ME3201 Machine Design Project

Semester 5

Sledgehammer

Final Report

By

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1 Introduction

A mechanical device called a gearbox is used to modify the motor's speed (RPM) or enhance the output torque. The gearbox has a motor shaft attached to one end, and the internal arrangement of the gears produces an output torque and a certain speed dictated by the gear ratio.

The simplest definition of a gearbox is a mechanical device or element made up of a gear train or series of gears that are housed in a housing. In actuality, the word "gearbox" itself describes the object as such. A gearbox functions like a gear system in the most fundamental way. A drive, such as a motor, and a load can change each other's torque and speed.

There are several different types of gears that can be used in gearboxes, including bevel and helical gears, worm gears, and planetary gears. The gear revolves around a shaft that is supported by roller bearings. A gearbox is a mechanical device used to enhance torque while reducing speed. It transfers energy from one device to another.

Conveyor belts, industrial machinery, machine tools, and practically any application involving rotating motion that calls for a change in torque and speed all use gearboxes.

Figure 1:Sample Gearbox

2 Problem Description

<u>Problem identification:</u>

Therefore, want to apply high force if they use sledgehammers.

- These are heavy tools and using them for extended periods can lead to muscle injuries.
- Sledgehammer mastery takes time and repetition. Aspiring craftspeople must put in a lot of work to hone their abilities, comprehend the intricacies of the tool, and develop an acute sense of detail.
- The ability to deliver precise and controlled blows with a sledgehammer is one of the main challenges.
- Lifting and swinging a large sledgehammer repeatedly requires considerable physical strength and endurance.

The project aims to design the gearbox for a sledgehammer. Therefore, the function of the machine is to apply the various loads hitting the surface. If required apply to various loads when changing the hammer upward upward-moving angle and hammer downward moving velocity. So, nine categories of forces can used. this hammer moves up and down for one impact line. Precision is high in the machine. This machine is mostly useful for traditional blacksmiths. This machine has three forward gears and one reverse gear.

❖ The simple design of this machine is shown below.

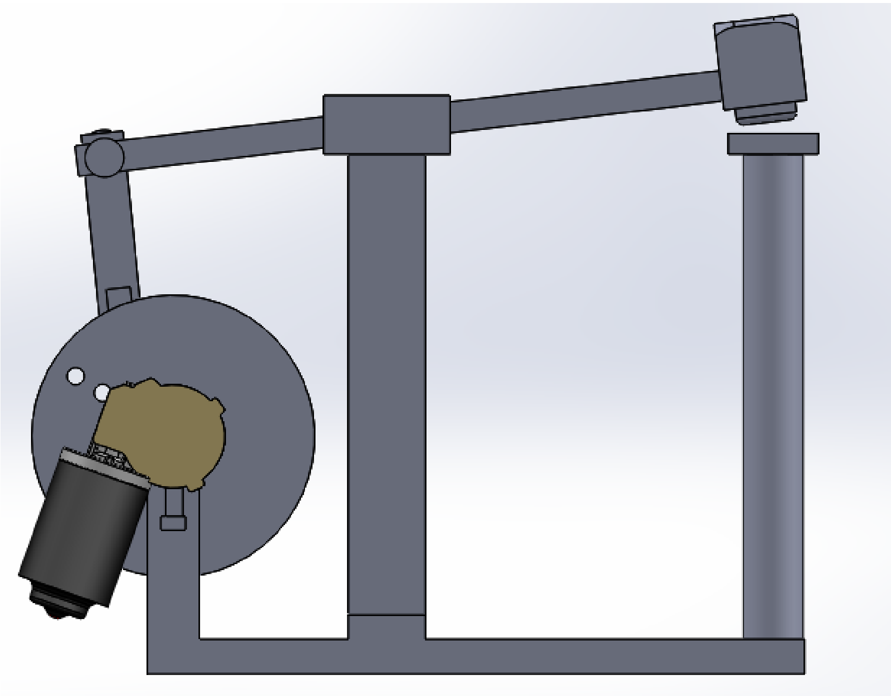


Figure 2: 3D Art of Sledgehammer design

3 Background Study

Some inventions stand out as crucial turning points that have advanced cultures in the history of human ingenuity. I am steady in many areas for the design machine project ME3201.So a got the idea for the hammer used in forging. Especially traditional blacksmiths. Therefore, traditional blacksmiths use many hammers if crush the metal various metal. Someone big and someone small. So, I can get the point of my design project. I focus on the big hammer used by blacksmiths.

So, traditional blacksmiths use many hammers thought history. Therefore, apply a large force on the metal if can use to trip the hammer.

The trip hammer stands out among them as an example of human creativity, demonstrating how a straightforward but clever operating principle can transform entire industries and alter how we live and work. The trip hammer's influence endures through the years, influencing metallurgy, agriculture, and countless other fields. It is historically rooted but timeless in its value.

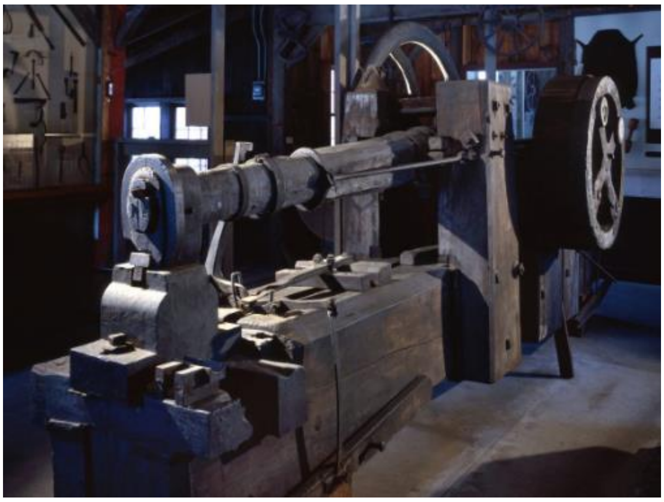


Figure 3: real world sledgehammer

The fundamental idea is that a hammer that is greater than what a man can lift is raised using some form of power in some way so that when it is dropped, it will do so with more force than a man can muster on his own. The drop to hammer in gravitational force effects. The hammer is up to use the potential energy to rotate the wheel. if rotational motion converts to linear motion, it applies the hammer.

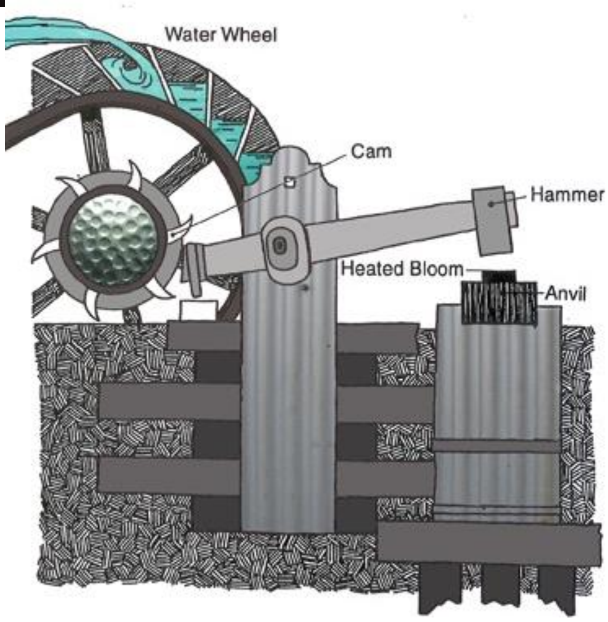


Figure 3: Hydraulic sledgehammer

So modern blacksmiths use the power hammer. The power hammer is a direct ancestor of the

Figure 4: Hydraulic Hammer

trip hammer, but it differs in that it accelerates the ram during the downward stroke and stores potential energy in a system of mechanical linkages and springs, compressed air, or steam. This hammer's cost is high. This is used in industrial factory applications. This is not used by traditional blacksmiths. There is face many difficulties for use this hammer, including noise and vibration, heat management, material selection, not controlling any ages, etc.

Therefore, traditional blacksmiths use the sledgehammer.

The sledgehammer is a representation of the might and inventiveness of humans because of its enormous size, unyielding force, and wide range of uses. The sledgehammer has established itself as a crucial instrument in many fields, whether it be in construction, destruction, or even symbolically as a representation of powerful impact.

Figure 5: Blacksmith forging

The sledgehammer has been crucial in influencing how humans have built, destroyed, and altered their surroundings throughout history. Its beginnings can be found in antiquity when crude versions of this tool were used for a variety of jobs that required a lot of force. The sledgehammer's design and functionality evolved alongside human skill, giving rise to specialized variations that can be used for a variety of tasks.

- Sledgehammer with Swage Block

- Cross peen hammer

- Straight Pein Hammer

- Rounding hammer

- Blacksmith's Hand Hammer

- Ball peen hammer

4 Design Calculations and Materials Selection

4.1Components of sledgehammer

The sledgehammer is powered by 110V single phase 2800 rpm 0.75Kw synchronous induction motor. The following diagram can be used to visualise the parts of the power transmission system.

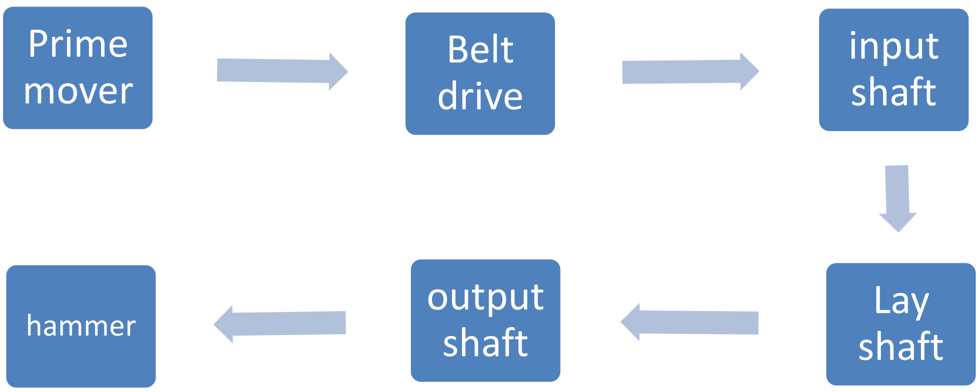


Figure 6: Power flow diagram

4.2 General parameters of the sledgehammer

Table 1: Parameters of the Sledgehammer

<u>Hammer arm</u>	
Material	**Boxwood**
Density	900 Kg/m^3
Weight	0.5Kg
length	80cm
Cross-section area	32cm^2

(Width-4cm, length-8cm	
<u>Hammerhead</u> Material Weight Density Square cross-section area Hight	 **High carbon steel** 3.5Kg 7850Kg/m^3 64cm^2 7cm
Hammer contact time	0.01s (generally sledgehammer)
Centroid of hammer	78.5cm
Moment of inertia in hammer	2.57Kgm^2
Forward gears	3
Reverse gears	1

4.3 Gear Calculations

Assumptions:

20^0 Full depth involute system is assumed.

Gears are assumed to be carefully cut.

Power loss is neglected in the spur gears.

Tooth Profile Selection

Two different types of gear profiles are regularly used when manufacturing gear teeth. They are,

- Involute gears
- Cycloidal gears

A point on a tangent that rolls on a circle without slipping creates an involute of that circle, as does a point on a taut thread that is unwound from a reel.

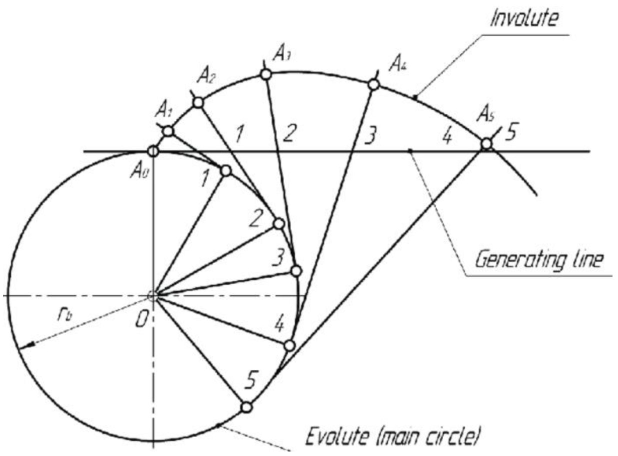


Figure 7: Involute teeth diagram

The toothed gear profile used in mechanical clocks is called a cycloidal gear profile, as opposed to the involute gear form utilized for most other gears. The foundation for the gear tooth profile is made up of the epicycloid and hypocycloid curves, which are produced by rolling a circle around the outside and inside of another circle, respectively.

A hypothetical circle called the pitch circle, can be constructed around the centre of each gear where its teeth make contact when two toothed gears mesh. The curves of the teeth outside the pitch circle are known as addenda, and the curves of the tooth gaps inside the pitch circle are known as dedendum. The addendum of one gear fits into the dedendum of the other gear.

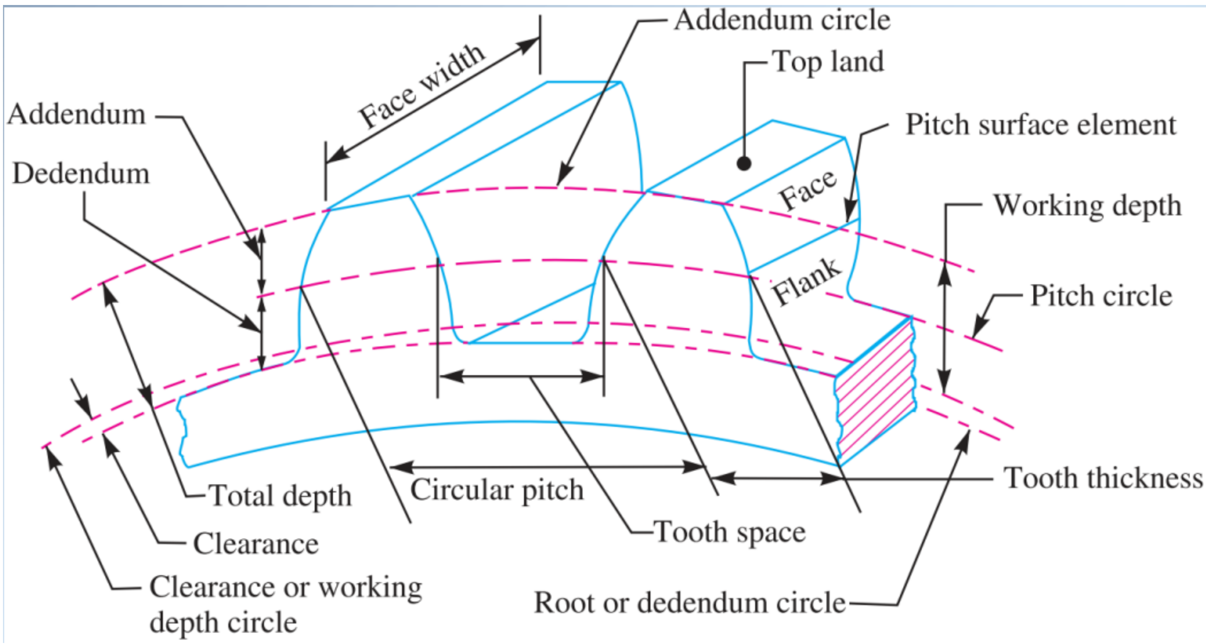


Figure 8: Terms related to gears

<u>Abbreviations for the terms used in gear calculations.</u>

T: - Number of teeth in the gear

d: - pitch circle diameter

$m = p / \pi$: - module of the gear

$\pi d = p T$ = circumference of the pitch circle

$d = (p\,T / \pi) = (p / \pi)\,T = m$

S. No.	Particulars	$14\frac{1}{2}°$ composite or full depth involute system	20° full depth involute system	20° stub involute system
1.	Addendum	1*m*	1*m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*

Table 2: Gear standard properties

A 20^0 full-depth involute system was chosen for the gear system during the design phase. Value of the module 2 mm is ultimately chosen as the optimal module for the gearbox after several iterations depending on the dynamic and wear loads.

Table 3:Values of gear parameters

Particulars	**Values for our case(mm)**
Addendum	2
Dedendum	2.5
Working depth	4
Minimum total depth	4.5
Tooth thickness	3.1416
Minimum clearance	0.5
Fillet radius at root	0.8

Calculation of gear teeth for input & lay shaft gears

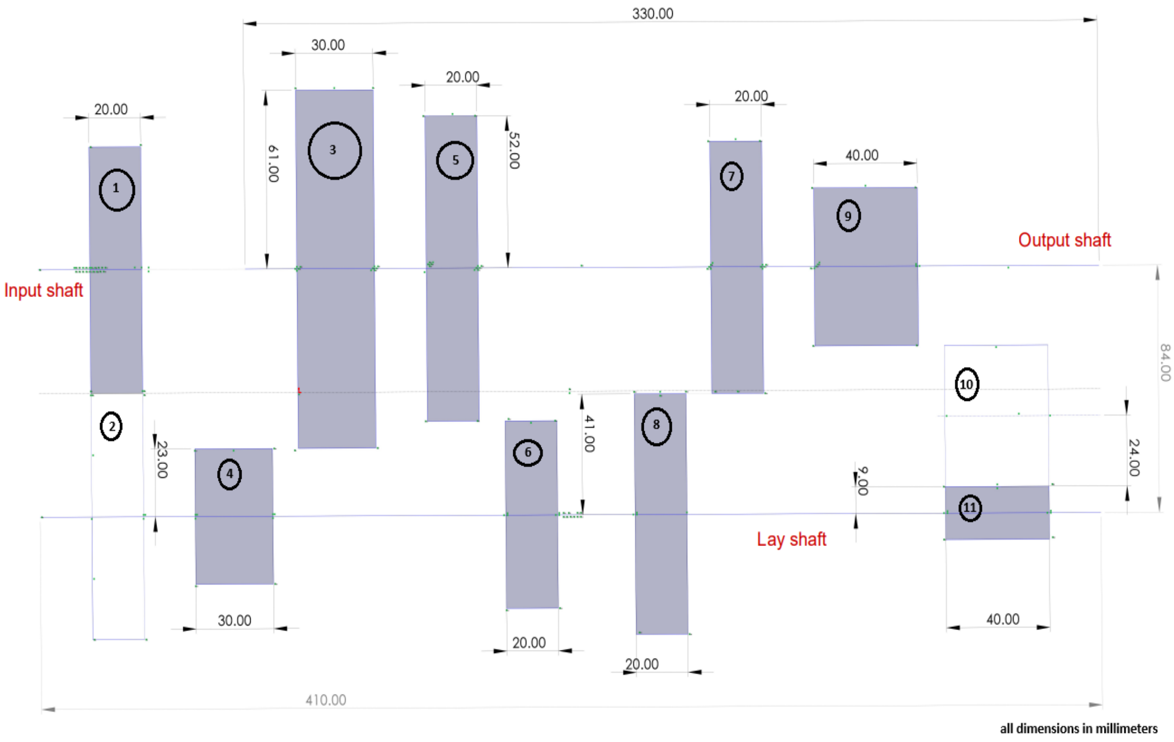


Figure 9: Gear arrangement

data:

Power: - 500W

Input speed in RPM: - 150 rpm

There is no speed reduction in input shaft and lay shaft.

Formulas used in Calculations:

$$G = \text{Gear ratio} = \frac{rpm\ of\ the\ pinion}{rpm\ of\ the\ gear\ wheel} = \frac{No.of\ \ teeth\ in\ wheel}{No\ of\ teeth\ in\ pinion}$$

$$T_p \geq \frac{2\ Aw}{G\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)(\sin\phi)^2}-1\right]}$$

Pitch circle diameter of the gear = d = mT

$\frac{0.154 - 0.912}{T}$, for 20^0 full depth system

Strength of the teeth = $(\sigma_0\ y)\ b\ m\ \pi\ C_v$

The pitch line velocity (v) = ω d / 2

Velocity factor (C_v) = 3 / (3+v)

Minimum no of teeth on the pinion to avoid interference.

Gear interference will take part a significant role in the gear wheel design process. Therefore, it is essential to calculate the number of teeth in order to prevent interference. The following equation is used to determine the minimal number of gear teeth for the pinon:

$$T_p \geq \frac{2\,Aw}{G\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)(\sin \emptyset)^2}-1\right]}$$

$$= \quad 18.48$$

$$= \quad 19$$

Using the gear ratio, we can calculate number of teeth in the gear using following equation.

$$T_g = T_p \times G$$

For interference, a pinion is required to have a minimum of 19 teeth. As a result, 42 teeth are selected as the number of teeth on the input pinion. To determine the centre distance, the input and lay shaft gears should be used. The results may then be applied to the remaining gears.

Determining the centre distance

No. of teeth of gear = 42

Pitch circle diameter of pinion = m × T = 2 × 42 = 84mm

Number of teeth of gear = 42 (because there is no speed reduction in input and lay shaft)

Pitch circle diameter of gear = 2 × 42 = 84mm

Table 4: Parameters of pinion gear

	Pinion	Gear
No. of teeth	42	42
Pitch circle diameter(mm)	84	84
Addendum(mm)	2	2
Dedendum(mm)	2.5	2.5
Face width(mm)	30	30

$$[(P.C.D)1 + (P.C.D)2]/2 = 84mm$$

As a result, 84mm is chosen as the central distance between the input shaft and the lay shaft.

The number of teeth in the other gear wheels has been determined using this centre distance and the gear ratios. Final diameters should match the obtained centre distance value by rounding off the numbers.

Table 5: Parameters of Gears

Gear	Minimum teeth for avoid interference	PCD of pinion (mm)	PCD of gear (mm)	No. of teeth in pinion	No. of teeth in gear	Face width (mm)	Centre distance (mm)
Input and Lay shaft	19	84	84	42	42	30	84
1^{st}	23	46	122	23	61	30	84
2^{nd}	21	64	104	32	52	20	84
3^{rd}	19	82	86	41	43	20	84
Reverse	15	36	108	18	54	40	84

Determining the teeth strength

The process of designing the gear wheel and obtaining the right materials are necessary. The gear wheel's material will affect every computed value. The material of the gear wheels is directly linked to parameters like size, face width, wear resistance, etc.

Table 6:Properties of commonly used gear materials.

Material (1)	*Condition* (2)	*Brinell hardness number* (3)	*Minimum tensile strength (N/mm²)* (4)
Malleable cast iron			
(*a*) White heart castings, Grade B	—	217 max.	280
(*b*) Black heart castings, Grade B	—	149 max.	320
Cast iron			
(*a*) Grade 20	As cast	179 min.	200
(*b*) Grade 25	As cast	197 min.	250
(*c*) Grade 35	As cast	207 min.	250
(*d*) Grade 35	Heat treated	300 min.	350
Cast steel	—	145	550
Carbon steel			
(*a*) 0.3% carbon	Normalised	143	500
(*b*) 0.3% carbon	Hardened and tempered	152	600
(*c*) 0.4% carbon	Normalised	152	580
(*d*) 0.4% carbon	Hardened and tempered	179	600
(*e*) 0.35% carbon	Normalised	201	720
(*f*) 0.55% carbon	Hardened and tempered	223	700
Carbon chromium steel			
(*a*) 0.4% carbon	Hardened and tempered	229	800
(*b*) 0.55% carbon	"	225	900
Carbon manganese steel			
(*a*) 0.27% carbon	Hardened and tempered	170	600
(*b*) 0.37% carbon	"	201	700
Manganese molybdenum steel			
(*a*) 35 Mn 2 Mo 28	Hardened and tempered	201	700
(*b*) 35 Mn 2 Mo 45	"	229	800
Chromium molybdenum steel			
(*a*) 40 Cr 1 Mo 28	Hardened and tempered	201	700
(*b*) 40 Cr 1 Mo 60	"	248	900

The sledgehammer is typically made to operate for 8 to 10 hours a day with medium shocks.

As a result, we can determine from data sheets that the device's service factor will be 1.54.

C_S = 1.54

<u>Considering 1st gear (gear 7 and 8)</u>

Allowable static stress = (σ_0)p = 210 MPa

Lewis Form Factor of the pinion (y_p) = 0.154 – 0.912 / T

= 0.154 – 0.912/ 23

=0.114

The pitch line velocity (v) = ω_1 d_1 / 2

$$= 2\pi * \frac{150}{60} * \frac{46}{2} * 10^{-3}$$

= 0.361 m/s

Therefore, velocity factor (C_v)= $\frac{3}{3+v}$

$$= \frac{3}{3+0.361}$$

= 0.8925

tangential load (W_T) = $\frac{p}{v} \times c_s$

$$= \frac{500}{0.361} \times 1.54$$

= 2132.96N

Static tooth load

$W_S = \sigma_e$ y b m π

= $130*10^6 * 0.1143*30*2* \pi *10^{-6}$

= 2802.02N

Calculating the dynamic load

$$W_D = W_T + W_I = W_T + \frac{21v(bC+WT)}{21v+\sqrt{bC+WT}}$$

- W_D = Total dynamic load
- W_T = Steady load due to transmitted torque
- W_I = Increment load due to dynamic action
- v = Pitch line velocity
- b = Face width of gears
- C = A deformation or dynamic factor

Table 7: Values of deformation factor

Material		*Involute tooth form*	*Values of deformation factor (C) in N-mm*				
			Tooth error in action (e) in mm				
Pinion	*Gear*		0.01	0.02	0.04	0.06	0.08
Cast iron	Cast iron	$14\frac{1}{2}°$	55	110	220	330	440
Steel	Cast iron		76	152	304	456	608
Steel	Steel		110	220	440	660	880
Cast iron	Cast iron	20° full depth	57	114	228	342	456
Steel	Cast iron		79	158	316	474	632
Steel	Steel		114	228	456	684	912
Cast iron	Cast iron	20° stub	59	118	236	354	472
Steel	Cast iron		81	162	324	486	648
Steel	Steel		119	238	476	714	952

Both steady load and dynamic load can be determined using the formula mentioned above and the values obtained in the previous sections.

<u>Considering 1st gear (gear 7 and 8)</u>

- 20^0 full depth involute gear form
- Pitch line velocity = 0.361 m/s

• Face width of gear = 30 mm

• Module of the gears = 2mm

• Deformation factor = 66.6

W_I = 300.98N

W_D = 2432.27N

As a result, the calculated static tooth load based on the equation is more than the total dynamic load. Therefore AISI 4340 Alloy steel (Ni, Cr, Mo) can be selected as the gear material.

Determining the Limiting Load for wear

$$W_w = D_p \text{ x b x Q x K}$$

• W_w = Maximum or limiting load for wear

• D_p = Pitch circle diameter of the pinion

• b = Face width of the pinion

• Q = Ratio factor = $2T_G / (T_G + T_P)$

• V.R. = Velocity ratio = T_G / T_P

• K = Load-stress factor where

K = $\frac{(\sigma es)^2 * \sin(\emptyset)}{1.4}\left(\frac{1}{Ep} + \frac{1}{Eg}\right)$

• σ_{es} = Surface endurance limit

• φ = Pressure angle

• E_P = Young's modulus for the material of the pinion

• E_G = Young's modulus for the material of the gear

K = $\frac{(630*10^6)^2*\sin(20)}{1.4} * \left(\frac{1}{32*10^9} + \frac{1}{32*10^9}\right)$

= 6060169.415

W_W = (0.046*0.03*1.447*6060169.415)

= 12101.30 N

Since dynamic load is lower than the wear load and allowable load, as a result, the pinion is secure according to the conclusions of the earlier calculations. In addition, the Lewis equation shows that the wheel is stronger than the pinion. The wheel is therefore safe as well.

By applying the method described above, we can determine the static and dynamic loads as well as the wear-limiting loads for all pinion gears. As a result, it will guarantee that spur gears are safe.

