Discussion Group: X

ME3200 Machine Design Project

Semester 5

Final Report

By

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Introduction

In this machine design project, we are developing a gear box for a industrial application to make the system perform better. In this project I selected a table saw as my industrial application, Table saw is widely used wood cutting tool in industry. It has a circular saw for cutting the wood. It is mostly driven by electric motor.

With a knowledge in design of machine elements we can calculate required bearing size, shaft design, gears, dog clutch and belt drives.

Problem Description

If you look table saw, electric power is directly transmitting energy through the sprocket and chain to the output shaft, such case speed of the cutting teeth is high, and torque is limited. There are various types of wood material is used in cutting operation, so not all the time we need same amount of torque so with reduction of speed we can increase torque so that we can vary its output torque hence we can use it improved performance and more efficiently.

In case of cutting wood with high hardness and have big diameter while cutting with high speed and lower torque, teeth may be stuck in it, so to avoid we can reduce speed and increase torque so that we can do the operation with better operation. In case of the cutting teeth is stuck within wood, then if we try to get back cutting teeth manually, those could cause damage to the cutting teeth hence replacement is costly and difficult job.

so, if we introduce a gear box into the table saw we can increase the torque so we can cut high hardness wood with better technique and with reduced speed of machine we can use it more efficiently. with reverse gear function we could operate the machine in opposite direction so that we can remove the stuck of cutting teeth without hurting the blade. It reduces our workload and saves money. We can get various amount of torque in the output shaft according to our need with help of varying gear teeth ratio.

Background study

The gearbox is a mechanical device used to increase the output torque or to change the speed (RPM) of a motor. The shaft of the motor is connected to one end of the gearbox and through the internal configuration of gears of a gearbox, provides a given output torque and speed determined by the gear ratio.

gearbox is that it is a contained gear train, or a mechanical unit or component consisting of a series of integrated gears within a housing.

The gears are mounted on shafts that are carried by roller bearings and rotate about them. The gearbox is a mechanical method of transferring energy from one device to another and is used to increase the torque while reducing the speed at the same time.

Gearbox consists of following main parts and that are,

- Clutch shaft
  A clutch shaft is a shaft that takes power from the engine to supply another shaft. The clutch shaft or driving shaft is connected through the clutch and when the clutch is engaged, the driving shaft also rotates.
- Counter shaft
  The counter shaft is a shaft that connects directly to clutch shaft. It can be run at engine speed or below engine speed according to gear ratio.

- Main shaft
  It rotates at different speeds and provides the necessary torque to the vehicle.
- Bearings
  Bearings are required to support the rotating part and reduce friction. The gear box has both a counter and main shaft which is supported by bearing.
- Gears
  Gears are used to transmitting the power from one shaft to another shaft.

Gear box types

There are two types of gear box

- Based on manual transmission
  1. Constant mesh gear box
     In this type of gearbox, all gears are constantly in mesh and dog clutches are used for engaging and disengaging the gears.

  2. Sliding mesh gear box

     It is the simplest type of gearbox. In this gearbox, spur gears are used. The Figure shows the construction of a sliding mesh type transmission having three forward and one reverse speeds.

- Based on automatic transmission
  1.Epicyclic gear
  2.Hydraulic gear converter

Initial Design and Material selection

1.Prime mover

To pick the prime mover for this application, we need to consider about parameters involve in the machine such as power requirement for our all type of load, time and conditions we needed to operate.

For power sources there are few options mainly we can consider fuel type engine and electric motor. In this machine I would like to go with electric motor because this is an domestic type. We use this application indoor so if gases emitted during operation it will stuck in it or we need system to remove the gases. Hence better to pick one emit less gases.

Fuel engines emit gases and electric motor will not emit and since this machine is fixed, we don’t need to carry and power supply can be got continuously so using electric motor won’t be problem

Electric motors have high efficiency compared to fuel engine.

Things we need to look into when selecting electric motor,

- Required RPM
- Power supply type
- Bearing and belt types

Wood hardness factor

- This is the property rely on material of the wood for hardness of wood.
  It given by
  HF=specific gravity*2.20

Type	Specific gravity Value
Softwood	0.35
Cedar-Western	0.31
Red	0.45
Douglas Fir-Coast	0.38
Pine-Ponderosa	0.37
Spruce-White	0.65
Hardwood	0.55
Birch-Yellow	0.56
Maple	0.60
Oak-White	

Type of wood	Hardness factor	Gear used for this wood
Soft wood	0.0– 0.5	3rd gear
Engineering wood	0.5 – 1.0	2nd gear
Hard wood	1.0 – 1.5	1st gear

Examples of tress of soft wood

- Pine
- Cedar
- Redwood
- Douglas fir

Examples of Engineering wood

- Plywood
- Particle board
- Blockboard
- Medium Density fiber board

Examples of tress of hard wood

- Maple
- Oak
- Walnut
- Hickory

Parameters of bench saw	dimension
Our maximum cutting depth of the woodcutter is	
Number of Teeth. (T)	120
Diameter (D)	10inch
Saw Thickness (t)	1.5mm
Kerf with (K)	3mm/0.11inch
Gullet Area(G)	2 inch square

Gullet area

It is the space between teeth of saw, to carry away waste such as saw dust and wood particles.

Kerf width

Width of the cut by saw or in other words width of the teeth

Finding Maximum Horsepower required for the saw?

Horsepower = Gullet Area(G) x Number of teeth(T) x Saw Speed(V) x Bite Factor(B) x Kerf Factor (KF) X Wood Hardness Factor (HF) x Face Width Factor (FF) x 0.003

Feed per tooth = (Carriage speed*12) / (saw rpm*number of teeth)

carriage speed in feet per min, this speed we move the saw, Let carriage speed 250ft/min

= (250*12)/(3200RPM*120)

=0.007

Bite Factor = 0.5 + 4x Feed per tooth(in)

= 0.5+(4*0.007)

= 0.528

Kerf Width Factor = 0.11 x 2.91

= 0.3201

Maximum Cant face width = ((Gullet Area/Feed per tooth) x 0.70)+ 3

= (2*0.70/0.007)+3

= 203

Face Width Factor = 12/max cant face width

(12/203) = 0.0591

Our maximum Saw RPM is 3200.

Where we operate at third gear for soft hard wood with highest Carriage Speed.

H.P = G x T x V x BF x KF x HF x 0.6741 x 0.003

= 2*120*3200*0.528*0.3201*0.5*(0.0591) *0.003

=11.63HP

We must select the motor higher power than the required since there will be loss, let motor has efficiency of 85%.

So power input will be = 11.63*(100/85)

=13.67HP

Therefore, my selection for the power mover is 14Hp squirrel cage electric motor.

2.Power transmission

If we kept the motor near by cutting saw, there can be wood particles will go to motor and I also felt motor makes unnecessary sound and vibrations near you so if make a power transmission system it will be better. there are different kind of power transmission

I am going to select v-type belt it is common available belt drive. And, with increased torque will be useful in smooth cutting. Belt drive will be connected to input shaft of motor and input shaft of gear box.

