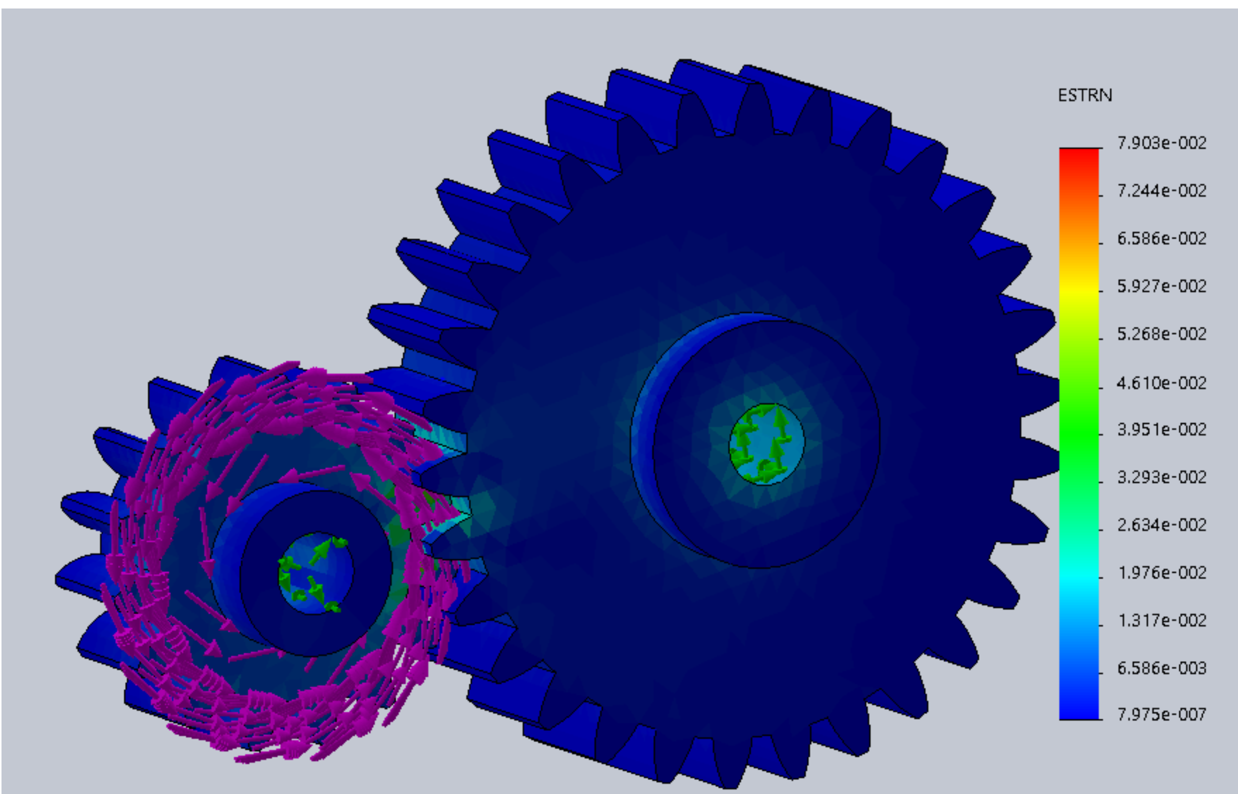
Fr = 363.895 N ≈ 363.90 N

Material For the Gear = AISI 1006 HR

Tooth Factor of Safety = $n_t = \frac{s_y}{2\,\tau_{max}}$

$$n_t = \frac{165 \times 10^6}{2(28.4 \times 10^6)}$$

$$n_t = 2.90$$



Gear Strain

Estimate shaft's lifetime under cyclic loading

Number of Cycles

Shaft IN

Ka = 0.833

Kb = 0.928

kc = 1

kd = 1

ke = 0.868

Kf = 1

Se' = 73.5875

Se = 49.37609632

Shaft OUT

Ka = 0.833

Kb = 0.935

kc = 1

kd = 1

ke = 0.868

Kf = 1

Se' = 73.5875

Se = 49.74854533

Auxiliary parts

Choose proper bearings from manufacturer catalogues, modeling casing (3D CAD) etc.

