Design of Two Stage Reduction Gearbox

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Abstract - *Gears and gear drives have been fundamental components in various systems and machines for centuries. However, mastering gear design remains a complex skill. Continuous advancements in gear design have been driven by the ongoing demand for more cost-effective, quieter, lighter, and efficient machinery. An industrial gearbox facilitates the transfer of mechanical energy between devices, optimizing torque while reducing speed. By adjusting speed and torque, gearboxes convert energy into usable forms, contributing to smoother industrial operations. In this paper, a two-stage reduction gearbox with a total speed reduction ratio of 2.57 is presented.*

***Key Words*: Gearbox, Speed Reduction, Two-stage, Industrial machinery, Power Transmission**

1. INTRODUCTION

A reduction gearbox allows the input speed to be lowered, resulting in a slower output speed while maintaining or enhancing the output torque. The reduction gear system comprises a set of rotating gears connected to an output shaft. High-speed input from the wheelwork is transmitted to these gears, where the speed and torque are modified. The number of gears required in the system depends on the specific output speed required for the application. This assembly is typically referred to as a reduction gearbox and can involve one or two stages, depending on the desired output speed.

Two-stage reduction gearboxes are commonly applied in situations involving high-speed operations. In this configuration, the pinion is attached to the input shaft via a key, and it is then connected to an intermediate gear, known as the first reduction gear. This gear is coupled to a second low-speed pinion using an additional shaft, which is then linked to the second reduction gear mounted on the output shaft.

The primary components of this gearbox include the gears, shafts, bearings, casing, oil, and oil seals.

1.1 Gears

Gears are rotating, toothed components used to transmit power and motion between shafts by the engagement of their teeth. Gears enable adjustments in speed, torque, and direction of the power source. The teeth on interacting gears are identical in shape, and when multiple gears are arranged in a sequence, they form a gear train or transmission. By manipulating the gear ratio, gears typically provide a mechanical advantage through torque variation.

The gear ratio (i) represents the relationship between the number of teeth on a gear and its pinion. It is given by the equation:

$$i = \frac{n_p}{n_g} = \frac{z_g}{z_p}$$

Where:
n_p = speed of pinion (rpm)
n_g = speed of gear (rpm)
Z_p = number of teeth on pinion
Z_g = number of teeth on gear

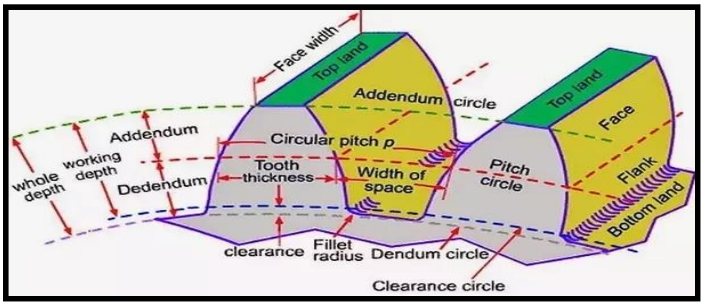

Fig -1: *Terms used in Gears*
Source: *Mechtics*

Gears are typically categorized into four major types: spur, helical, bevel, and worm gears. This project uses spur gears for a speed reduction ratio of 2.57 in an industrial gearbox.

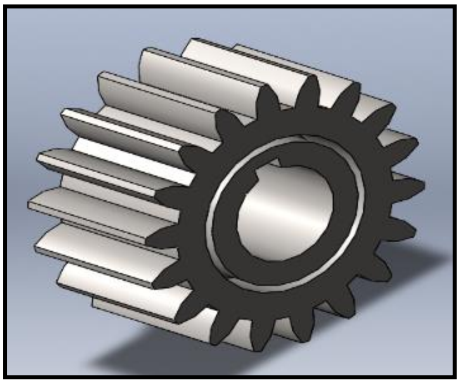

Fig-2*: Spur Gear (Designed in Catia V5)*

In spur gears, the teeth are cut parallel to the axis of the shaft, making them suitable only when the shafts are parallel. The gear tooth profile follows an involute curve and remains consistent across the width of the gear. Spur gears apply radial loads to the shafts and are commonly found in devices like washing machines, screwdrivers, and wind-up alarm clocks. However, they are known for being noisy due to the way the teeth engage, which also causes vibrations. For this reason, spur gears are generally not used in applications like automobiles, where noise and vibration control are critical.