My required rpm will be 3200 rpm, so ratio of “V” Belt drive is 3500/3200 = 1.094

3.Gear ratio Selection

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All length measurements are in inch

Kerf Factor $= \text{Kerf width} \times 2.91$

$= 0.11\,\text{inch} \times 2.91$

$= 0.3201$

Feed per tooth $= \dfrac{\text{Carriage Speed} \times 12}{\text{Saw rpm} \times \text{No. of tooth}}$

$= \dfrac{250 \times 12}{3200\,\text{rpm} \times 120}$

$= 0.007$

Bite Factor $= 0.5 + (4 \times \text{Feed per tooth in})$

$= 0.5 + (4 \times 0.007)$

$= 0.528$

Max Cont Face width $= \dfrac{\text{gullet area} \times 0.7}{\text{feed per tooth}}$

$= \dfrac{2 \times 0.7}{0.007}$

$= 200$

Face width factor $= 12/\text{max cont face width}$

$= 12/200 = 0.06$

Since I am connecting third gear with input shaft it has same rpm as input shaft

- Angular velocity of 3^{rd} gear is 3200 RPM
- Angular velocity of 2^{nd} year is (HF=1.0)

  Let other parameters does not change,

  H.P = 2*120*W2*0.528*0.3201*1.0*(0.059) *0.003

  W2 will be 1917.45 RPM

- Angular velocity of 3rd Gear is when HF=1.5

  H.P =2*120*W2*0.528*0.3201*1.0*(0.059) *0.003

  So W3 will be = 1278.32RPM

Gear ratio

For the 3^{rd} gear is 3500/3500 = 1

For the 2^{nd} gear is 3500/1917.5=2

For the 3^{rd} gear is 3500/1278.31= 3

For reverse gear is 3500/800 = 5

.

4.Selecting suitable gear box type

There are three varieties available

- Constant mesh gear box
- Sliding mesh gear box
- synchromesh gearbox

Constant Gear box	Sliding Gear box
It consists of spur gear and helical gear	It consists of spur gear
Main shaft gears are always mesh with countershaft gears	Main shaft gears are not in mesh constantly with counter shaft gears.
Torque transmission is high	Torque transmission is low
Size of the gear box is small compared to sliding gear box.	Size of gear box is large.
Transmission is smooth	Transmission is not smooth
Less noise	Higher noise

I have decided to go with constant mesh type, and manual transmission system, since we operate this equipment at high rpm we need smooth and easy gear transmission for that constant mesh type will suit.

5.Gear type

There two varieties of gear type we can consider for this purpose namely spur gear and helical gear.
Spur gear in high-speed application it will be noisy and create vibrations, but helical gear has better effect.
In this case if we get vibrations occur it will become difficult to handle the machine, so it's better to choose helical gear for this purpose and constant mesh gear box type generally use helical gears.
So my selection for gear type is helical gear

6.Layout

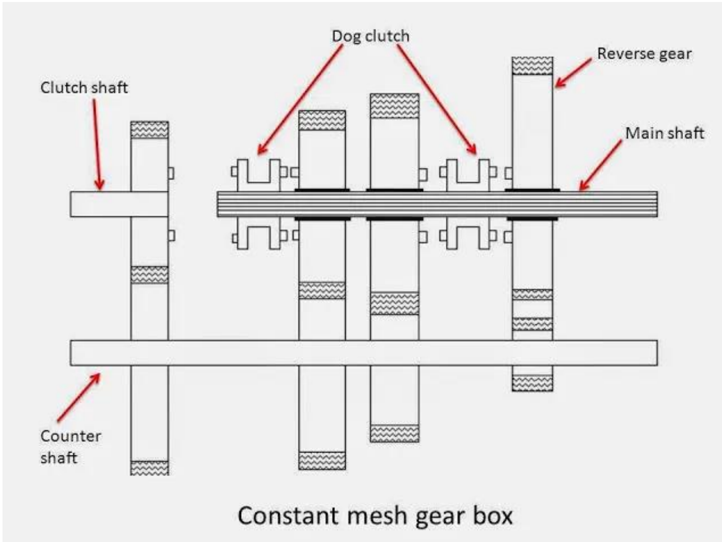
Constant mesh gearbox generally consists of 3 shafts such as input shaft,
the layshaft, and Main Shaft.
Other than that, there will be two dog clutch and gear lever will be available.
Input shaft
This is the shaft that get that power from v belt drive to gear box.

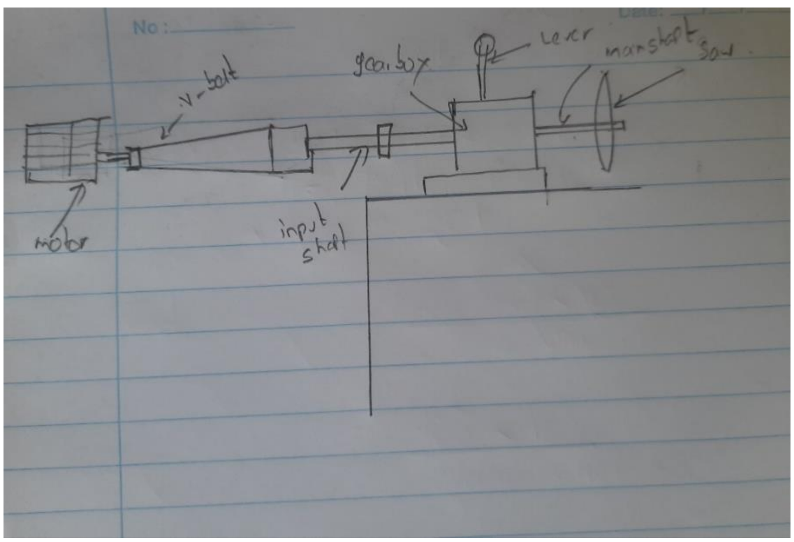
- Lay shaft
  In this shaft where all constant mesh gears are in fixed in the shaft

- Main shaft
  This shaft will provide output power to saw.

- Dog clutch
  this is used to reduce speed or changing Gear, gear which relates to dog clutch will Be the one drive the main shaft, this done by engaging and disengaging of gears on the main shaft

- Gear lever
  This lever will be used by operator to change gear manually.





7.Lubrication

- It will help to reduce friction between metals and enable smooth transmission
- can reduce heat dissipation

Factors needed to consider is

- ✓ Speed
- ✓ Materials used

- ✓ Temperature
- ✓ Load
- ✓ vibration
- ✓ Type of motion

Lubrication can be applied by several types.

- ➢ Drip oil feed
  Where the system operated on low speed and low load and low to moderate speed have bearings where small quantity of oil at regular intervals is expected.

- ➢ Splash oil feed
  Splash oil feed is a term applied to a variety of conducting lubricated bushings or pistons. Oil is splashed on bearings or pistons from the action of various moving parts regularly dipped in lubricating oil.

- ➢ Force oil feed
  High speed or high load equipment can develop high temperature from friction. To protect equipment from such high temperature from friction. To protect equipment from such high temperatures, a high flow of oil is needed. In force oil feed lubrication system, the pressurized oil from oil pump is directed to the rotating component.

- ➢ Grease lubrication
  Grease are semi solid lubricants. They are used instead of oil when lubricant must stay in one place or stay adhered to the part. Greases do not leak out as easily as oils. Greases are also used when the component cannot be lubricated often and are not accessible during operation.

In this we do not need lubrication continuously, this machine is not high load and high speed so lubricant can be applied at regular intervals and this machine won’t make high heat and not operated for long time also. So for this purpose I felt drip oil feed will be better as it applied for low load and oil will feed to machine in regular intervals.