Table 8: Gear loadings

Gears	Velocity Factor (C_v)	Face width(mm)	Tangential tooth load (W_T)	Increment load due to dynamic action (W_I)	Total Dynamic Load (W_D)	Static tooth load(W_s)	Design is safe or not considering strength
Input & lay shaft	0.819	30	1167.13	337.12	1504.25	3241.58	safe
1st	0.892	30	2131.29	300.98	2432.27	2802.02	safe
2nd	0.854	20	1505.05	326.84	1831.9	2058.35	safe
3rd	0.823	20	1195.60	336.75	1532.35	2152.40	safe
reverse	0.913	40	2723.31	278.35	3001.67	3376.16	safe

Table 9: Safety conditions of the gears

Gears	Load stress factor (K)	Limiting load for wear (W_w)	Design is safe or not considering wear load
Input & lay shaft	6060169.41	15271.62	safe
1st	6060169.41	12102.79	safe
2nd	6060169.41	9647.88	safe
3rd	6060169.41	10144.95	safe
reverse	6060169.41	13089.96	safe

Details of the gear wheels

Gears	No. of teeth	Pitch circle diameter	Module(mm)	Material
Gear 1	42	84	2	AISI 4340 Alloy steel
Gear 2	42	84	2	AISI 4340 Alloy steel
Gear 3	43	86	2	AISI 4340 Alloy steel
Gear 4	41	82	2	AISI 4340 Alloy steel
Gear 5	52	104	2	AISI 4340 Alloy steel
Gear 6	32	64	2	AISI 4340 Alloy steel
Gear 7	61	122	2	AISI 4340 Alloy steel
Gear 8	23	46	2	AISI 4340 Alloy steel
Gear 9	54	108	2	AISI 4340 Alloy steel
Gear 10	20	40	2	AISI 4340 Alloy steel
Gear 11	18	36	2	AISI 4340 Alloy steel

4.3 Pulley Calculation

Pulley that connected to input shaft

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Belt thickness = 5 mm

Belt width(b) = 40 mm

Rpm of the pulley = 150 rpm

Assumption:

Pulley material Grey cast iron.

Density = 7150 kg/m^3

Compressive stress = 752 MPa

Centrifugal stress = 0.8 MPa

Centrifugal stress,

$$\sigma_t = \rho \times v^2$$

$$\sigma_t = \rho \times \left(\frac{\pi D N}{60}\right)^2$$

Pulley diameter (D) can be find as,

$$D = \sqrt{\frac{\sigma_t}{\rho}} \times \frac{60}{\pi N}$$

$$D = \sqrt{\frac{0.8 \times 10^6}{7150}} \times \frac{60}{\pi \times 150}$$

$$D = 1.34m = 1340\ mm \approx 1400\ mm$$

Because of belt width is 40mm,

Table 10

Belt width in mm	*Width of pulley to be greater than belt width by (mm)*
upto 125	13
125-250	25
250-375	38
475-500	50

According to the above table the pulley is greater than 13% by belt width. The pulley width can be considered by,

$$B = 1.13b$$

$$B = 1.13 \times 40 = 45.2mm$$

Thickness of the pulley (t):

Because of using single belt,

$$t = \frac{D}{300} + 2$$

$$t = \frac{1400}{300} + 2 = 6.667 \approx 7mm$$

Number of arms:

600mm < 1400 mm < 1500mm of Diameter of the pulley,

∴We can use 6 arms with a safety condition.

Diameter of Hub(d_1):

$$d_1 = 1.5d + 25$$

$$d_1 = 1.5 \times 25 + 25 = 62.5mm$$

Pulley that connected to the motor

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Belt thickness = 5 mm

Belt width(b) = 40 mm

Rpm of the pulley = 1500 rpm

Assumption:

Pulley material Grey cast iron.

Density = 7150 kg/m^3

Compressive stress = 752 MPa

Centrifugal stress = 0.8 MPa

Centrifugal stress,

$$\sigma_t = \rho \times v^2$$

$$\sigma_t = \rho \times \left(\frac{\pi D N}{60}\right)^2$$

Pulley diameter (D) can be find as,

$$D = \sqrt{\frac{\sigma_t}{\rho}} \times \frac{60}{\pi N}$$

$$D = \sqrt{\frac{0.8 \times 10^6}{7150}} \times \frac{60}{\pi \times 1500}$$

$$D = 0.134m = 134\ mm \approx 140\ mm$$

Because of belt width is 40mm,

Table 11: Standard width of pulley

Belt width in mm	Width of pulley to be greater than belt width by (mm)
upto 125	13
125-250	25
250-375	38
475-500	50

According to the above table the pulley is greater than 13% by belt width. The pulley width can be considered by,

$$B = 1.13b$$

$$B = 1.13 \times 40 = 45.2mm$$

Thickness of the pulley (t):

Because of using single belt,

$$t = \frac{D}{300} + 2$$

$$t = \frac{140}{300} + 2 = 2.46 \approx 3mm$$

Number of arms:

100mm < 140 mm < 200mm of Diameter of the pulley,

∴We can use 3 arms with a safety condition.

Diameter of Hub(d_1):

$$d_1 = 1.5d + 25$$

$$d_1 = 1.5 \times 25 + 25 = 62.5mm$$

4.4 Belt Calculation

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Input pulley width = 45.2 mm

Assumption:

Diameter of pulley that connected to input shaft (d_1) = 1400mm

Diameter of pulley that connected to motor (d_2) = 140mm

Belt material - Leather oak tanned.

Used **Open flat belt** for the power transmission.

Belt thickness = 5 mm

Belt width(b) = 40 mm

Centre distance (x) between two pulleys = 900mm

Fiction Coefficient of belt with pulley material (μ) = 0.25

Assuming No slipping and no creep of the belt.

Length of the belt (L):

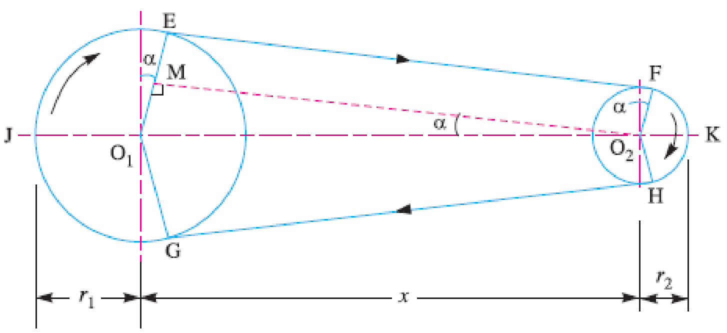


Figure 10: Layout of Pulleys with open belt system

$$L = \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x}$$

$$L = \frac{\pi}{2}(1400 + 140) + 2 \times 900 + \frac{(1400 - 140)^2}{4 \times 900} = 4660mm = 4.66mm$$

Finding the arc angle (α):

$$\sin \alpha = \frac{d_1 - d_2}{2x}$$

$$\sin \alpha = \frac{1400 - 140}{2 \times 900} = 0.7$$

$$\alpha = 44.42°$$

$$\alpha_{rad} = 0.775\, rad$$

Power transmitted through belt & Tensions:

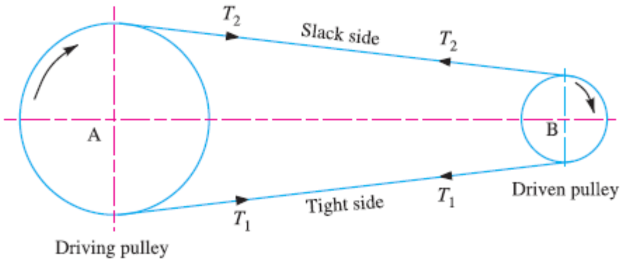


Figure 11: Standard width of pulley

$$P = (T_1 - T_2)v$$

$$T\omega = (T_1 - T_2) \times {}^{d_1}/_{2} \times \omega$$

$$(T_1 - T_2) = \frac{T}{{}^{d_1}/_{2}}$$

$$(T_1 - T_2) = \frac{31.83}{0.7} = 45.17N$$

For maximum power of V belt,

$$2.3\ln\left(\frac{T_1}{T_2}\right) = \mu\theta$$

$$\theta = (180° + 2 \times 44.42°) \times \frac{2\pi}{360} = 0.469$$

$$2.3\ln\left(\frac{T_1}{T_2}\right) = \ 0.25 \times 0.469$$

$$\frac{T_1}{T_2} = 1.661$$

$$T_1 = 1.661T_2$$

$$(1.661T_2 - \ T_2) = 45.17$$

$$T_2 = 68.49N$$

$$T_1 = 114.36N$$

4.6 Shaft Calculation

Input shaft

Data:

Torque = 31.83N

Speed = 150 rpm

Assumption:

Shaft material – 50C 12 steel

Table 12: mechanical properties of steel used for shaft

Indian standard designation	*Ultimate tensile strength, MPa*	*Yield strength, MPa*
40 C 8	560 - 670	320
45 C 8	610 - 700	350
50 C 4	640 - 760	370
50 C 12	700 Min.	390

The weight of the shaft is negligible.

Table 13: shaft Properties

50 C 12 Steel (AISI 1065)		
Density	7.85	g/cm3
Ultimate tensile strength	700	MPa
Modulus of Rigidity	80	GPa
Allowable Shear Stress	525	MPa

Input shaft configuration:

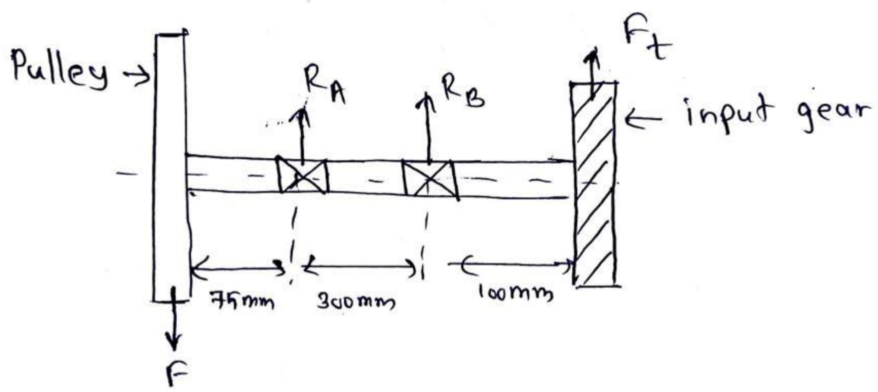


Figure 12: Input Shaft

Finding Loads on bearing:

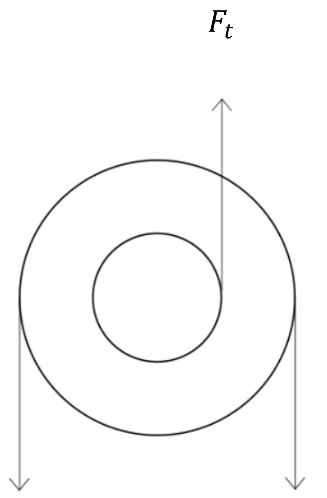


Figure 13: Pulley sketch

T_1 T_2

$$T_1 + T_2 = F$$

$$F = 183.25N$$

$$F_t = {}^{T}\!\big/_{(^{D}/_{2})}$$

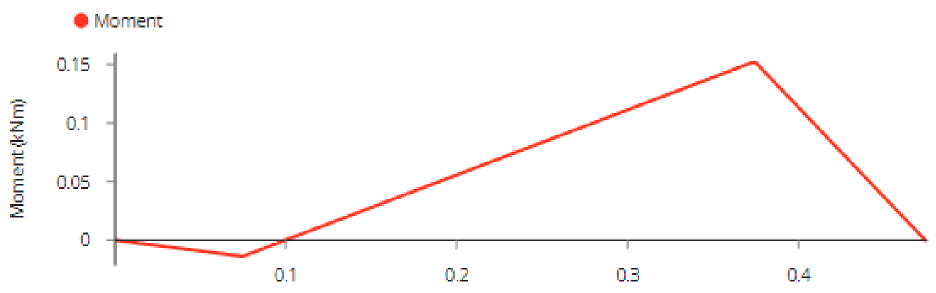
$$F_t = {}^{31.83}\!\big/_{(^{0.042}/_{2})} = 1515.71N$$

$$R_A + R_B = -1332.46N$$

$$F \times 75 + R_B \times 300 + F_t \times 400 = 0$$

$$\therefore\ R_B = -2066.75N$$

$$\therefore\ R_A = 734.29N$$



Graph 1: moment graph

$$Maximum\ bending\ moment(M) = 110.2Nm$$

For calculate the twisting moment (T_E):

$$T_E = \sqrt{(K_m M)^2 + (K_t T)^2}$$

Table 14: recommended value for K_m and K_t

Nature of load	K_m	K_t
1. Stationary shafts		
(*a*) Gradually applied load	1.0	1.0
(*b*) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(*a*) Gradually applied or steady load	1.5	1.0
(*b*) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(*c*) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

Because of a rotating steady load applied shaft,

$$K_m = 1.5, K_t = 1$$

$$T_E = \sqrt{(1.5 \times 110.2)^2 + (1 \times 31.83)^2} = 168.33Nm$$

Equivalent bending Moment of the shaft,

$$M_E = \frac{1}{2}(K_m M + T_E)$$

$$M_E = \frac{1}{2}(1.5 \times 110.2 + 168.3) = 166.845Nm$$

$$\tau_{max} \geq \frac{16}{\pi d^3}\sqrt{(K_m M)^2 + (K_t T)^2}$$

$$d \geq \frac{16}{\pi \times 525 \times 10^6}\sqrt{(1.5 \times 110.2)^2 + (1 \times 31.83)^2}$$

$$d \geq 12mm$$

According to the standard diameter catalogue we can get the input shaft diameter as 25mm.

Lay Shaft

Lay shaft configuration:

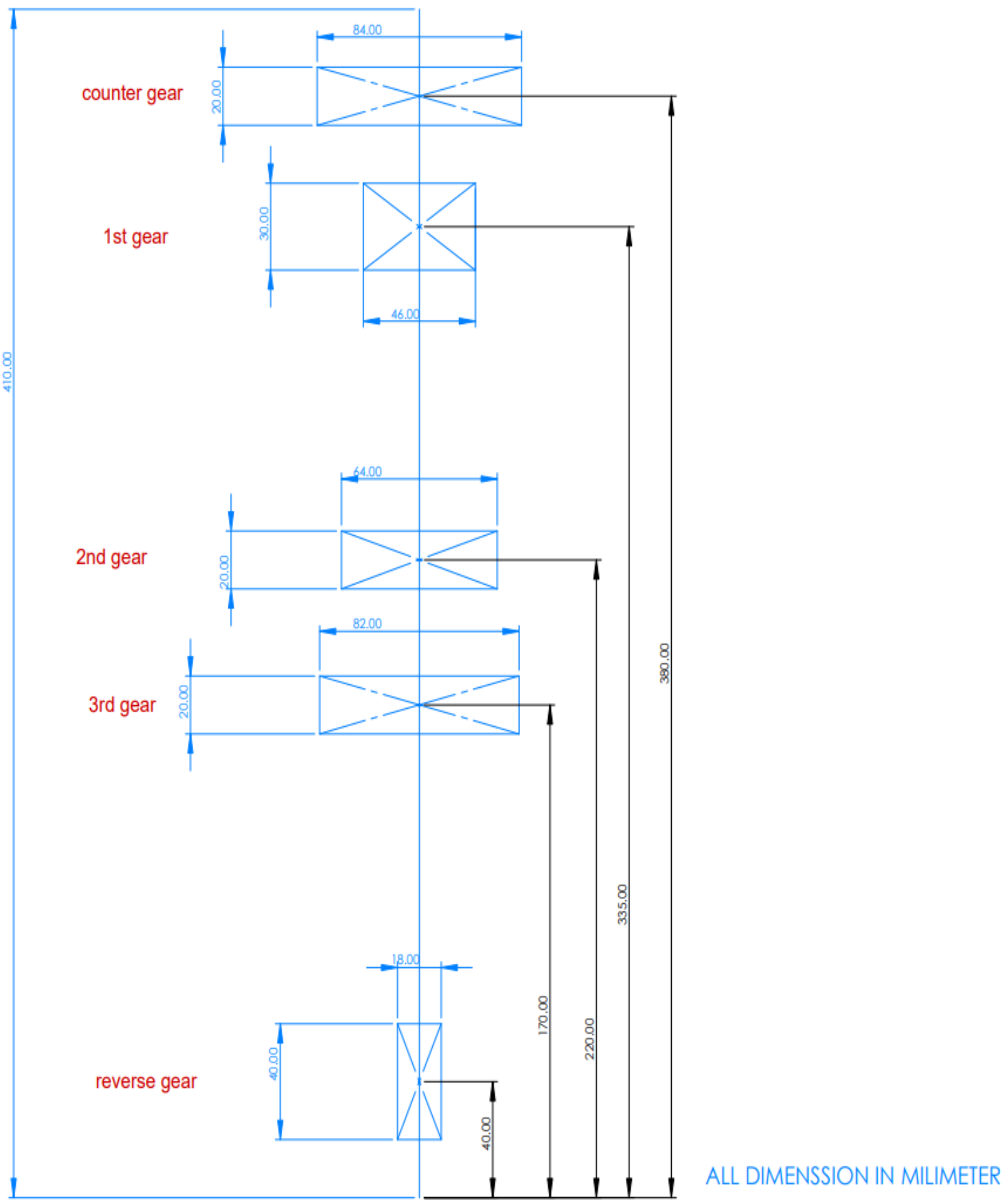


Figure 14: Lay shaft

Maximum force acting on the lay shaft and bending Momentum:

Data

<u>mass of gears</u>

Counter gear- 0.8kg

1st gear- 0.3kg

2nd gear- 1.25kg

3rd gear- 0.8kg

Reverse gear- 1.6kg

Table 15: Gear weights

Catalog No.	Module	No. of teeth	Shape	Bore	Hub dia.	Pitch dia.	Outside dia.	Face width	Hub width	Total length	Web thickness	Web O.D.
				A_{H7}	B	C	D	E	F	G	H	I
KMSGA2-15 KMSGB2-15**	m2	15	S1	12 15	24	30	34	20	10	30	—	—
KMSGA2-18 KMSGB2-18		18	S1	12 15	30	36	40	20	10	30	—	—
KMSGA2-20 KMSGB2-20		20	S1	15 18	32	40	44	20	10	30	—	—
KMSGA2-24 KMSGB2-24		24	S1	15 18	35	48	52	20	10	30	—	—
KMSGA2-25 KMSGB2-25		25	S1	16 20	35	50	54	20	10	30	—	—
KMSGA2-30 KMSGB2-30		30	S1	18 22	40	60	64	20	10	30	—	—
KMSGA2-35 KMSGB2-35		35	S1	18 22	40	70	74	20	10	30	—	—
KMSGA2-36 KMSGB2-36		36	S1	18 22	40	72	76	20	10	30	—	—
KMSGA2-40 KMSGB2-40		40	S1	20 25	45	80	84	20	10	30	—	—
KMSGA2-45 KMSGB2-45		45	S1	20 25	45	90	94	20	10	30	—	—
KMSGA2-48 KMSGB2-48		48	S1	22 28	50	96	100	20	10	30	—	—
KMSGA2-50 KMSGB2-50		50	S1	22 28	50	100	104	20	10	30	—	—
KMSGA2-55 KMSGB2-55		55	S1	25 30	55	110	114	20	10	30	—	—
KMSGA2-60 KMSGB2-60		60	S1	25 30	55	120	124	20	10	30	—	—
KMSGA2-70 KMSGB2-70		70	S1	25 30	55	140	144	20	10	30	—	—
KMSGA2-80 KMSGB2-80		80	S2	30 35	60	160	164	20	10	30	13	144
KMSGA2-100 KMSGB2-100		100	S2	35 40	80	200	204	20	10	30	13	174

Keyway	Allowable torque (N·m)		Allowable torque (kgf·m)		Backlash (mm)	Weight (kg)	Catalog No.
WidthxDepth	Bending strength	Surface durability	Bending strength	Surface durability			
4 x 1.8 5 x 2.3	73.1	35.7	7.46	3.64	0.10~0.20	0.12 0.10	KMSGA2-15 KMSGB2-15**
4 x 1.8 5 x 2.3	97.2	53.5	9.91	5.46	0.10~0.20	0.19 0.17	KMSGA2-18 KMSGB2-18
5 x 2.3 6 x 2.8	114	67.6	11.6	6.89	0.10~0.20	0.22 0.20	KMSGA2-20 KMSGB2-20
5 x 2.3 6 x 2.8	148	101	15.1	10.3	0.10~0.20	0.32 0.30	KMSGA2-24 KMSGB2-24
5 x 2.3 6 x 2.8	157	110	16.0	11.2	0.10~0.20	0.33 0.31	KMSGA2-25 KMSGB2-25
6 x 2.8 6 x 2.8	201	161	20.5	16.5	0.12~0.22	0.48 0.45	KMSGA2-30 KMSGB2-30
6 x 2.8 6 x 2.8	246	223	25.1	22.7	0.12~0.22	0.64 0.61	KMSGA2-35 KMSGB2-35
6 x 2.8 6 x 2.8	255	236	26.0	24.1	0.12~0.22	0.67 0.64	KMSGA2-36 KMSGB2-36
6 x 2.8 8 x 3.3	292	294	29.7	30.0	0.12~0.22	0.84 0.79	KMSGA2-40 KMSGB2-40
6 x 2.8 8 x 3.3	338	377	34.5	38.4	0.12~0.22	1.05 1.00	KMSGA2-45 KMSGB2-45
6 x 2.8 8 x 3.3	349	411	35.6	41.9	0.12~0.22	1.20 1.14	KMSGA2-48 KMSGB2-48
6 x 2.8 8 x 3.3	367	448	37.4	45.7	0.12~0.22	1.29 1.24	KMSGA2-50 KMSGB2-50
8 x 3.3 8 x 3.3	412	548	42.0	55.8	0.14~0.24	1.56 1.51	KMSGA2-55 KMSGB2-55
8 x 3.3 8 x 3.3	457	658	46.6	67.1	0.14~0.24	1.84 1.79	KMSGA2-60 KMSGB2-60
8 x 3.3 8 x 3.3	547	909	55.8	92.7	0.14~0.24	2.48 2.43	KMSGA2-70 KMSGB2-70
8 x 3.3 10 x 3.3	610	1150	62.2	117	0.14~0.24	2.55 2.49	KMSGA2-80 KMSGB2-80
10 x 3.3 12 x 3.3	785	1820	80.1	186	0.14~0.24	4.16 4.09	KMSGA2-100 KMSGB2-100

Forces & bending moments acting on the layshaft

tangential loads and radial loads,

Between Input shaft and lay shaft:

F_r = 435.16N

F_t = 1195.60N

Between 1st gear and lay shaft:

F_r = 775.29N

F_t =2131.3N

Between 2nd gear and lay shaft:

F_r = 547.9N

F_t = 1505.05N

Between 3rd gear and lay shaft:

F_r = 435.16N

F_t = 1195.6N

Between Reverse gear and lay shaft

F_r = 1540.56N

F_t = 1697.64N

<u>Reverse gear force calculation</u>

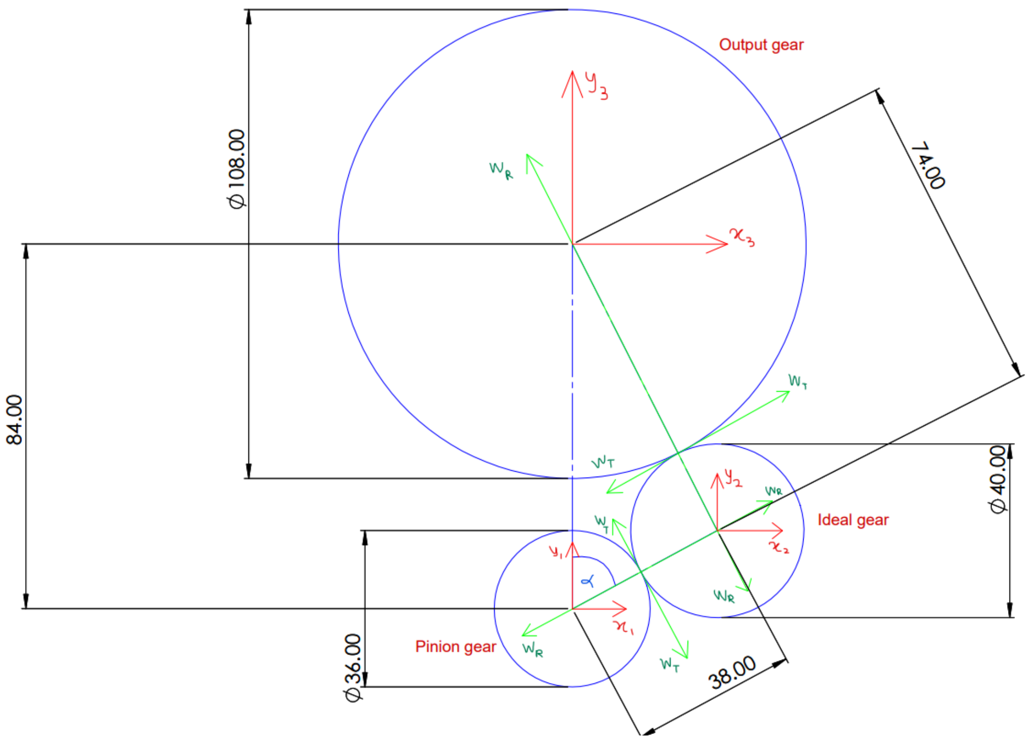


Figure 15:Positions and dimension of the reverse gear

$$\alpha = 62.8$$

$$W_T = 2723.31N$$

$$W_R = W_T \times tan\,(20)$$

$$W_R = 991.2N$$

Pinon gear & idler gear

Pinion Gear

$$x_1 = W_R Cos\theta - W_T Cos\alpha$$

$$y_1 = -W_R Sin\alpha + W_T Sin\alpha$$

$$x_1 = 1697.64N$$

$$y_1 = 1540.56N$$

Idler Gear

$$x_2 = W_R Sin\alpha + W_R Cos\alpha + W_T Sin\alpha + W_T Cos\alpha$$

$$x_2 = 5030.4N$$

$$y_2 = -W_R Sin\alpha - W_T Sin\alpha + W_T Cos\alpha$$

$$y_2 = 1605.76N$$

Gear

$$x_3 = -W_R Cos\alpha - W_T Sin\alpha$$

$$x_3 = -2875.22N$$

$$y_3 = W_R Sin\alpha - W_T Cos\alpha$$

$$y_2 = 363.22N$$

<u>Formula</u>

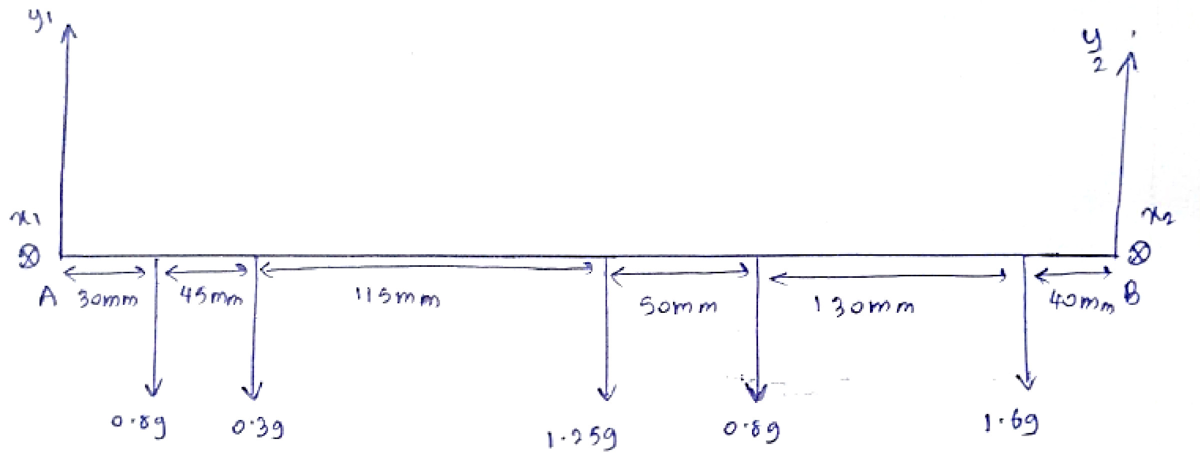
Total bending moment

$$M = \sqrt{{M_x}^2 + {M_y}^2}$$

Load of bearing

$$R = \sqrt{X^2 + Y^2}$$

<u>The total bending moment of the shaft when considering the gear mass</u>



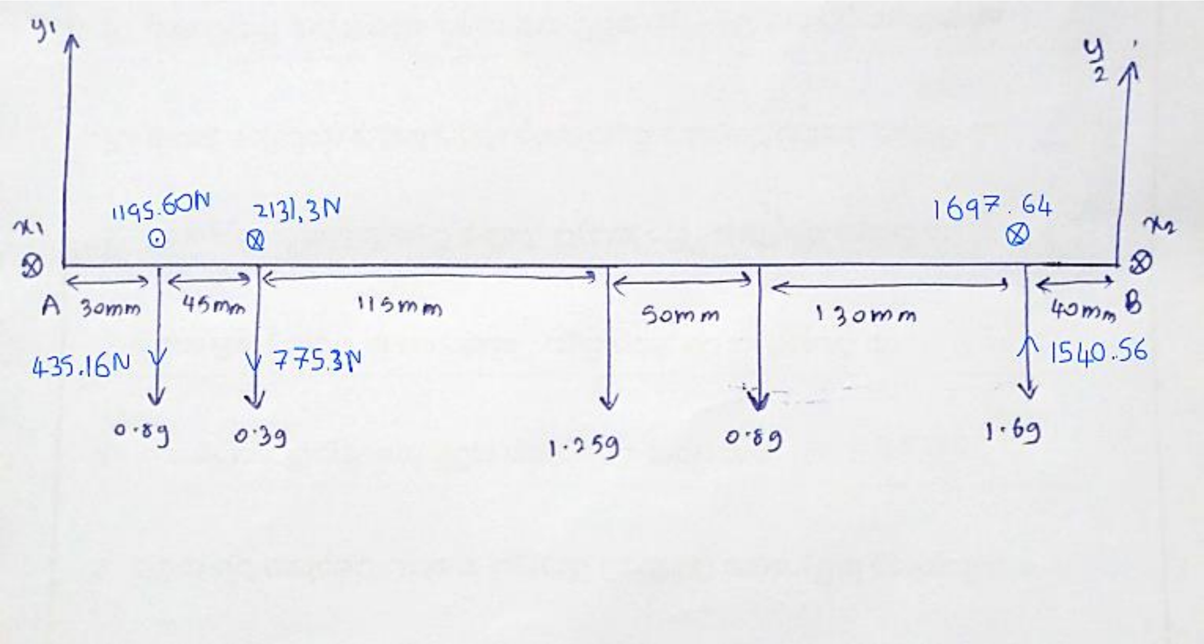
$$M_A = 0.8g \times 30 + 0.3g \times 75 + 1.25g \times 190 + 0.8g \times 240 + 1.6g \times 370$$

$$M_A = 10477.05Nmm$$

$$M_B = 1.6g \times 40 + 1.25g \times 220 + 0.8g \times 170 + 0.3g \times 335 + 0.8g \times 380$$

$$M_B = 8627.8Nmm$$

1st Gear



Consider point A.