The 20° pressure angle system is used because of several benefits:
(a) It reduces the likelihood of undercutting.
(b) It minimizes interference.
(c) The increased pressure angle slightly broadens the tooth base, enhancing its strength and load capacity.
(d) It offers a longer contact length between teeth.

1.2 Shafts

A shaft is a rotating component responsible for transmitting power from one part to another. The power is transferred by a tangential force, and the resulting torque helps deliver energy to other machines connected to the shaft. Shafts are typically stepped, with a larger diameter in the middle and smaller diameters at both ends where bearings are placed. These steps create shoulders to hold transmission elements like gears, pulleys, and bearings in position. The rounded portion between sections of different diameters is called a fillet, and it is designed to reduce stress concentrations caused by abrupt changes in the shaft's cross-section. Components are mounted onto the shaft using keys or splines. Effect of stress- concentration due to abrupt change in the cross-section. The various members are mounted on the shaft using keys or splines.

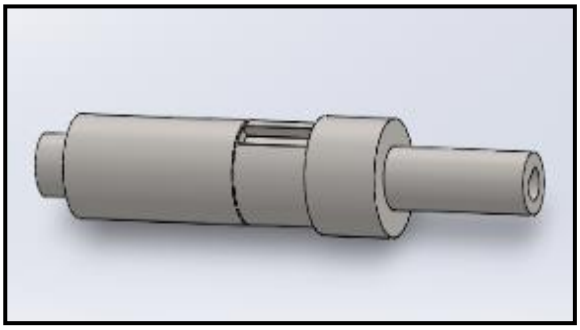

Fig-3*: Shaft (Designed in Catia V5)*

Shafts are subjected to axial tensile forces, bending moments, torsional moments, or a combination of these forces. Typically, shafts endure combined bending and torsional stresses, and shaft design involves selecting the appropriate diameter to ensure strength and rigidity.

1.3 Bearing

A bearing is a mechanical component that facilitates relative motion between two parts with minimal friction. Its key functions include ensuring smooth shaft or axle rotation, supporting and holding the shaft or axle in position, and transferring forces acting on the shaft to the frame or foundation.

Rolling Contact Bearing:

A rolling contact bearing is composed of four main parts: the inner and outer races, a rolling element (such as a ball, roller, or needle), and a cage that holds the rolling elements in place and evenly distributes them around the shaft's periphery. Bearings are categorized based on the type of rolling element used, including ball bearings, cylindrical roller bearings, taper roller bearings, and needle bearings. Additionally, bearings are classified by the load direction they handle, either as radial bearings or thrust bearings

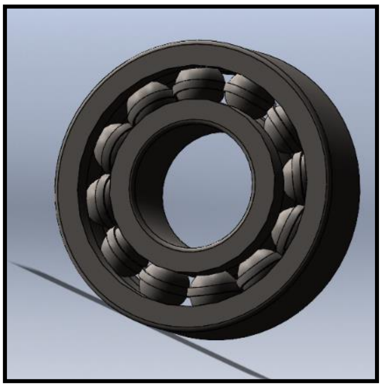

Fig-4*: Rolling contact bearing (Designed in Catia V5)*

Applications of Rolling Contact Bearings:
Rolling contact bearings are commonly used in the following areas:

1. Spindles of machine tools
2. Front and rear axles of automobiles
3. Gearboxes
4. Small electric motors
5. Rope sheaves, crane hooks, and hoisting drums

1.4 Casing

The casing serves as a structural enclosure that holds the shafts in place and houses all the gears, ensuring they function together without interference. It also features mounting points for installation in a powertrain assembly and absorbs the load imposed by the power source. If an engine powers the system, the casing also transmits vibrations. To prevent failure, the casing design includes intricate contours.