Calculation Part

Gear ratio and RPM

Design Calculation
ME 3200

Gear box design

(01) gear

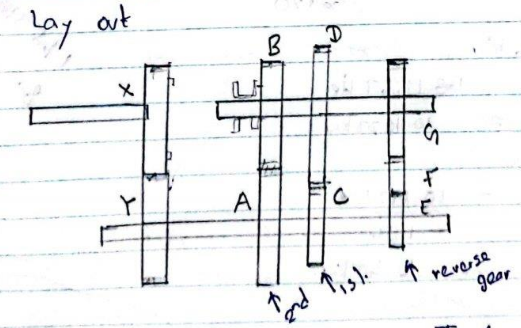
* Mass of gear wheel is negligible
* Centre distance between Lay shaft and output shaft is ~~100mm~~ 150 mm
* Pressure angle - 20°
* module - 2.5mm
* material we took is cast iron with yield stress 196 ×10⁶ pa

For Forward gears

* For reverse gear

Min Teeth required for gear to interface $\quad T_p \geq \dfrac{2A_w}{\sqrt{G^2+\sin^2\phi\,(1+2G)}-G}$

Lay out



X $\quad$ For $T_x/T_y = 1.3$

$$T_x \geq \frac{2\times 1}{\sqrt{1.3^2+\sin 20\,(1+2\times1.3)}-1.3}$$

$$\geq 13.08$$

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Gear Calculation

$$160 = \frac{(D_x + D_y)}{2}$$

$$D_x / D_y = \cancel{1.3}\ 1/1.3$$

$D_x - 139.1304\ mm$

$D_y - 180.1304\ mm$

$$T_x = D_x / m = \frac{139.1304\ mm}{2.5\ mm} = 55.65$$

$$T_y = 72.34$$

$$\text{Pitch line velocity} = r\omega = \frac{139.1304}{2} \times 10^{-3}\ m \times 3200\ rpm \times \frac{2\pi}{60}\ rads^{-1}$$

$$= 23.3114\ ms^{-1}$$

$$C_v = \frac{3}{3+v} = \frac{3}{3 + 23.3114\ ms^{-1}} = 0.1140$$

Required Power $= 8.6759\ kW$

design Power $= C_s \times 8.6759\ kW$

$= 1.54 \times 8.6759\ kW$

$= 10411.08\ W$

$= 10.4108\ kW$

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$P_t = F_t \cdot v$

$F_t = \frac{10.4108\ kW}{23.3114\ ms^{-1}}$

$= 446.6066\ N$

Max load pinion gear can sustain $= \sigma_0 \cdot C_v \cdot b \cdot m \cdot \pi y$

Let $\sigma_0 - 240 MPa$

$b - 10 \times m$

$y = 0.154 - \frac{0.912}{T}$

So

$= 0.154 - \frac{0.912}{55.65}$

$= 0.1375$

So Max Load $= 240 \times 10^6 Pa \times 0.1140 \times 10 \times (2.5 \times 10^{-3} m)^2 \times \pi \times 0.1375$

$= 739.39 N$

$\therefore 446.6066 N < 739.39 N$

So Pinion can sustain

For A, B set

Max Load $= 1101.06 N$ Pitch line $v = 14.08 ms^{-1}$

$F_t = 738.51 N$

$F_t <$ Max. Load

For C, D set

Max Load $= 924.56 N$ Pitch line $v = 18.06 ms^{-1}$

$F_t = \cancel{73}\ 576.49 N$

$F_t <$ Max Load

Gear calculation results

gear set	Aw	m	G	sin Φ	rpm of pinion	Tmin	center distance	diameter of pinion	diameter of wheel	teeth of pinion	Teeth of wheel	RPM of wheel
input shaft	1	2.5	1.3	0.342	3200	13.07597	160	139.1304	180.8696	55.65217	72.34783	2461.538
1st gear	1	2.5	1.925604	0.342	2461.538	17.59421	160	109.3791	210.6209	43.75165	84.24835	1278.32
2nd gear	1	2.5	1.283756	0.342	2461.538	16.3016	160	140.12	179.88	56.04801	71.95199	1917.45

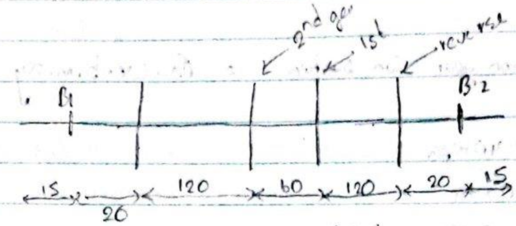
Reverse gear

Reverse gear		diameter	teeth	Pitch v	Ft	rpm
idler gear	1	48.49474	19.3979		1253.818	3270.154
lay pinion	1	64.42526	25.7701	8.303505	1253.818	2461.538
main wheel	1	158.5853	63.4341		1253.818	1000

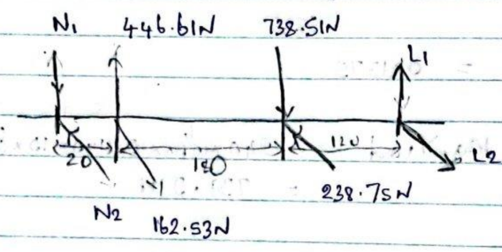
Shaft calculation

Calculation for shaft design

(01) Lay Shaft



1st gear



$Ft = 446.61$

$Fr = \tan 20 \times 446.61$

$= 162.53N$

$B_1)$ $446.61N \times 20 - 738.51N \times 200 + L_1 \times 340 = 0$

$L_1 = 408.146N$

$B_1)$ $162.53N \times 20 - 238.75 \times 200 + L_2 \times 340 = 0$

$L_2 = 148.53N$

$\uparrow$ $-N_1 + 446.61N - 738.51N + 408.146N = 0$

$N_1 = 116.24N$

$\rightarrow$ $-N_2 + 162.53N - 238.75N + 148.53N$

$N_2 = 42.31N$

Reaction at B_2 $= \sqrt{408.14^2 + 148.83^2}$

$= 434.34N$

at B_1 $= \sqrt{116.24^2 + 42.31^2}$

$= 123.71N$

B.M at gear X $= 123.71N \times 20 \times 10^{-3} m$

$= 2.47 Nm$

at gear C $= 434.34N \times 120 \times 10^{-3} m$

$= 60.81 Nm$

Max B.M at C $= 60.81 Nm$

Torque on lay shaft $= P/\omega$

$= \dfrac{10.41 kW}{2461.538 \times \frac{2\pi}{60}}$

$= 40.39 Nm$

$$\frac{\tau_{max}}{\text{Safety Factor}} \geq \frac{16 \sqrt{(K_m \cdot M_b)^2 + (K_T T)^2}}{\pi d^3}$$

$\tau_{max} = 345 \times 10^6$, $K_m = 1.5$

Safety Factor $= 2$ $\quad$ $K_T = 1$

$$d^3 \geq \frac{16 \times 2 \sqrt{(1.5 \times 60.81)^2 + (1 \times 40.39)^2}}{\pi \times 345 \times 10^6}$$

$$d \geq 14.34 \text{ mm}$$

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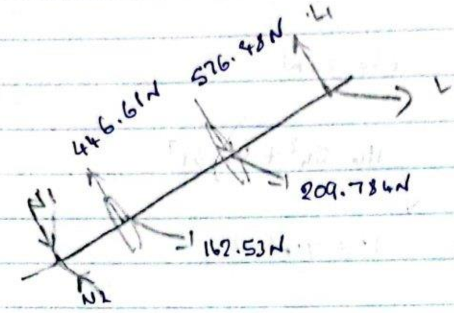
2^{nd} gear