Bearing IN and OUT

Bore ≈ 1.5 cm = 15 mm

Thickness = 5 mm

Number of balls = 26

Outer Diameter = 21 mm

Casing

Top:

Height = 9.5 cmon

Length = 25 cm

Width = 10.75 cm

Wall thickness = 1.5 cm

Bottom:
Height = 9.5 cm

Length = 23 cm

Width = 8.75 cm

Wall thickness = 0.5 cm

Bolts

Overall Length = 28.58 mm

Head Height = 3.58 mm

Thread Length = 16.31 mm

Thread Diameter = 5 mm

Oil level window

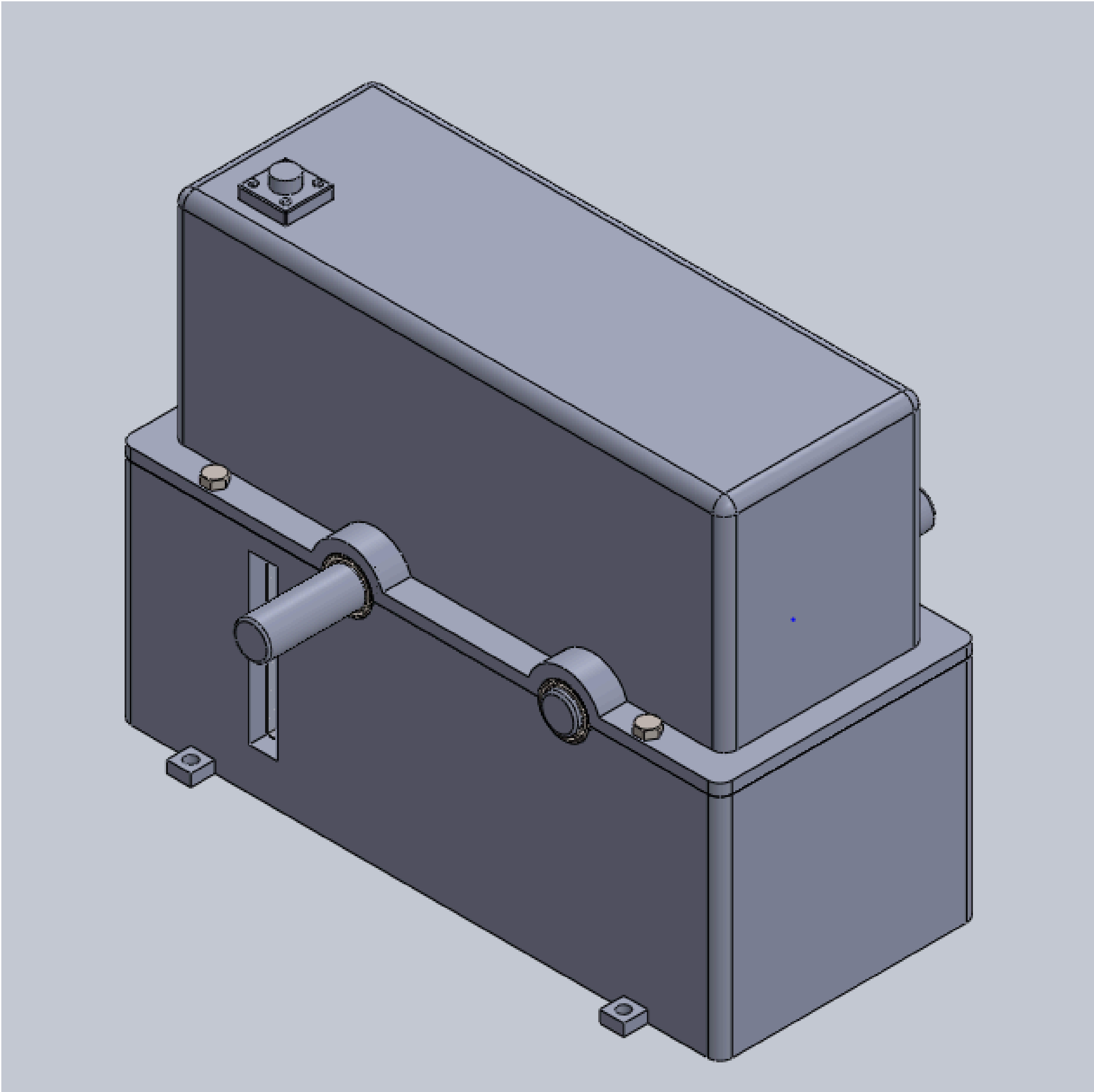
