

Design and Analysis of an 8-Speed Gear box.

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Abstract - Lathe Machine is used in all the engineering applications and in the college Workshops. Lathe machine is used to perform all the basic operations such as drilling, cutting, tapping, turning, etc. with the help of different tools placed in the work environment There are 2 main types of Lathe machines: Capstan Lathe and Turret Lathe. Turret Lathes use a turret head to perform operations. A turret head is a vertical cylindrical revolving tool-holder for bringing different tools into action successively in the lathe machine. The turret head needs to rotate at different speeds to perform different operations on the workpiece. This is controlled by a gearbox. A gearbox which converts high speed input into various output speeds is called a multi speed gearbox. A 6-speed gearbox is usually used. But 8-speed and 9-speed gearboxes give a wider range of speeds and efficiency. **The aim of this project is to design and analyze 8-speed gearbox which meets the functional requirements of a turret gearbox. Design and Analysis have been conducted and a detailed study has been performed for various components in the gearbox.**

Key Words: Lathe, Gearbox, Turret, Gears, Shaft, Ray Diagram, etc.

1. INTRODUCTION

Ever since the development of the first gearbox by Sturtevant in 1904 for automobile application, the utilisation of this technology has evolved marking its presence in various industrial sectors like power generation, petroleum refining, mining and material handling, metal processing and so on. Considering the metal processing and machining aspect, a wide range of industrial gearboxes are used in lathe machines. These gearboxes offer a vivid range of operating speed, and the main aim of this project is to design an 8-Speed gearbox for using in turret lathe machines.

1.1 Deciding Input Requirements

Speed Requirement: After a comprehensive study on speed ranges (N_{min} and N_{max}) for turret lathe, we arrived at a generic average value of 100rpm to 1100rpm. These values are optimal for further processing and give a better design result as compared to other experimental values as well.

Torque Requirement: Turret lathe gearboxes usually require a power of approximately 5-8kW [2].

Motor Selection: Motor selection has been done taking power and input speed conditions into account. An

approximate 8.5HP(6.33kW) and 650rpm motor has been selected. The values will be explained further on in the project.

1.2 Deciding Material

After trial and error with various materials in the design process and opinions of industry professionals, materials for the following components were decided mainly based on strength and durability.

Gear: All the gear are made from 'Grey Cast Iron'.

Shaft: All the shafts have been made of the material 'Steel FeE 580'.

Keys and keyways: All the keys have been made out of 'Steel 50C4'.

Casing: The casing has been made from 'Cast Iron'.

2. DETAILED DESIGN PROCEDURE

2.1 Derivation of Structural Formula

The structural formula gives information about various aspects of the gearbox like number of shafts, number of gears and a basic idea of transmission pairs. It tells about transmission range as well. The procedure followed to reach the structural formula is explained below.

Law of Discretizing Speeds: Discretizing speeds between minimum speed and maximum speed is very important to know our output speeds. These speeds are decided by three basic laws as depicted in the table below.

Sr.No	Comparison Parameter	Arithmetic Progression	Geometric Progression	Harmonic Progression
1	Definition	In arithmetic progression, the difference between any two successive spindle speeds is constant.	In geometric progression, the ratio of any two successive spindle speeds is constant.	In harmonic progression, the difference between reciprocal of any two successive speeds is constant.
2	Z th Spindle Speed	$n_z = \frac{n_{max} - n_{min}}{(z - 1)}$	$\phi = \left[\frac{n_{max}}{n_{min}} \right]^{\frac{1}{z-1}}$	$n_z = \frac{n_{min}}{[1 - (z - 1)C]n_{max}}$
3	Good in	High spindle speed range	High spindle speed range	Low spindle speed range
4	Poor in	Low spindle speed range	Low spindle speed range	High spindle speed range

Fig -1: Laws of discretizing speeds.

Justification of using Geometric Progression: The speeds in gear boxes can be arranged in arithmetic progression (A.P.), geometric progression (G.P.), harmonic progression (H.P), and logarithmic progression (L.P.). However, when the speeds are arranged in G.P., it has the following advantages.

- The speed loss is minimum i.e., Speed loss = Desired optimum speed – Available speed
- The number of gears to be employed is minimum.
- G.P. provides a more even range of spindle speeds at each step.
- The layout is comparatively very compact.
- Productivity of a machining operation, i.e., surface area of the metal removed in unit time, is constant in the whole speed range.
- G.P. machine tool spindle speeds can be selected easily from preferred numbers. Because preferred numbers are in geometric progression.

Deciding Speed Ratio from Standard Series: In gearbox design a set of preferred step ratio or preferred numbers is used to obtain the series of output speed of gearbox. The preferred step ratio is mentioned as basic series named as R5, R10, R20, R40 and R80. Each basic series has a specific step ratio. In our case, we have N_{min} as 100rpm and N_{max} as 1100rpm.

$$\text{Thus, } \phi = \left(\frac{1100}{100}\right)^{\frac{1}{7}} = 1.4$$

The Step Ratio 1.4 comes under R20 series.

Thus, the spindle speeds calculated are:

$$N_1 = 100 * \phi^0 = 100\text{rpm}$$

$$N_2 = 100 * \phi^1 = 140\text{rpm}$$

$$N_3 = 100 * \phi^2 = 196\text{rpm} = 200\text{rpm}$$

$$N_4 = 100 * \phi^3 = 274.4\text{rpm} = 280\text{rpm}$$

$$N_5 = 100 * \phi^4 = 392\text{rpm} = 400\text{rpm}$$

$$N_6 = 100 * \phi^5 = 560\text{rpm}$$

$$N_7 = 100 * \phi^6 = 784\text{rpm} = 780\text{rpm}$$

$$N_8 = 100 * \phi^7 = 1097\text{rpm} = 1100\text{rpm}$$

These are subject to change based on number of teeth chosen.

Deriving Structural Formula:

Let,

n = Number of speeds available at the spindle.

p1, p2, p3..... = stage numbers in gearbox.

X1, X2, X3..... = Characteristics of the stages.

Then, structural formula is given as:

$$N = p_1(X_1).p_2(X_2).p_3(X_3).p_4(X_4)$$

Were,

$$X_1 = 1; X_2 = p_1; X_3 = p_1 \times p_2; X_4 = p_1 \times p_2 \times p_3$$

Table -1:

SR NO:	Number of speeds	Preferred Structural Formula
1	6 Speeds	3 (1) . 2 (3) 2 (1) . 3 (2)
2	8 Speeds	2 (1) . 2 (2) . 2 (4) 4 (1) . 2 (4)
3	9 Speeds	3 (1) . 3 (3)
4	12 Speeds	3 (1) . 2 (3) . 2 (6) 2 (1) . 3 (2) . 2 (6) 2 (1) . 2 (2) . 3 (4)

For 8 – Speed Gearbox:

$$2 \times 2 \times 2 = 8$$

Therefore,

$$P_1 = 2, X_1 = 1$$

$$P_2 = 2, X_2 = 2$$

$$P_3 = 2, X_3 = 2 \times 2 = 4$$

Then, structural formula = 2 (1) . 2 (2) . 2 (4)

Determining number of Shafts, Gears and Transmissions: Calculation of No. of shafts to draw kinematic arrangement:

$$\text{No. of shafts} = \text{No. of stages} + 1.$$

That is,

If 3 stages are there in gearbox design, then no. of shaft will be = 3 + 1 = 4 shafts.

As per gear kinematics,

$$\text{Shaft 1} = p_1 \text{ gears} = 2 \text{ gears.}$$

$$\text{Shaft 2} = (p_1 + p_2) \text{ gears} = 4 \text{ gears.}$$

$$\text{Shaft 3} = (p_2 + p_3) \text{ gears} = 4 \text{ gears.}$$

$$\text{Shaft 4} = p_1 \text{ gears} = 2 \text{ gears.}$$

Limiting Value of (lg)stage:

2.2 Structural Ray Diagram

Drawing Structural Ray Diagram: The ray diagram is graphical representation of the drive arrangement in general form. In other words, the ray diagram is a graphical representation of the structural formula, as shown in figure. It provides the following data on the drive:

- The number of stages (a stage is a set of gear trains arranged on two consecutive shafts).