$F_b = 576.48N$

$F_r = \tan 20 \cdot 576.48N$

$= 209.784N$

$B_1)$ $446.61N \times 20 - 576.48N \times 140 + L_1 \times 340 = 0$

$L_1 = 211.11N$

$B_2)$ $162.53N \times 20 - 209.78 \times 140 + L_2 \times 340 = 0$

$L_2 = 76.81N$

$\uparrow$ $-N_1 + 446.61N + 576.48N + 211.1 = 0$

$N_1 = 81.23N$

$\rightarrow$ $-N_2 + 162.53N - 209.78N + 76.81N = 0$

$N_2 = 29.56N$

Reacion at $B2 = \sqrt{211.1^2 + 76.81^2} = 224.65N$

$M = \sqrt{29.56^2 + 81.23^2} = 86.44N$

B.M at gear x = $\cancel{224.65N}$ $86.44N \times 20 \times 10^{-3} m$

$= \cancel{44.93Nm}$ $1.73 Nm$

at gear A = $\cancel{86.44N}$ $224.65N \times 200 \times 10^{-3} m$

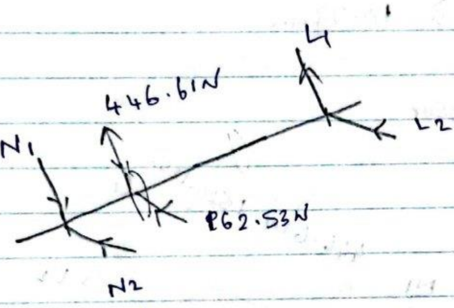
$= \cancel{1.73Nm}$ $44.93N$

Max B.M at A = 44.93Nm

$$d^3 \geq \frac{16 \times 2}{345 \times 10^6} \sqrt{(1.5 \times 44.93)^2 + (1 \times 40.39)^2}$$

$$d \geq 13.24 \text{ mm}$$

3^{rd} gear



B_1) $446.61N \times 20 \times 10^{-3} + L_1 \times 340 = 0$

$$L_1 = 26.28N$$

B_2) $162.53 \times 20 \times 10^{-3} + L_2 \times 340 = 0$

$$L_2 = 9.57N$$

$\uparrow \quad 446.61N + 26.28N - N_1 = 0$

$$N_1 = 472.88N$$

$\rightarrow \quad 162.53N - N_2 + 9.57N = 0$

$$N_2 = 172.01N$$

Reaction at B_2 $= \sqrt{26.28^2 + 9.57^2}$

$$= 27.96N$$

B_1 $= \sqrt{172.01^2 + 472.88^2 N}$

$$= 503.22N$$

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$$B.M \text{ at } x = 503.22\,N \times 20 \times 10^{-3}\,m$$

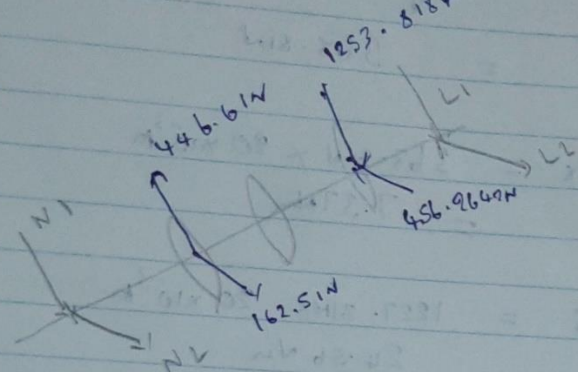
$$= 10.06\,Nm$$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^6} \sqrt{(1.5 \times 10.06)^2 + (1 \times 40.39)^2}$$

$$\geq 10.79\,mm$$

Lay shaft

Reverse gear



$F_t = 1253.818N$

$F_r = \tan 20 \cdot F_t$

$= 456.2642N$

B_1) $446.61N \times 20 - 1253.818N \times 320 + L_1 \times 340 = 0$

$L_1 = 1153.793N$

B_1) $162.51N \times 20 - 553.174 \times 320 + L_2 \times 340 = 0$

$L_2 = 419.86N$

$\uparrow$ $446.61N + N_1 + 1153.793N - 1253.81 = 0$

$N_1 = 346.59N$

$\longrightarrow$ $162.51N - N_2 + 419.86N - 456.26N = 0$

$N_2 = \cancel{346.59N}\ 126.121N$

Reaction at $B_2 = \sqrt{1153.79N^2 + 419.86N^2}$

$= 1227.813N$

Reaction at B1 $= \sqrt{346.59^2 N + 126.12^2 N}$

$= 368.81N$

B.M at X $= 368.81N \times 20 \times 10^{-3} m$

$= 7.37 Nm$

at E $= 1227.81N \times 20 \times 10^{-3} m$

$= 24.56 Nm$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^{16}} \sqrt{(1.5 \times 24.56)^2 + (1 \times 40.39)^2}$$

$\geq 11.72 mm$

Lay shaft Results

1st gear

forces	vertical	horizontal	Reaction	B.M
B2	408.1464	148.5245	434.3306	60.80628
B1	116.2433	42.30095	123.7008	2.474016
Max B.M	60.80628			
Torque on lay shaft			40.38876	
Dmin	14.33399			

2nd gear

vertical	horizontal	Reaction	B.M
211.1065	76.82164	224.6497	44.92995
81.22491	29.55775	86.4358	1.728716
Max B.M	44.92995		
Torque on lay shaft	40.38876		
Dmin 13.23773			

3rd Gear

3rd Gear

vertical	Horizontal	Reaction	B.M
26.27097	9.560006	27.95635	8.946033
472.8775	172.0801	503.2143	10.06429
Max B.M	8.946033		
Torque on lay shaft	40.38876		
Dmin	10.79095		

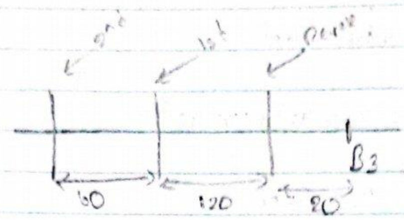
Reverse gear

Reverse gear

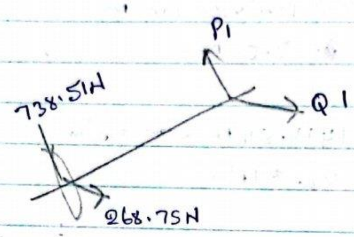
	vertical	horizontal	Reaction	B.M
	1153.793	419.8651	1227.813	24.55625
	346.5816	126.121	368.8161	7.376321
Max B.M		24.55625		
Torque on lay shaft		40.38876		
Dmin		11.72981		