$$y_2 \times 410 = 10477.05 + 435.16 \times 30 + 775.3 \times 75 - 1540.56 \times 370$$

$$y_2 = -1190.86N$$

$$x_2 \times 410 = 1697.6 \times 370 + 2131.3 \times 75 - 1195.6 \times 30$$

$$x_2 = 1834.36N$$

Consider point B

$$y_1 \times 410 = 8627.8 + 775.3 \times 335 + 435.16 \times 380 - 1540.56 \times 40$$

$$y_1 = 908.1N$$

$$x_1 \times 410 = 1697.6 \times 40 + 2131.3 \times 335 - 1195.6 \times 380$$

$$x_1 = 800.16N$$

Load of point A bearing

$$R = \sqrt{{x_1}^2 + {y_1}^2}$$

$$R = 1204.86N$$

Load of point B bearing

$$R = \sqrt{{x_2}^2 + {y_2}^2}$$

$$R = 2187.01N$$

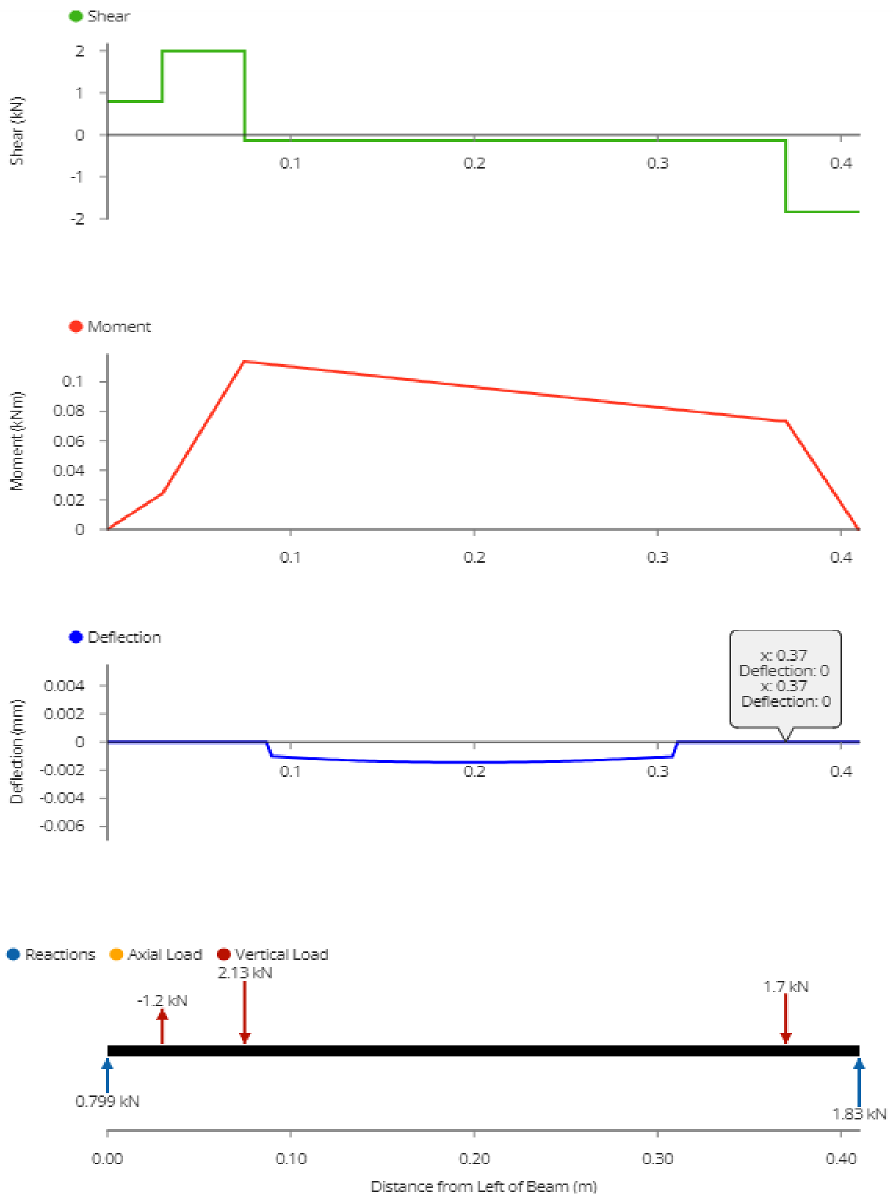


Figure 16: $\boldsymbol{1st\ Gear\ X\ direction\ Graphs}$

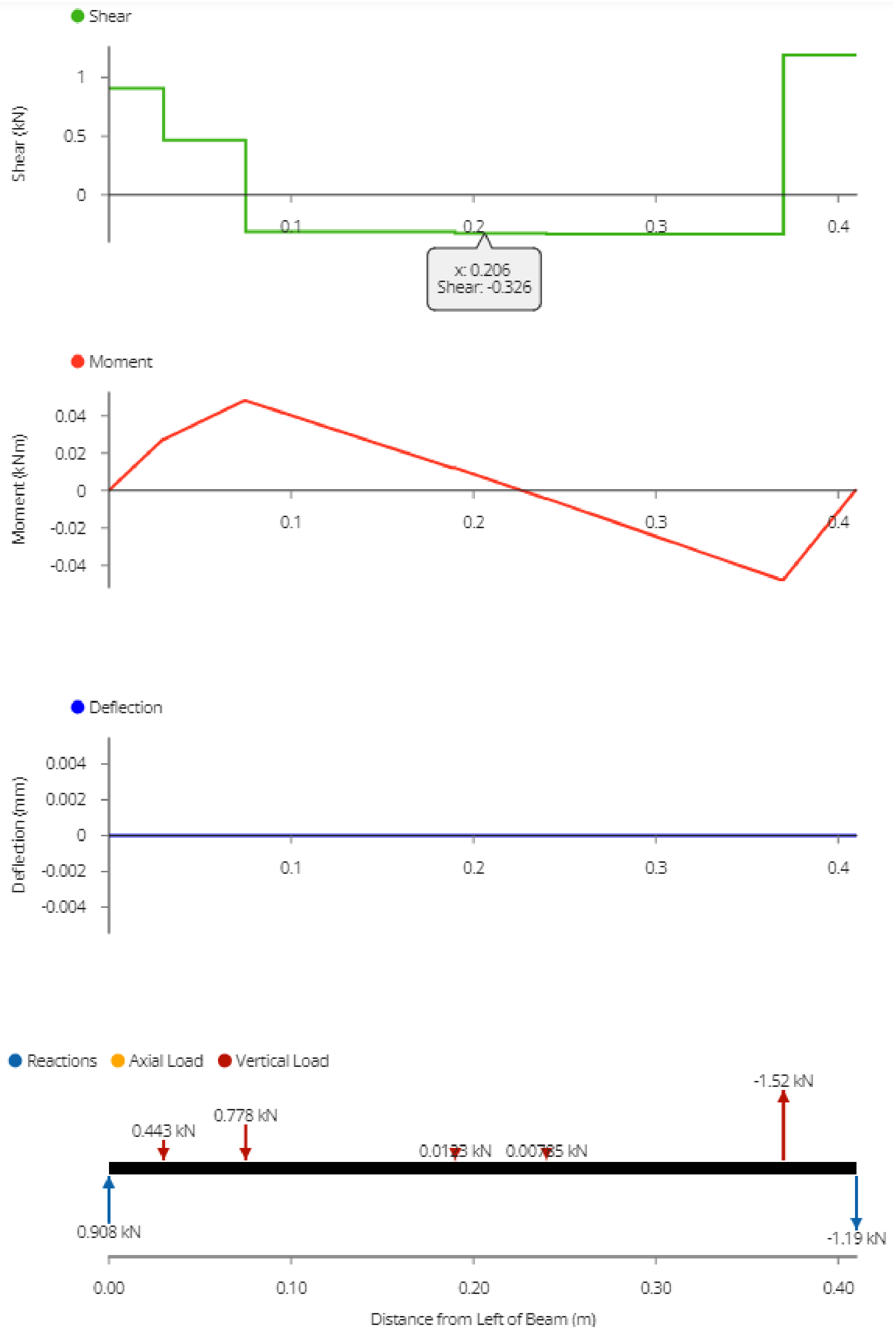


Figure 17: $\mathbf{1st\ Gear\ Y\ direction\ Graphs}$

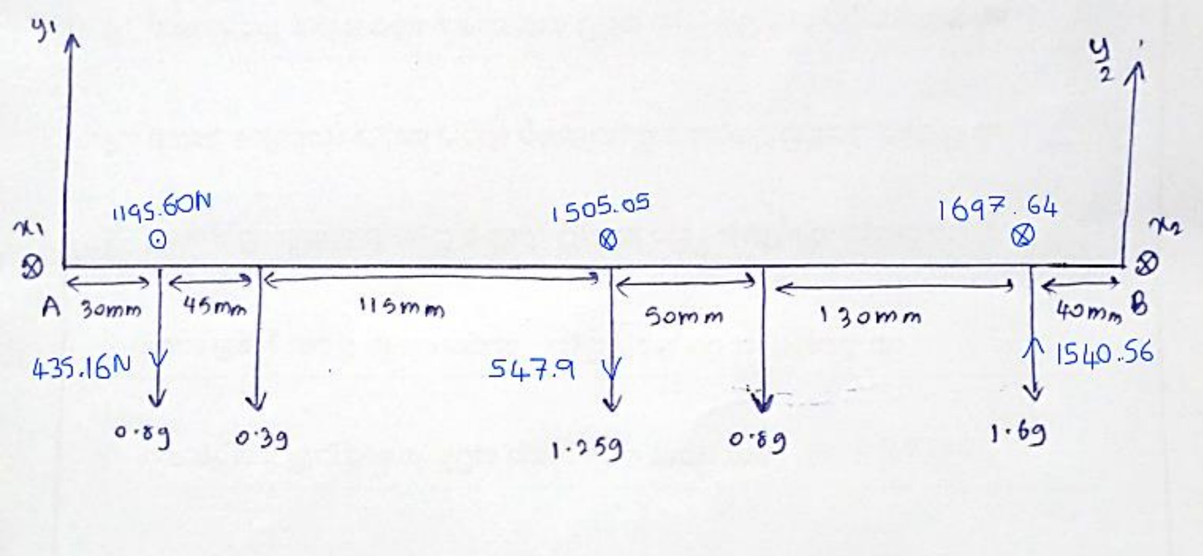
$$M_1 = 114.31Nm$$

$$M_2 = 481.2Nm$$

Maximum moment

$$M = \sqrt{114.31^2 + 481.2^2} = 494.324Nm$$

2nd Gear



Consider point A.

$$y_2 \times 410 = 10477.05 + 435.16 \times 30 + 547.9 \times 190 - 1540.56 \times 370$$

$$y_2 = -1080.6N$$

$$x_2 \times 410 = 1697.6 \times 370 + 1505.05 \times 190 - 1195.6 \times 30$$

$$x_2 = 3530N$$

Consider point B

$$y_1 \times 410 = 8627.8 + 547.9 \times 220 + 435.16 \times 380 - 1540.56 \times 40$$

$$y_1 = 568.7N$$

$$x_1 \times 410 = 1697.6 \times 40 + 1505.05 \times 220 - 1195.6 \times 380$$

$$x_1 = 1480.5N$$

Load of point A bearing

$$R = \sqrt{{x_1}^2 + {y_1}^2}$$

$$R = 1585.96N$$

Load of point B bearing

$$R = \sqrt{{x_2}^2 + {y_2}^2}$$

$$R = 3691.69N$$

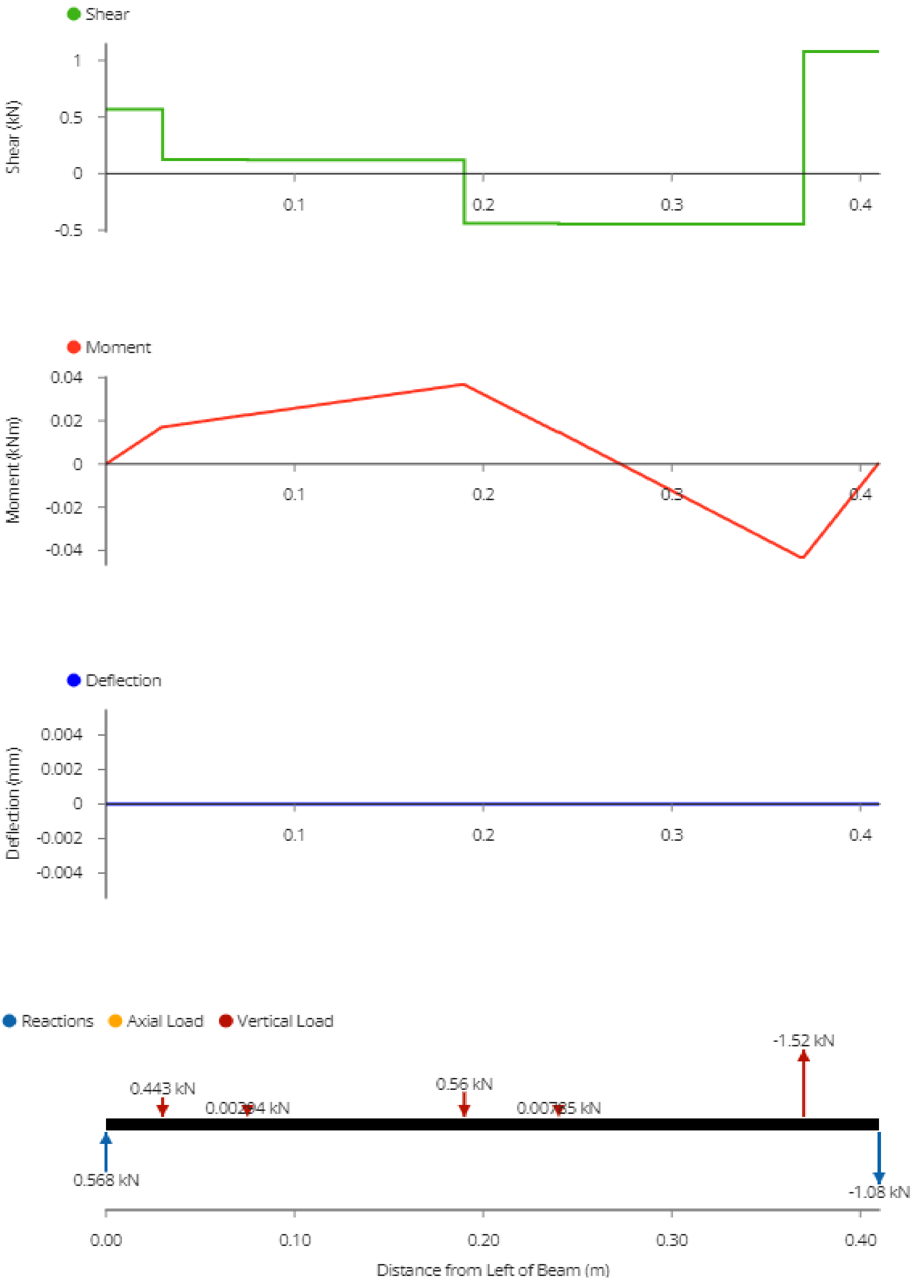


Figure 18: ***2nd Gear Y direction Graphs***

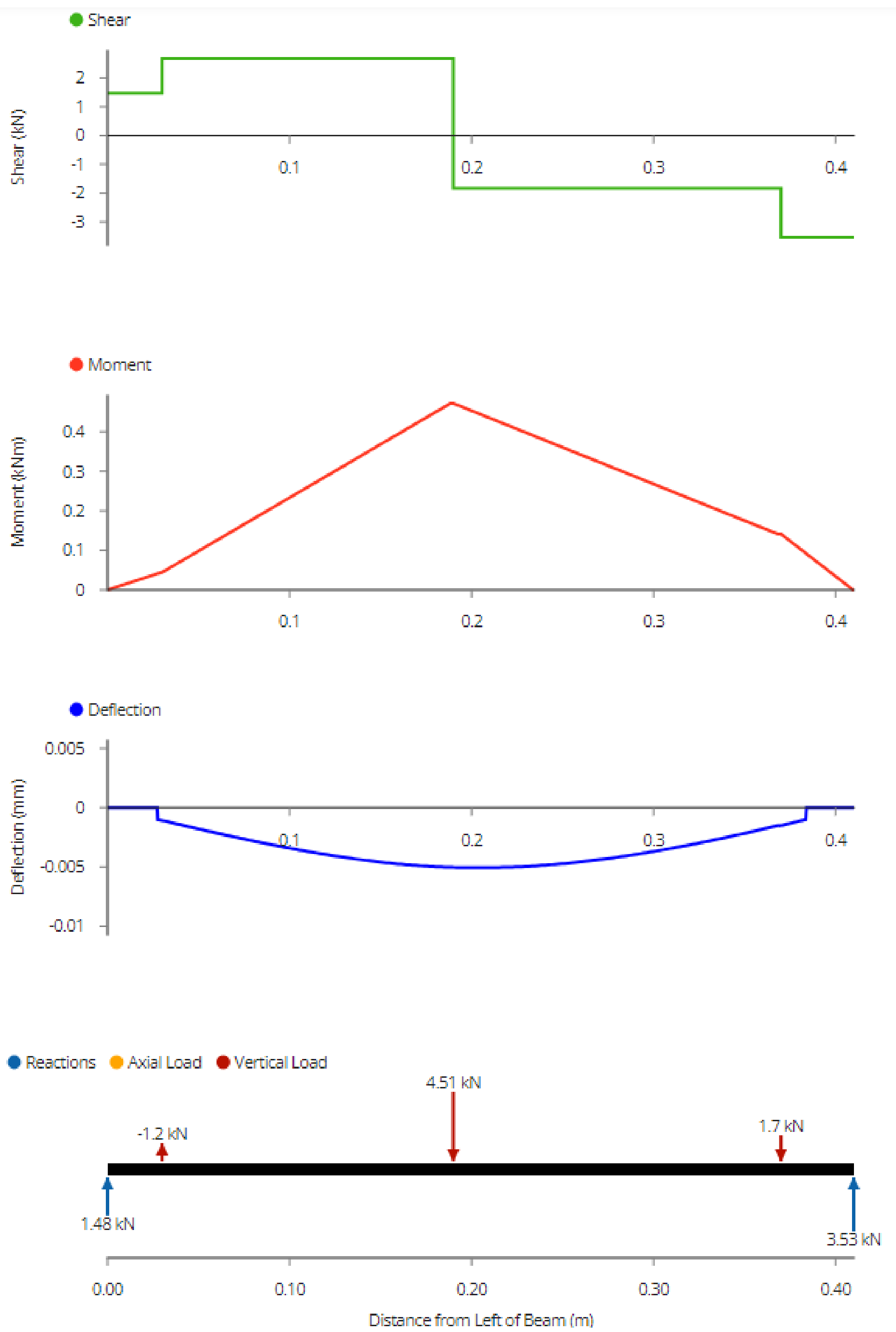


Figure 18: $\boldsymbol{2nd\ Gear\ X\ direction\ Graphs}$

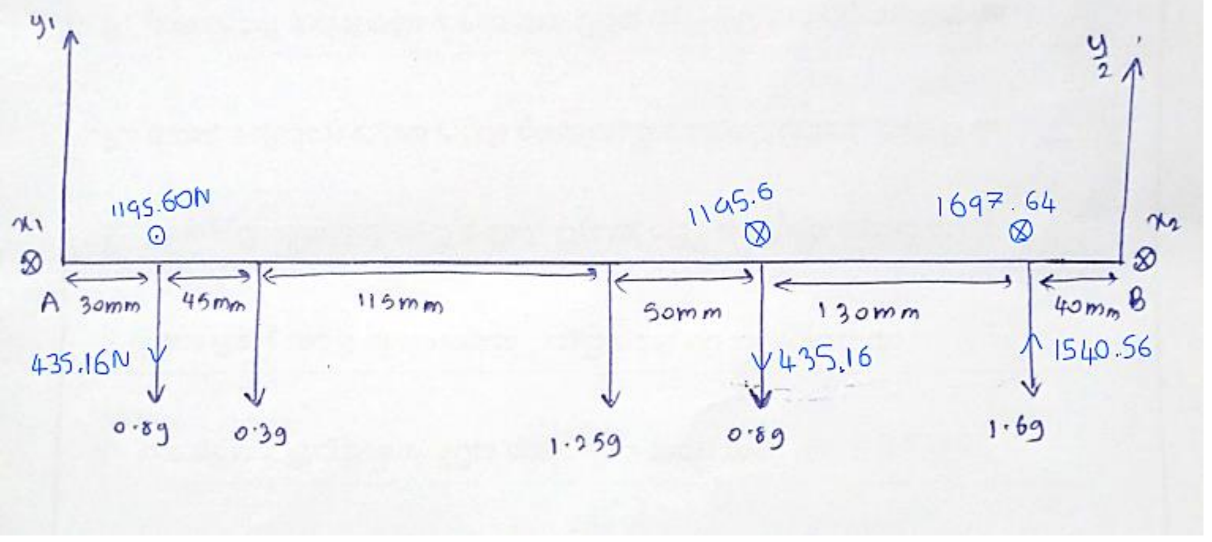
$$M_1 = 43.2Nm$$

$$M_2 = 472.1Nm$$

Maximum moment

$$M = \sqrt{43.2^2 + 472.1^2} = 474.17Nm$$

3rd Gear



Consider point A.

$$y_2 \times 410 = 10477.05 + 435.16 \times 30 + 435.16 \times 240 - 1540.56 \times 370$$

$$y_2 = -1080.2N$$

$$x_2 \times 410 = 1697.6 \times 370 + 1195.6 \times 240 - 1195.6 \times 30$$

$$x_2 = 2140.3N$$

Consider point B

$$y_1 \times 410 = 8627.8 + 435.16 \times 170 + 435.16 \times 380 - 1540.56 \times 40$$