Height = 7 cm

Length = 1.2 cm

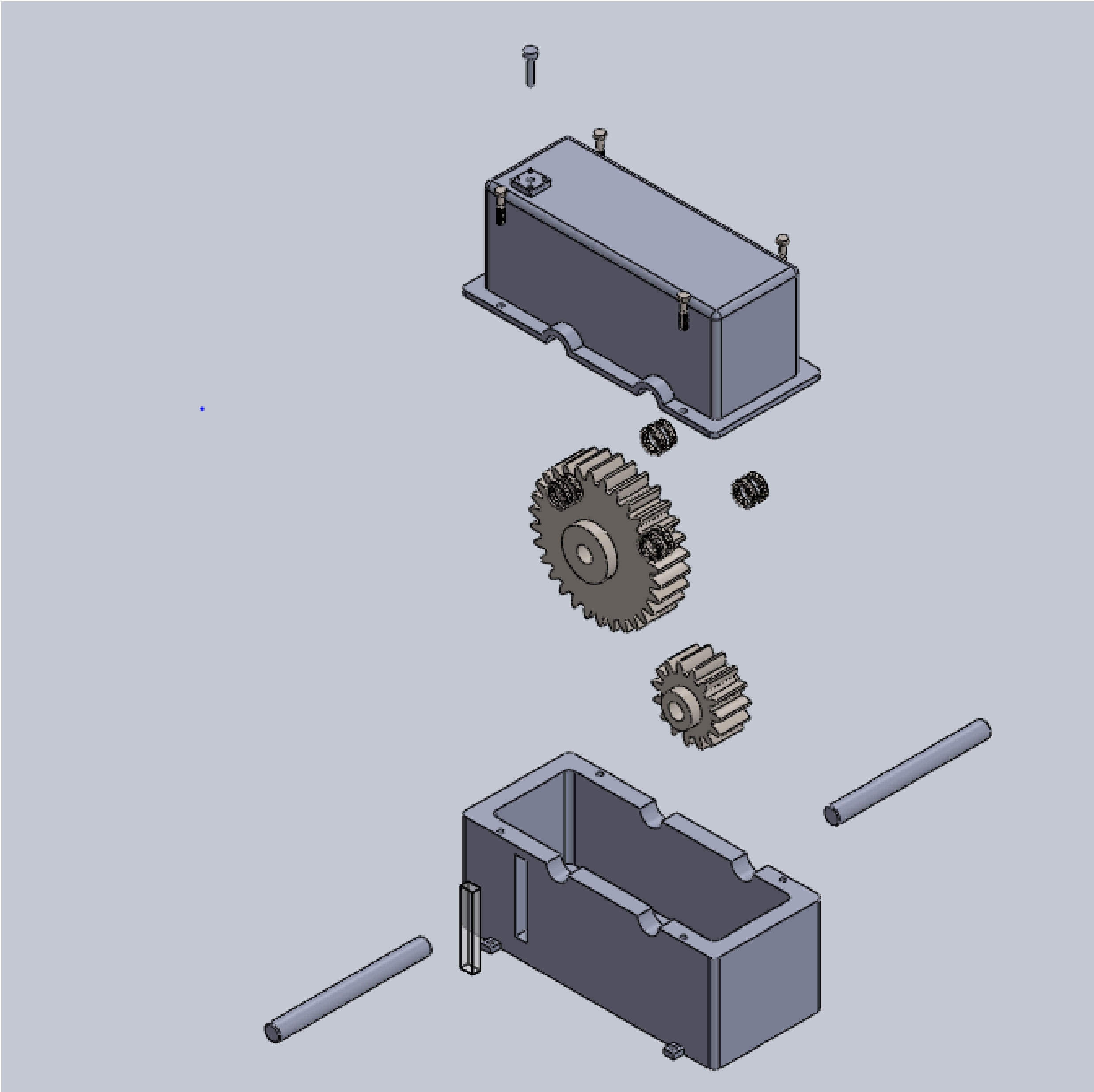
Thickness = 0.75 cm

Assembly illustration

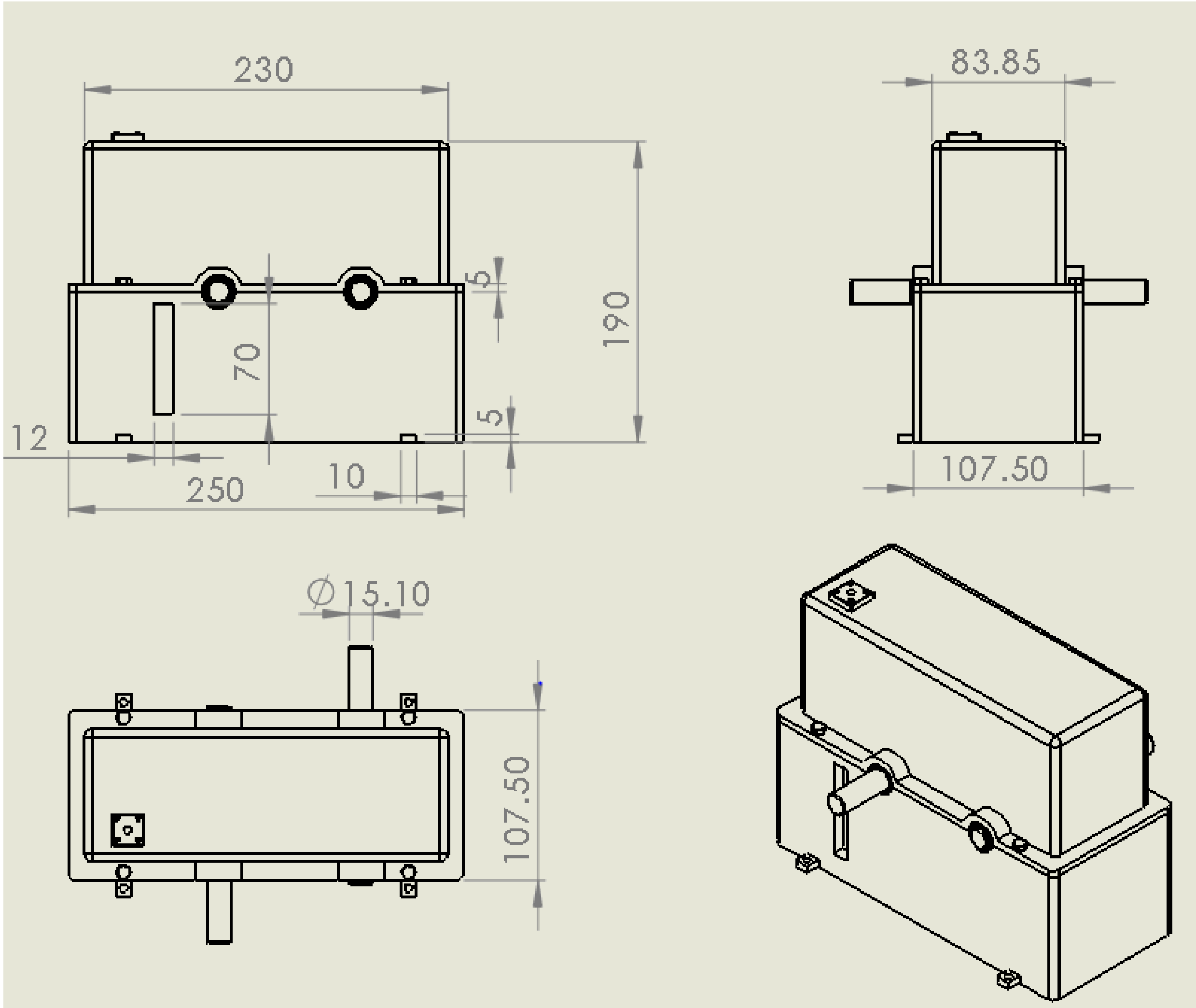
show the assembly model of the final design.