Main shaft Calculation

Main shaft



1st gear



$P_1 = 738.51N$

$Q_1 = -268.75N$

$$\text{Reaction} = \sqrt{738.51^2 + 268.75^2}$$

$$= 785.89N$$

$$B.M \text{ at } D = 785.89N \times 0.140 \times 10^{-3} m$$

$$= 110.02 Nm$$

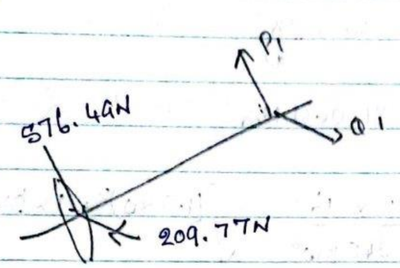
$$rpm = 1278.32 \text{ rpm}$$

$$\text{Torque} = \frac{10.411 kW}{1278.32 \times 2\pi/60} = 77.78 Nm$$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^6} \sqrt{(1.5 \times 110.0)^2 + (1 \times 77.78)^2}$$

$$d \geq 17.5294\,mm$$

2^{nd} gear



$P_1 = 576.49N$

$Q_1 = 209.77N$

$$\text{Reaction} = \sqrt{576.49^2 + 209.77^2}$$

$$= 613.4719\,N$$

$$B.M \text{ at } B = 613.4719N \times 200 \times 10^{-3} m$$

$$= 122.69\,Nm$$

$$\text{Torqe} = \frac{10.411kW}{1917.45rpm \times \frac{2\pi}{60}}$$

$$= 51.8492\,Nm$$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^6} \sqrt{(1.5 \times \cancel{110.0}\, 122.69)^2 + (1 \times 51.85)^2}$$

$$d \geq 17.8056\,mm$$

3^{rd} gear.

* At 3^{rd} gear, mainshaft will be directly connect to input shaf so no bending stress, only Torsion stress occur.

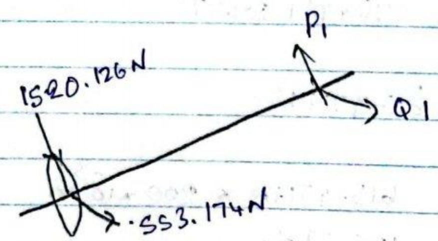
$$Torque = \frac{10.411 kW}{3200 \times \frac{2\pi}{60}}$$

$$= 31.0681 Nm$$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^6} \sqrt{(1.5 \times 0)^2 + 1 \times (31.0681)^2}$$

$$d \geq 9.71$$

Reverse gear



$$P_1 = 1520.126N$$
$$Q_1 = -553.174N$$

$$\text{Reaction at bearing } B_3 = \sqrt{1520.126^2 + 553.174^2}$$

$$= 1617.648 N$$

$$B.M \text{ at } a = 1617.648 N \times 20 \times 10^{-3} m$$
$$= 32.34 Nm$$

$$\text{Torque} = \frac{10.411\,\text{kW}}{700 \times 2\pi/60}$$

$$= 142.0264\ \text{Nm}$$

$$d^3 \geq \frac{2 \times 16}{345 \times 10^6} \sqrt{1.5 \times (32.34)^2 + 1 \times (142.0264)^2}$$

$$d \geq 16.4251\ \text{mm}$$

From above cases d_{min} is 17.8056 mm

Main shaft Calculation Results

1st gear

	1st gear			
	vertical	horizontal	reaction	B.M
B1	738.5096	268.7436	785.8878	110.0243
MAX B.M				
Torque				77.77277
Dmin				17.52951

2nd gear

2nd Gear

vertical		horizontal	reaction	B.M
	576.4881	209.784	613.4719	122.6944
Torque				51.84933
	Dmin			17.80577

3rd gear

3rd gear

Torsion	B.M
31.06828	0

Torque	31.06828
Dmin	9.716266

Reverse gear

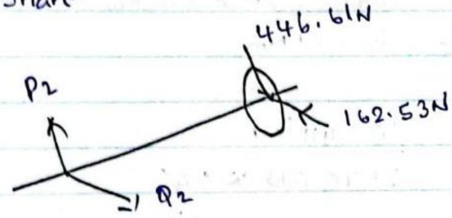
Reverse gear

vertical	horizontal	Reaction	B.M
1253.818	456.2642	1334.255	26.68509

Torque	110.465
Dmin	15.13788

Input shaft Calculation

Input shaft



$P_2 = 446.61N$; $Q_2 = 162.53N$

Reaction at bearing $B_4 = \sqrt{446.61^2 + 162.53^2}$

$= 475.258\ N$

B.M at x $= 475.258N \times 20 \times 10^{-3} m$

$= 9.5051 Nm$

Torque $= 31.068\ Nm$

$d^3 \geq \frac{2 \times 16}{245 \times 10^6} \sqrt{(1.5 \times 9.5051)^2 + (1 \times 31.068)^2}$

$d \geq 10.0307 mm$

d min is 10.0307mm

Input shaft Calculation Results

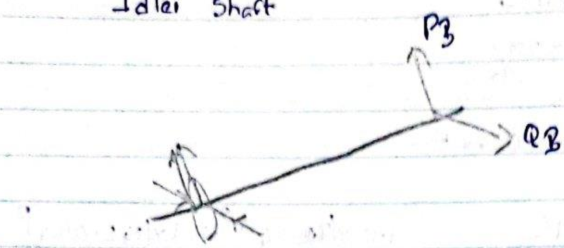
	Input shaft Calculation		
B1	vertical	horizontal	Reaction
	446.6065	162.5201	475.258
Max B.M			9.50516
Torque			31.06828
Dmin			10.03074

Idler shaft

No :

Date: ..../..../

Idler shaft



$P_3 = 1520.126N \times 2 \quad = 3040.253N$

$Q_3 = 0$

Reaction $= 3040.253N$

B.M at F $= 3040.253N \times 20 \times 10^{-3}m$

$= 60.81Nm$

Torque $= \dfrac{10.411 kw}{3270.073 \times 2\pi/60}$

$= \cancel{60} \rightarrow 30.41Nm$

$d3 \geq \dfrac{2 \times 16}{345 \times 10^6} \sqrt{(1.5 \times 60.81)^2 + (1 \times 30.41)^2}$

$d \geq 14.1588mm$

d Min is 14.1688 mm

Idler shaft Calculation Results

	Idler shaft calculation		
B1	vertical	Horizontal	Reaction
	2507.635	0	2507.635
Max B.M			50.1527
Torque			30.40178
Dmin			13.38048

Calculations For Bearings

Bearing For Lay shaft

Max. Load Force $(F_{r\,max})$ = 503.2143 N at 3rd gear

In B1

In B2 = 1494.536N at reverse gear

Assumptions

* working duration = 6hr/day × 250 day/year × 8 year
= 12000 hrs

Life of bearing = $12 \times 10^3 \times 60 \times 246.154$ rpm
= 177.23088×10^6 cycles.

B1

equivalent dynamic bearing load :

$$P = XF_r + YF_a$$

no axial load $F_a = 0$

so $F_a/F_r = 0$ and $F_a/C_{or} = 0$

$e = 0.18$ so $0 < 0.18$

$\therefore X = 1, Y = 0$

$P = 1 \times 503.2143N + 0$

$= 503.2143N$

For 90% reliability

$$L_{10} = \left(\frac{C_r}{P}\right)^3$$

$$(177.23086)^{1/3} = \frac{C_r}{503.2143N}$$

$$C_r = 5.617 \times 503.2143$$
$$= 2826.61N$$

B2

$$P = 1494.536N$$

$$C_r = 5.617 \times 1494.536N$$
$$= 8394.89N$$

From lay shaft calculation dmin shaft is 14.34mm

here I can select 28mm and above bearings.