$$y_1 = 454.12N$$

$$x_1 \times 410 = 1697.6 \times 40 + 1195.6 \times 170 - 1195.6 \times 380$$

$$x_1 = 445.2N$$

Load of point A bearing

$$R = \sqrt{445.2^2 + 454.12^2}$$

$$R = 635.94N$$

Load of point B bearing

$$R = \sqrt{2140.3^2 + 1080.2^2}$$

$$R = 2397.43N$$

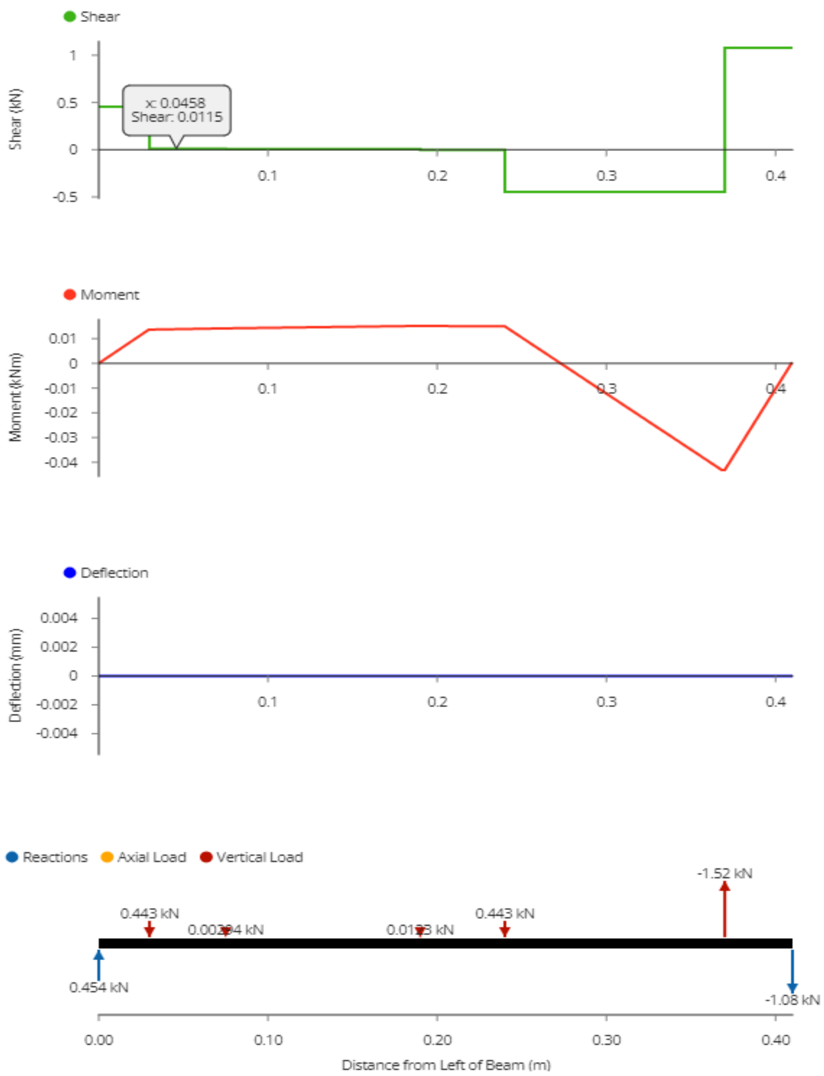


Figure 19: ***3rd Gear Y direction Graphs***

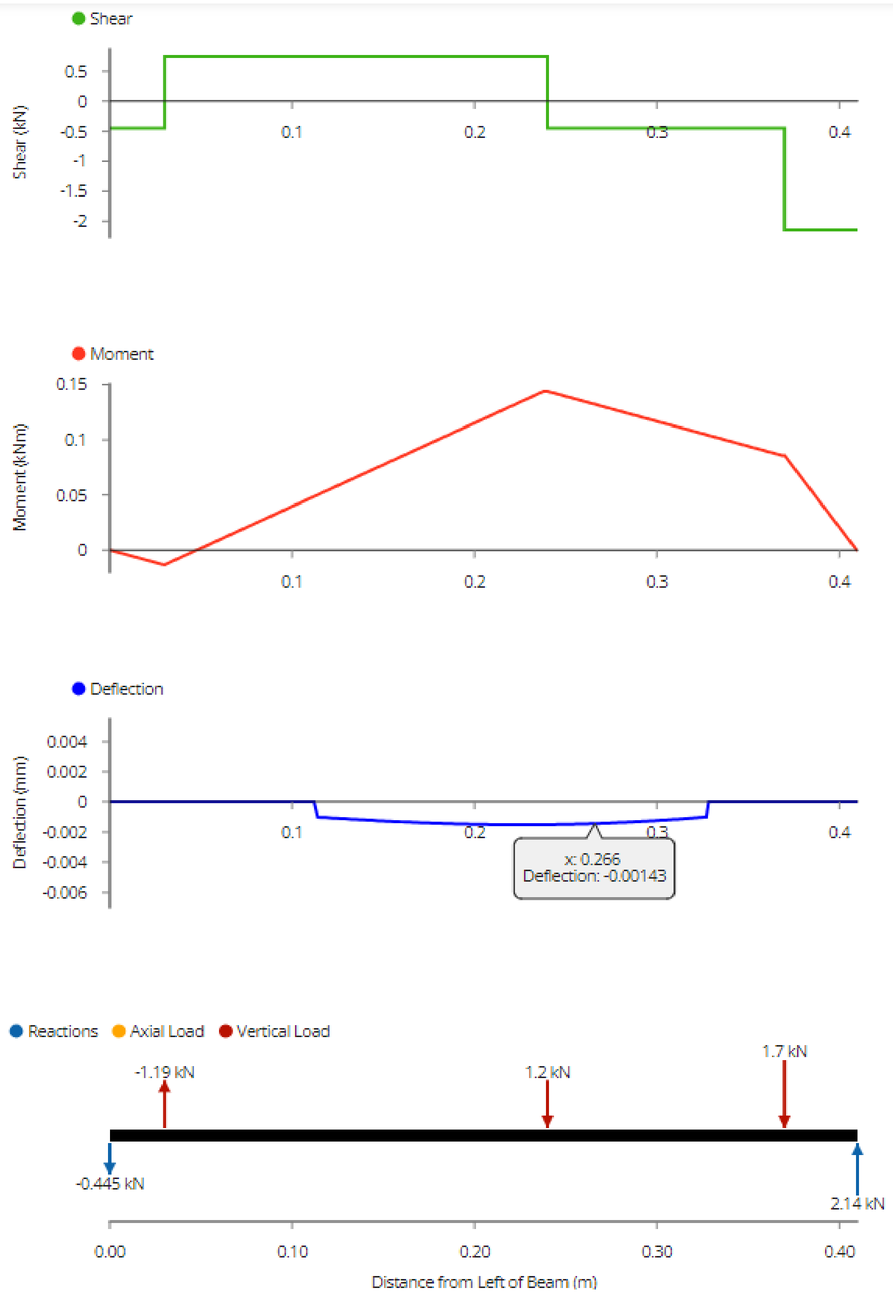


Figure 20: $\mathbf{3rd\ Gear\ X\ direction\ Graphs}$

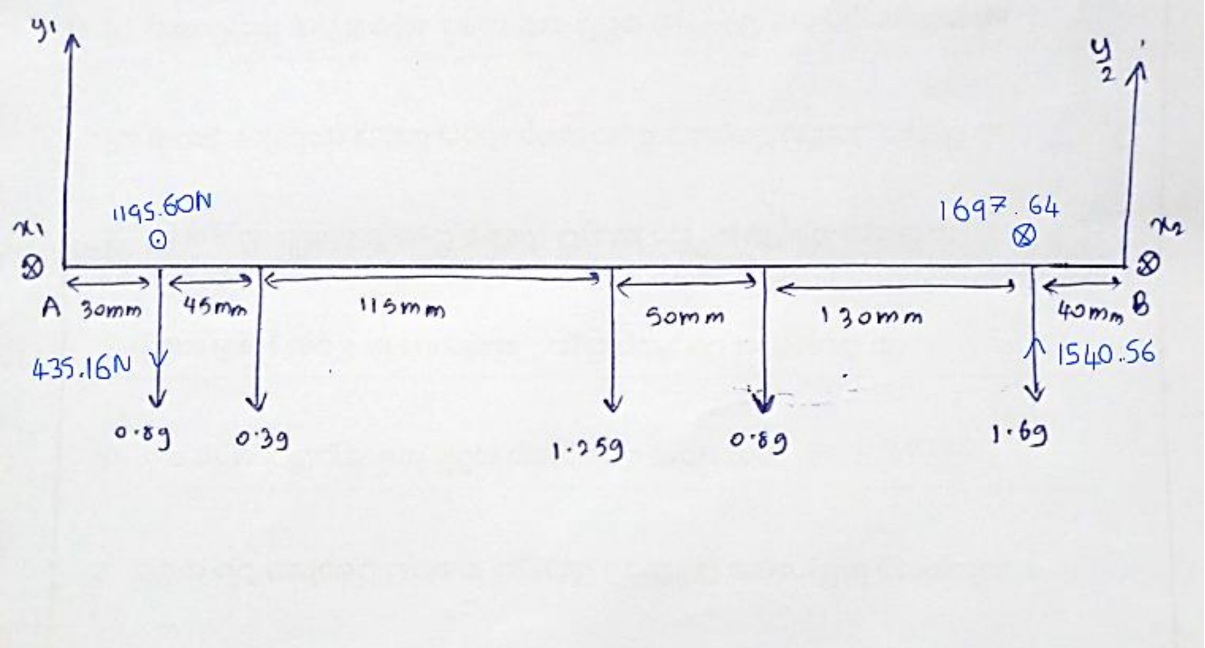
$$M_1 = 43.1Nm$$

$$M_2 = 144.1Nm$$

Maximum moment

$$M = \sqrt{43.1^2 + 144.1^2} = 150.53Nm$$

Reverse Gear



Consider point A.

$$y_2 \times 410 = 10477.05 + 435.16 \times 30 - 1540.56 \times 370$$

$$y_2 = -1330.1N$$

$$x_2 \times 410 = 1697.6 \times 370 - 1195.6 \times 30$$

$$x_2 = 1440.32N$$

Consider point B

$$y_1 \times 410 = 8627.8 + 435.16 \times 380 - 1540.56 \times 40$$

$$y_1 = 247.2N$$

$$x_1 \times 410 = 1697.6 \times 40 - 1195.6 \times 380$$

$$x_1 = 941.2N$$

Load of point A bearing

$$R = \sqrt{{x_1}^2 + {y_1}^2}$$

$$R = 973.12N$$

Load of point B bearing

$$R = \sqrt{{x_2}^2 + {y_2}^2}$$

$$R = 1960.53N$$

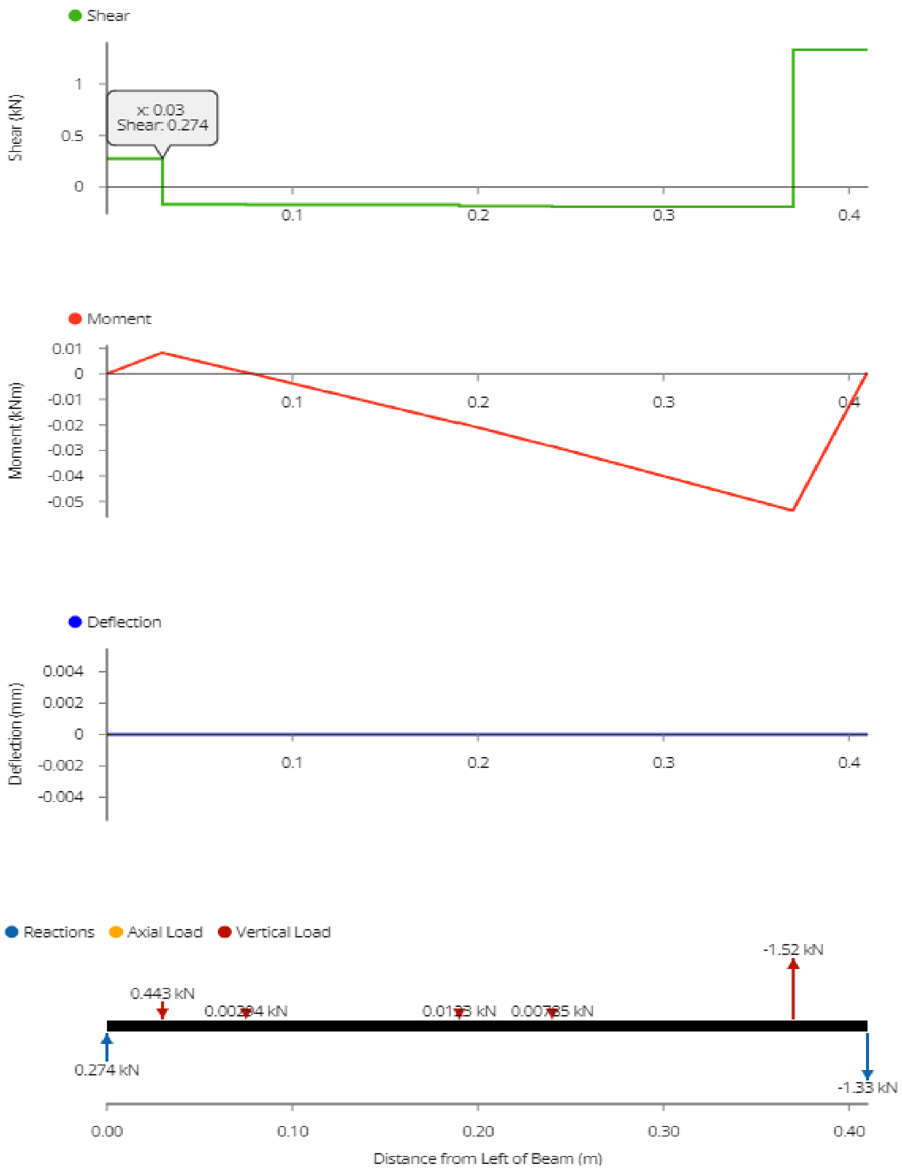


Figure 21: ***Reverse Gear Y direction Graphs***

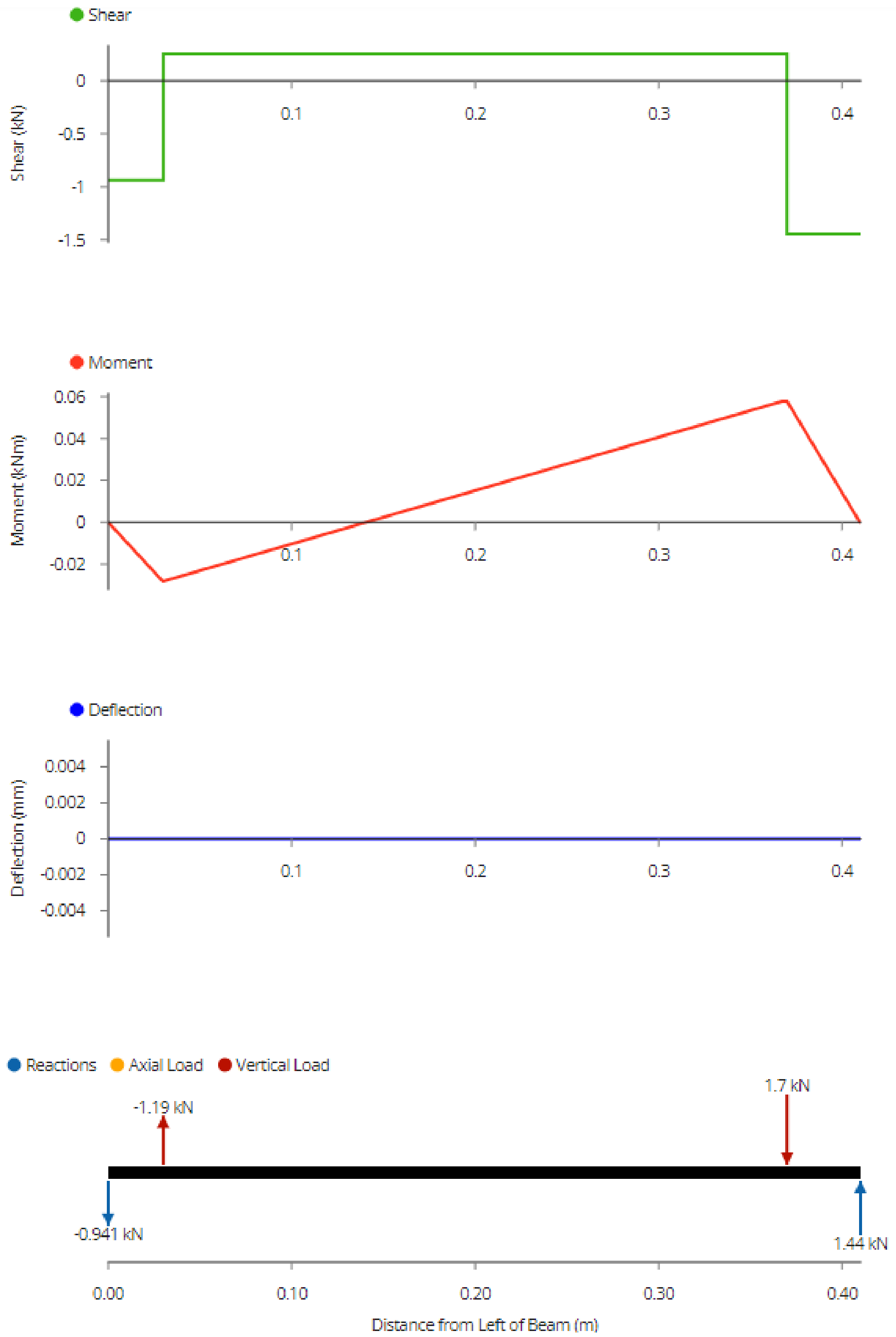


Figure 22: ***Reverse Gear X direction Graphs***

$$M_1 = 53.3Nm$$

$$M_2 = 57.8Nm$$

Maximum moment

$$M = \sqrt{53.3^2 + 57.8^2} = 78.63Nm$$

Maximum bending moment = 494.32Nm

Maximum force on the A point bearing = 1585.96N

Maximum force on the B point bearing = 3691.69N

Lay shaft diameter Calculation

Assumption:

Shaft material = 50C 12 steel (AISI 1065

Indian standard designation	*Ultimate tensile strength, MPa*	*Yield strength, MPa*
40 C 8	560 - 670	320
45 C 8	610 - 700	350
50 C 4	640 - 760	370
50 C 12	700 Min.	390

Figure 22: mechanical properties of steel used for shaft

50 C 12 Steel (AISI 1065)		
Density	7.85	g/cm3
Ultimate tensile strength	700	MPa
Modulus of Rigidity	80	GPa
Allowable Shear Stress	525	MPa

Figure 28: properties of shaft material

Tensile stress= 700Mpa

Shear stress= 0.75×700

= 525Mpa

Torque of the lay shaft:

$$T = \frac{500}{150 \times 2\pi \times \frac{1}{60}}$$

$$T = 31.83Nm$$

Nature of load	K_m	K_t
1. Stationary shafts		
(a) Gradually applied load	1.0	1.0
(b) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(a) Gradually applied or steady load	1.5	1.0
(b) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(c) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

Figure 29: recommended value for K_m *and* K_t

Because of a rotating steady load applied shaft,

$$K_m = 1.5, K_t = 1$$

$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2}$$

$$T_e = \sqrt{(1.5 \times 494.32)^2 + (1 \times 31.83)^2} = 742.16Nm$$

Equivalent bending Moment of the shaft,

$$M_e = \left(K_m \times M + \sqrt{(K_m \times M)^2 + (K_t \times T)^2}\right)$$

$$M_e = 741.82Nm$$

$$\tau \geq \frac{16 \times T_e}{\pi \times d^3}$$

$$525 \times 10^6 \geq \frac{16 \times 742.16}{\pi \times d^3}$$

$$d \geq 19.3mm$$

$$\sigma \geq \frac{32 \times M_e}{\pi \times d^3}$$

$$700 \times 10^6 \geq \frac{32 \times 741.8}{\pi \times d^3}$$

$$d \geq 22.1mm$$

According to the standard diameter catalogue we can get the lay shaft diameter as 25mm.

Output Shaft

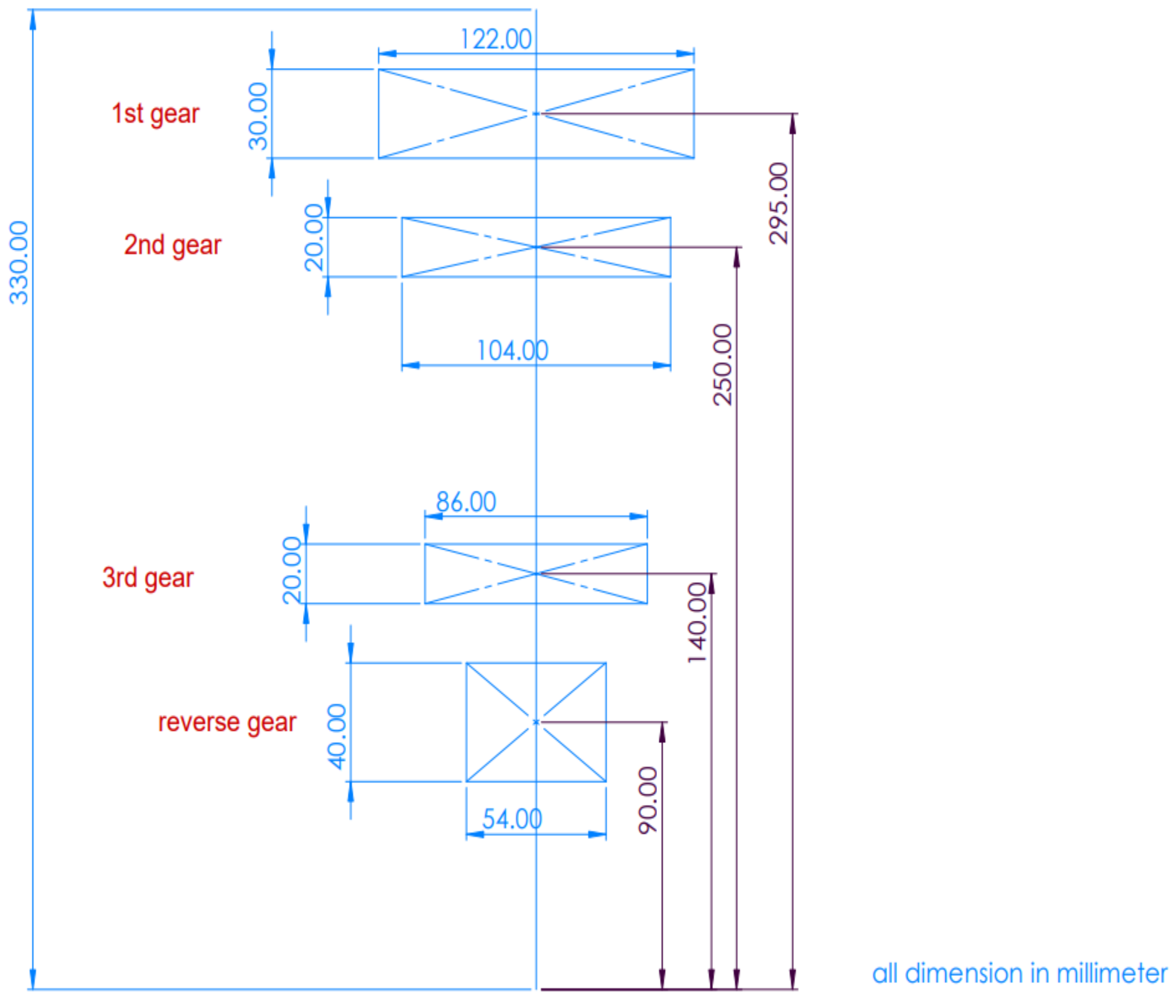


Figure 30

<u>Formula</u>

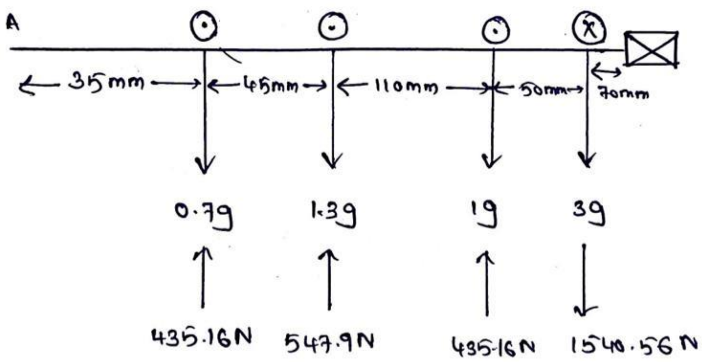
Total bending moment

$$M = \sqrt{{M_x}^2 + {M_y}^2}$$

Load of bearing

$$R = \sqrt{X^2 + Y^2}$$

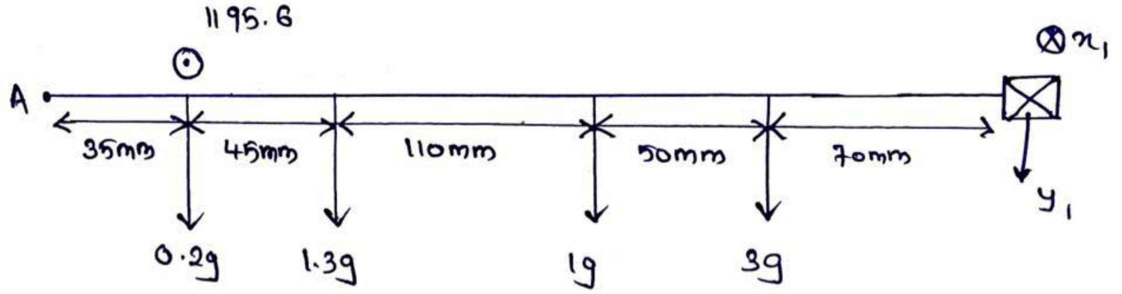
The total bending moment of the shaft when considering the gear mass



$$M_A = 0.2g \times 35 + 1.3g \times 65 + 1g \times 175 + 3g \times 225$$

$$M_A = 9236.115Nm$$

1st Gear



Consider point A.

$$y_1 \times 330 = -9236.11 + 435.16 \times 35$$

$$y_2 = 18.16N$$

$$x_1 = 1195.6$$

Load on bearing

$$R = \sqrt{{x_1}^2 + {y_1}^2}$$

$$R = 1195.73N$$

Torque of the output shaft:

$$T = \frac{500}{57.3 \times 2\pi \times \frac{1}{60}}$$

$$T = 83.32Nm$$

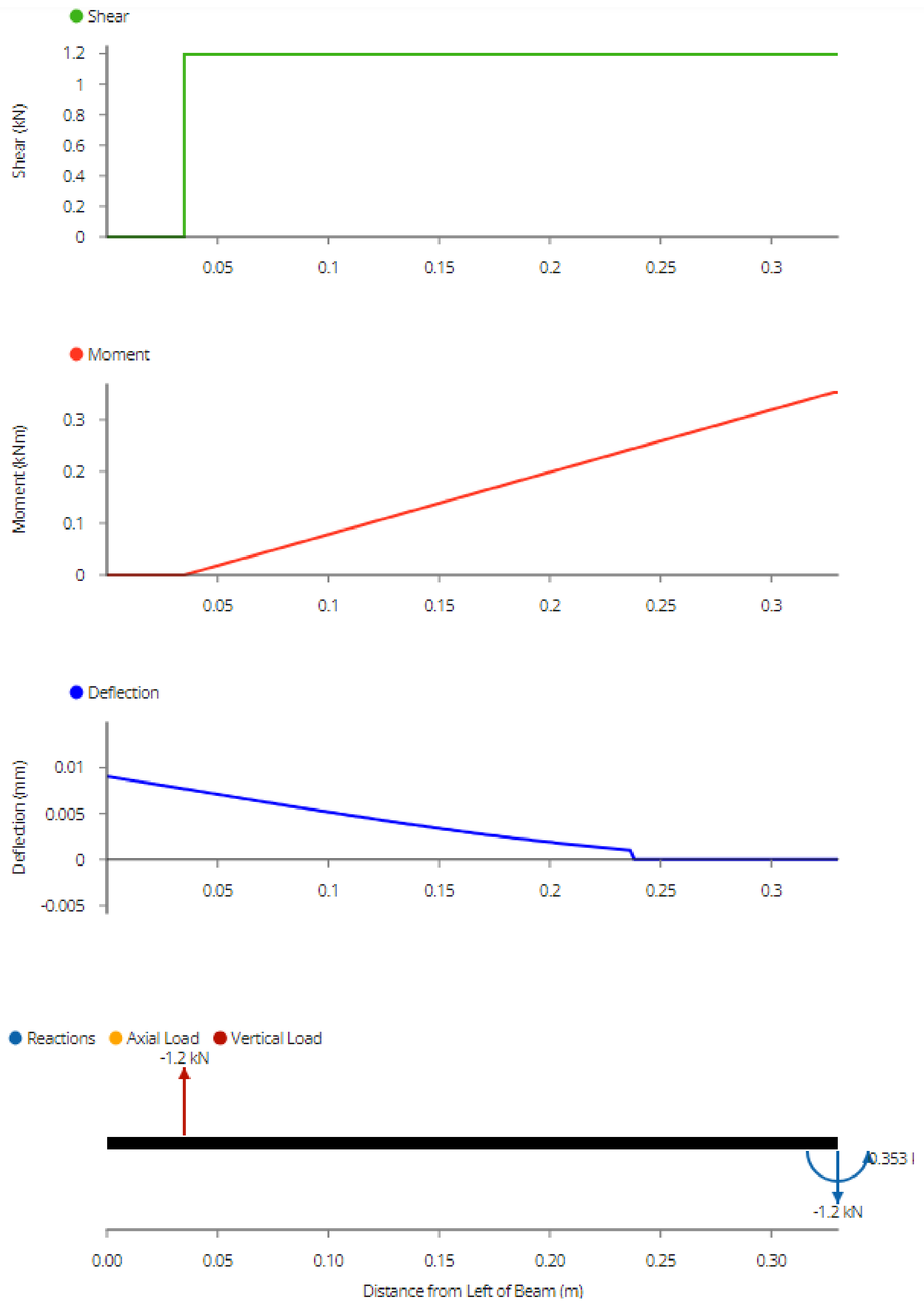


Figure 31: 1*st Gear X direction Graphs*

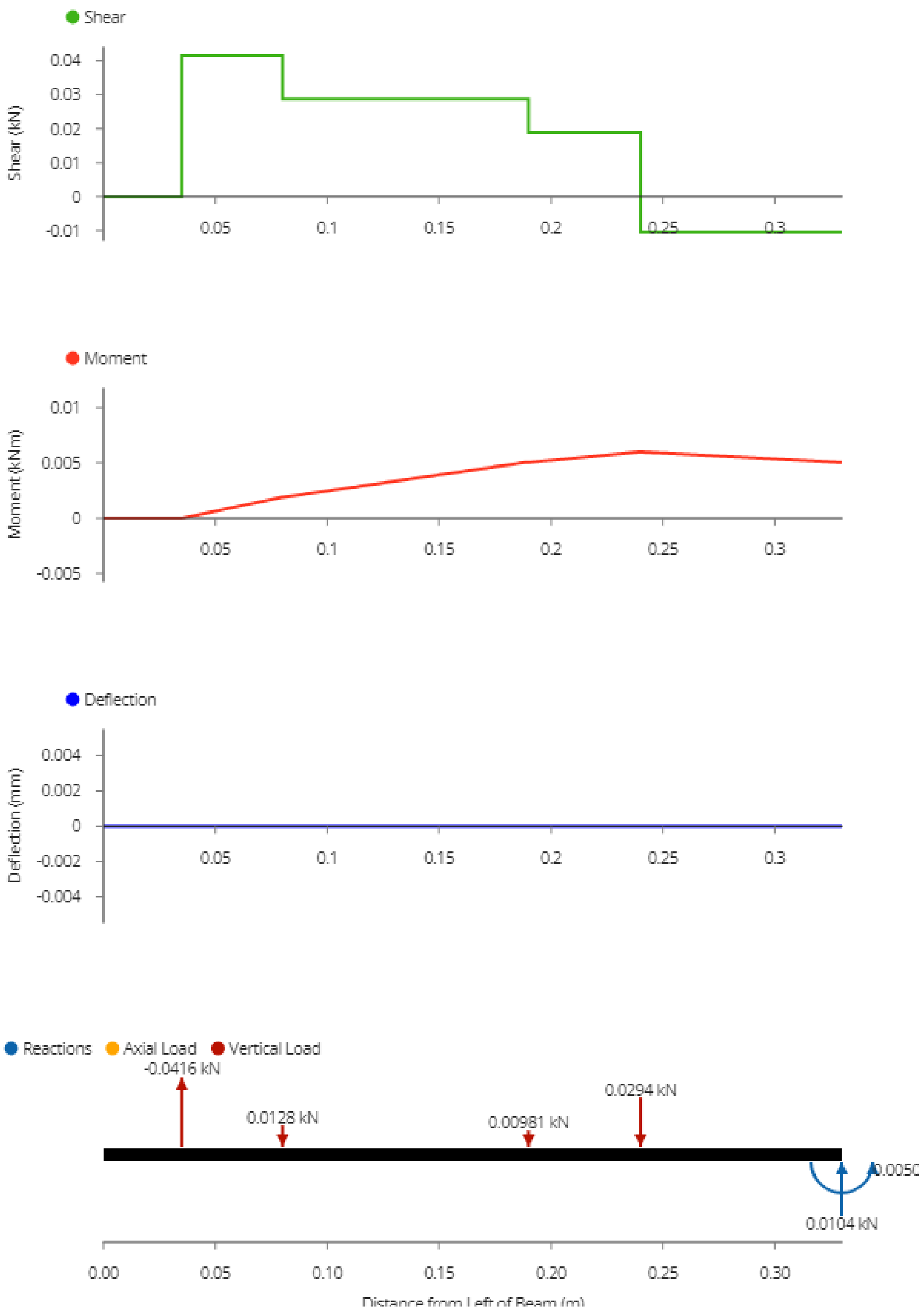


Figure 32: $1st\ Gear\ Y\ direction\ Graphs$

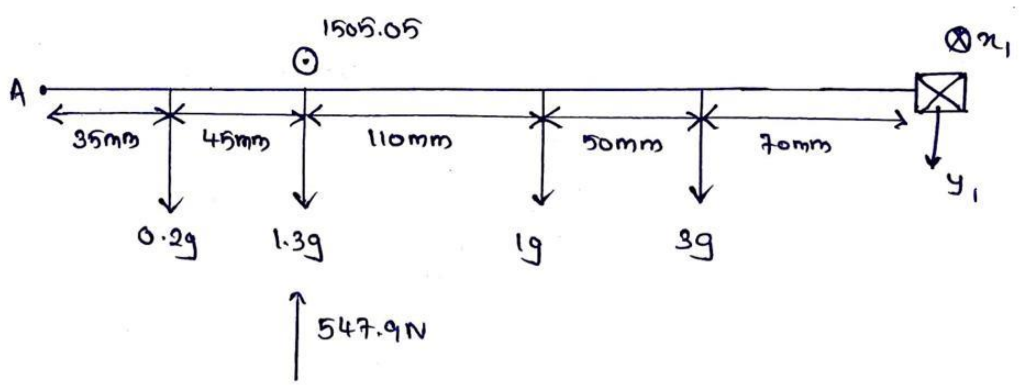
$$M_1 = 6Nm$$

$$M_2 = 1195.64Nm$$

Maximum moment

$$M = \sqrt{6^2 + 1195.64^2} = 353.05Nm$$

2nd Gear



Consider point A.