- The number of speeds in each stage.
- The order of kinematic arrangement of the stages.
- The specific values of all the transmission ratios in the drive.
- The total number of speeds available at the spindle.

Procedure: The following is the procedure to plot structural ray diagram:

- In this diagram, shafts are shown by vertical equidistant and parallel lines.
- The speeds are plotted vertical on a logarithmic scale with $\log \phi$ as a unit.
- Transmission engaged at definite speeds of the driving and driven shafts are shown on the diagram by rays connecting the points on the shaft lines representing these speeds.
- Figure shows the ray diagram for a 9-speed gear box, having the structural formula, $Z = 2 (1). 2 (2). 2(4)$.

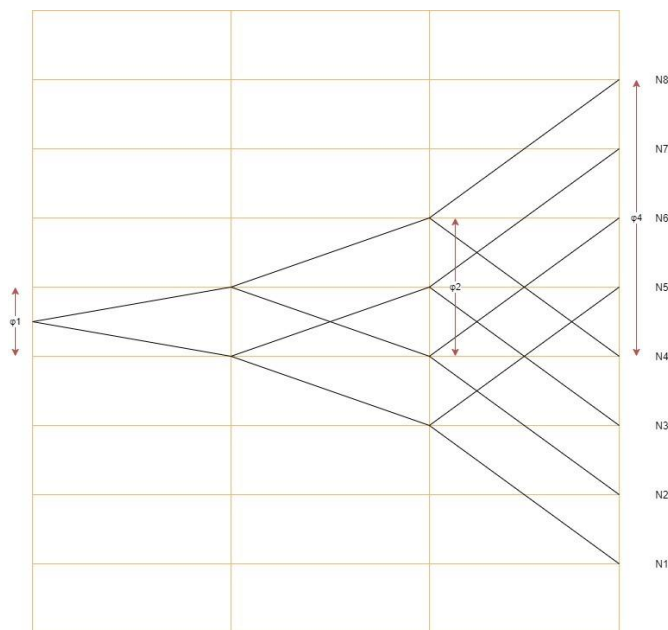


Fig -2: Structural Ray Diagram

2.3 Speed Ray Diagram:

Significance: A speed ray diagram is a diagram depicting all possible exact values of speeds at which the multi speed gearbox can run. It gives us a clear picture of gear ratios as well.

Plotting Speed Diagram: A basic procedure which is followed is as follows-

- Draw $u+1$ vertical line at convenient distance where $u =$ Number of stages (for above formula $u=4$ three transmission group) Note : first vertical line

represents the transmission from motor shaft and the rest represent the transmission groups of speed box. 2).

- Draw array of horizontal lines is equal to the number of speed steps z of speed intersecting the vertical lines distance of $\log \phi$ from each other.

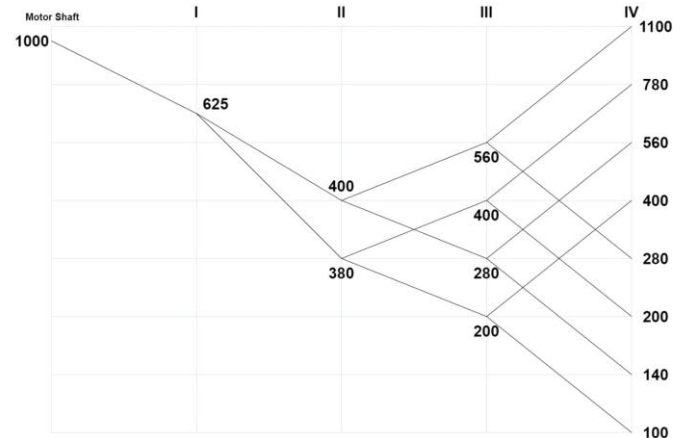


Fig -3: Speed Ray Diagram

2.4 Preliminary Gearbox assembly:

Basic Assembly Diagram Optimisation: The following diagram shows the basic gearbox assembly where G_n is the number of the n^{th} gear and Z_n is number of teeth of G_n^{th} gear.

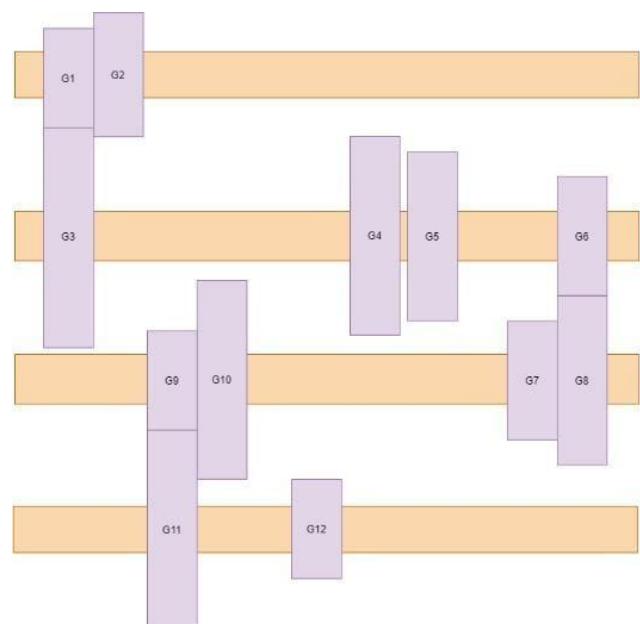


Fig -4: Preliminary Gearbox assembly

2.5 Determining Gear Ratios and Number of teeth:

Approximating Gear Ratio from Speed Ratio and Constraints: As we have seen in the picture above, there are 12 gears of which we need to find the number of teeth. There

are a few constraints to take care of after which we can easily select number of teeth from the design data book.

The summation of number of teeth of the meshing gears between two shafts should be exactly equal.

$$\frac{d_1}{2} + \frac{d_3}{2} = \frac{d_2}{2} + \frac{d_4}{2} = cd$$

$$d = mz$$

m = constant for all

$$\text{thus, } \frac{z_1}{2} + \frac{z_3}{2} = \frac{z_2}{2} + \frac{z_4}{2}$$

$$\text{thus, } z_1 + z_3 = z_2 + z_4$$

$$\text{similarly, } z_5 + z_7 = z_6 + z_8$$

$$\text{and } z_9 + z_{11} = z_{10} + z_{12}$$

Therefore, by selecting standard number of teeth from design data book to satisfy the above given gear ration equation, we get the following number of teeth per gear:

Table -2:

Gear Number	Number of Teeth
01	Z01 = 20
02	Z02 = 25
03	Z03 = 45
04	Z04 = 40
05	Z05 = 24
06	Z06 = 34
07	Z07 = 34
08	Z08 = 24
09	Z09 = 20
10	Z10 = 40
11	Z11 = 40
12	Z12 = 20

Corrected final speeds:

Old Speed (rpm)	New Speed (rpm)	Difference (rpm)	% Change
100	98	2	2%
140	137	3	2.1%
200	196	4	2%
280	276	4	1.4%
400	392	8	2%
560	551	9	1.6%
780	787	7	0.8%
1100	1106	6	0.5%

2.6 Determining final gear dimensions:

Determination of module:

Choice-1	1.0	1.25	1.5	2.0	2.5	3.0	4.0
Preferred	5.0	6.0	8.0	10	12	16	20
Choice-2	1.125	1.375	1.75	2.25	2.75	3.5	4.5
	5.5	7	9	11	14	18	

The standard proportions of the gear tooth in terms of module *m*, for 20° full depth system are rewritten here.

$$\text{addendum } (h_a) = (m)$$

$$\text{dedendum } (h_f) = (1.25 m)$$

$$\text{clearance } (c) = (0.25 m)$$

$$\text{working depth } (h_k) = (2 m)$$

$$\text{whole depth } (h) = (2.25 m)$$

$$\text{tooth thickness } (s) = (1.5708 m)$$

$$\text{tooth space} = (1.5708 m)$$

$$\text{fillet radius} = (0.4 m)$$

Fig - 5: Determination of Module

By our calculations, we got a module of 2.8. Rounding this off, we chose a module of 3mm.

Module m = 3mm.