For B2

I Select 30mm, $C_r = 10200 > 8394.81N$

bearing NO - 6006

For B1

I select 20mm and above

Select 28mm, $C_r = 2826.61N > 3100$

bearing NO - 6804

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Bearing for main shaft

Max. radial fore $(F_{rmax}) = 1617.648N$ at reverse gear

Life of bearing $= 12 \times 10^3 \times 60 \times \cancel{80}\,700$ rpm
$= 504 \times 10^6$ cycles.

$F_a = 0$, so $F_a/F_r = 0$ and $F_a/C_{0r} = 0 < e = 0.18$

$\therefore X = 1, Y = 0$

$$P = 1 \times 1617.648N$$
$$= 1617.648N$$

For 90% reliability

$$L_{10} = \left(C_r/P\right)^3$$

$$C_r = (504)^{1/3} \times 1617.648$$
$$= 12873.43$$

I can seld 35mm and above

I Selad 35mm bearing NO - 6207

$12873.43 < 19700$

Bearing for Input shaft

Max. radial force $(F_{r\,max}) = 475.258N$

rpm = 3200 rpm

Life of bearing $= 12 \times 10^3 \times 3200 \times 60$

$= \cancel{384}\ 2304 \times 10^6$ cycles.

$F_a = 0$ so $X = 1,\ Y = 0$

$\therefore P = 475.258N$

For 90% reliability.

$L_{10} = \left[\frac{C_r}{P}\right]^3$

$(2304)^{1/3} = \frac{C_r}{475.258N}$

$C_r = 6277.07$

I can select 25 mm above

I select 28 mm bearing NO 60/28

$6277.07N < \cancel{6}\ 9600N$

Bearing For idler shaft

Max radial Force = 3040.253 N

rpm = 3270.073 rpm

duration = $\frac{12 \times 10^3}{20}$ hrs

Life of bearing = $\frac{12 \times 10^3 \text{ hrs}}{20} \times 60 \times \cancel{3040.253}$ 3270.073 rpm

= 117.723×10^6 cycles

$F_a = 0$, so $x = 1$, $Y = 1$

$P = \cancel{117}$ 3040.253 N

For 90% reliability

$$L_{10} = \left(\frac{C_r}{P}\right)^3$$

$$(117.723)^{1/3} = \frac{C_r}{3040.253\text{N}}$$

$$C_r = 14900.35 \text{ N}$$

I can select 35 mm and above

I Select 40 mm bearing NO - 6208

22400N > 14900.35N = Cr

No. ....

Date: ..../..../....

Calculation for keys

Lay shaft

Power = 10.411 kw

rpm = 2461.54 rpm

length = 340 mm

Safety Factor = 2

* let key also made using shaft material

τ_{max} = 345 Mpa

σ_T = 690 Mpa

Torque on Lay shaft $= \dfrac{10.411\text{ kw}}{2461.54 \times \frac{2\pi}{60}}$

$= \cancel{30.41\text{ Nm}}\ \cancel{31.0681\text{ Nm}}\ 40.39\text{ Nm}$

width of key should be greater than d/4

$w \geq \frac{d}{4}$

$w \geq \frac{30\text{mm}}{4}$

$\geq 7.5\text{mm}$

* key I have selected gib-head rectangular sunk key

From the standard Proportions of gib-head key Catalogue :

For d = 30mm $\Rightarrow$ width of key = 10mm > 7.5mm

Thickness of key = 8 mm

Shearing strength of key :

$$T = l \times w \times \tau \times d/2$$

T - Torsional strength
d - diameter of keyshaft

For Safety,

$$l \times w \times \tau \times d/2 \geq T$$

$$l \geq \frac{T}{w \times \tau \times d/2}$$

$$\tau_{all} = \frac{\tau_{max}}{\text{Safety Factor}}$$

$$\geq \frac{40.39\text{ Nm}}{10\text{mm} \times \frac{345\text{ Mpa}}{2} \times \frac{30\text{mm}}{2}}$$

$$\geq 1.96\text{ mm}$$

Crushing strength of key

$$l \geq \frac{T}{t/2 \times \tau_{crush} \times d/2}$$

$$\tau_{crush} = \frac{\sigma_y . T}{\text{Safety Factor}}$$

$$= 690\text{mpa}/2$$

$$\geq \frac{40.39\text{ Nm}}{8\text{mm}/2 \times 345\text{ Mpa} \times 30\text{mm}/2}$$

$$\geq 4.207\text{ mm}$$

$\therefore$ We can take length of key ≥ 4.207 mm

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No :

Date: / /

Maximum strength of ~~of~~ shaft with keyway

$$T = \frac{\pi}{16} \times \tau_{all} \times d^3 \times e$$

e - shaft strength factor
h - depth of key

$$e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

$$e = 1 - 0.2\left(\frac{10}{30}\right) - 1.1\left(\frac{8/2}{30}\right) \qquad h = \tfrac{1}{2}$$

$$= 0.78667$$

$$\text{So} \quad T = \frac{\pi}{16} \times \frac{345\,mpa}{2} \times (30 \times 10^{-3})^3 \times 0.786667$$

$$= \cancel{719408.130\,N} \quad 713.31\,N$$

$$\text{Shear strength of key} = l \times w \times \tau \times d/2$$
$$= 30 \times 10 \times \frac{345}{2} \times 30/2 \times 10^6 \times 10^{-9}$$

$$= 776.25\,N.$$

$$\frac{\text{Shear strength of key}}{\text{normal strength of key shaft}} = \frac{\cancel{776.25\,N}}{\cancel{719408.130\,N}} =$$

$$= \frac{776.25\,N}{713.31\,N} = 1.089 > 1$$

Idler shaft

Input shaft

$$\text{Torque} = \frac{10.411\,kW}{3200\,rpm \times 2\pi/60}$$

$$= 31.0681\,Nm$$

diameter of shaft = 30mm

Width of key $\geq d/4$

≥ 7.5 mm

From the standard Proportions

$w = 10mm$, $t = 8mm$

Shear strength =

$$l \geq \frac{T}{w \times \tau \times d/2}$$

$$\geq \frac{31.0681\,Nm}{10 \times \frac{345}{2} \times 30/2}$$

$$l \geq 1.21\,mm$$

For Crushing strength

$$l \geq \frac{T}{t/2 \times \tau_{crush} \times d/2}$$

$$\geq \frac{31.0681\,Nm}{8mm/2 \times 345\,Mpa \times 30mm/2}$$

$$\geq 1.51\,mm$$

$\therefore\ d \geq 1.51mm$

~~Width of gear wheel~~ =

Shaft strength with keyway = Same as in Lay shaft = ~~71.4408-130N~~ 776.25N (same diameter)

Shear strength of key $= d \times w \times \tau \times d/2$

$= \cancel{30}\ 25mm \times 10mm \times \frac{345}{2} MPa \times 30m/2$

$= 64.6875 \times 10^{-3} N \Rightarrow 646.875N$

$776.25N > 646.875N$

$\therefore$ Shaft and key are safe.