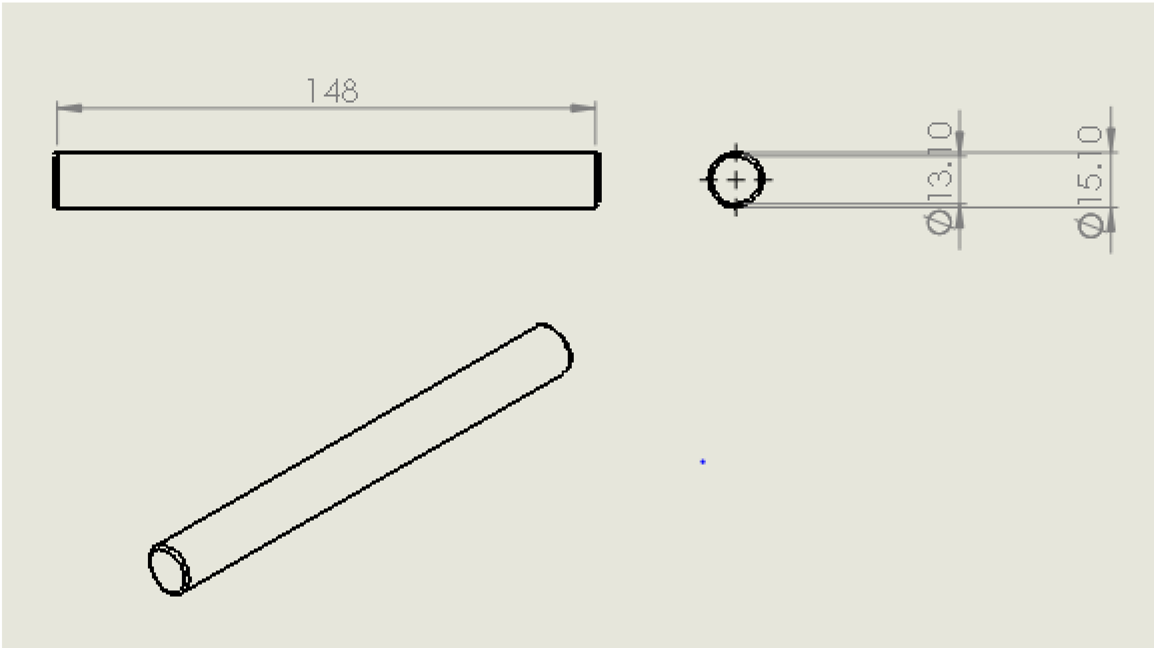
Assembly collapsed view



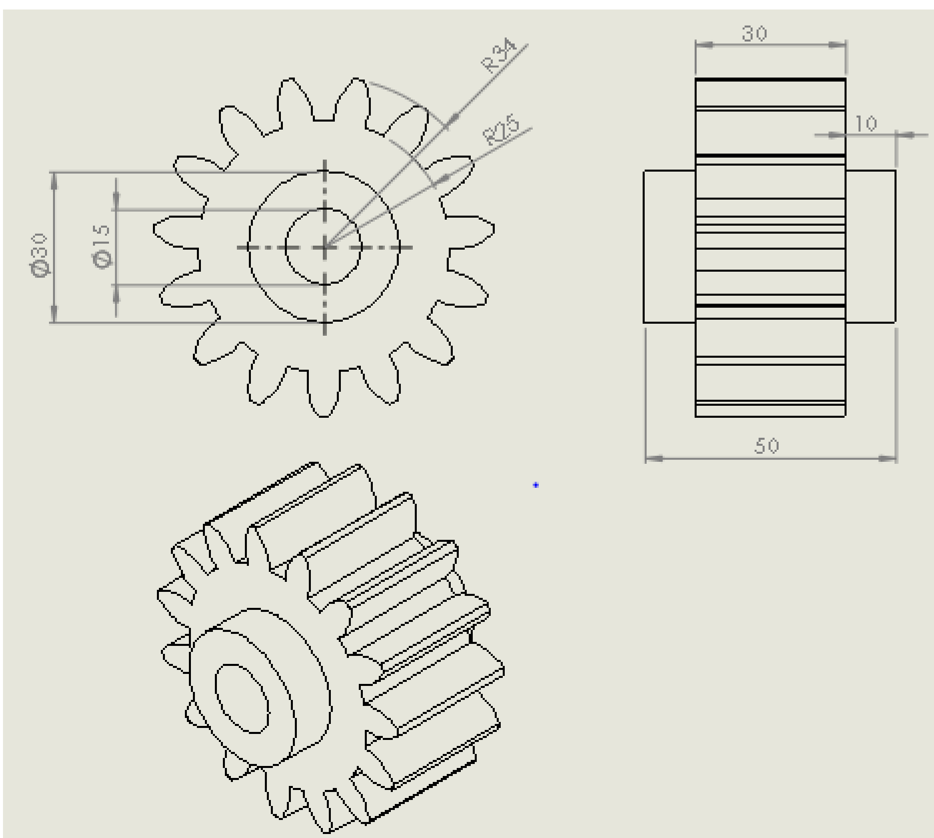
Assembly Exploded view



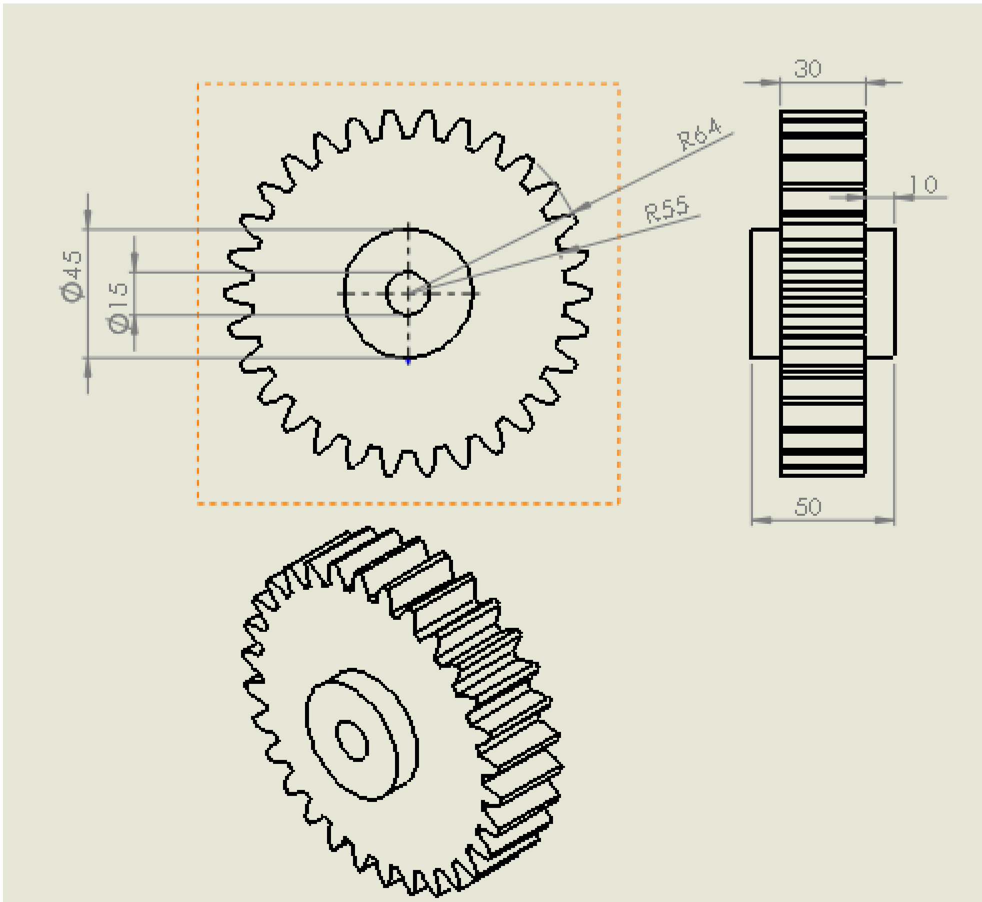
Assembly Drawing



Shaft for both gears



Gear IN



Gear OUT

Maintenance plan

discuss maintenance plan, for example lubrication for components.

Effective lubrication is extremely critical to all gearboxes. Improper lubrication is among the leading causes of gearbox failure. It helps prevent gear and bearing failures, maintains a sufficient oil flow, and keeps oil clean and free of foreign materials. When lubrication problems do occur, they can cause failures like scoring and galling. Scoring and galling are generally caused by oil film breakdown, which results in metal-to-metal contact, and high temperatures, which cause tooth surface damage. If a gear continues to operate without adequate lubrication, the gear tooth profiles will degrade to the point where replacement is the only remedy. Not maintaining the correct oil-fill level, operating the gearbox with dirty or contaminated oil and other poor lube practices also can ruin your equipment.

In helical or spur-gear speed reducers, the operating hours between oil changes can be extended with the use of synthetic lubricants.

Therefore, for our spur gearbox, we plan to use synthetic lubricants, have oil level visual checks, and oil changes. The visual checks will occur every 2 months. It will include running the unit without load for a short time to circulate the lubricant thoroughly, then stopping the unit and re-checking the oil level after allowing the unit to stand for 10 minutes. Additionally, Check the surface of the gearbox or variator for cleanliness (dust/dirt layer must not be greater than 1mm) and check the gearbox for mechanical damage.

For oil changes and overhauling, the following factors should be used to determine the frequency at which these are carried out:

- Oil temperature - unit operating under load.
- Type of oil.
- Environment - humidity, dust, etc.
- Operating conditions - shock, loading, etc.

For the proposed spur gearbox design, the presumed operating temperature condition, considering the environment and the type of oil (Synthetic), is 80° C; therefore, the time period for oil changes is every 26000 operating hours or every six months. The operating conditions for shock and loading are not taken into account because the load and shock does not vary in our given parameters.

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