$$y_1 \times 330 = -9236.11 + 547.9 \times 80$$

$$y_1 = 104.83N$$

$$x_1 = 1505.05N$$

Load on bearing

$$R = \sqrt{{x_1}^2 + {y_1}^2}$$

$$R = 1585.96N$$

Torque of the output shaft:

$$T = \frac{500}{95.5 \times 2\pi \times \frac{1}{60}}$$

$$T = 49.29Nm$$

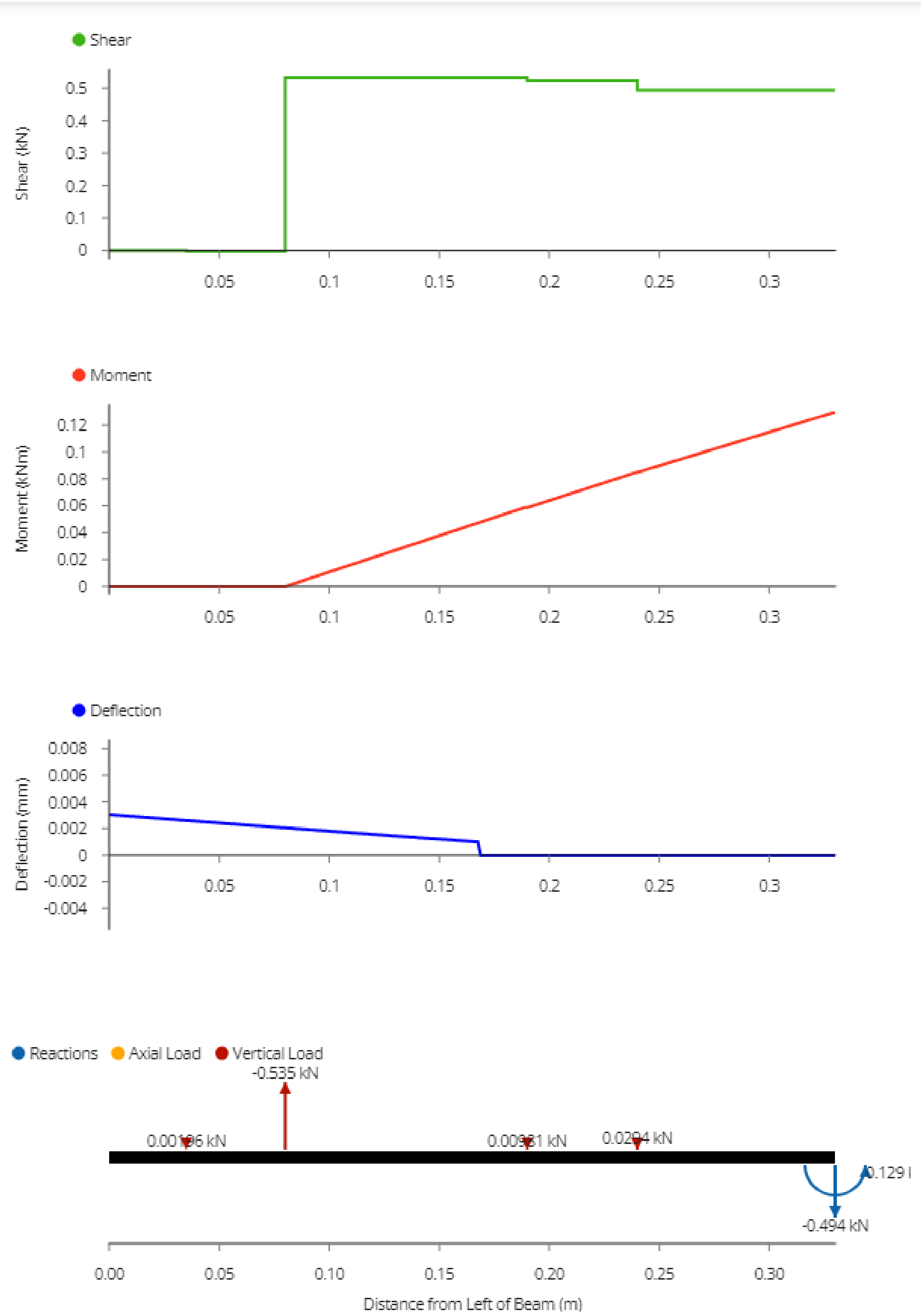


Figure 33: *2nd Gear Y direction Graphs*

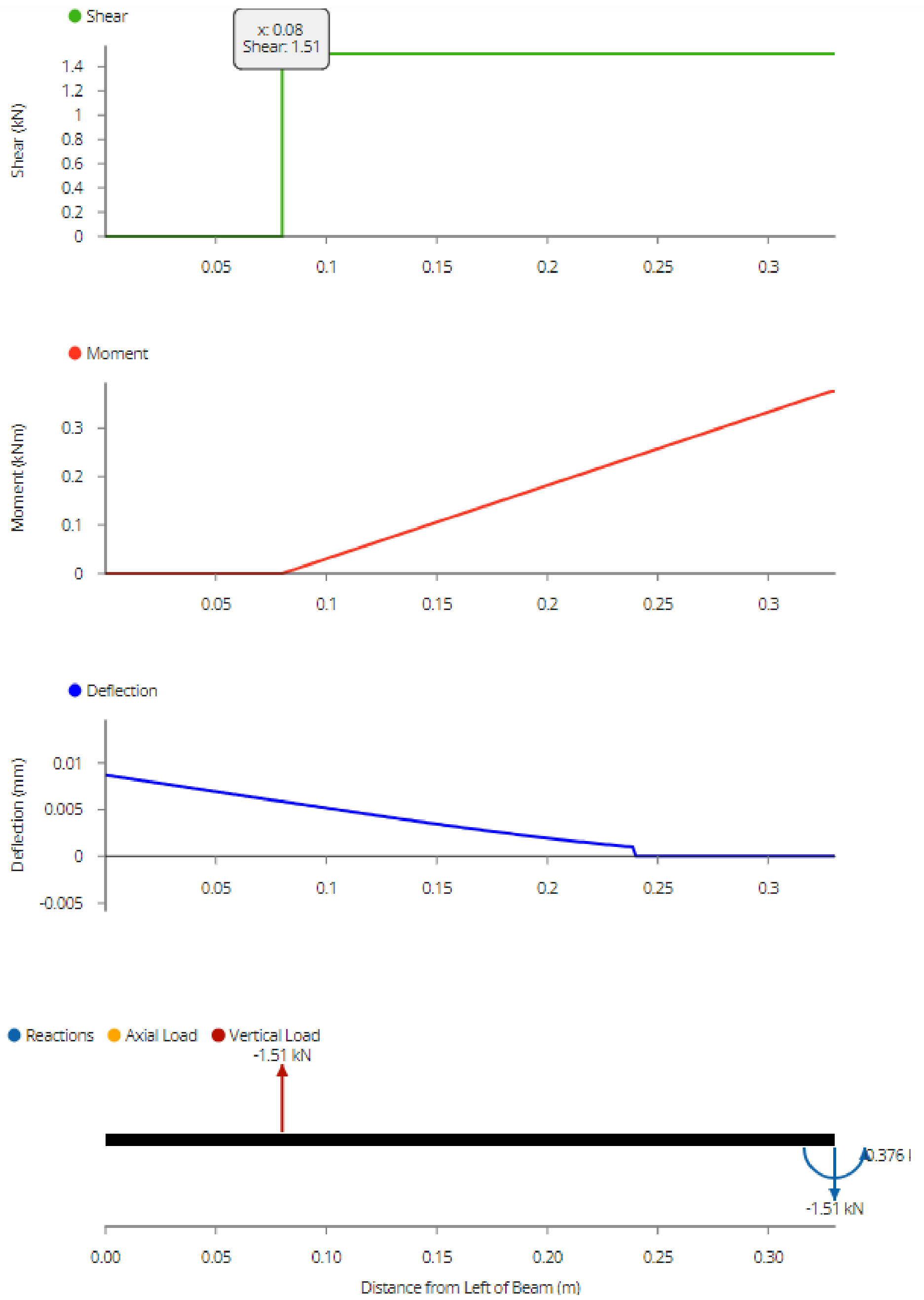


Figure 34: *2nd Gear X direction Graphs*

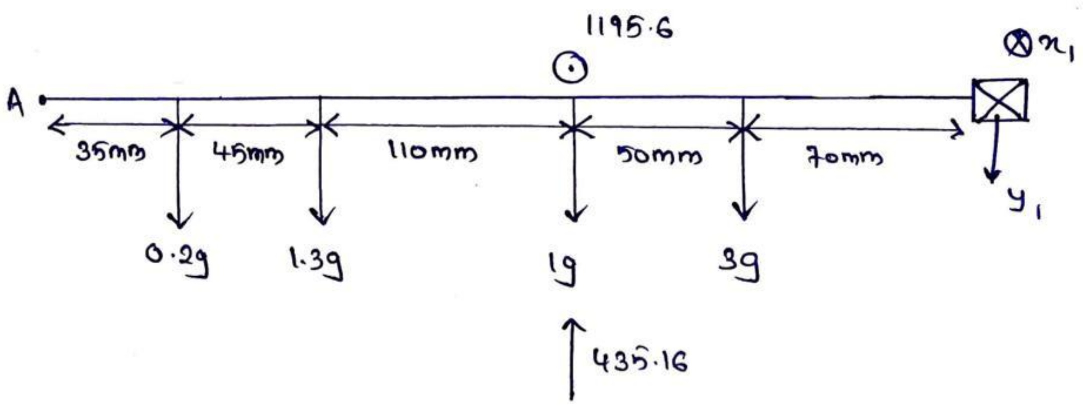
$$M_1 = 129.1Nm$$

$$M_2 = 376Nm$$

Maximum moment

$$M = \sqrt{129.1^2 + 376^2} = 397.5Nm$$

3rd Gear



Consider point A.

$$y_1 \times 330 = -9236.11 + 435.16 \times 190$$

$$y_1 = 222.5N$$

$$x_1 = 1195.6N$$

Load on bearing

$$R = \sqrt{1195.6^2 + 222.5^2}$$

$$R = 1216.12N$$

Torque of the output shaft:

$$T = \frac{500}{143.9 \times 2\pi \times \frac{1}{60}}$$

$$T = 33.18Nm$$

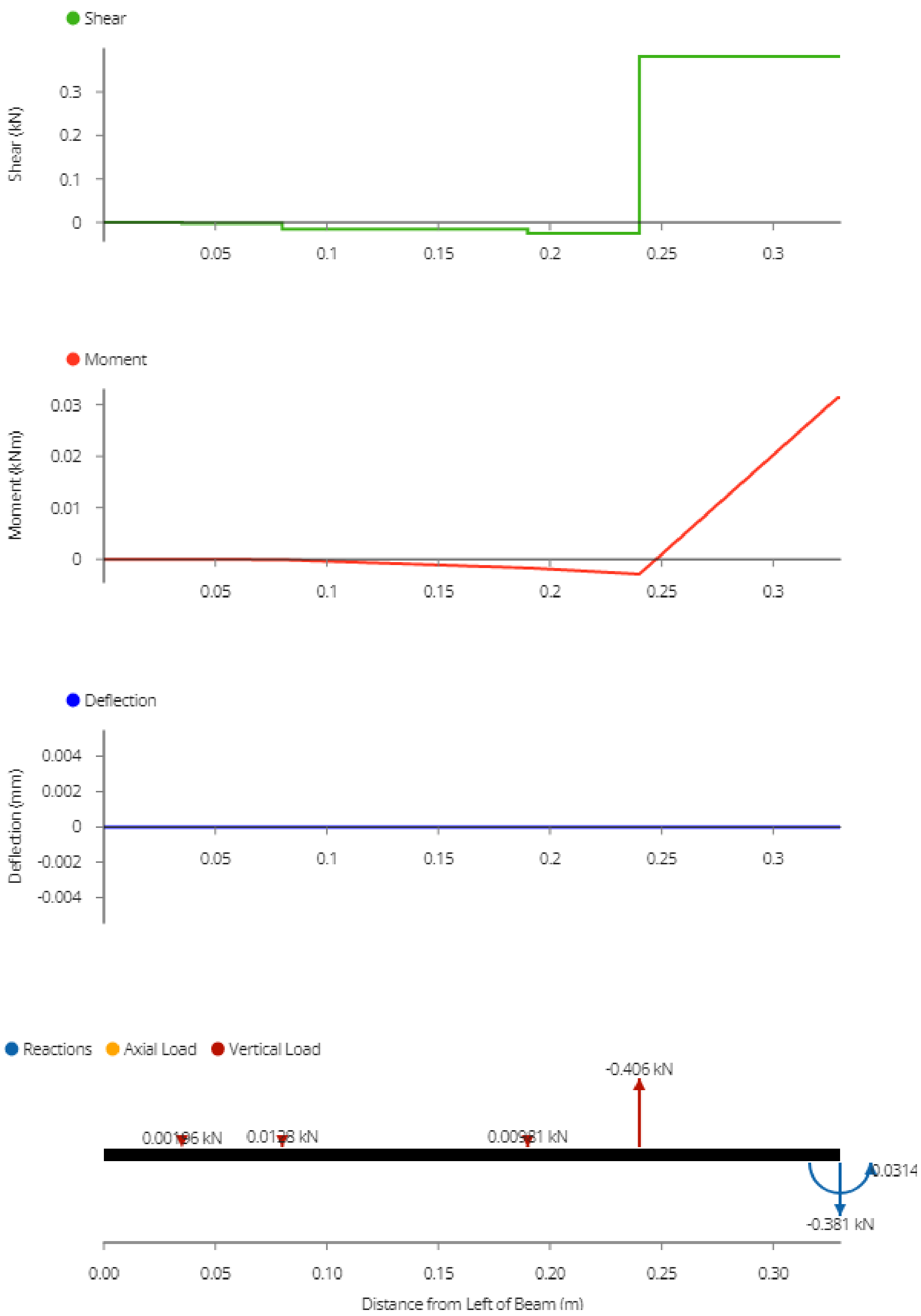


Figure 35: *3rd Gear Y direction Graphs*

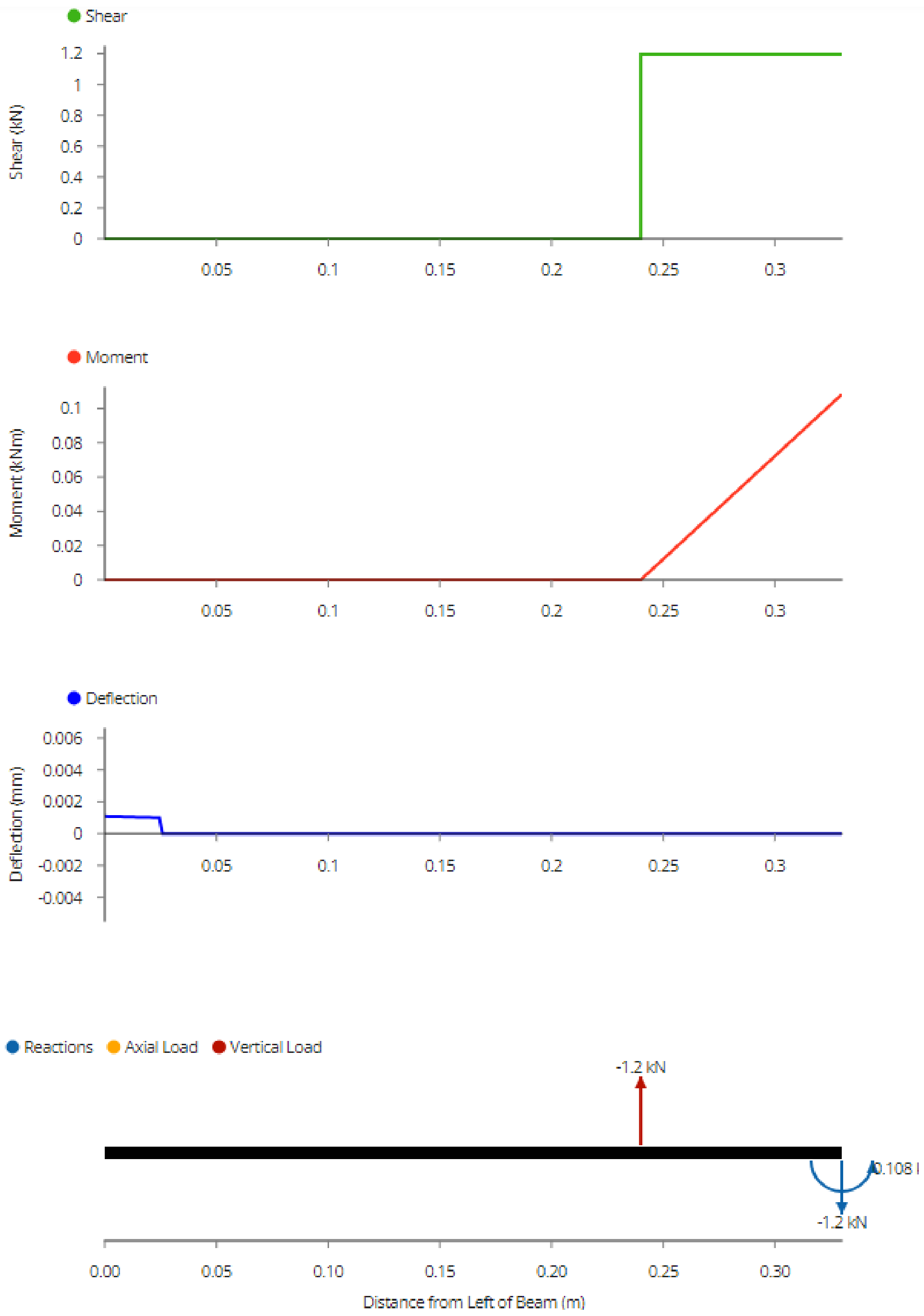


Figure 36: *3rd Gear X direction Graphs*

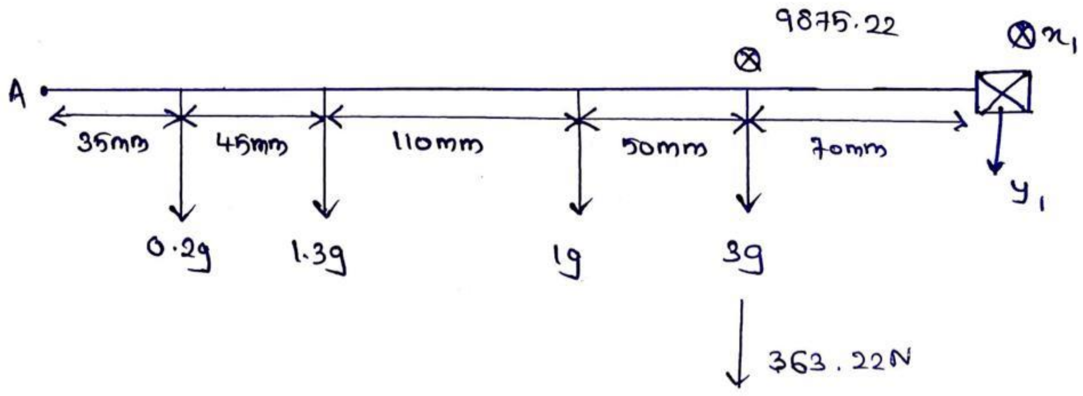
$$M_1 = 53.1Nm$$

$$M_2 = 167.2Nm$$

Maximum moment

$$M = \sqrt{53.1^2 + 167.2^2} = 175.42Nm$$

Reverse Gear



$$y_1 = 417.2N$$

$$x_1 = 2875.22N$$

Load on bearing

$$R = \sqrt{2875.22^2 + 417.2^2}$$

$$R = 2905.33N$$

Torque of the output shaft:

$$T = \frac{500}{150 \times 2\pi \times \frac{1}{60}}$$

$$T = 31.83Nm$$

Torque of the output shaft:

$$T = \frac{500}{143.9 \times 2\pi \times \frac{1}{60}}$$

$$T = 33.18Nm$$

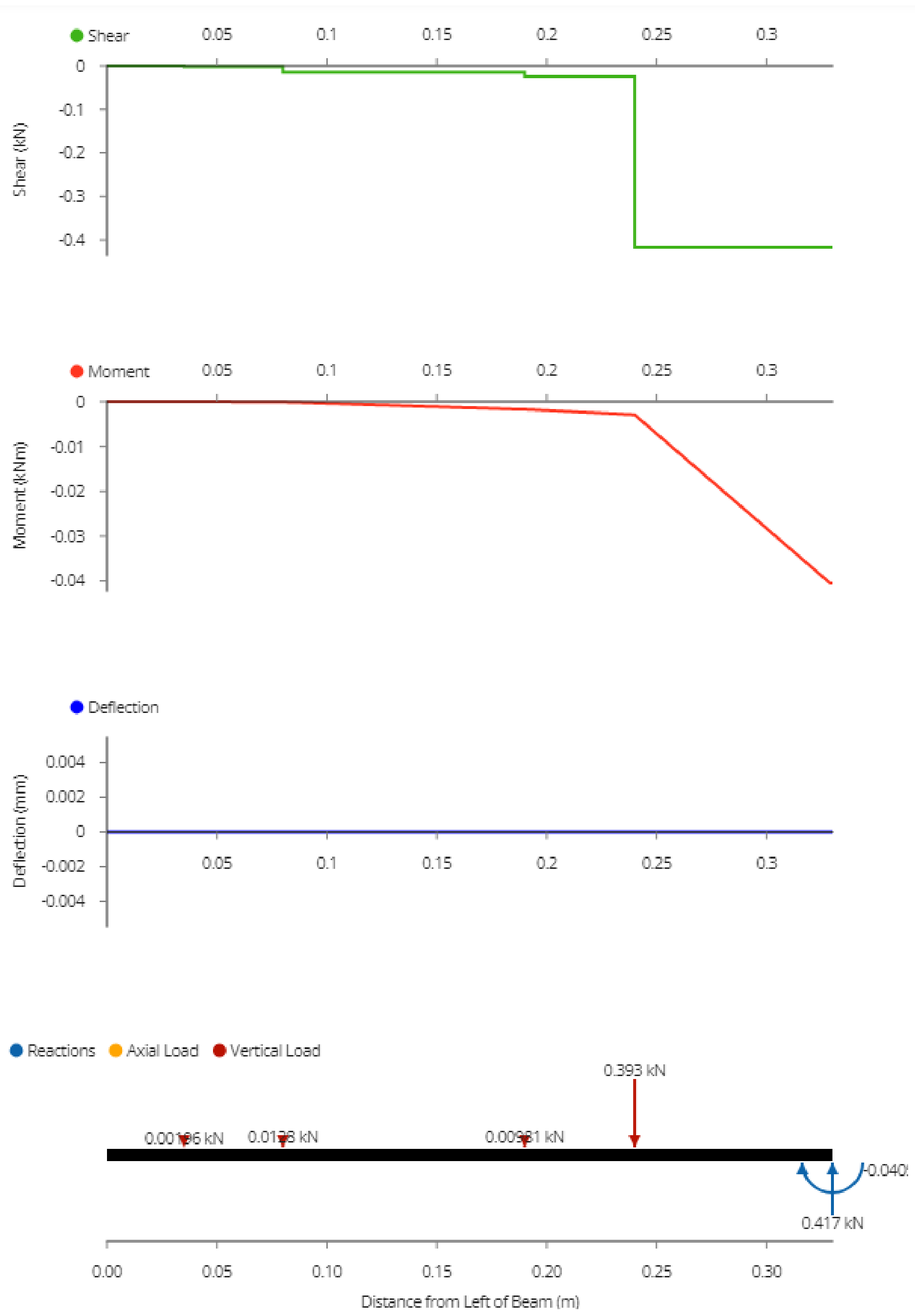


Figure 37: *Reverse Gear Y direction Graphs*

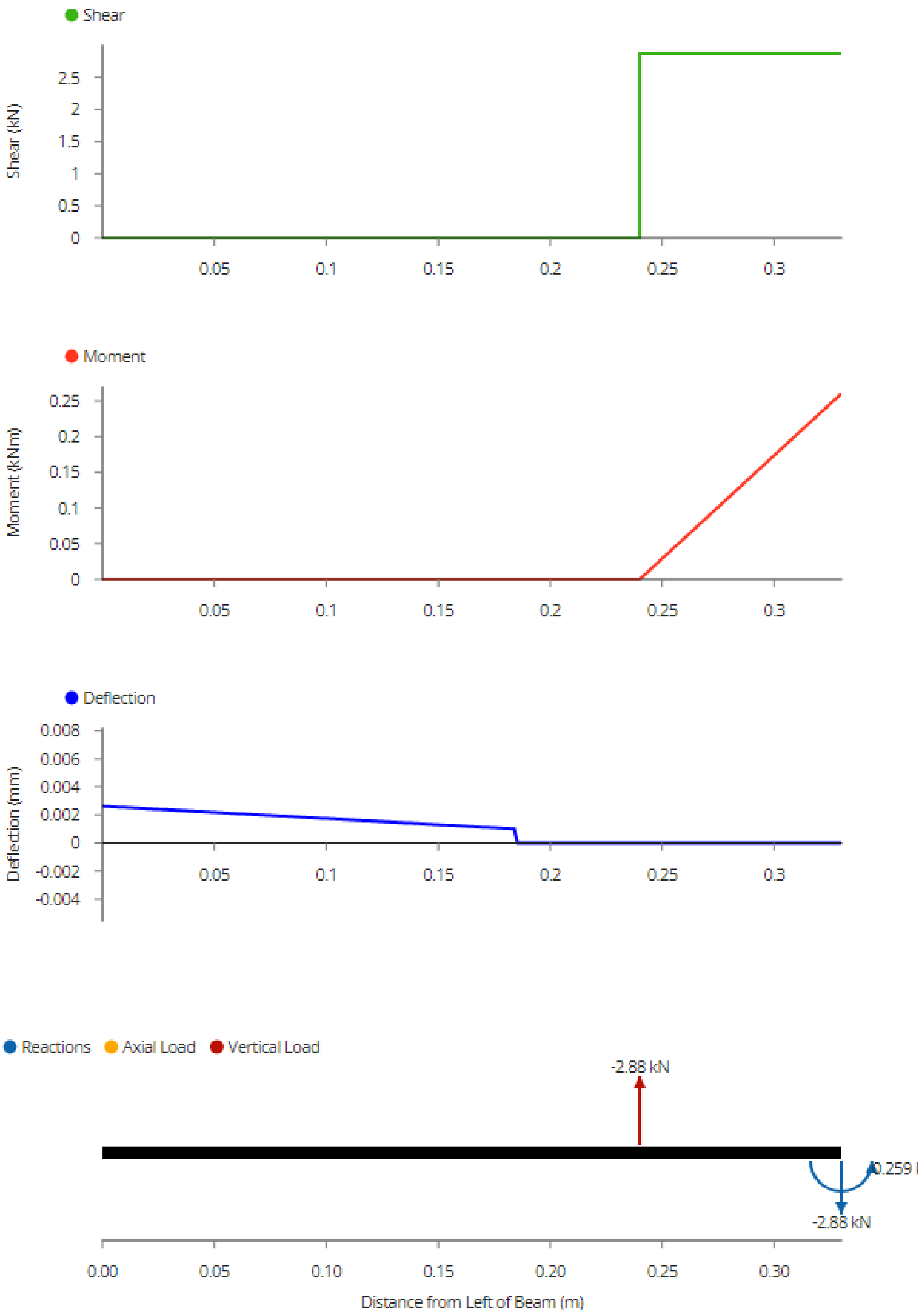


Figure 38: *Reverse Gear X direction Graphs*

$$M_1 = 259.2Nm$$

$$M_2 = 40.5Nm$$

Maximum moment

$$M = \sqrt{259.2^2 + 40.5^2} = 262.14Nm$$

Maximum bending moment = 397.54Nm

Maximum Torque = 83.32Nm

Maximum force on bearing = 2905.33N

Output shaft diameter Calculation

Assumption:

Shaft material = 50C 12 steel (AISI 1065)

Indian standard designation	*Ultimate tensile strength, MPa*	*Yield strength, MPa*
40 C 8	560 - 670	320
45 C 8	610 - 700	350
50 C 4	640 - 760	370
50 C 12	700 Min.	390

Figure 39: mechanical properties of steel used for shaft

50 C 12 Steel (AISI 1065)		
Density	7.85	g/cm3
Ultimate tensile strength	700	MPa
Modulus of Rigidity	80	GPa
Allowable Shear Stress	525	MPa

Figure 40: properties of shaft material

$$Tensile\ stress = \ 700Mpa$$

$$Shear\ stress = \ 0.75 \times 700 = 525MPa$$

Nature of load	K_m	K_t
1. Stationary shafts		
(a) Gradually applied load	1.0	1.0
(b) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(a) Gradually applied or steady load	1.5	1.0
(b) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(c) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

Figure 41: recommended value for K_m and K_t

Because of a rotating steady load applied shaft,

$$K_m = 1.5, K_t = 1$$

$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2}$$

$$T_e = \sqrt{(1.5 \times 397.54)^2 + (1 \times 83.32)^2} = 602.122Nm$$

Equivalent bending Moment of the shaft,

$$M_e = \left(K_m \times M + \sqrt{(K_m \times M)^2 + (K_t \times T)^2}\right)$$

$$M_e = 599.176Nm$$

$$\tau \geq \frac{16 \times T_e}{\pi \times d^3}$$

$$525 \times 10^6 \geq \frac{16 \times 602.122}{\pi \times d^3}$$

$$d \geq 18mm$$

$$\sigma \geq \frac{32 \times M_e}{\pi \times d^3}$$

$$700 \times 10^6 \geq \frac{32 \times 599.176}{\pi \times d^3}$$

$$d \geq 22.58mm$$

According to the standard diameter catalogue we can get the output shaft diameter as 25mm.