IDler shaft

Torque $= \dfrac{10.411 kW}{3270.073 rpm \times 2\pi/60} = 30.41 Nm$

diameter of shaft = 40mm

Width of key $\geq d/4$

$\geq 10mm$

From the standard Proportions,

$W = 12mm$, $t = 8mm$

Shear strength

$$l \geq \frac{T}{w \times \tau \times d/2}$$

$$l \geq \frac{30.41\,Nm}{\frac{345\,Mpa}{2} \times 12\,mm \times 40/2\,mm}$$

$$l \geq 0.73\,mm$$

Crushing strength

$$l \geq \frac{T}{t/2 \times \tau_{crush}\, d/2}$$

$$\geq \frac{30.41\,Nm}{8mm/2 \times \frac{345\,Mpa}{2} \times \frac{40}{2}}$$

And width of gear = 1.101mm

So $l \geq 1.101mm$

Shaft strength with key way

$$= \frac{\pi}{16} \times \frac{345\,Mpa}{2} \times$$

$$= \frac{\pi}{16} \times \tau_{all} \times d^3 \times e$$

$$e = 1 - 0.2 \times \left(\frac{12}{40}\right) - 1.1 \left(\frac{8}{40}\right)$$

$$= 0.72$$

$$T = \frac{\pi}{16} \times \frac{345\,Mpa}{2} \times 0.72 \times (40 \times 10^{-3})^3$$

$$= 1560.75\,N$$

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Shear strength of key $= l \times w \times \tau \times \frac{d}{2}$

$= 30 \times 10 \times \frac{345 \times 10^6}{2} \times \frac{\cancel{25}\,40}{2}$

$= 1035\ N$

max shear strength > max shaft strength.

∴ Design is safe.

Calculation For belt-drive (v-belt)

Assumptions

(1) belts are made of standard synthetic rubber (density $= 10^3\ kg\,m^{-3}$)

(2) $\mu = 0.28$

(3) $\sigma_{all} = 2\ Mpa$

motor Power = ~~10.441 kw~~ 8675.9 kw , Speed = 3500 rpm

Efficiency = 85 %

Transmitted Power = 7.374 kw

Input shaft speed = 3200 rpm

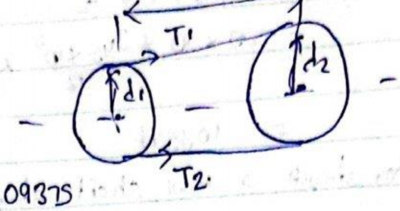
Power is in range of (2 - 15) range, we can use B type belt

Min Pitch diameter of pulley = 125 mm (d_1)

Top width =

thickness =

Find



Speed reduction ratio = 1.

$= 1.09375$

diameter $= 1.0938 \times 125\,mm$

$= 136.72\,mm$

Assume Center distance = 800mm

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

$$= \frac{136.72 - 125}{2 \times 800}$$

$$= 7.325 \times 10^{-3}$$

$$\alpha = 0.4197^{\circ}$$

Angle of contact $= \theta = 180 - 2\alpha$

$= 179.16^{\circ}$

$= 3.13\,rad$

For B type belt ⇒ groove angle $2\beta = 32^{\circ}$

$\beta = 16^{\circ}$

$$2.3 \log(T_1/T_2) = \mu\,\theta \cdot \operatorname{cosec}\beta$$

$$2.3 \log\left|\frac{T_1}{T_2}\right| = 0.28 \times 3.13\,rad \times \operatorname{cosec} 16^{\circ}$$

$$= 3.1794$$

$$\log(T_1/T_2) = 1.38$$

$$T_1/T_2 = 23.98$$

Velocity of belt $= \frac{\pi d_1 N_1}{60}$

$$= \frac{\pi \times 0.125 \times 3500 \text{ rpm}}{60}$$

$$= 22.91 \, ms^{-1}$$

$$\text{Length of belt} = \pi \times \left(\frac{d_2+d_1}{2}\right) + 2C + \frac{(r_1-r_2)^2}{2C}$$

$$= \pi \times 130.86\,mm + 2 \times 800 + \frac{(136.72-125)^2}{2 \times 2 \times 800}$$

$$= 411.11 + 1600 + 0.04$$

$$= \cancel{1734.045\,mm}\ 2011.16\,mm$$

From standard length we can choose ~~1760 mm~~ 2180 mm For B-type belt

$$\cancel{1760}\ 2180\,mm = \pi \left(\frac{130.86}{2}\right) + 2C + \frac{(136.72-125)^2}{4 \times C}$$

$$C = 834.43\,mm$$

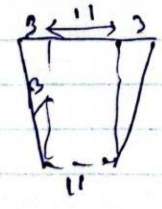
$$\text{mass of belt per unit length} = \frac{1.89\,N}{9.81}$$

$$= 0.192\,kg$$

$$\text{Tension} \Rightarrow T = mv^2$$

$$= 0.192\,kg \times 22.91^2\,s^{-1}$$

$$= 100.78\,N.$$



$$h = \frac{3}{\tan\beta}$$

$$= 10.46$$

$$A = \frac{1}{2} \times (17+11) \times h$$

$$= 0.146 \times 10^{-3}\,mm^2$$

$$\text{Max Tension} = \sigma_{all} \times A$$

$$= 2 \times 10^{6} \times 0.146 \times 10^{-3}$$

$$= 292\,N$$

No: ________ Date: ___/___/___

Tension on high side T_1 = $292N - 100.78N$

= $191.22N$

$T_1 / T_2 = 23.98$

$T_2 = 7.9742N$

Power transmitted by belt drive $= (T_1 - T_2)v$

$= (191.2N - 7.97N) \cdot 22.91ms^{-1}$

$= 4.197\,kW$

No. of belts needed $= \frac{10.411kW}{4.197kW}$

$= 2.40$

$= 3$

So 3 belts needed.