Bearing sockets are integrated into the casing, allowing for the installation of bearings, which then accommodate the shafts. The casing also has an inlet for adding gear oil, and it should be airtight to maintain the cooling efficiency of the oil. Typically, gearbox casings are designed as a two-piece, symmetrical structure to facilitate easy opening and closing during maintenance. High precision in casing manufacturing is crucial to avoid misalignment issues. Cast iron has long been the preferred material for producing casings.

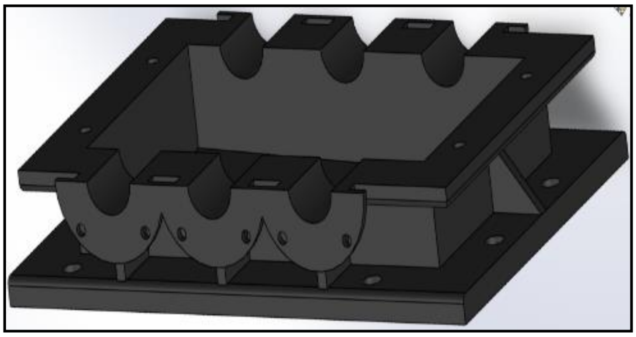

Fig-5: *Casing (Designed in Catia V5)*

1.5 Oil and Oil seals

Gear oil is a lubricant specially designed for use in transmissions, transfer cases, and differentials in various machines, including vehicles. It has a high viscosity and often contains organosulfur compounds. Oil seals serve to prevent contaminants like dirt from entering and keep oil or grease from escaping. A typical oil seal consists of a rigid outer metal ring and a flexible inner sealing component, chemically bonded to the metal ring for durability.

2. Design Calculations:

2.1 Gears

The design of gears primarily depends on the module and the number of teeth. While the module must be the same for gears in mesh, the number of teeth can vary to achieve speed reduction or increase. The first step in gear design is determining the number of teeth on each gear. Standard gear systems, based on the pressure angle, provide guidelines for this. For this project, a 20° pressure angle is used. According to this standard, a minimum of 18 teeth on the pinion is required to avoid undercutting and interference, which is why this value is chosen.

The motor's input angular speed (Np) is 1440 rpm, and the output angular speed (Ng) is 560 rpm. Thus, the total speed reduction ratio (i) is:

$$i = 1440 / 560 = 2.57$$

This gearbox design includes two stages, meaning the speed reduction occurs in two phases. The gear ratio at each stage is equal, given that the number of teeth on both pinions and gears in each stage is the same. The gear ratio at each stage is the geometric mean of the total reduction ratio.

Thus, gear ratio at each stage, i.e., *i'* is,

$$i' = (i)^{1/2}$$

$$i' = (2.57)^{1/2}$$

$$i' = 1.603$$

For pinion,

Number of teeth, $Z_p = 18$

For gear,

$$Z_g / Z_p = i$$

$$Z_g = i \times Z_p$$

$$Z_g = 29$$

At first stage,

$$N_g / N_p = Z_p / Z_g$$

$$N_p = 1440,\ Z_p = 18,\ Z_g = 29$$

$$N_g = 893.79 = 900 \text{ rpm}$$

At second stage,

$N_g / N_p = Z_p / Z_g$

N_p **= 900,** Z_p **= 18,** Z_g **= 29**

N_g **= 560 rpm**

At the second stage, the rotational speed is lower, but since the total power is determined by torque, the angular velocity (W) must remain constant. This leads to an increase in torque at the second stage compared to the first. As a result, the second stage becomes more critical than the first, making it the focus of the analysis.

The service factor (Cs) is selected as 1.25, based on the system's configuration, which involves a uniform driving machine (electric motor) and a medium shock driven machine (main drive for machine tools).