Determination of Gear diameter:

Gear Number	Diameter of Gear (mm)
1	D1 = m*Z1 = 60
2	D2 = m*Z2 = 75
3	D3 = m*Z3 = 135
4	D4 = m*Z4 = 120
5	D5 = m*Z5 = 72

6	$D6 = m \cdot Z6 = 102$
7	$D7 = m \cdot Z7 = 102$
8	$D8 = m \cdot Z8 = 72$
9	$D9 = m \cdot Z9 = 60$
10	$D10 = m \cdot Z10 = 120$
11	$D11 = m \cdot Z11 = 120$
12	$D12 = m \cdot Z12 = 60$

4	34	14.98
5	40	21.98
6	45	28.616

Determination of Gear dimensions:

We have selected 20 full depth involute teeth system. As we input our module, system of teeth and number of teeth into Solid works, it gives us the remaining outputs.

	14.5° full depth system	20° full depth system	20° stub system
Pressure angle	14.5°	20°	20°
Addendum	m	m	0.8 m
Dedendum	1.157 m	1.25 m	m
Clearance	0.157 m	0.25 m	0.2 m
Working depth	2 m	2 m	1.6 m
Whole depth	2.157 m	2.25 m	1.8 m
Tooth thickness	1.5708 m	1.5708 m	1.5708 m

Fig - 6: Types of teeth system

By considering 20-degree full depth system:

- Addendum (h_a) = $m = 3\text{mm}$
- Dedendum (h_f) = $1.25 m = 3.75\text{mm}$
- Clearance (c) = $0.25 m = 0.75\text{mm}$
- Working depth (h_k) = $2 m = 6\text{mm}$
- Whole depth (h) = $2.25 m = 6.75\text{mm}$
- Tooth thickness (s) = $1.5708 m = 4.7\text{mm}$
- Tooth space = $1.5708 m = 4.7\text{mm}$
- Fillet radius = $0.4 m = 1.2\text{mm}$

Weights of individual gears:

Post obtaining the dimensions required for each gear, we design them on modelling software and then obtain the weights as follows:

Sr. No.	Gear with teeth	Weight(N)
1	20	4.28
2	24	6.76
3	25	7.595

Design of shaft:

Torque input to individual shafts: The selected motor provides an input power of 6.33kW to the initial shaft. The first shaft rotates at 625rpm. From the formula,

$$T = \frac{60P}{2\pi N}$$

We get Torsional moment of first shaft $T1 = 96.71 \text{ N-m}$. Similarly, finding out all the possible torsional moments of all shafts, we consider the maximum one for calculation purposes. The results are tabulated as follows:

Shaft Number	Torque acting(N-m)							
	1	96.71						
2	42.98				60.44			
3	30.33	60.88		85.62	42.66			
4	15.16	60.66	30.44	121.76	42.81	171.24	21.33	85.32

The torques marked in red are the highest torques acting on the respective shafts. This be our torsional moments to be considered during design.

Calculation and Direction of Gear Forces: There are 3 main types of forces acting when a gear pair meshes. They are as follows:

- **Tangential Force F_t :** The force acting tangential to the gear surface at the point of contact with meshing gear.
- **Radial force F_r :** The force acting towards the centre of the gear at the point of contact with the meshing gear.
- **Axial Force F_a :** The force that acts in the z-direction along the shaft due to the angle of the teeth is the axial force. As our teeth don't have an angle with respect to the gear base, this force for our consideration, is 0.

As our gearbox is a spur gear system, we will consider the tangential and radial forces of all gears for calculation purposes.

For each gear

$$F_t = \frac{2000T}{d} \quad F_r = \frac{F_t}{\tan \alpha}$$

Gear	Force Type	Value (N)
G1	Tangential force Ft	3223.6
	Radial Force Fr	1173.3
G2	Tangential force Ft	2758.9
	Radial Force Fr	938.65
G3	Tangential force Ft	322.2
	Radial Force Fr	117.28
G4	Tangential force Ft	362.5
	Radial Force Fr	131.93
G5	Tangential force Ft	426.47
	Radial Force Fr	115.22
G6	Tangential force Ft	604.16
	Radial Force Fr	219.89
G7	Tangential force Ft	427.78
	Radial Force Fr	115.69
G8	Tangential force Ft	301.96
	Radial Force Fr	109.90
G9	Tangential force Ft	513.33
	Radial Force Fr	186.83
G10	Tangential force Ft	256.67
	Radial Force Fr	93.41
G11	Tangential force Ft	513.33
	Radial Force Fr	186.83
G12	Tangential force Ft	1026.6
	Radial Force Fr	373.67

Modified Weight of Gear: From Solid works, we have the weight of the solid gear. As there will be a shaft of diameter 'd' through this, we must reduce that much weight. Therefore, giving a new modified weight to work with.

Let,

W_{new} = Modified Weight in Newton

W_{old} - Solid works output weight in Newton

W_r - Weight of Inner part to be removed in Newton.

d - diameter of shaft in m

f - face width of gear in m

ρ - density of gear in kg/m³

g - acceleration due to gravity in m/s²

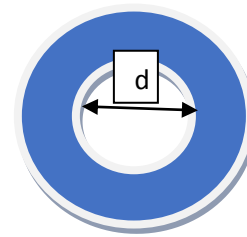


Fig -1: Name of the figure

$$W_{new} = W_{old} - W_r$$

W_r = Weight of cylindrical part of gear

$$W_r = \left\{ \left[\left(\frac{\pi}{4} d^2 \right) * f \right] * \rho \right\} * g$$

Calculating Maximum Bending Moment for each Shaft:

Our aim right now is to calculate maximum bending moment acting on the shaft for the calculation of shaft diameter. This can be done in terms of 'd' as all values in our SFD are in terms of 'd'.

The maximum bending moment is area between the SFD and the shaft axis above the shaft axis and the point of maximum bending is the point where the SFD graph crosses the shaft axis. Thus, calculating the areas, we get the following results.

Shaft	Maximum Bending Moment (Mb)	Maximum Torsional Moment (Mt)
1	252261.7	346544.50
2	282504.30	264387.4
3	294134.50	357548.00
4	263150.23	346753.82

Determining Shaft Diameter: We will be designing the shaft using ASME code of shaft design. According to this code, the permissible shear stress τ_{max} for the shaft without keyways is taken as 30% of yield strength in tension or 18% of the ultimate tensile strength of the material, whichever is minimum. Therefore, $\tau_{max} = 0.30 S_{yt}$ or $\tau_{max} = 0.18 S_{ut}$ (whichever is minimum).

If keyways are present, the above values are to be reduced by 25 per cent. According to the ASME code, the bending and torsional moments are to be multiplied by factors k_b and k_t respectively, to account for shock and fatigue in operating condition. The ASME code is based on maximum shear stress theory of failure.

$$\tau_{max} = \frac{16}{\pi d^3} \sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

were,

kb = combined shock and fatigue factor applied to bending moment

kt = combined shock and fatigue factor applied to torsional moment

M_b = Maximum Bending moment acting on shaft

M_t = Maximum torsional moment acting on shaft

Application	<i>k_b</i>	<i>k_t</i>
(i) Load gradually applied	1.5	1.0
(ii) Load suddenly applied (minor shock)	1.5–2.0	1.0–1.5
(iii) Load suddenly applied (heavy shock)	2.0–3.0	1.5–3.0

Fig -7: Shock and fatigue factors

For the gearbox, it is a minor shock application. Thus, the load is applied suddenly. We have taken as intermediate value of $k_b = 1.5-2.0$ and $k_t = 1.0-1.5$.

Our shaft material is 'Steel FeE 580 ($S_{ut} = 770$ and $S_{yt} = 580$ N/mm²).

Calculating permissible shear stress,

$$0.30 S_{yt} = 0.30(580) = 174 \text{ N/mm}^2$$

$$0.18 S_{ut} = 0.18(770) = 138.6 \text{ N/mm}^2$$

The lower of the two values is 138.6 N/mm² and there are keyways on the shaft.

$$\tau_{max} = 0.75(138.6) = 103.95 \text{ N/mm}^2$$

Combining values of τ_{max} from both above equations,

$$d^3 = \frac{16}{\pi * \tau_{max}} \sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

Using max bending moment and torsional moment values for each shaft, we get the following shaft diameters which are taken in accordance with the standard shaft table.