Spline Design

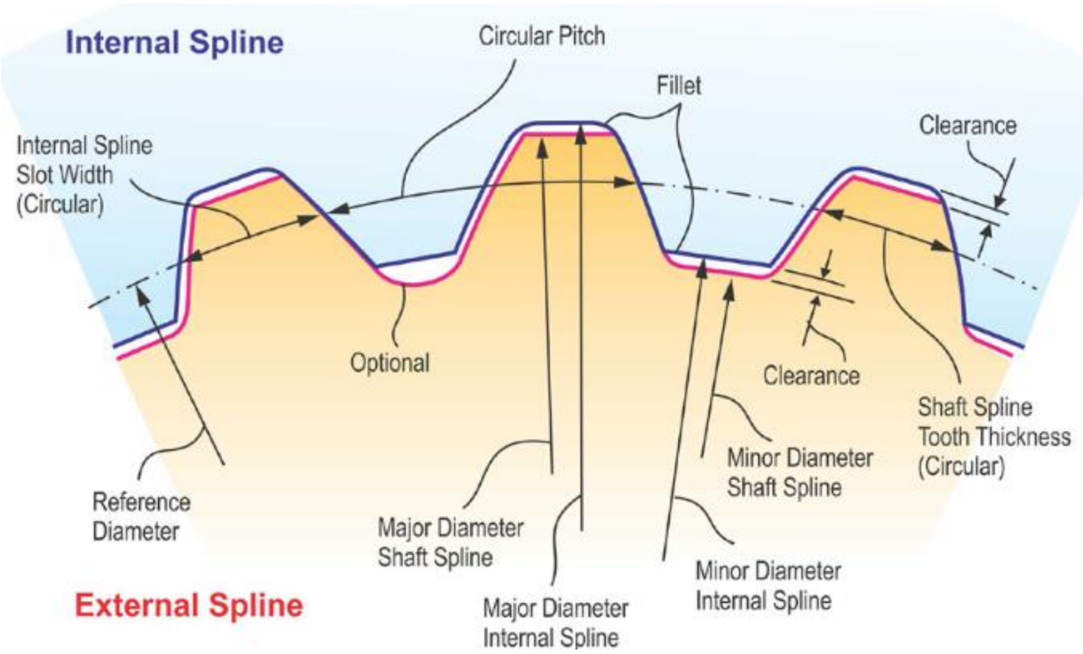


Figure 42 : Design parameters of splines

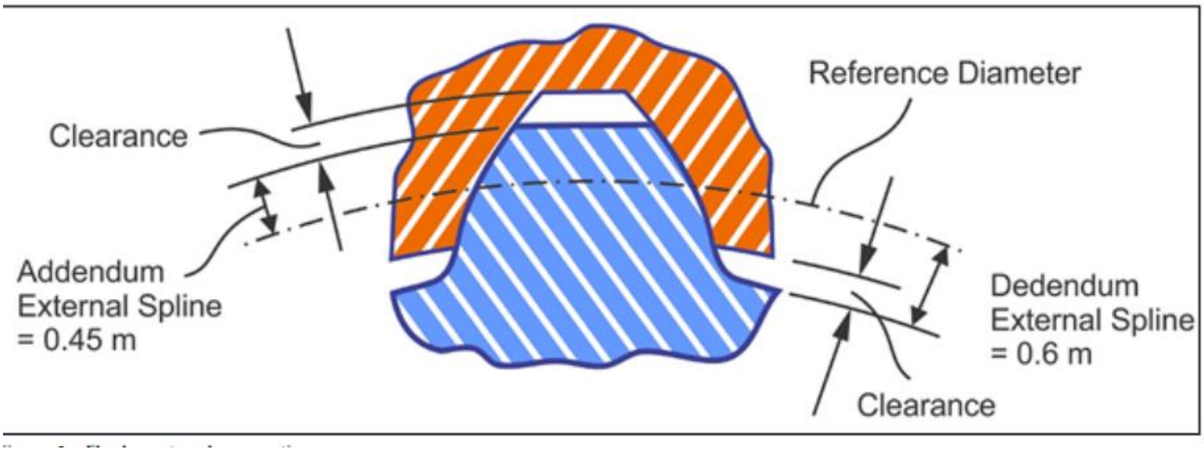


Figure 43 : Flank cantered connection

Data:

Diameter of the output shaft = 25 mm

Minor diameter of the external spline = 25 mm

Assumption:

The same material is used to create both the shaft and the splines.

The spline has a total of 10 teeth.

The spline's pressure angle is 37.5^0 degrees.

Major diameter of the external spline = $1.25 \times D = 31.25\ mm$

$$Tooth\ depth = \frac{D_{major} - D_{minor}}{2} = 3.125\ mm$$

$$Tooth\ depth = Addendum\ of\ external\ spline + Dedendum\ of\ external\ spline$$

$$Addendum\ of\ external\ spline = 0.45 \times m \qquad ; m - Gear\ module$$

$$Dedendum\ of\ external\ spline = 0.60 \times m \qquad ; m - Gear\ module$$

$$Tooth\ depth = 0.45m + 0.60m = 3.15$$

$$Gear\ module = m = 3$$

$Spline\ pitch\ diameter = 3 \times 10 = 30\ mm$

$Torque\ allowed\ for\ the\ spline = \dfrac{\pi \times d_p \times d_s \times L \times \tau}{16}$

$Allowable\ shaft\ torque = \dfrac{\pi \times d_s^3 \times \tau}{16}$

$Torque\ allowed\ for\ the\ spline = Allowable\ shaft\ torque$ -----------------------(1)

Using (1),

Spline length = L = 20.83 mm

$\underline{\underline{L \approx 21\ mm}}$

4.7Bearing Calculation

Input Shaft

Data:

Input shaft diameter = 25mm

Maximum rpm = 150rpm

Load on Upper bearing = 734.29N

Load on Bottom bearing =2066.75N

Assumption:

Working time = 6hrs per days 300 days per year

Total hours per 1.5 years (L_H) = 2700h

K = 3 for ball bearings

Single raw cylindrical roller bearings are selected.

The rating time(L) can be find by,

$$L = 60 \times N \times L_H$$

$$L = 60 \times 150 \times 2700 = 24.3 \times 10^6$$

Finding the dynamic load by (C),

$$L = \left(\frac{C}{W}\right)^K \times 10^6$$

$$C = W \times \left(\frac{L}{10^6}\right)^{1/K}$$

For upper bearing,

$$C = 734.29 \times \left(\frac{24.3 \times 10^6}{10^6}\right)^{1/3}$$

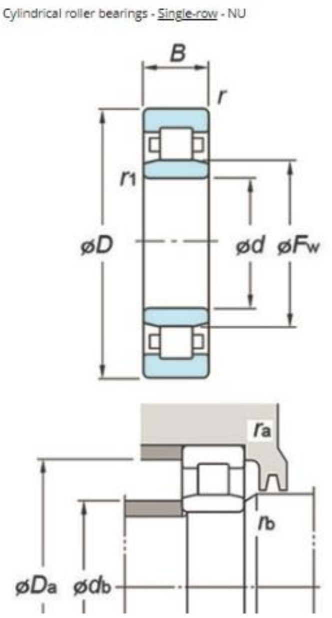
$$C = 2126.84N = 2.126\ kN$$

For bottom bearing,

$$C = 2066.75 \times \left(\frac{24.3 \times 10^6}{10^6}\right)^{1/3}$$

$$C = 5986.27N = 5.986kN$$

NU205R



Specifications (Boundary dimensions ...etc)

d	25 mm	Ew	- mm
D	52 mm	Basic load ratings : Cr	36.7 kN
B	15 mm	Basic load ratings : C0r	27.7 kN
r(min.)	1 mm	Fatigue load limit : Cu	3.75 kN
r1(min.)	0.6 mm	Limiting speeds(Grease lub.)	13000 min-1
Fw	31.5 mm	Limiting speeds(Oil lub.)	15000 min-1
		sd1	- mm
		B1	- mm
		B2	- mm

Figure 44

Selected Bearing

Bearing Types	Cylindrical roller bearings Single-row NU
Bearing No.	NU205R Products detail
Boundary dimensions	25 x 52 x 15 mm
Dynamic load rating	36.7 kN
Static load rating	27.7 kN

Dynamic loads are satisfying we can take NU205R single raw cylindrical roller bearing that used for input shaft.

Lay Shaft

Data:

Lay shaft diameter = 25mm

Maximum rpm = 150rpm

Load on Upper bearing = 1585.96N

Load on Bottom bearing = 3691.69N

Assumption:

Working time = 6hrs per days 300 days per year

Total hours per 1.5 years (L_H) = 2700h

K = 3 for ball bearings

Single raw cylindrical roller bearings are selected.

The rating time(L) can be find by,

$$L = 60 \times N \times L_H$$

$$L = 60 \times 150 \times 2700 = 24.3 \times 10^6$$

Finding the dynamic load by (C),

$$L = \left(\frac{C}{W}\right)^K \times 10^6$$

$$C = W \times \left(\frac{L}{10^6}\right)^{1/K}$$

For upper bearing,

$$C = 1585.96 \times \left(\frac{24.3 \times 10^6}{10^6}\right)^{1/3}$$

$$C = 4593.682N = 4.593\ kN$$

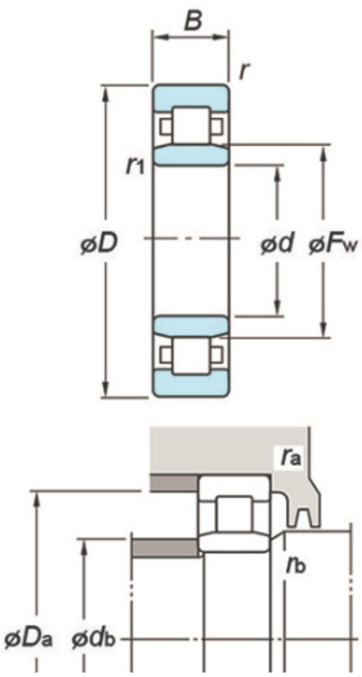
For bottom bearing,

$$C = 3691.69 \times \left(\frac{24.3 \times 10^6}{10^6}\right)^{1/3}$$

$$C = 10692.86N = 10.692kN$$

NU1005

Cylindrical roller bearings - Single-row - NU



Specifications (Boundary dimensions ...etc)

d	25 mm
D	47 mm
B	12 mm
r(min.)	0.6 mm
r1(min.)	0.3 mm
Fw	30.5 mm

Ew	- mm
Basic load ratings : Cr	17.8 kN
Basic load ratings : C0r	13.1 kN
Fatigue load limit : Cu	2.25 kN
Limiting speeds(Grease lub.)	15000 min-1
Limiting speeds(Oil lub.)	18000 min-1
sd1	- mm
B1	- mm
B2	- mm

Figure 46

Selected Bearing

Bearing Types	Cylindrical roller bearings Single-row NU
Bearing No.	NU1005 Products detail
Boundary dimensions	25 x 47 x 12 mm
Dynamic load rating	17.8 kN
Static load rating	13.1 kN

Figure 47

Dynamic loads are satisfying by above bearing.

∴ we can take NU1005 single raw cylindrical roller bearing that used for input shaft.

Output Shaft

Data:

Output shaft diameter = 25mm

Maximum rpm = 143.2rpm

Load on bearing = 2905.33N

Assumption:

Working time = 6hrs per days 300 days per year

Total hours per 1.5 years (L_H) = 2700h

K = 3 for ball bearings

Single raw cylindrical roller bearings are selected.

The rating time(L) can be find by,

$$L = 60 \times N \times L_H$$

$$L = 60 \times 143.2 \times 2700 = 23.1984 \times 10^6$$

Finding the dynamic load by (C),

$$L = \left(\frac{C}{W}\right)^K \times 10^6$$

$$C = W \times \left(\frac{L}{10^6}\right)^{1/K}$$

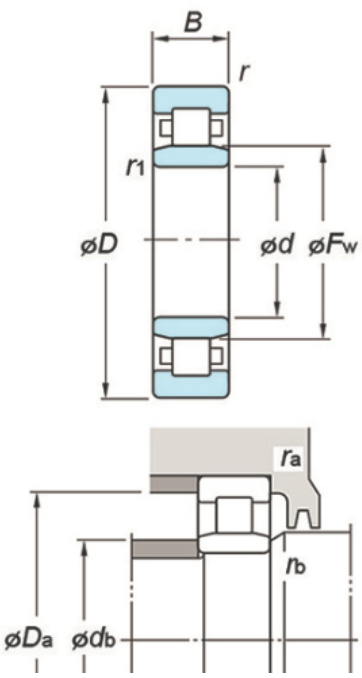
For bearing,

$$C = 2905.33 \times \left(\frac{23.1984 \times 10^6}{10^6}\right)^{1/3}$$

$$C = 8286.061N = 8.286kN$$

NU1005

Cylindrical roller bearings - Single-row - NU



Specifications
(Boundary dimensions ...etc)

d	25 mm	Ew	- mm
D	47 mm	Basic load ratings : Cr	17.8 kN
B	12 mm	Basic load ratings : C0r	13.1 kN
r(min.)	0.6 mm	Fatigue load limit : Cu	2.25 kN
r1(min.)	0.3 mm	Limiting speeds(Grease lub.)	15000 min-1
Fw	30.5 mm	Limiting speeds(Oil lub.)	18000 min-1
		sd1	- mm
		B1	- mm
		B2	- mm

Figure 48

Selected Bearing

Bearing Types	Cylindrical roller bearings Single-row NU
Bearing No.	NU1005 Products detail
Boundary dimensions	25 x 47 x 12 mm
Dynamic load rating	17.8 kN
Static load rating	13.1 kN

Figure 49

Dynamic loads are satisfying by above bearing.

∴ we can take NU1005 single raw cylindrical roller bearing that used for input shaft.

Idler Shaft

Data:

Idler shaft diameter = 25mm

Maximum rpm = 150rpm

Load on Upper bearing = 5280.47N

Assumption:

Working time = 6hrs per days 300 days per year

Total hours per 1.5 years (L_H) = 2700h

K = 3 for ball bearings

Single raw cylindrical roller bearings are selected.

The rating time(L) can be find by,

$$L = 60 \times N \times L_H$$

$$L = 60 \times 150 \times 2700 = 24.3 \times 10^6$$

Finding the dynamic load by (C),

$$L = \left(\frac{C}{W}\right)^K \times 10^6$$

$$C = W \times \left(\frac{L}{10^6}\right)^{1/K}$$

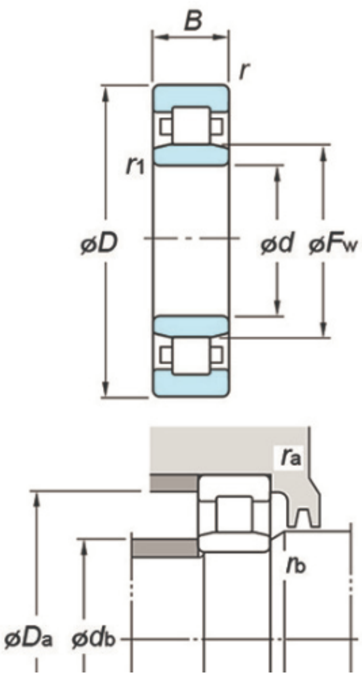
For bearing,

$$C = 5280.47 \times \left(\frac{24.3 \times 10^6}{10^6}\right)^{1/3}$$

$$C = 15294.72N = 15.294kN$$

NU1005

Cylindrical roller bearings - Single-row - NU



Specifications (Boundary dimensions ...etc)

d	25 mm
D	47 mm
B	12 mm
r(min.)	0.6 mm
r1(min.)	0.3 mm
Fw	30.5 mm

Ew	- mm
Basic load ratings : Cr	17.8 kN
Basic load ratings : C0r	13.1 kN
Fatigue load limit : Cu	2.25 kN
Limiting speeds(Grease lub.)	15000 min-1
Limiting speeds(Oil lub.)	18000 min-1
sd1	- mm
B1	- mm
B2	- mm

Figure 50

Selected Bearing

Bearing Types	Cylindrical roller bearings Single-row NU
Bearing No.	NU1005 Products detail
Boundary dimensions	25 x 47 x 12 mm
Dynamic load rating	17.8 kN
Static load rating	13.1 kN

Figure 51

Dynamic loads are satisfying by above bearing.

∴ we can take NU1005 single raw cylindrical roller bearing that used for input shaft.

4.8 Key Calculation

Input shaft

4.1.1 Key of pulley that connected to input shaft

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Input pulley width = 45.2 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = $d/_4$ = $25/_4$ = 6.25 mm

$$\therefore w \text{ and } t \text{ are not sufficent for the sequence}$$

Shaft diameter (mm) upto and including	*Key cross-section*		*Shaft diameter (mm) upto and including*	*Key cross-section*	
	Width (mm)	*Thickness (mm)*		*Width (mm)*	*Thickness (mm)*
6	2	2	85	25	14
8	3	3	95	28	16
10	4	4	110	32	18
12	5	5	130	36	20
17	6	6	150	40	22
22	8	7	170	45	25
30	10	8	200	50	28
38	12	8	230	56	32
44	14	9	260	63	32
50	16	10	290	70	36
58	18	11	330	80	40
65	20	12	380	90	45
75	22	14	440	100	50

Figure 52: proportions of standard parallel, tapered angd gib head keys

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T = l \times w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = 0.592mm$$

For crushing,

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = 0.854\ mm$$

Input pulley width 45.2mm >>> 0.854mm

∴Let’s take the key length as 30mm.

Key for input gear

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Input gear width = 20 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = ${}^{d}/_{4}$ = ${}^{25}/_{4}$ = 6.25 mm

Thickness (t) = $^{d}/_{6}$ = $^{25}/_{6}$ = 4.16 mm

$$\therefore w \text{ and } t \text{ are not sufficent for the sequence}$$

Shaft diameter (mm) upto and including	Key cross-section		Shaft diameter (mm) upto and including	Key cross-section	
	Width (mm)	Thickness (mm)		Width (mm)	Thickness (mm)
6	2	2	85	25	14
8	3	3	95	28	16
10	4	4	110	32	18
12	5	5	130	36	20
17	6	6	150	40	22
22	8	7	170	45	25
30	10	8	200	50	28
38	12	8	230	56	32
44	14	9	260	63	32
50	16	10	290	70	36
58	18	11	330	80	40
65	20	12	380	90	45
75	22	14	440	100	50

Figure 53: proportions of standard parallel, tapered angd gib head keys

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T = l \times w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = 0.592mm$$

For crushing,

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = 0.854\, mm$$

Input gear width 20mm >>> 0.854mm

∴Let’s take the key length as 20mm.

Lay shaft

Key for First gear

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

1^{st} gear width = 30 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = $d/_4$ = $25/_4$ = 6.25 mm

Thickness (t) = $d/_6$ = $25/_6$ = 4.16 mm

$$\therefore w \text{ and } t \text{ are not sufficent for the sequence}$$

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T \; = l \times \; w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = \; 0.592mm$$

For crushing,

$$T \; = l \times \; \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = \; 0.854 \; mm$$

1^{st} gear width 30mm >>> 0.854mm

∴Let's take the key length as 20mm.

Key for Second gear

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

2^{nd} gear width = 20 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = $d/4$ = $25/4$ = 6.25 mm

Thickness (t) = $d/6$ = $25/6$ = 4.16 mm

$$\therefore w \ and \ t \ are \ not \ sufficent \ for \ the \ sequence$$

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T = l \times w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = 0.592mm$$

For crushing,

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = 0.854\ mm$$

2nd gear width 20mm >>> 0.854mm

∴Let's take the key length as 12mm.

Key for Third gear

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

3rd gear width = 20 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = $d/_4$ = $25/_4$ = 6.25 mm

Thickness (t) = $d/_6$ = $25/_6$ = 4.16 mm

$$\therefore w\ and\ t\ are\ not\ sufficent\ for\ the\ sequence$$

Shaft diameter (mm) upto and including	Key cross-section		Shaft diameter (mm) upto and including	Key cross-section	
	Width (mm)	Thickness (mm)		Width (mm)	Thickness (mm)
6	2	2	85	25	14
8	3	3	95	28	16
10	4	4	110	32	18
12	5	5	130	36	20
17	6	6	150	40	22
22	8	7	170	45	25
30	10	8	200	50	28
38	12	8	230	56	32
44	14	9	260	63	32
50	16	10	290	70	36
58	18	11	330	80	40
65	20	12	380	90	45
75	22	14	440	100	50

Figure 54: proportions of standard parallel, tapered angd gib head keys

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T = l \times w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = 0.592mm$$

For crushing,

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = 0.854\, mm$$

3rd gear width 20mm >>> 0.854mm

∴Let's take the key length as 12mm.

Key for Reverse gear

Data:

Torque = 31.83N

Shaft diameter (d) = 25mm

Reverse gear width = 40 mm

Assumption:

Key material AISI 4340 Alloy steel

Shear stress (τ_{max}) = 430×10^6 MPa

Crushing stress = 745×10^6 MPa

Using a rectangular tapered sunk key for the sequence.

For rectangular sunk keys,

Width (w) = $^{d}/_{4}$ = $^{25}/_{4}$ = 6.25 mm

Thickness (t) = $^{d}/_{6}$ = $^{25}/_{6}$ = 4.16 mm

$$\therefore w \text{ } and \text{ } t \text{ } are \text{ } not \text{ } sufficent \text{ } for \text{ } the \text{ } sequence$$

By above table,

Lets' take w = 10mm & t = 8mm (Key of shaft diameter 30mm)

For shearing,

$$T = l \times w \times \tau_{max} \times \frac{d}{2}$$

$$l = \frac{T}{w \times \tau_{max} \times \frac{d}{2}}$$

$$l = \frac{31.83}{10 \times 430 \times \frac{25}{2}}$$

$$l = 0.592 mm$$

For crushing,

$$T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$l = \frac{T}{\frac{t}{2} \times \sigma_c \times \frac{d}{2}}$$

$$l = \frac{31.83}{\frac{8}{2} \times 745 \times \frac{25}{2}}$$

$$l = 0.854\, mm$$

Reverse gear width 40mm >>> 0.854mm

∴Let's take the key length as 20mm.

4.9Lubrication method

Selecting a lubrication method

In use to 30^0 C,1 atm conditions in this application.

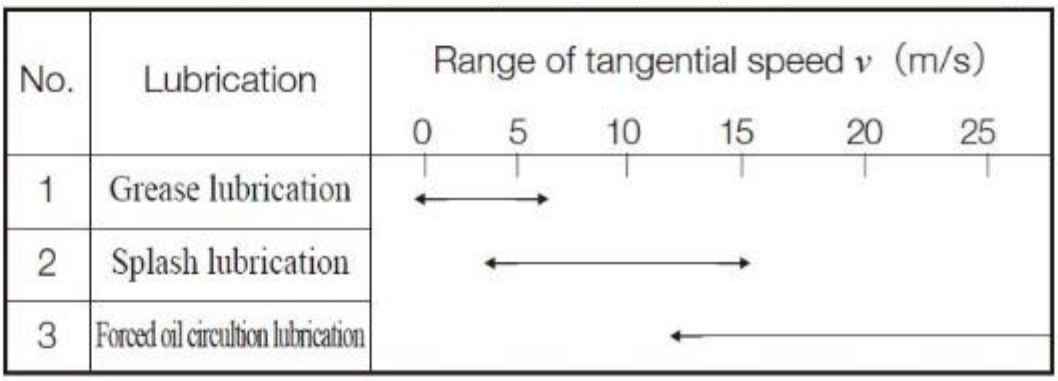
Generally, there are three lubrication method in use

- Oil bath method
- Force oil circulation
- Grease lubrication

Maximum tangential speed can be get by the 3rd gear of the output shaft,

$$V = \frac{d}{2} \times \frac{2\pi N}{60}$$

$$V = \frac{122}{2} \times \frac{2\pi \times 143.2}{60} = 0.91474\, m/s$$



Lubricantion method range table

By above chart Grease lubrication is prefer for the system.

Lubrication oil selection

For spur, helical and beveled enclosed gears
For oils with VI=90 (recommendations are empirical)

Temp °C	Pitch line velocity, m/s²							
	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

Refer toASTM 9005-E02 for charts corresponding to oils with higher VIs and for worm gear viscosity selection (typically ISO VG 200 to 680 depending on temperature and speed).

Notes:

1. (yellow) Consult gear, bearing and lubricant suppliers if a viscosity grade of less than 32 or greater than 3200 is indicated.
   (blue) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
   (white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
   (red) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.
2. This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval.
3. Determine pitch line velocity of all gear sets.Select viscosity grade for critical gear set taking

Viscosity grade table

Type			Usage
For Industrial Usage	1	ISO VG 32 ISO VG 46 ISO VG 68 ISO VG 100 ISO VG 150 ISO VG 220 ISO VG 320 ISO VG 460	Mostly used for lightly loaded enclosed-gears in general type of machines.
	2	ISO VG 68 ISO VG 100 ISO VG 150 ISO VG 220 ISO VG 320 ISO VG 460 ISO VG 680	Mostly used for middle or heavily loaded enclosed-gears in general types of machines or rolling machines.