Since the pitch line velocity is not known at this point, the velocity factor (Cv) cannot be determined directly. However, assuming a pitch line velocity of 5 m/s, the velocity factor is calculated as:

Cv = (3 / 3 + V) = (3 / 3 + 5) = 3 / 8

For calculation of module(m) based on bending strength we have the following formula,

$$m = \left[\frac{60 \times 10^6}{\pi}\left\{\frac{(\mathrm{kW})C_s(fs)}{znC_v\left(\frac{b}{m}\right)\left(\frac{S_{ut}}{3}\right)Y}\right\}\right]^{1/3}$$

As we are using the same material for pinion and gear, the pinion will be weaker as the Lewis form factor Y is lower for the pinion.
Now,

(power) kW = 5 kW

Cs = 1.25

Fs = 1.5

Zp = 18

N = 900 rpm

Cv = 3 / 8

B / m = 10 mm

Sut = 750 N / mm2

Y = 0.308

After putting all these values in the equation, we get module m as, m = 3.38 = 4 mm

Gear Dimensions,

m = 4 mm

dp = m x Zp = 4 x 18 = 72

dg = m x Zg = 4 x 29 = 116

b = 10, m = 40 mm

Calculation of Bending Strength:

$$S_b = mb\sigma_b Y$$

For calculating the bending force exerted on the pinion tooth, we have the following formula,

Here,
m=4mm, **b=**40mm, **σ** = 750 / 3 N / mm2, Y = 0.308

After substituting these values in the equation, we get,
Sb = 12320 N

Calculation of wear strength:

For calculation of wear strength, we have the following formula,

$$S_w = bQd'_p K$$

Here, **b** = 40 mm, **Q** = 2 X Zg / (Zg + Zp) = 1.234, **dp** = 72

K = 0.16 (BHN / 100)2 (For EN31 Steel) = 1.85

Thus, S_w **= 6574.752 N**

Now, Sb = 12320 N and Sw = 6574.752 N

It is clear that $S_w < S_b$, Thus, further analysis must be done based on Sw.

Calculation of dynamic load and thus effective load using Buckingham equation:

For calculating moment exerted on gear tooth, we have the formula,

Here, kW = 5 kW, n = 900 rpm

Thus, **M_t = 53051.65 N-mm**

$$M_t = \frac{60 \times 10^6(\text{ kW})}{2\pi n}$$

Here, **M_t = 53051.65 N – mm**, **dp = 72**

Thus, **P_t = 1473.66 N**

For dynamic load, assuming grade 4 type of gears,

e = 3.2 + 0.25 x Φ

Where, Φ = m + 0.25 x (d) ½

Thus, ep = 4.73 , eg = 4.87

Now, e = ep + eg

Thus, e = 9.60 µm

Now**, pitch line velocity, V is calculated as,**

$$v = \frac{\pi d' n}{60 \times 10^3}$$

d = 72 mm, n = 900 rpm

Thus,
V = 3.39 m / s

Dynamic load by Buckingham equation is given by,

$$P_d = \frac{21v(\text{Ceb} + P_t)}{21v + \sqrt{(\text{Ceb} + P_t)}}$$

Here,

V = 3.39 m / s

C = 11400

E = 9.60 µm

P_t = 1473.66 N

Thus,
P_d = 2820.64 N

Now, **Peff = C_s x P_t + P_d**
Here,
Pt = 1473.66 N
P_d = 2820.64 N
C_s = 1.25

Thus,
Peff = 4662.71 N

Now, **S_w = 6574.752 N** & **Peff = 4662.71 N**

Here,
Fos = S_w / Peff = 1.41

This indicates that the design meets the required standards, confirming that the chosen module is appropriate for this application. Therefore, using this module, the remaining gear dimensions will be calculated.

Final Dimensions:

1) **Module: 4 mm**
2) **Face width: 40 mm**
3) **Addendum: 4 mm**
4) **Dedendum: 5 mm**
5) **Clearance: 1.5 mm**
6) **Working depth: 8 mm**
7) **Whole depth: 8.628 mm**
8) **Tooth thickness: 6.2832 mm**

2.2 Shafts

Input Shaft:

Input torque= power/ω= $\frac{5\times10^3}{2\times\pi\frac{1440}{60}}$

=33.15N

M_T= 33.15 Nm.