Diameters (mm)			
5	20	45	90
6	22	50	100
8	25	55	110
10	28	60	120
12	30	65	140
14	32	70	160
16	35	75	180
18	40	80	200

Fig -8: Shaft diameters

Shaft Number	Shaft diameter (mm)
1	28
2	30
3	30
4	28

2.7 Design of keys and keyways:

Type of key: There are various types of keys. We have selected **Rectangular Saddle Key**. Its design has been conducted as follows.

Design on load basis: The material for key is Steel 50C4 ($S_{yt} = 460$ N/mm²) and the factor of safety is taken as 3. For key material, the yield strength in compression can be assumed to be equal to the yield strength in tension. Keyways are on the second and fourth shaft.

$$S_{yc} = S_{yt} = 460 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{460}{3} = 153.33 \text{ N/mm}^2$$

According to maximum shear stress theory of failure,

$$S_{sy} = 0.5 * S_{yt} = 0.5(460) = 230 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{230}{3} = 76.67 \text{ N/mm}^2$$

b – breadth of key

h – height of key

l – length of key

d – diameter of shaft

Industry standard is to select a basic dimension of

$$b = h = d/4 = 30/4 = 7.5 = 7 \text{ mm}$$

$$l = \frac{2M_t}{\tau db} \quad \text{or} \quad l = \frac{4M_t}{\sigma_c dh}$$

Taking maximum value using both equations, we get key length = 28.7mm

Now selecting from standard table,

Shaft diameter		Key size	Keyway depth
Above	Up to and including	b × h	
6	8	2 × 2	1.2
8	10	3 × 3	1.8
10	12	4 × 4	2.5
12	17	5 × 5	3.0
17	22	6 × 6	3.5
22	30	8 × 7	4.0
30	38	10 × 8	5.0
38	44	12 × 8	5.0
44	50	14 × 9	5.5
50	58	16 × 10	6.0
58	65	18 × 11	7.0
65	75	20 × 12	7.5
75	85	22 × 14	9.0
85	95	25 × 14	9.0
95	110	28 × 16	10.0
110	130	32 × 18	11.0
130	150	36 × 20	12.0
150	170	40 × 22	13.0
170	200	45 × 25	15.0
200	230	50 × 28	17.0

Fig -9: Determination of keys and keyways

Thus, we select the key with dimension (b,h) as (8,7) and a length of 30mm.



Fig -10: Shafts

2.8 Selection of bearings:

Defining Bearing Life: To select bearings, it is necessary to specify life of the bearings. This can be done from standard bearing selection table. The recommended bearing life for gear applications is given in the table below:

(i) Machines used intermittently such as lifting tackle, hand tools and household appliances	4000–8000 h
(ii) Machines used for eight hours of service per day, such as electric motors and gear drives	12 000–20 000 h
(iii) Machines used for continuous operation (24 h per day) such as pumps, compressors and conveyors	40 000–60 000 h

Fig -11: Bearing life.

The values given in the above table are only for general guidance. For a particular application, the designer should consider the experience, the difficulties faced by the customer in replacing the bearing and the economics of breakdown costs.

As we are relatively new to the designing aspect, we have taken an estimated 15,000 hr (L_{10h}) due to it being an intermediate value for a service machine.

Selection from Manufacturer’s Catalogue: To select the final bearings, we will need radial force acting. As there is no axial force, total force $P = Fr$. Diameter of all 4 shafts are taken into consideration. Then using max speed of all shafts, we calculate the rated bearing life L_{10} by the formula,

$$L_{10} = \frac{60 \cdot N \cdot L_{10h}}{10^6}$$

After we get this value for all shafts, we calculate the dynamic load capacity C by,

$$C = P(L_{10})^{\frac{1}{3}}$$

Thus, we get the values of C as follows:

Shaft Number	Dynamic Load Capacity (N)
1	9682
2	12500
3	5411
4	3700

Based on these values of C, we must select a bearing from the manufacturer’s catalogue whose dynamic load capacity is more than our requirement. The table for selection is given below.

Principal dimensions (mm)			Basic load ratings (N)		Designation
d	D	B	C	C ₀	
12	21	5	1430	695	61801
	28	8	5070	2240	6001
	32	10	6890	3100	6201
	37	12	9750	4650	6301
15	24	5	1560	815	61802
	32	9	5590	2500	6002
	35	11	7800	3550	6202
	42	13	11400	5400	6302
17	26	5	1680	930	61803
	35	10	6050	2800	6003
	40	12	9560	4500	6203
	47	14	13500	6550	6303
20	32	7	2700	1500	61804
	42	8	7020	3400	16404
	42	12	9360	4500	6004
	47	14	12700	6200	6204
25	52	15	15900	7800	6304
	72	19	30700	16600	6404
	37	7	3120	1960	61805
	47	8	7610	4000	16005
30	47	12	11200	5600	6005
	52	15	14000	6950	6205
	62	17	22500	11400	6305
	80	21	35800	19600	6405
30	42	7	3120	2080	61806
	55	9	11200	5850	16006
	55	13	13300	6800	6006
	62	16	19500	10000	6206
	72	19	28100	14600	6306

Fig -12: Bearing selection

The following table tells us which bearings are sufficient to satisfy our requirement and are thus selected.

Shaft Number	Bearing Number	Principal Dimensions(d,D,B) (all mm)
1	6006	30,55,13
2	6006	30,55,13
3	6006	30,55,13
4	16006	30,55,9

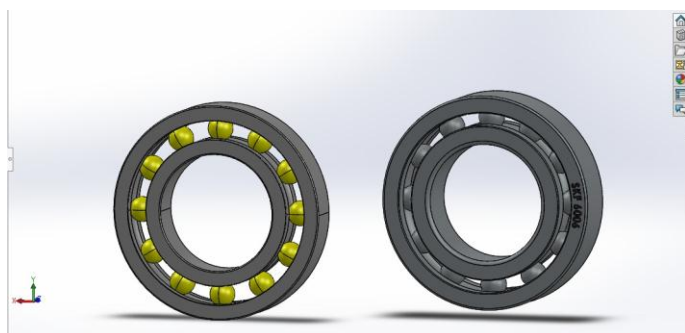


Fig -13: Selected bearings

2.9 Casing Design:

Casing consideration: Following steps have been taken into consideration while designing the casing of the gearbox.

- Compact
- Weight optimisation
- Leak proof (bearing fitted in casing)

Visual Representation: The part of the casing which holds the gearbox looks as follows.

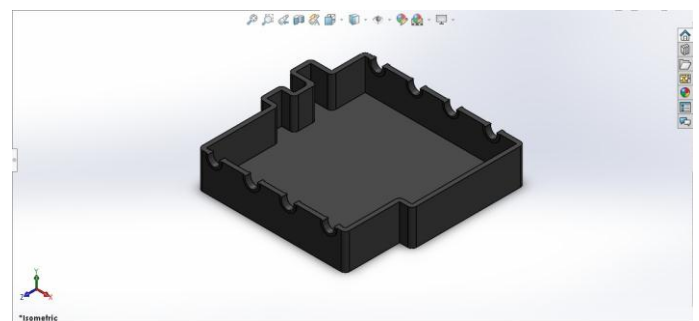


Fig -14: Gearbox casing

2.10 Final Gearbox assembly:

Final Gearbox assembly: The final Gearbox assembly with corrected values is as follows:

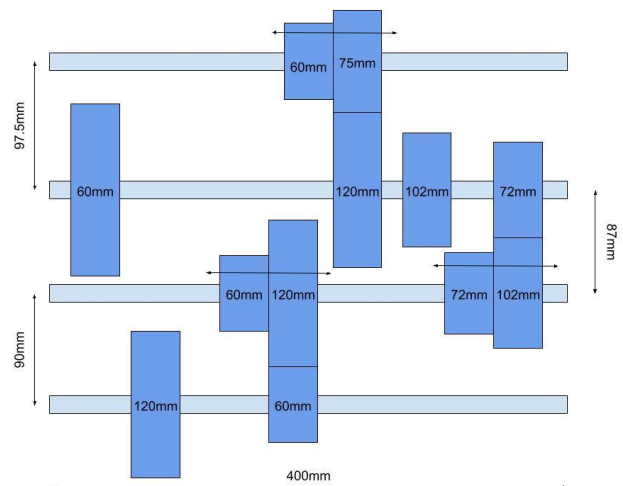


Fig -15: Schematic of final assembly

3. 3D Modelling and Analysis of gearbox.