Types of Gear Oils and the Usage

According to above tables because of the pitchline velocity is 1 -2 m/s range the lubricant that choose can be used for bulk stabilization. Also, because of light weight and light load we can choose Type 1 category.

Rotation of Pinion (rpm)	Horsepower (PS)	Reduction Ratio below 10 cSt (40℃)	Reduction Ratio below 10 ISO Viscosity Grade	Reduction Ratio over 10 cSt (40℃)	Reduction Ratio over 10 ISO Viscosity Grade
Below 300	Less than 30	5 - 234	150, 220	180 - 279	220
	30 - 100	180 - 279	220	216 - 360	220,320
	More than 100	279 - 378	320	360 - 522	460
300 ~ 1,000	Less than 20	81 - 153	100,150	117 - 198	150
	20 - 75	117 - 198	150	180 - 279	220
	More than 75	180 - 279	220	279 - 378	320
1,000 ~ 2,000	Less than 10	54 - 117	68,100	59 - 153	68,100,150
	10 - 50	59 - 153	68,100,150	135 - 198	150
	More than 50	135 - 198	150	189 - 342	220,320
2,000 ~ 5,000	Less than 5	27 - 36	32	41 - 63	46
	5 - 20	41 - 63	46	59 - 144	68,100
	More than 20	59 - 144	68,100	95 - 153	100,150
More than 5000	Less than1	9 - 31	10,15,22	18 - 32	22,32
	1 - 10	18 - 32	22,32	29 - 63	32,46
	Less than10	29 - 63	32,46	41 - 63	46

Types of Gear Oils and the Usage

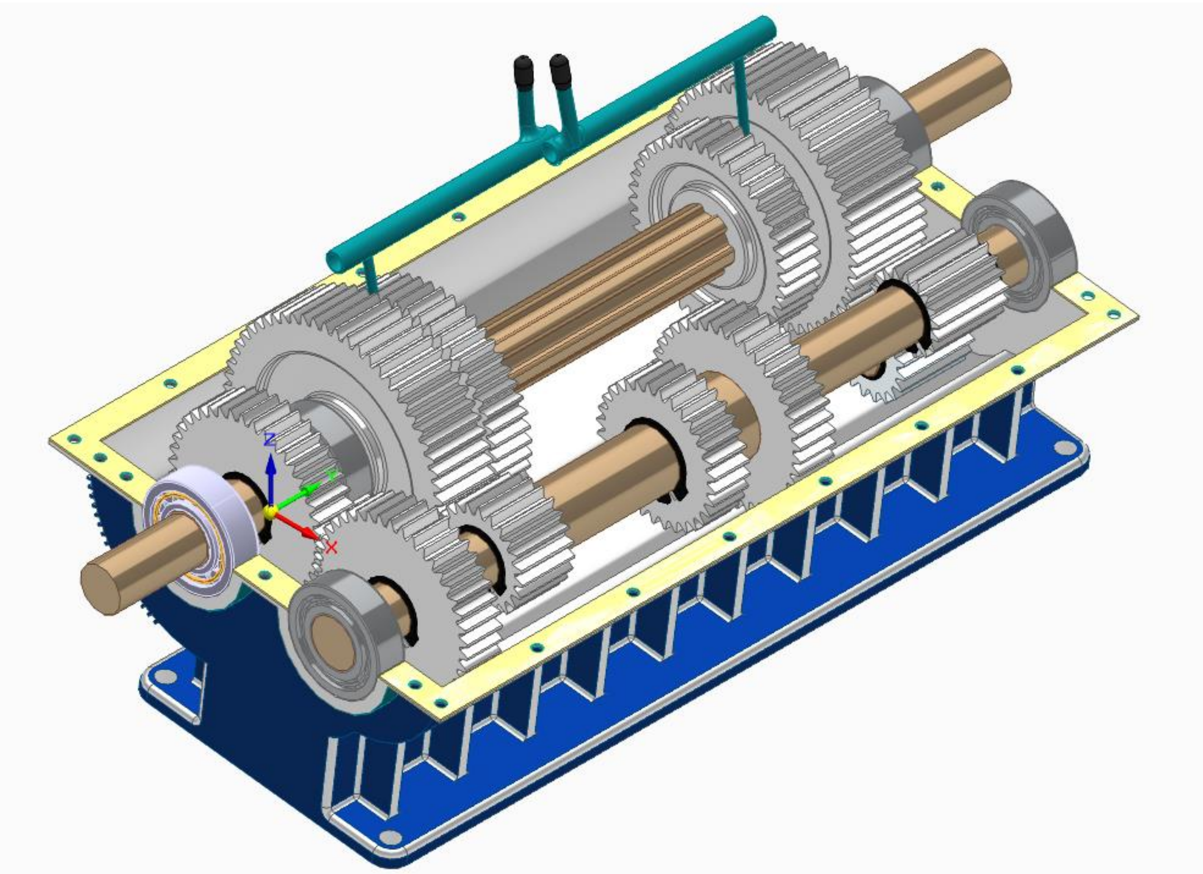
Power of the machine is 500W = 0.671hp.

Pinion velocity = 150rpm

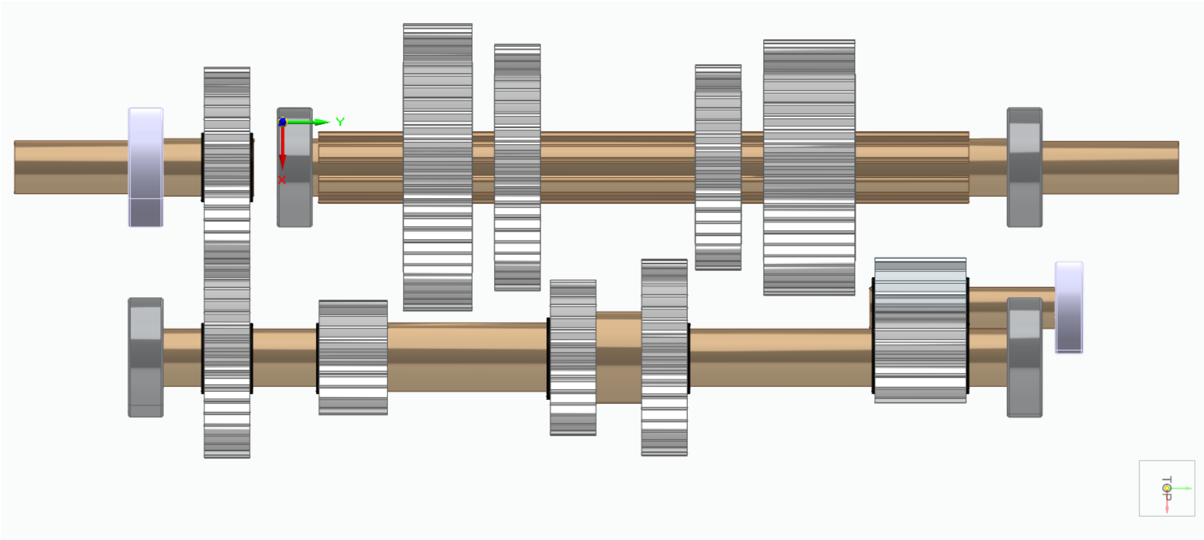
By above table as the lubrication should be used ISO VG 220 gear oil.

5 Final Design

Three forward gears and one reverse gear are found in the gearbox designed for the sledgehammer. One of the simplest gearbox designs to construct is the sliding mesh gearbox with spur gears used in this design. Eleven gear wheels, shafts, bearings, keys, a casing, and a number of other parts make up the gearbox, which transfers power from the main mover. gears with varying tooth counts and shaft configurations arranged in an organised manner. The gears are meshing when the output gears are moved right or left down a splined output shaft using an operator-operated gear lever.



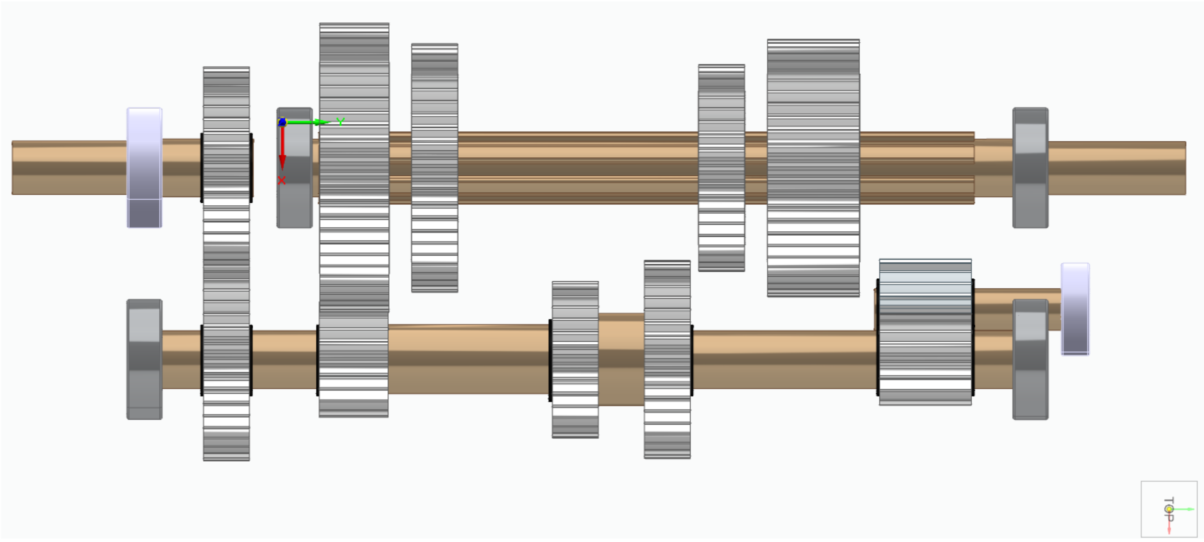
Neutral Position



Neutral position

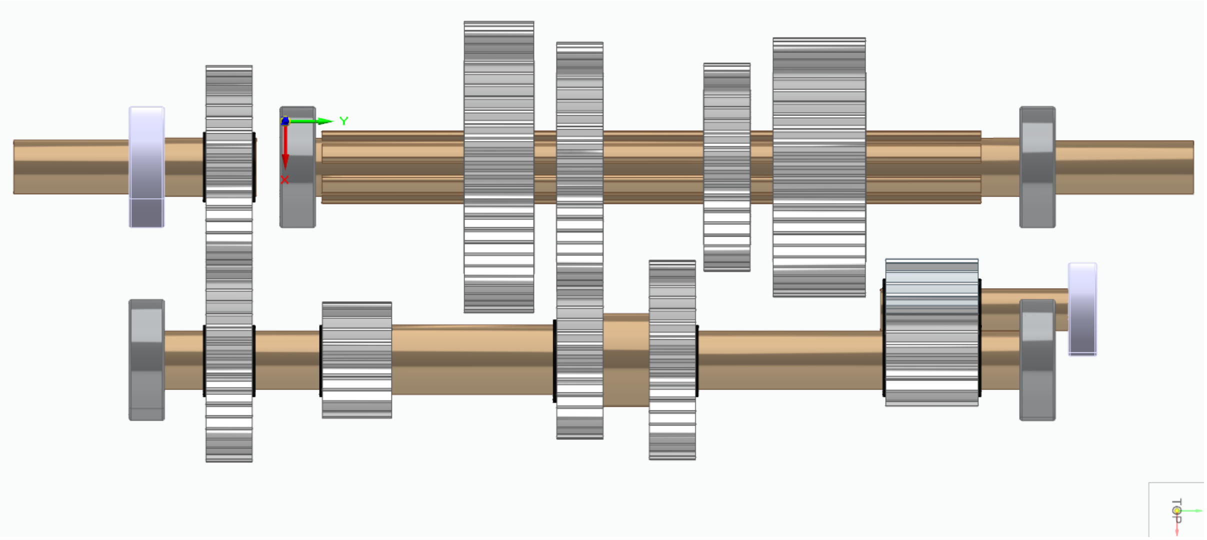
Initially, the gearbox is set to the neutral position. The input shaft pinion is meshing with the lay shaft input gear, and the lay shaft reverse pinion is meshing with the idler gear. Throughout the gearbox's operation, these gears will be meshed. The lay shaft and output shaft are separated. The sledgehammer does not receive power as a result. By applying a shifting fork to move the output shaft's gears from the neutral position to the left or right, different speeds can be achieved.

First gear

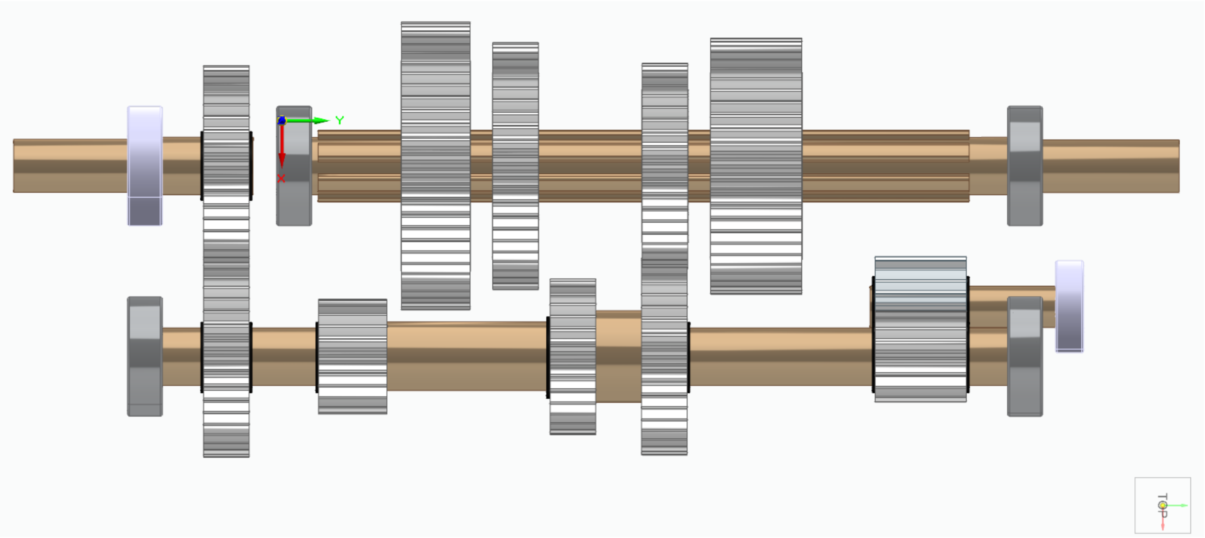


First gear

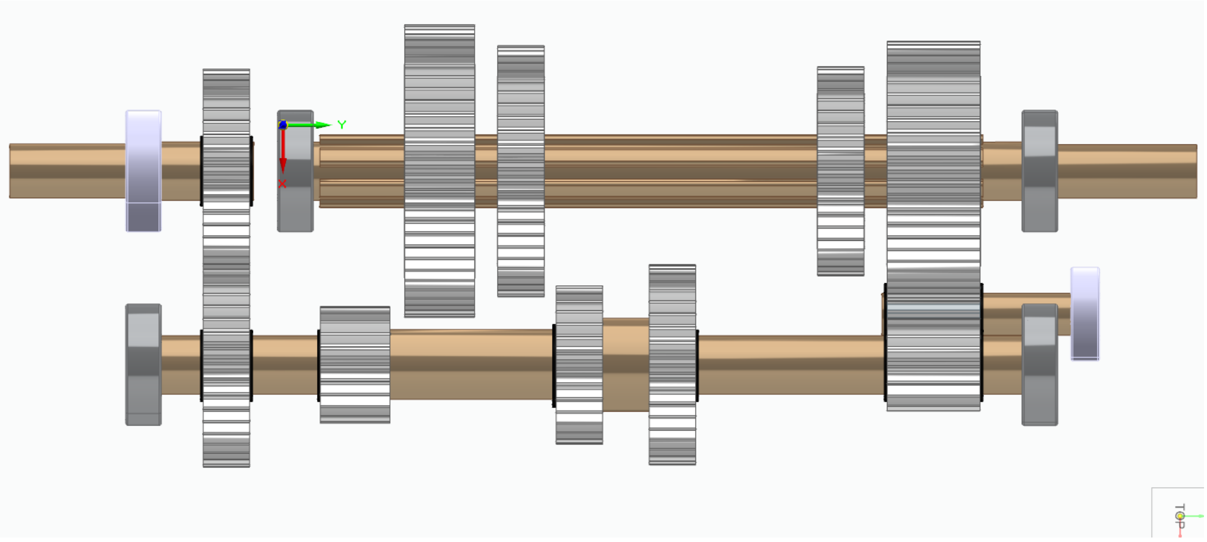
Second gear



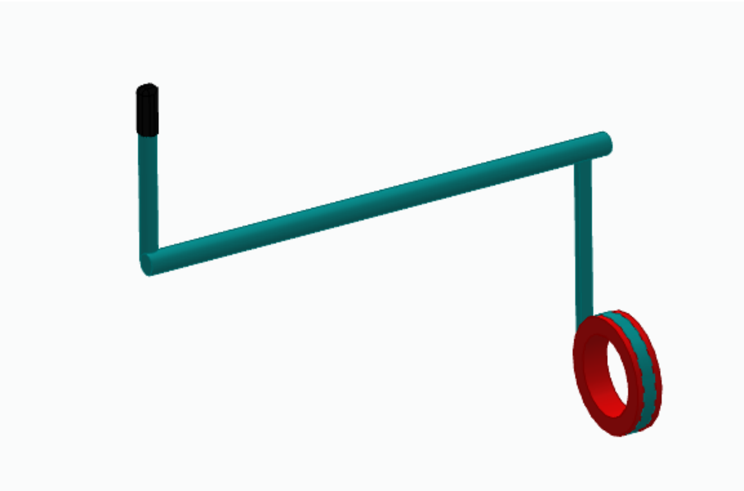
Third gear



Reverse gear



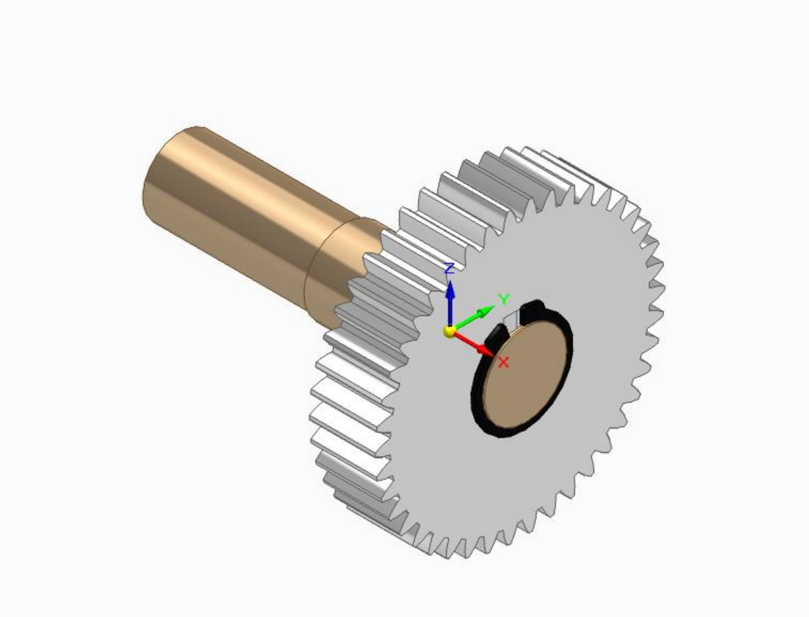
Shifting Mechanism



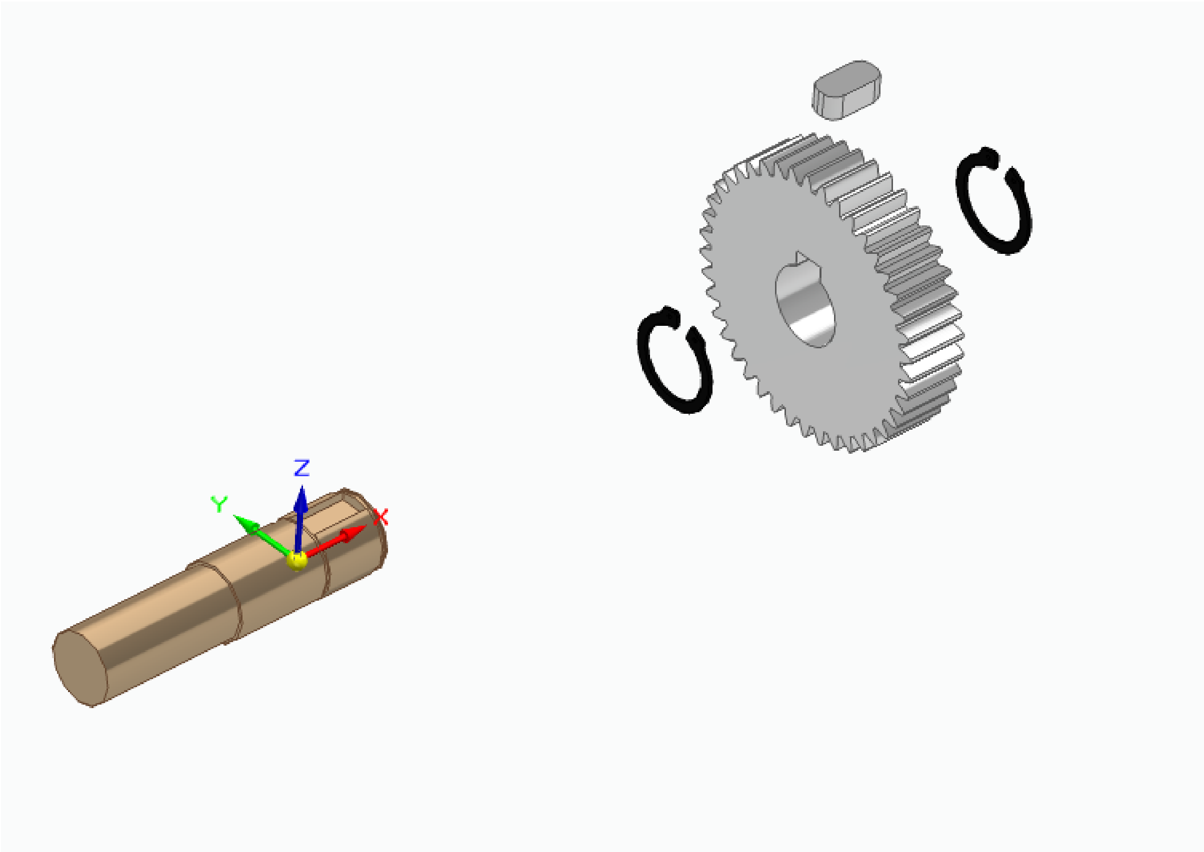
The purpose of the shifting mechanism is to change the gear in the output shaft, which can change the sledgehammer's speed as power is transferred from the prime mover to the wheels. The two gear connectors in the output shaft are linked to the shifting forks. The gears in the output shaft are connected by means of gear connectors. The ger connector will allow those two gears to move together as an assembly. These two components will be moved via the output shaft's splines by a shifting mechanism.

Gearbox Component

Input shaft

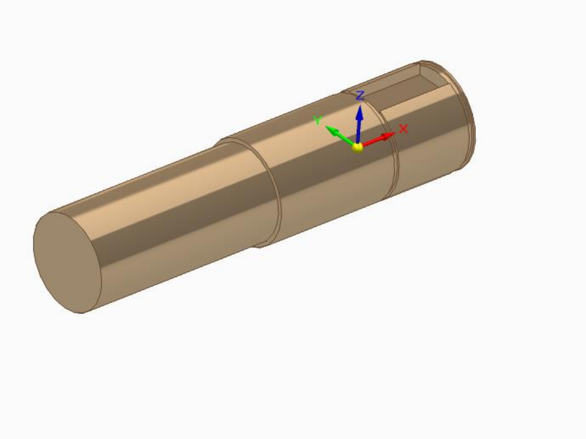


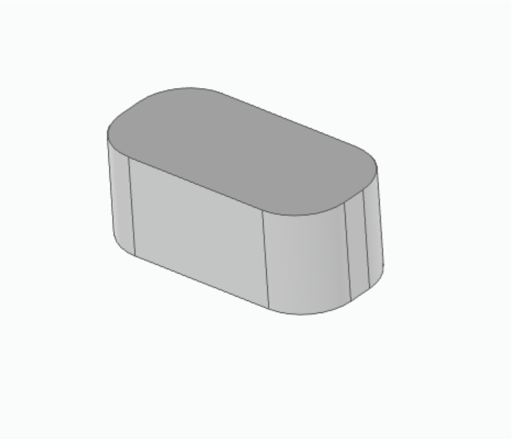
Input shaft



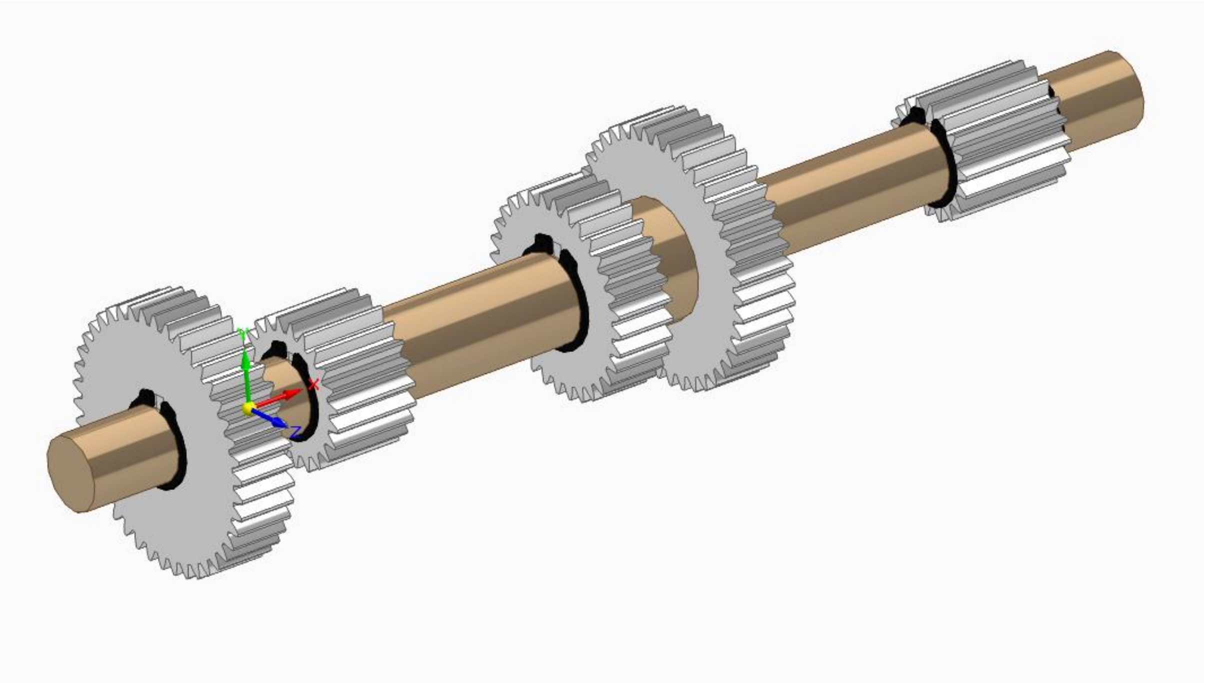
Exploded view of input shaft

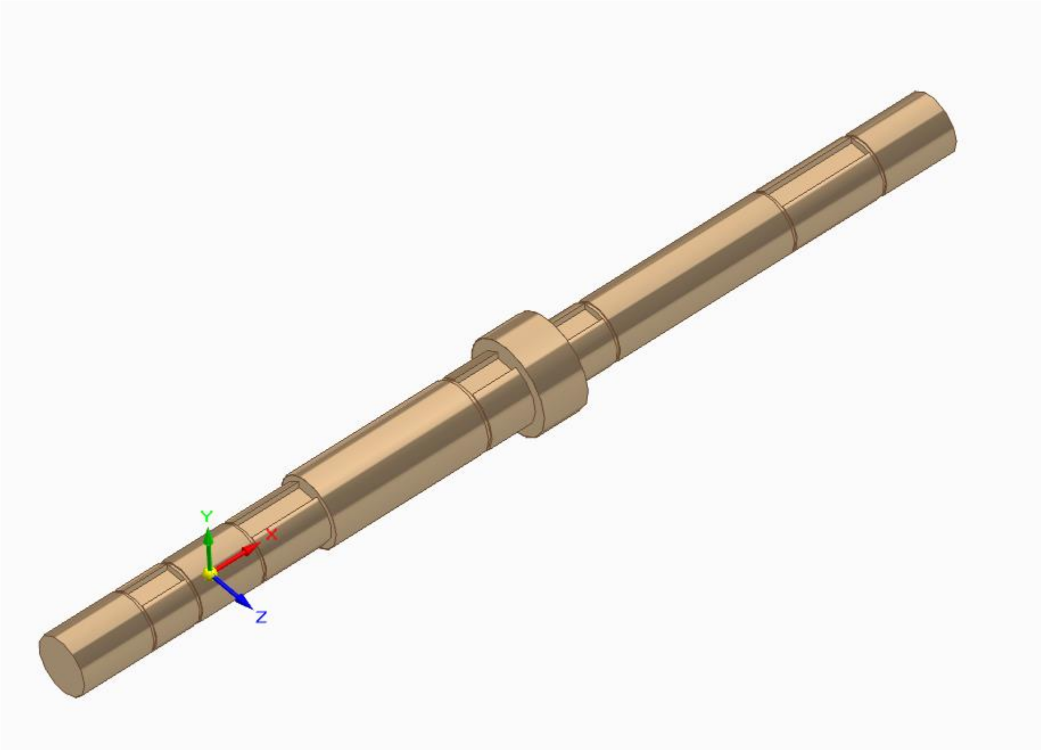
The prime mover is attached to the input shaft. The gear is connected to the shaft by a key. The shaft is fixed to the case using bearings.