M_T = Ft x (Dp/2)
[Dp is the pitch circle diameter of pinion=72mm]

Ft= 2x (MT/Dp)

Ft= 920

F =Ft/cos (20) = 979.765 N (20= Pressure angle)

Fr= Ft*tan (20) = 334.85 N

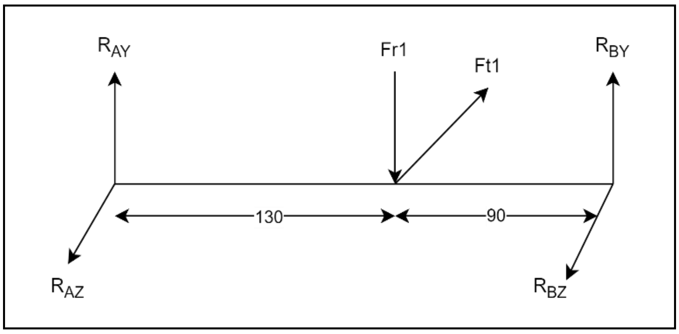

***Fig-6**: Loads on input shaft*

Source: Made using https://app.diagrams.net/

for XY plane
Taking moment at pt A;

$\mathbf{R_{BY}}$ x 220 = f_r x 130

$\mathbf{R_{BY}}$ =197.86 N.

Taking moment at pt B

$\mathbf{R_{AY}}$ x 220 = fr x 90

$\mathbf{R_{AY}}$ =136.98 N.

Bending moment in the X-Y plane

170< x <150

$\mathbf{R_{AY}}$ × 15 - $\mathbf{F_r}$ x 20 =20547Nmm

for XZ plane:

Taking moment about A;

$\mathbf{R_{BY}}$ X 220 = $\mathbf{F_t}$ x 130

$\mathbf{R_{BZ}}$=552.5N

$\mathbf{R_{AZ}}$ = 935 - 552.5 = 382.5N

Bending moment in X-Z plane;

170< x <150

$\mathbf{R_{AZ}}$ × 15 - $\mathbf{F_t}$ x 20 =57375Nmm

Resultant bending moment:

$\mathbf{M_B}=\sqrt{(205471^2 + (57375)^2}$ = 60943.168Nmm

For fully reversed loading:

$$\sigma_a = \frac{M \times y}{I} = \frac{32M}{\pi d^3} \text{ and } \quad \sigma_m = 0$$

$$\tau_m = \frac{T \times r}{J} = \frac{16 \times \sqrt{3}}{\pi d^3} \text{ and } \tau_a = 0$$

Von-Mises Criteria:

$$\sigma_a' = (\sigma_a^2 + 3t_a^2)^{1/2} = \left[\left(\frac{32K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_f T_d T_d^2}{\pi d^3}\right)^{1/2}\right]^{1/2}$$

$$\sigma_m' = (\sigma_m^2 + 3t_m^2)^{1/2} = \left[\left(\frac{32K_t M_m}{\pi d^3}\right)^2 + 3\left(\frac{16K_f T_T}{\pi d^3}\right)^2\right]^{1/2}$$

DE Goodman's Equation:

$$D_B = \left[\frac{32 \cdot SF}{\pi} \cdot 10^6 \cdot \sqrt{\left(\frac{SCF \cdot M_B}{S_n}\right)^2 + \frac{3}{4} \cdot \left(\frac{T_{In}}{S_y}\right)^2}\right]^{\frac{1}{3}} [\text{ mm}]$$

SF = safety factor; it is recommended as 2 commonly,

SCF = stress concentration factor; it is recommended as 3,

Sn = endurance strength; it equals 280 Mpa for C45 steel in accordance with EN10083-2 standard

Sy = yield strength; it equals 370 Mpa for C45 steel in accordance with EN-10083-2 standard. MB and Tin are the same as calculated earlier.