Now that we had all the parameters required for designing the gearbox on CAD tools, the CAD models for all the gears, shafts, bearings, and the casing was created. The CAD was modelled in Solid works software.

3.1 CAD Models – Gears and shafts

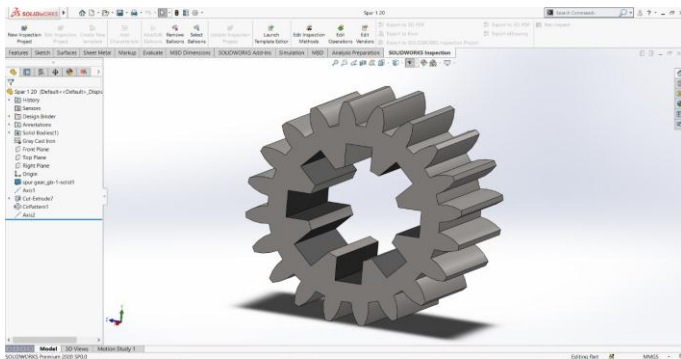


Fig -16: Shaft 1 Gear 1

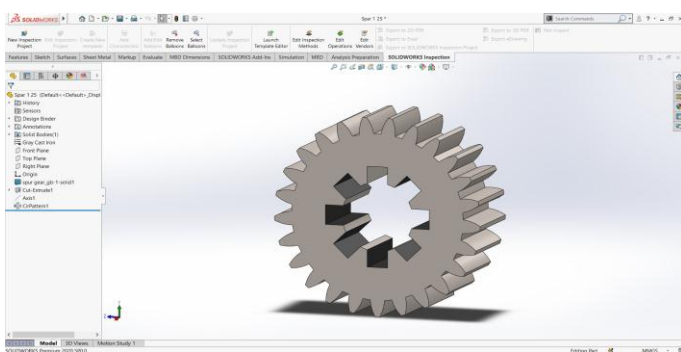


Fig -17: Shaft 1 Gear 2

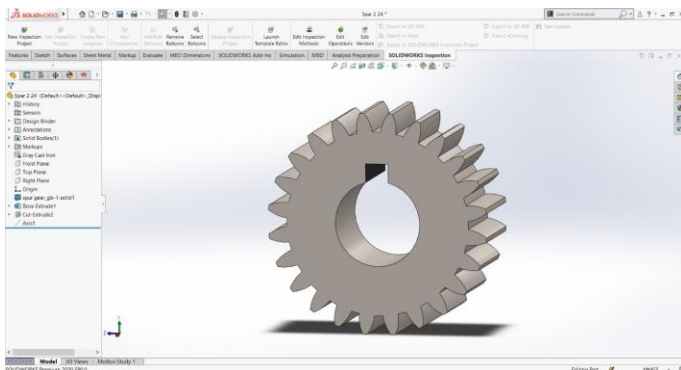


Fig -18 Shaft 2 Gear 1

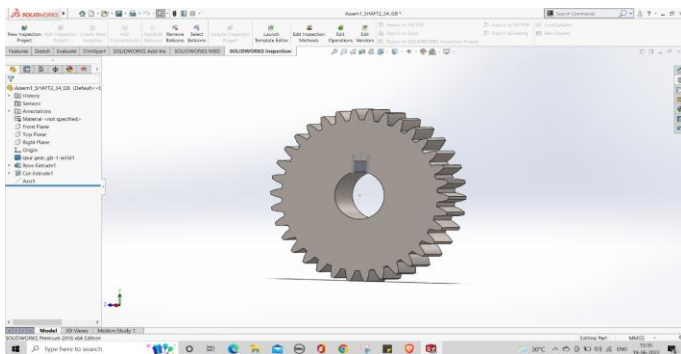


Fig -19: Shaft 2 Gear 2

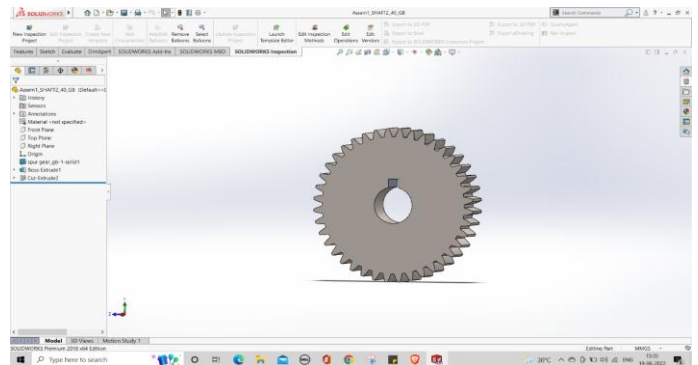


Fig -20: Shaft 2 Gear 3

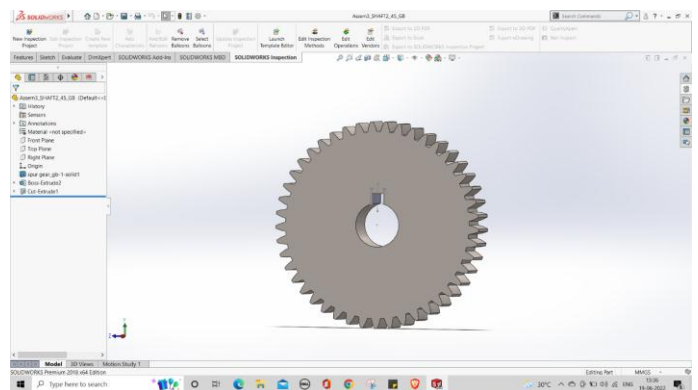


Fig -21: Shaft 2 Gear 4

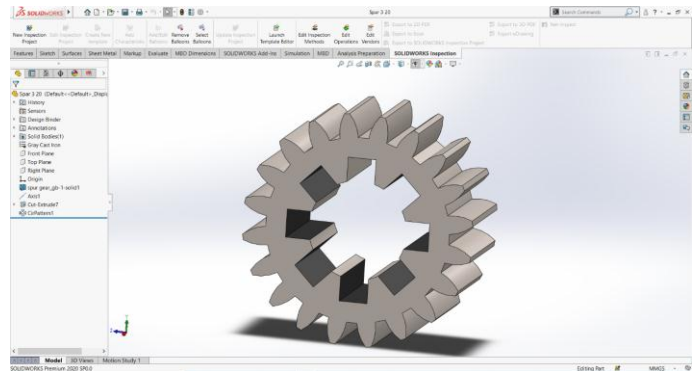


Fig -22: Shaft 3 Gear 1

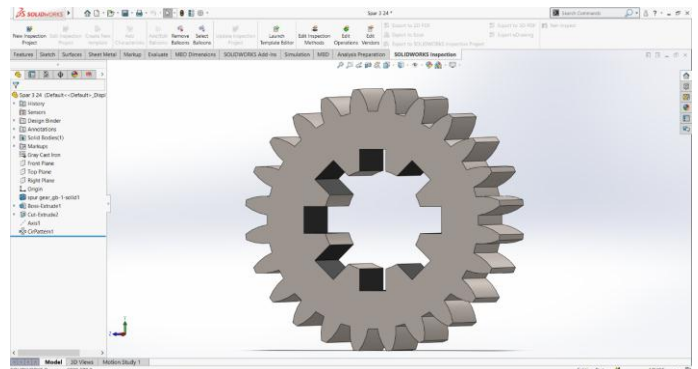


Fig -23: Shaft 3 Gear 2

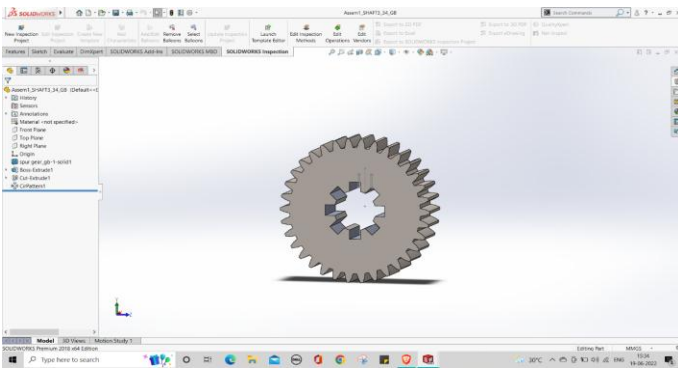


Fig -24: Shaft 3 Gear 3

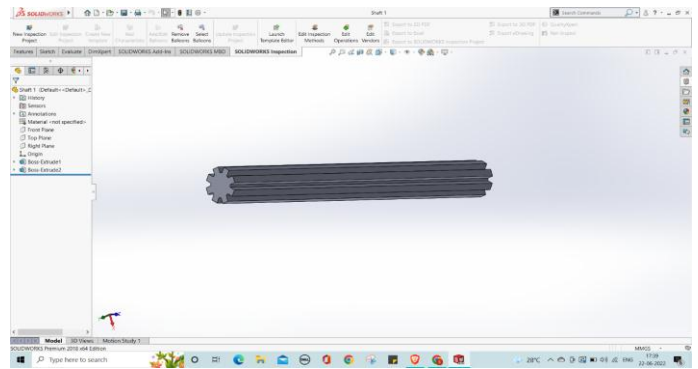


Fig -28: Shaft 1

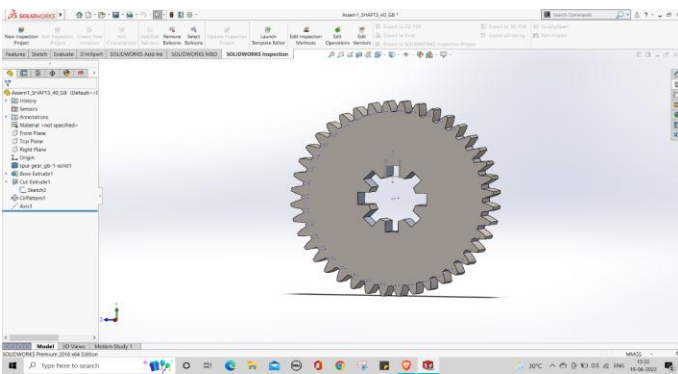