Input shaft

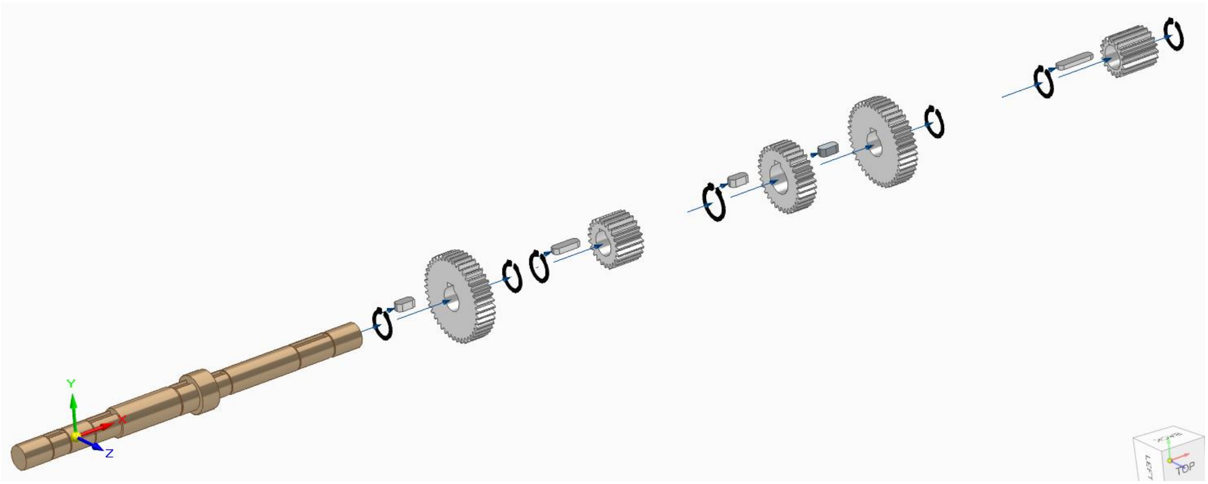

key of input shaft’s gear

Lay shaft


Lay shaft assembly



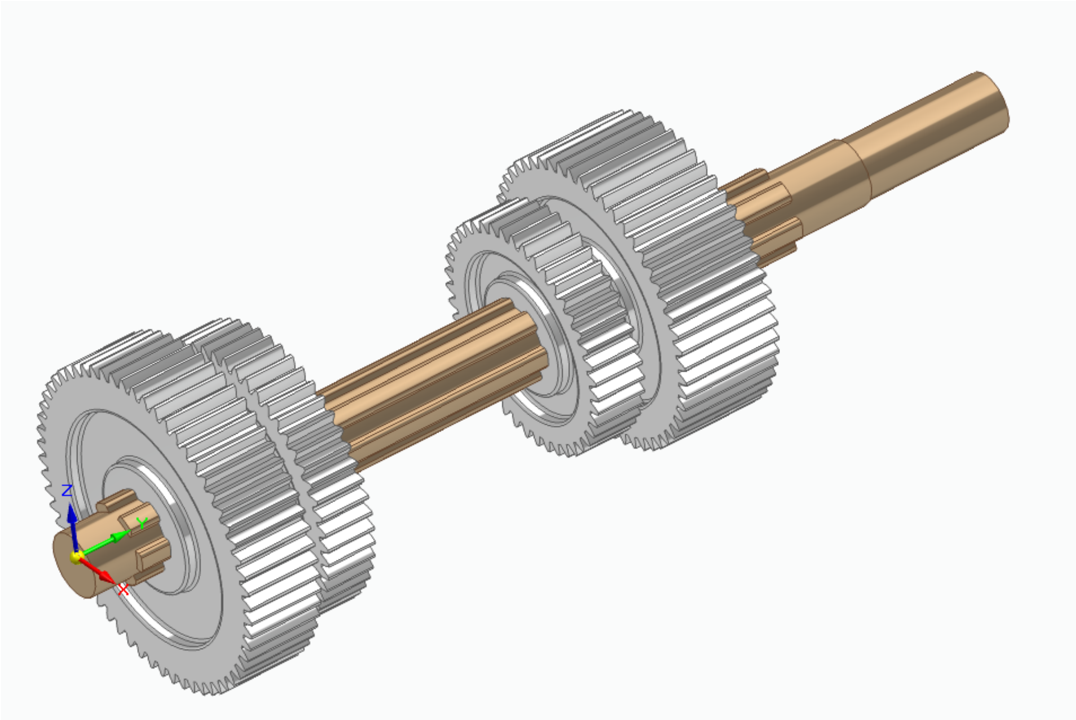
Lay shaft



Exploded View of Lay Shaft Assembly

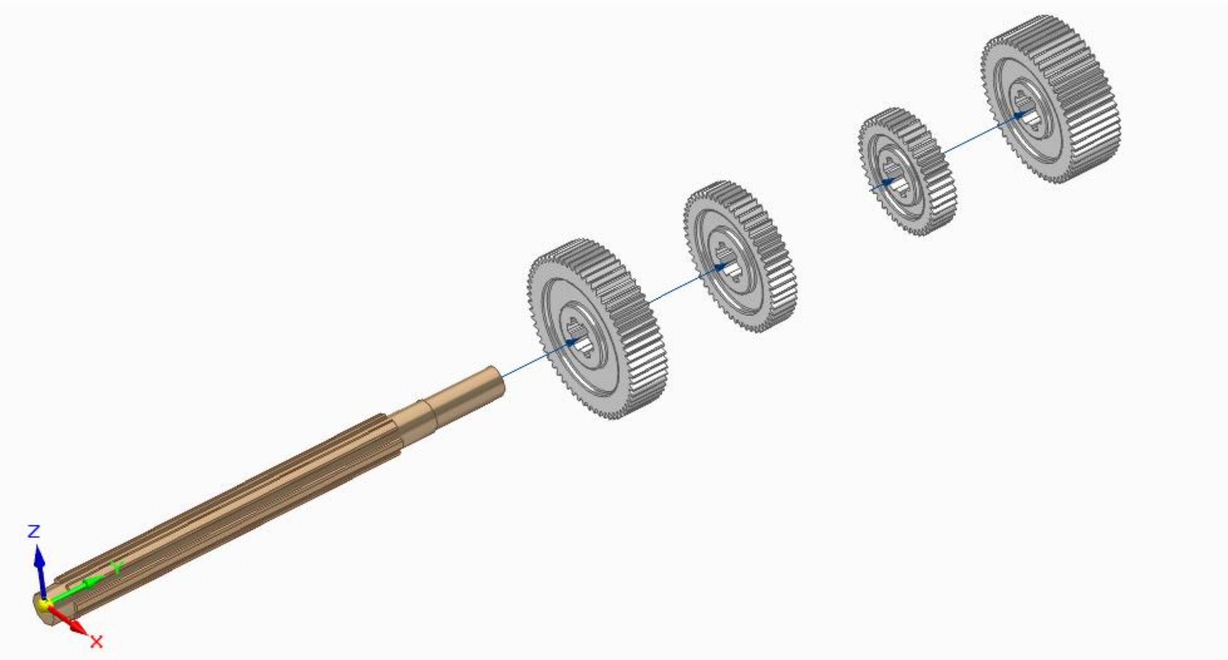
Considering the Lay shaft, its' gears that mounted on its' outer surface are rotating with a constant speed. The power of rotation has been transmitted directly to lay shaft by input shaft with meshing two gears on each other. This shaft serves as an intermediary shaft (between the input shaft and main shaft) that allows the main shaft's gears to mesh so that the proper output is transmitted to the final drive. The gear wheels are fastened to the shaft using keys. To stop the linear motion of the gears circlips has been used for each side with cutting a small grove. On the other half of each gear the diameter has been increased in advanced of stopping the linear motion.

Output shaft



Output shaft assembly

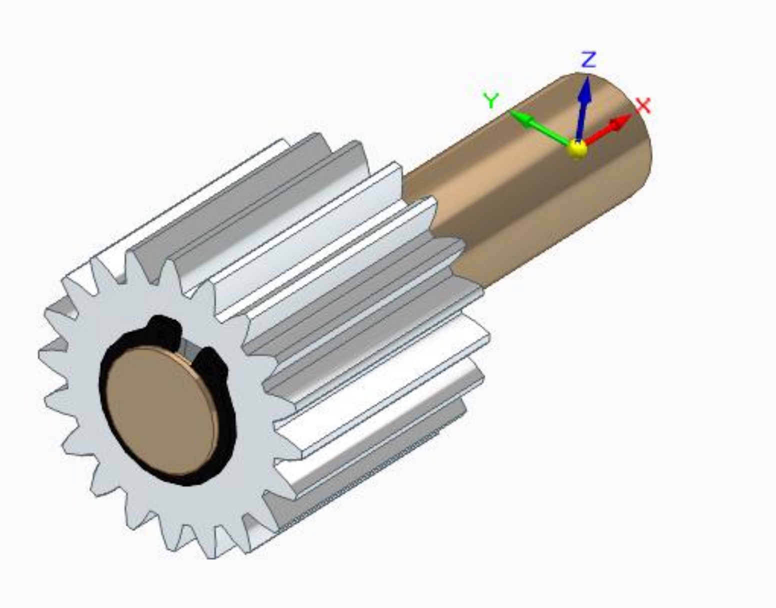
Output shaft



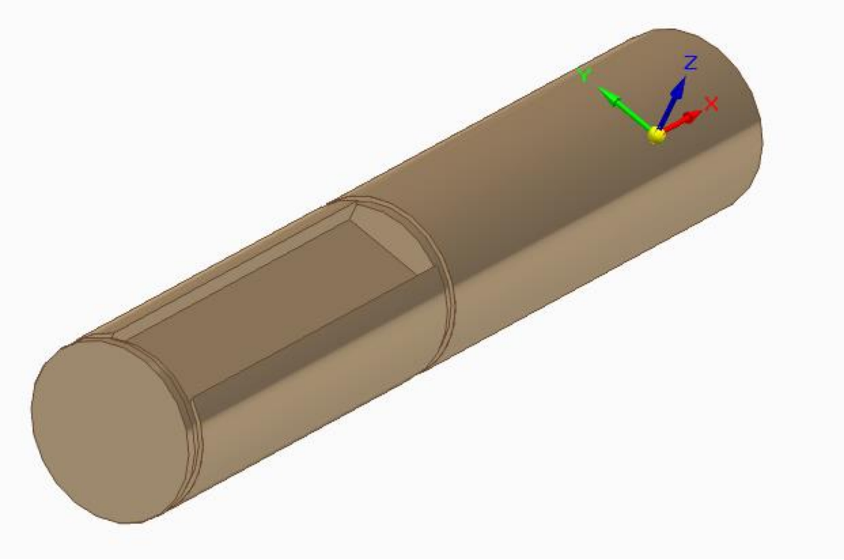
Exploded view of output shaft

The shaft of a gearbox that attaches to the driven or load component is called the output shaft. It connects to the lay shaft and transfers power to the sledgehammer. Compared to the input shaft, which is attached to the prime mover, it rotates at a different speed. The number of output shafts in a gearbox varies based on its design and intended purpose. There is only one output shaft in this design. The output shaft of a sliding mesh gearbox consists of a splined shaft because the gears on the shaft required to move left and right to change gears. Thus, to move the gear along the output shaft, splines are utilized.

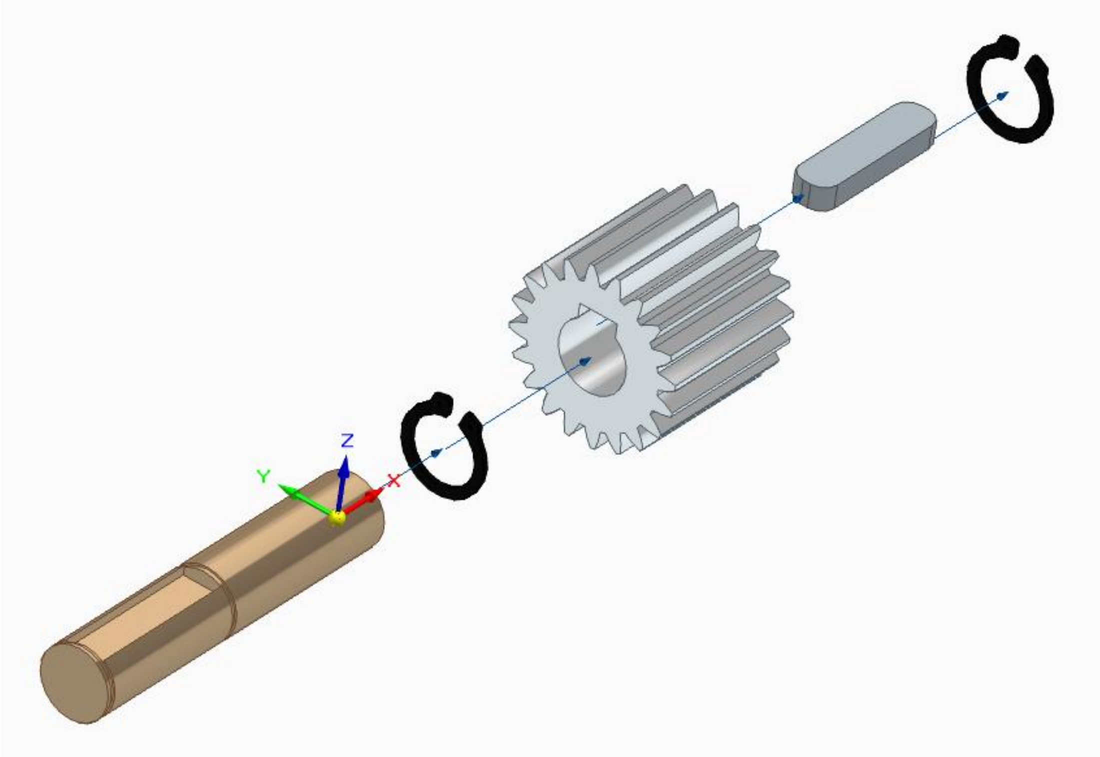
Idler shaft



Idler shaft assembly



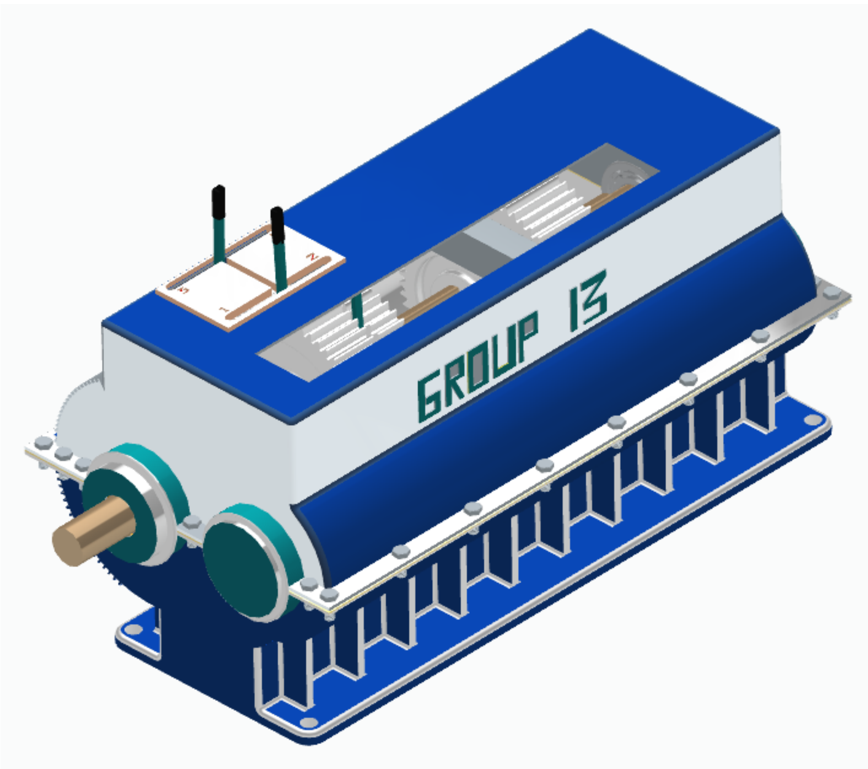
Idler shaft

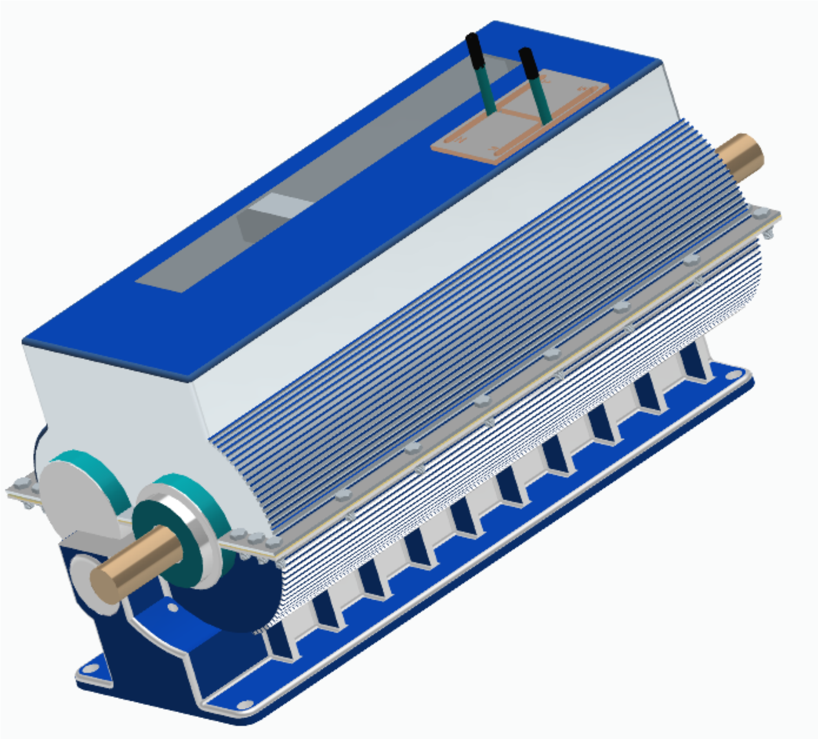


Exploded view of idler shaft

An idler shaft is a gearbox component that acts as a connection that allows the transfer of power between other gears in the gearbox. It is used to reverse the direction of rotation between the gears; it does not transmit power on its own.

Casing





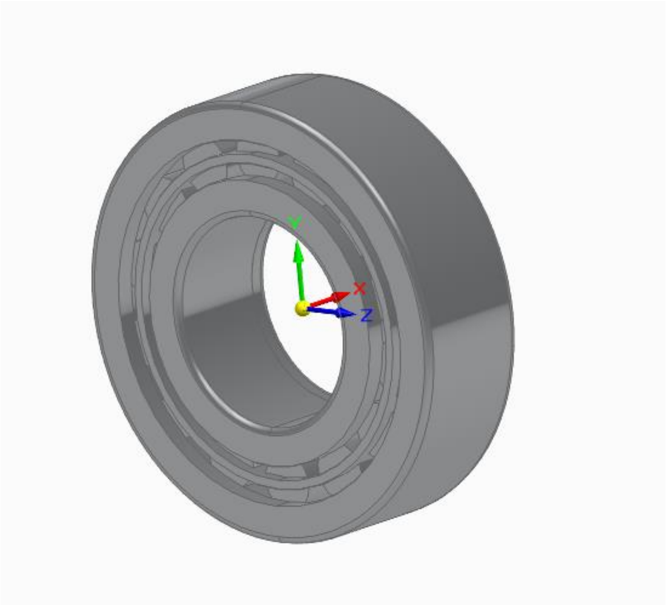
Sketch of casing

The gearbox casing is the component that protects and covers the gearbox's internal components. It frequently consists of metal to withstand the high stresses produced by the shaft's gears. The main purposes of a gearbox's casing are to contain any lubricants or other fluids that are used in the gearbox and to shield its internal components from outside elements like dust, debris, and other impurities. The casing may be useful in dissipating heat produced during operation in addition to supporting and aligning the internal components. In order to maintain the proper alignment and operation of the internal components, the gearbox's casing also contributes to its overall mechanical strength and helps to maintain its shape and dimensions. To reduce heat that generate in the power transmission process casing has been upgraded with fins in the upper body of the housing. Also, to increase the strength of the gear box, housing included ribs in the lower body.

Bearing

A gearbox's rotating shaft and other moving parts are supported and guided by bearings. Usually, bearings are made to minimize wear and friction between moving parts while simultaneously offering alignment and support. Within a gearbox, a variety of bearing types can be employed, such as needle, sleeve, ball, and roller bearings. The particular application and gearbox specifications will determine the type of bearing to be used. Roller bearings are used in this design.

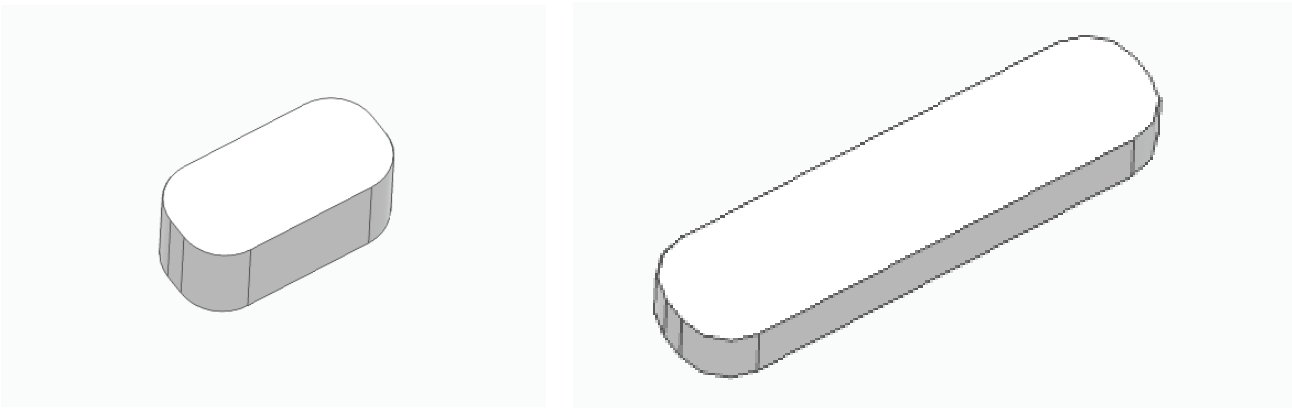
In a gearbox, bearings serve two primary purposes: they lower friction between moving parts and support the rotation of the shafts and gears. This contributes to the gearbox's increased longevity and efficiency. In order to ensure proper gear engagement and operation, bearings also aid in maintaining the proper alignment of the gears and shafts.



Keys

In a gearbox, keys serve two main purposes: they maintain the proper alignment of gears and other components and transfer torque and power from the input shaft to the output shaft. By inserting the keys into the keyways and locking the gears and shafts in place, this is accomplished. This guarantees that the gears mesh correctly and that the gearbox runs correctly and efficiently by preventing any undesired rotation or axial movement of the gear or other component.

This design creates use of seven keys. Five are located in the lay shaft gears, one in the idler shaft gear, and another in the input shaft pinion gear.



Sketches of keys

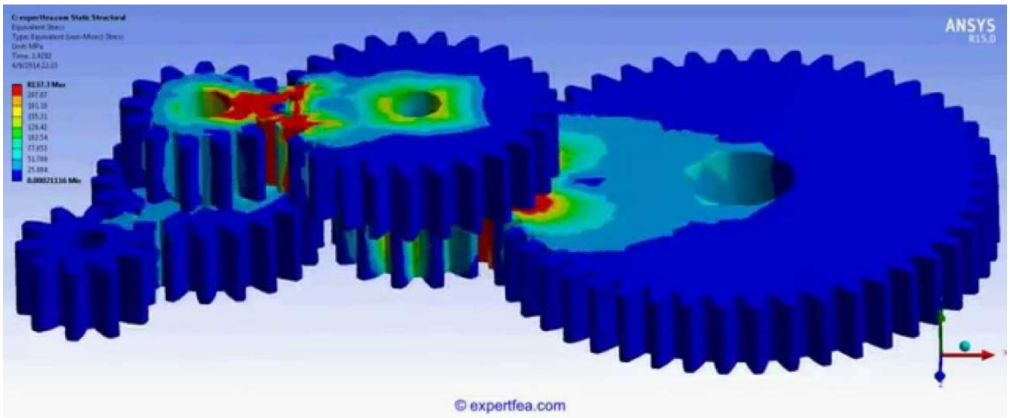
Oil seals

In gearboxes, oil seals are used to stop oil leaks from the gearbox and ensure that outside contaminants out of the gearbox. They are usually constructed of synthetic or rubber materials and are meant to fit around the gears' shafts to keep oil from escaping while still enabling the gears to rotate freely. An oil seal is typically composed of two parts: an inner flexible member that performs the sealing and is attached to the metal part using chemical adhesive agents, and an outer circular metal part.

6 Future work

6.1 Stress Analysis

Each equation was done in accordance with the applicable standards, and the relevant parts were designed in accordance with the standards as well. By performing a Finite Element Analysis (FEA), we can confirm that the previously determined values are accurate and think about additional design or material improvements that we can make to ensure the gearbox can withstand changes in power input up to a certain point. We can use the ANSYS software to finish this type of analysis, and it provides every necessary detail we need to check into before making any changes to the currently suggested design parameters.



FEA analysis for gears

6.2 Topology Optimization

We don't perform any steps to optimize a component that we design; instead, we only take into account the common modes of failure when designing the gearbox. Using mathematical modelling, we are able to optimise the components in topology optimization in accordance with specified parameters. Through appropriate topology optimization, we can,

• A lower ratio of stiffness to weight

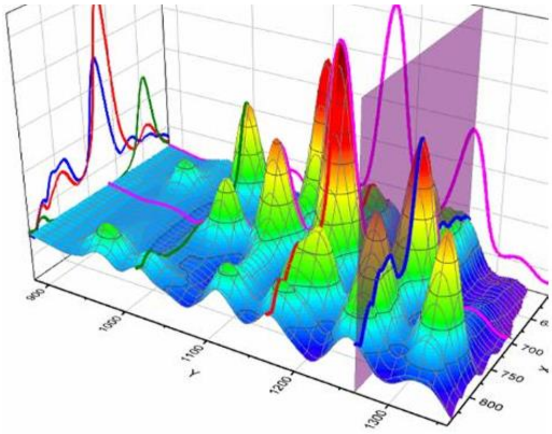
• An improved ratio of strain energy to weight

• A lower ratio of material volume to safety factor

• The ratio of natural frequency to weight.

6.3 Vibration Analysis

Every system experience vibrations while it is in operation, but these are not taken into account during the calculations. For this reason, it is advisable to perform a vibration analysis on the entire system before accepting the results. To do this, we can use the "DADiSP" program to identify the vibrations that each component experiences while it is in use.



Vibration analysis

7 References

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