D=23.74 mm

Similarly, D=24 mm for Intermediate shaft and D=26.38 mm for output shaft

2.3 Keys

According to international standards for square key,

b = h = d/4 = 28/4 = 7mm

l = 1.5 * d = 1.5 * 28 = 42mm

where b=h = side of square

l =length of key

d =diameter of shaft

2.4 Gear blank

For gear,
Pitch circle diameter =
d' = m x z = 4 x 29 = 116 mm
Addendum circle diameter =
da =m*(z+2) = 4 X 31=124mm
Dedendum circle diameter =
df = m*(z-2.5) = 4 *26.5 =106mm

Shaft diameter = d = 28mm
Face width = b = 40mm

So, for blank construction:
d1 = outer diameter of hub = 1.5 x d = 42mm
tr = thickness of rim = 2 x m = 8 mm
b1 = thickness of web = 0.25 x b = 0.25 x 40 = 10mm
d3 = inner diameter of rim = df-2tr = 106-16 = 90mm
d4 = diameter of holes in rim = (d3-d1)/4 = 12mm
d2 = PCD of holes = (d1+d3)/2 = 66mm

for pinion,
Pitch circle diameter = d' = m x z = 4 x 18 = 72 mm
Addendum circle diameter = da =m x (z+2) = 4 x 20 =80mm
Dedendum circle diameter = df = m x (z-2.5) = 4 x 15.5 = 62mm

Shaft diameter = d = 28mm

Face width = b = 40mm

So for blank construction:

d1 = outer diameter of hub = 1.5 x d = 42mm

tr = thickness of rim = 2 x m = 8 mm

b1 = thickness of web = 0.25 x b = 0.25 x 40 = 10mm

d3 = inner diameter of rim = df-2tr = 62-16 = 46mm

2.5 Bearing Selection

ID = 17 mm

Fr = max static load = 1178 N

For industrial gearbox

L10h = 15000 hours

L10 = 60 x n x L10h/10^6 = 504 million revolutions

As there is no axial load,

P = Fr

C = p x (L10) ^ (1/3) = 1178 x (504) ^ (1/3) =9375

From manufacturer's catalogue:

d = 17 mm

D = 40 mm

B = 12 mm

Designation for bearing is 6202.

2.6 Casing Dimensions

L = Overall length of gearbox (in mm)
L = 310

Wall thickness of housing:
t1 = (0.012 x L) + 5
t1 = 8.72 mm

Wall thickness of covering:
t2 = 0.8 to 1 times t1
t2 = 8.72 mm

Wall thickness of mating flanges:
t3 = 1.5 x t1
t3 = 13.08 mm (each flange is 13.08 mm thick)

Wall thickness of housing:
t4 = 2 x t1
t4 = 17.44 mm

Thickness of ribs:
t5 = 0.85 x t1
t5 = 7.41 mm

a = Centre dist. between input and output shafts
a = 94 mm

Diameter of foundation bolts:
d_f = (0.036 x a) + 12
d_f = 15.384 mm

Diameter of bearing bolts:
d_s = 0.75 x d_f
d_s = 11.538 mm

Diameter of bolts securing housing and covering:
d_s = 0.6 x d_f
d_s = 9.23 mm

Diameter of bolts of bearing caps:
d_b = 0.5 x d_f
d_b = 7.692 mm

Distance of foundation bolt axis from housing wall:
$d_w = (1.25 \times d_f) + 5$
$d_w = 24.23$ mm

Distance of bolt axis securing covering and housing from housing wall:
$d_w = (1.2 \times d_s) + 5$
$d_w = 16.076$ mm

Width of foundation flange = $d_w + d_f + 5 = 44.614$ mm

Width of flange connecting housing and covering = $1.2d_s + d_f + 5 = 31.46$ mm

2.6 Oil and Oil Seal:

Pitch line velocity = 3.39 m/s
Temp. in gearbox = maximum is 60°C
So, from the table, ISO VG 220 oil is required.

<u>Amount of oil required :</u>
Tooth depth(h) = 9mm
Oil level must be between - h to 3h
So, according to model and other dimensions,
Volume = (Length of housing) x (breadth of housing) x (height of housing - radius of gear + 2h)
Volume = (315) x (180) x [72 - 58 + (2 x 9)]
Volume = 1814400 mm^3
Volume = 1.8 Litres

So, approximately 1.8 Litres of gear oil is required for smooth functioning of this gearbox.