Fig -25: Shaft 3 Gear 4

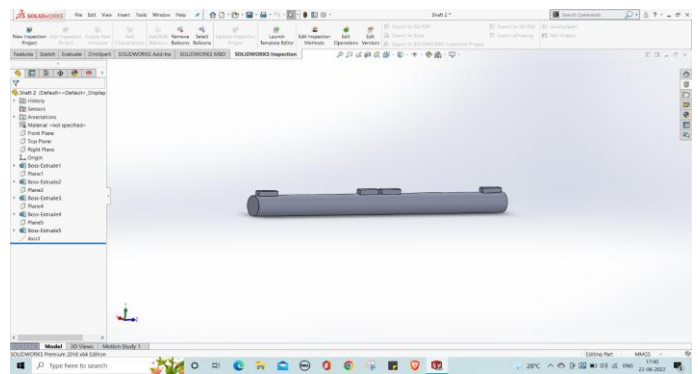


Fig -29: Shaft 2

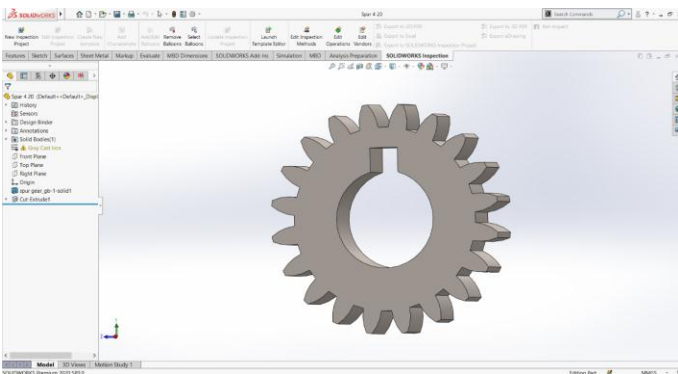


Fig -26: Shaft 4 Gear 1

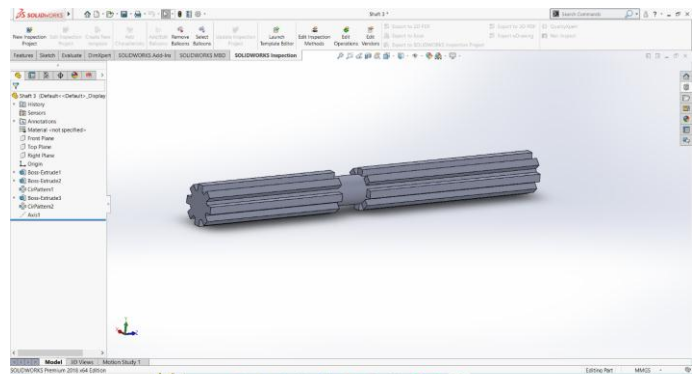


Fig -30: Shaft 3

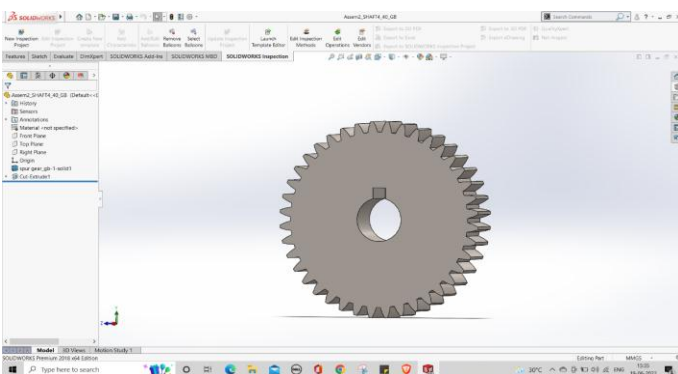


Fig -27: Shaft 4 Gear 2

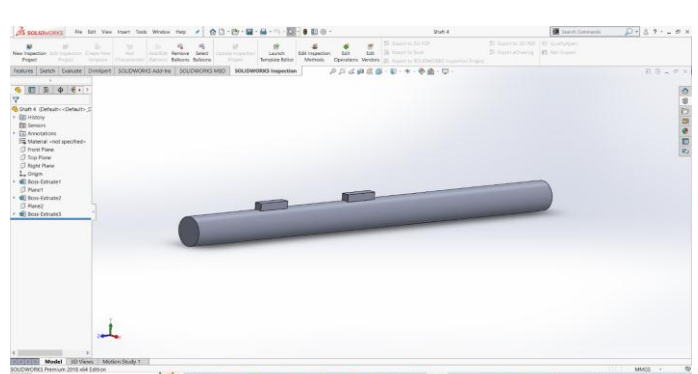


Fig -31: Shaft 4

3.2 Final 3D Model:

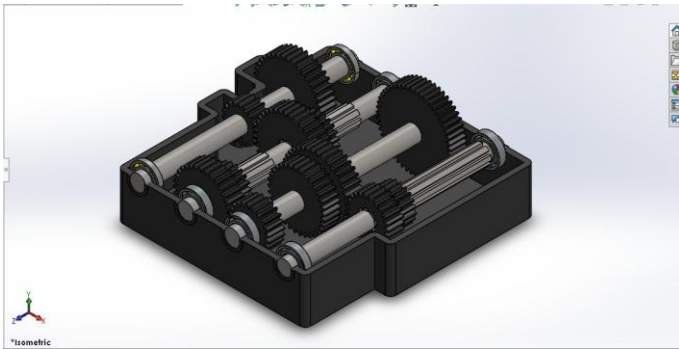


Fig -32: Final Gear box assembly

3.3 Gearbox and shaft analysis:

Analysis of Gearbox

The gear pairs have been analyzed on ANSYS 18.1 software. Analysis of components can be done in various ways like static, dynamic, transient, etc. depending up on application. Static analysis is concerned with determination of response of a gear to steady loads whose response remains unchanged with time. The response of the gear is expressed in terms of stress, strain, displacement. The tool used in static analysis is Static structural.