Temp °C	Pitch line velocity, m/s^2							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1000	680	320	220	150	100	100	68
75	1500	380	460	320	220	150	150	100
80	2200	1000	680	460	220	220	220	150
85	3200	1500	1000	460	320	220	220	150
90	3200	2200	1000	680	460	320	320	220
95		3200	1500	1000	460	460	320	220
100		3200	2200	1000	680	460	460	320

fig-7:Selection of oil
Source:https://www.machinerylubrication.com/Read/926/gear-oils

Oil Seal - Material is NBR Rubber as temperature is below 100°C.
Dimensions are - outer diameter of shaft x bore diameter x width = 16 x 40 x 10
So, #1333 Sized Oil Seal is selected.

3. RESULTS

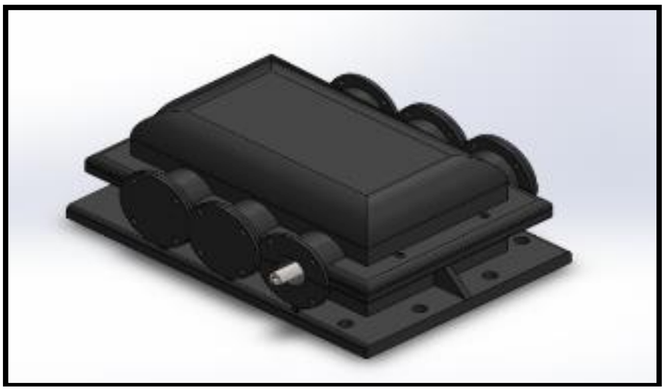

***Fig-7**: Gearbox*
Source: Designed in Catia V5

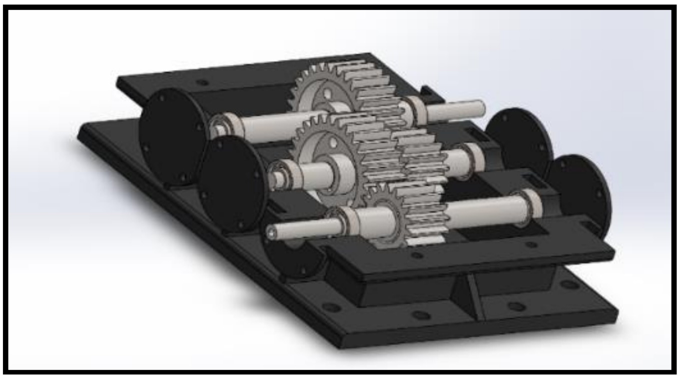

***Fig-8**: Gearbox*
Source: Designed in Catia V5

4. Bill of Materials

Bill of Materials			
Sr no.	Name of Components	Quantity	Material
1	Gear	4	EN31 Steel
2	Shaft	3	C45 Steel
3	Bearing	6	SAE 52100 Steel
4	Casing	2	Cast Iron
5	Key	3	Cast Steel
6	Gear Blank	4	EN31 Steel
7	Bearing Cap	4	Cast Iron
8	Oil Seals	2	NBR Rubber

4. CONCLUSIONS

A gearbox can function as either a reducer or an increaser, depending on the specific requirements of the machine. Staged gearboxes are developed to enhance system compactness and efficiency, with two-stage gearboxes being the most used in industrial applications. These gearboxes consist of four gears and three shafts. Research shows that for two-stage gearboxes, the most efficient configuration involves using identical pinions and gears, meaning that the gear ratio for each stage should be the same. Gears can be designed by performing precise calculations based on the system's requirements. The gear blank is crafted to withstand stress while minimizing mass, achieved by adding webs at strategic locations. Shaft design must consider assembly needs, leading to the use of stepped shafts designed with factors such as gear weight, applied torque, and radial force in mind. Keys are used to secure the gear to the shaft, and bearings are selected to minimize friction during shaft rotation, based on parameters such as total dynamic load and bore diameter. The casing design is primarily determined by the centre distance between the two shafts. By considering all these factors, an optimal gearbox can be designed, ensuring both required strength and minimal weight.

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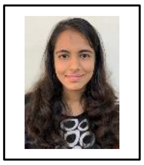
BIOGRAPHIES



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