Procedure: The finite element analysis procedure of the spur gear was given below:

1. A three-dimensional model of the spur gear was created using the pro/engineer CAD software.
2. The material properties were defined for gears.
3. The model was meshed using finite element software.
4. Boundary conditions for ANSYS Workbench as mentioned below.
5. Fixed displacement constraint was applied on gear.
6. Moment was applied on gear.
7. To arrest the displacement on x, y, z directions and rotations on x, y directions remote displacement constraint is applied on pinion surface.

Contacts – Frictionless contacts between the gear pairs and the gear and shaft pairs.

Boundary Conditions – Torques have been applied to the input gear surfaces.

Material – Cast Iron has been considered for all the gear pairs.

Deformation Results:

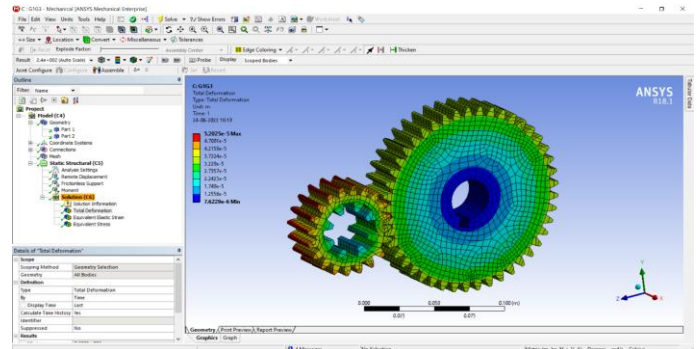


Fig -33: Total deformation for meshing pair 1

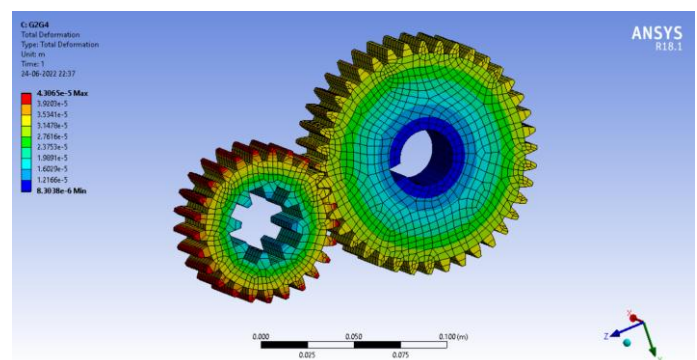


Fig -34: Total deformation for meshing pair 2

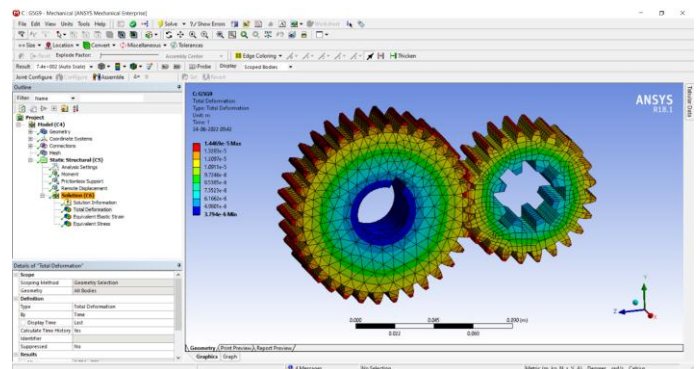


Fig -35: Total deformation for meshing pair 3

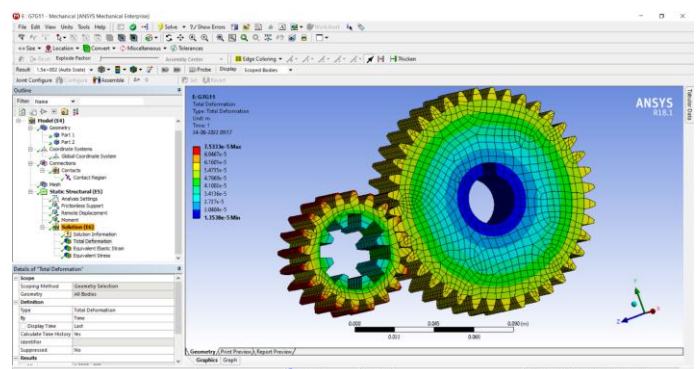


Fig -36: Total deformation for meshing pair 4

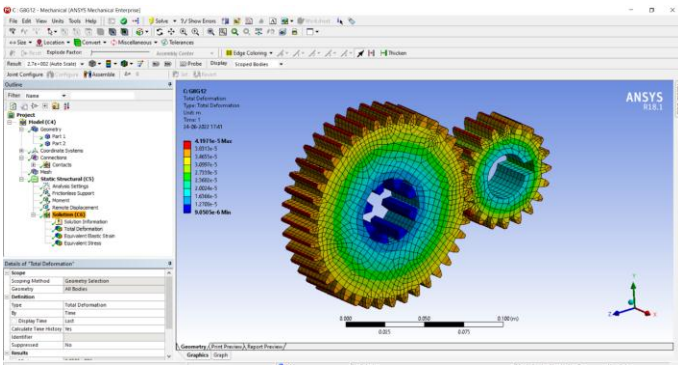


Fig -37: Total deformation for meshing pair 5

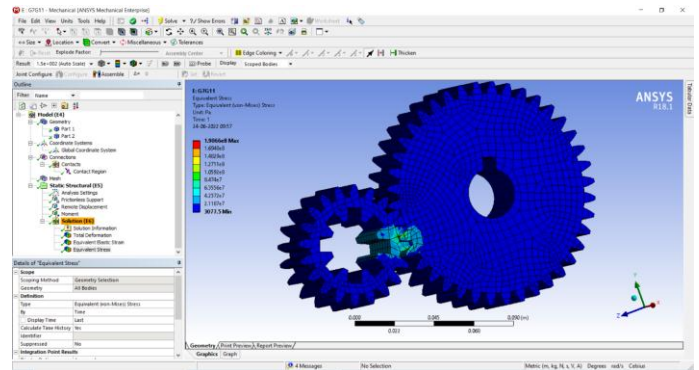


Fig -41: Equivalent stress for meshing pair 4

Equivalent Stress:

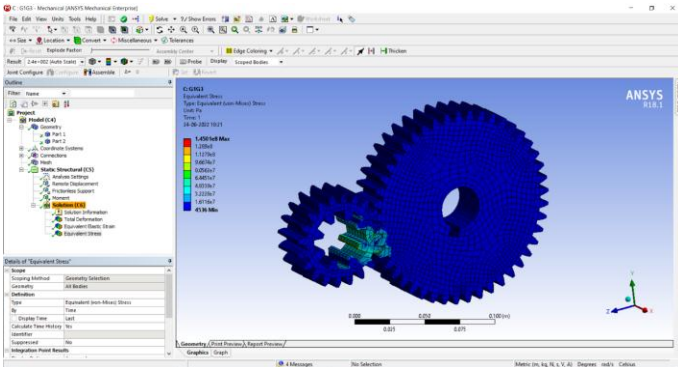


Fig -38: Equivalent stress for meshing pair 1

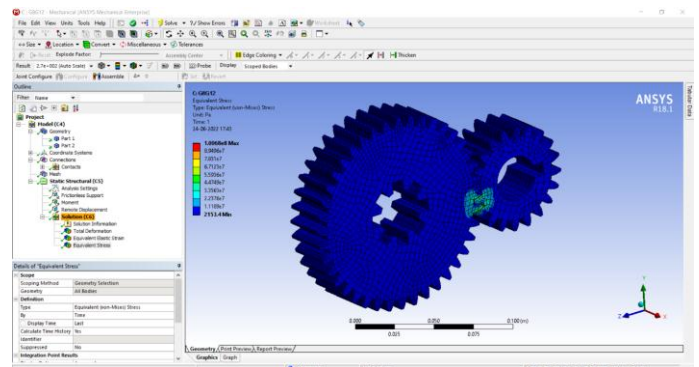


Fig -42: Equivalent stress for meshing pair 5

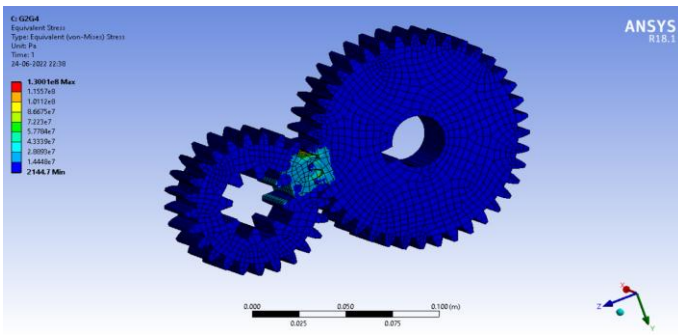


Fig -39: Equivalent stress for meshing pair 2

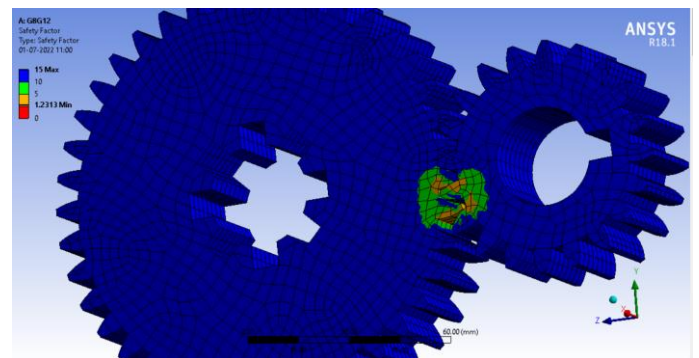


Fig -43: Factor of safety analysis

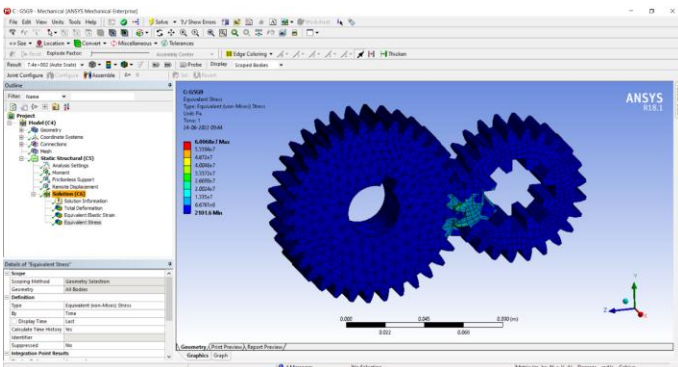


Fig -40: Equivalent stress for meshing pair 3

The value obtained for the FOS for all the gear pairs is mentioned below –

Gear Pair	FOS Min	FOS Max
1	1.3674	15
2	1.3896	15
3	1.2645	15
4	1.3564	15
5	1.2313	15

Hence, we can conclude that the design is within the permissible limits for the given material and geometry.

Analysis of shafts:

The tool used in static analysis is Static structural.

Procedure: The finite element analysis procedure of the shafts was given below:

1. A three-dimensional model of the shafts was created using the pro/engineer CAD software.
2. The material properties were defined for shaft.
3. The model was meshed using finite element software.
4. Boundary conditions for ANSYS Workbench as mentioned below.
5. Fixed displacement constraint was applied on gear.

Contacts – Fixed contacts have been provided at the ends of the shafts.

Boundary Conditions – Forces were applied on the shaft as Point loads and UDL.

Total Deformation:

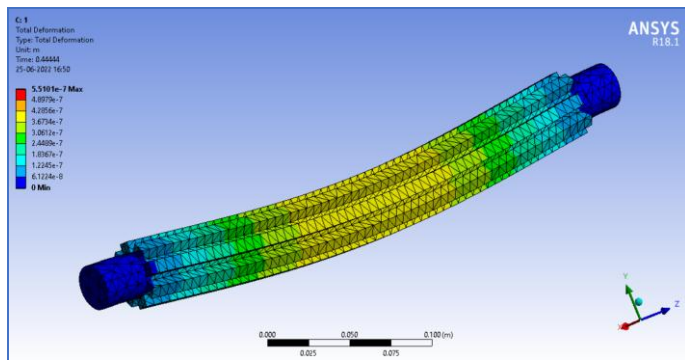


Fig -44: Shaft 1

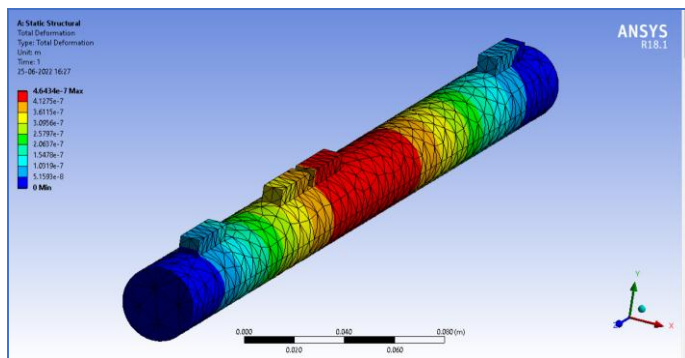


Fig -45: Shaft 2

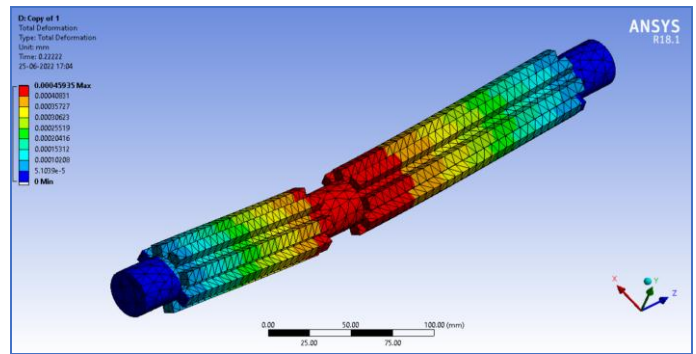


Fig -46: Shaft 3

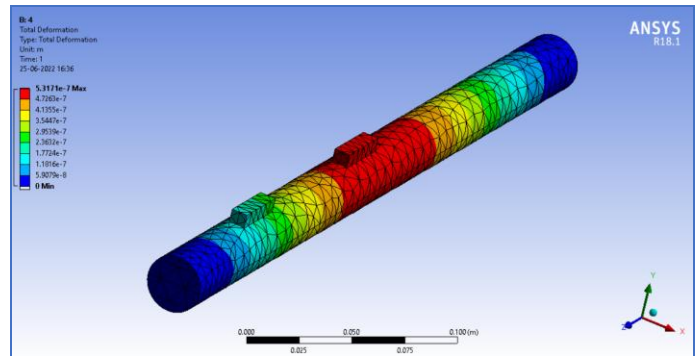


Fig -47: Shaft 4

4.0 Analysis Results summary:

4.1 Gear Analysis:

Pair	Min Def (m)	Max Def (m)
1	7.622e-6	5.202e-5
2	8.303e-6	4.303e-5
3	3.794e-6	1.446e-5
4	1.353e-6	7.533e-5
5	9.050e-6	4.197e-5

Pair	Min stress	Max stress
1	4536	1.45e8
2	2144.7	1.30e8
3	2101.6	6.01e8
4	3077.5	1.91e8
5	2153.4	1.01e8

4.2 Shaft Analysis

Shaft	Min def (m)	Max def (m)
1	0	5.51e-7
2	0	4.64e-7
3	0	4.59e-7
4	0	5.31e-7

Shaft	Min stress (Pa)	Max stress (Pa)
1	69.6	3.854e5
2	1590.6	2.763e5
3	189	3.017e5
4	666.45	2.814e5

Shaft	Min strain	Max strain
1	1.523e-9	1.938e-6
2	1.041e-9	1.394e-6
3	7.205e-9	1.571e-6
4	3.332e-9	1.414e-6

5. CONCLUSIONS

In conclusion, a standard design procedure for the design of this 8-speed gearbox was followed considering various research papers and textbooks. Normally, in lathe machine applications, 6 speed gearboxes are used and using an 8-speed gearbox will provide more operating speeds along with offering more smoother and uniform machining operation as the intervals between two successive speeds will reduce. Post designing, thorough analysis considering the working and static conditions of the gearbox was done and all critical parameters were found to be well within the permissible range.

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