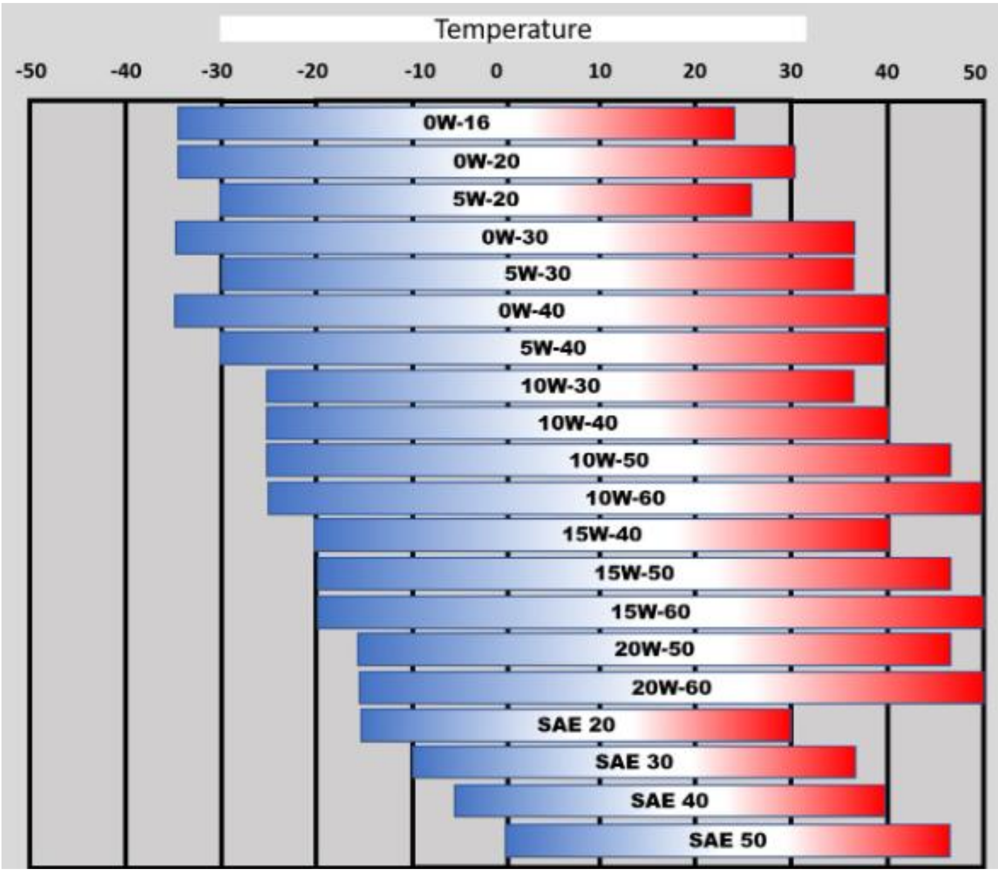
Lubrication Selection

Lubrication could be decided on max tangential velocity.

- Grease Lubrication applicable for 0 to 6ms^-1.
- Splash lubrication applicable for 4ms^-1 to 15ms^-1.
- Forced oil circulation lubrication applicable for above 15ms^-1.

so, in this case max tangential velocity is 23ms^-1 therefore we can use forced oil lubrication method for table saw.

Which oil need to be used can be found by operating temperature of table saw.



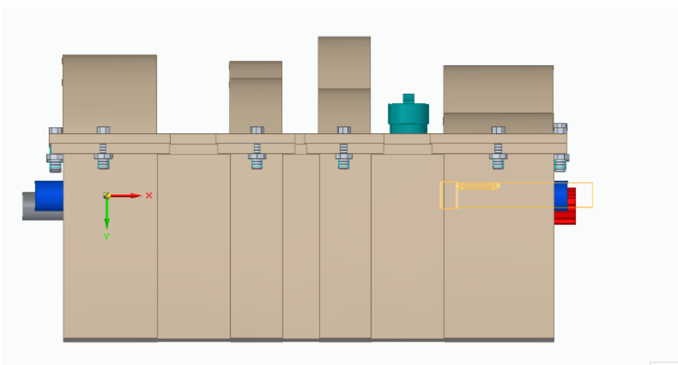
I consider the average operating temperature as 50 degrees so my pick for lubrication is SAE 20W-60 semisynthetic base.

Final Design

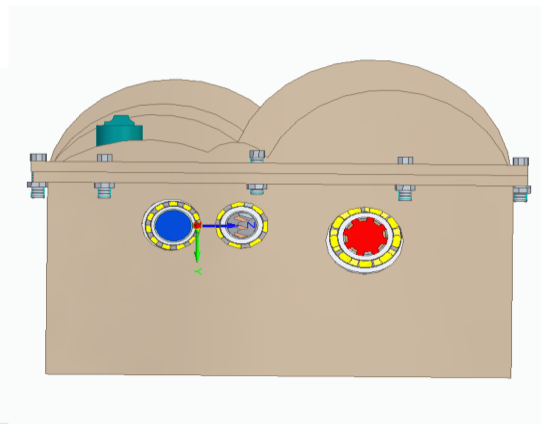
By the above calculations I designed the constant mesh gear box for table saw application using solid edge software. In the constant mesh gear box has gears, shaft, spline, dog clutch and housing etc. are listed below.

Speed variation in main shaft happen due to the dog clutch where it clamped with the required gear in main shaft.

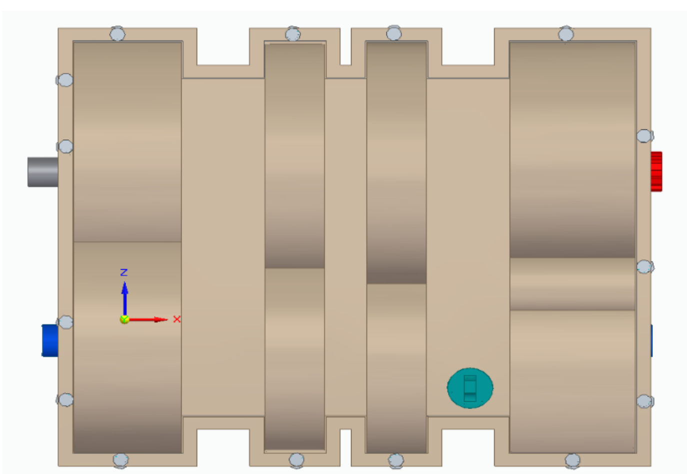
It helps for smooth gear change when compared to sliding mesh type.



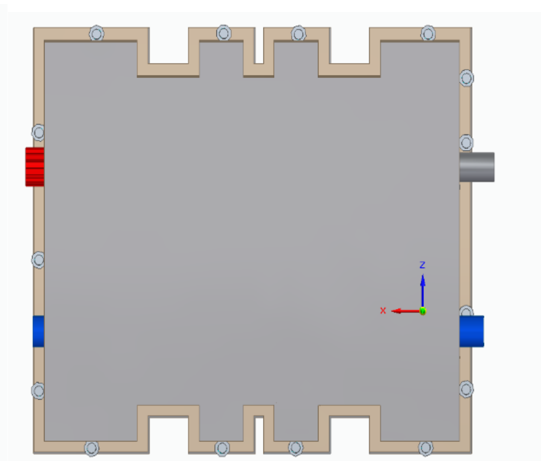
Front view



Back view

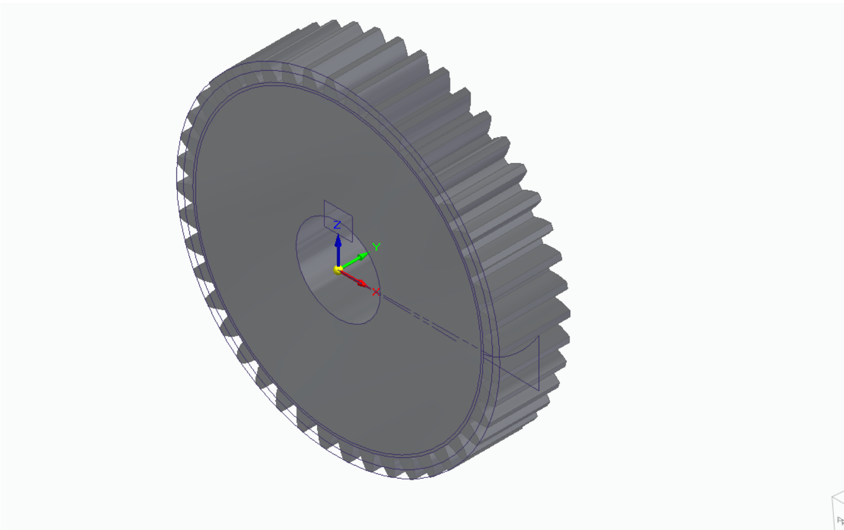


Top view

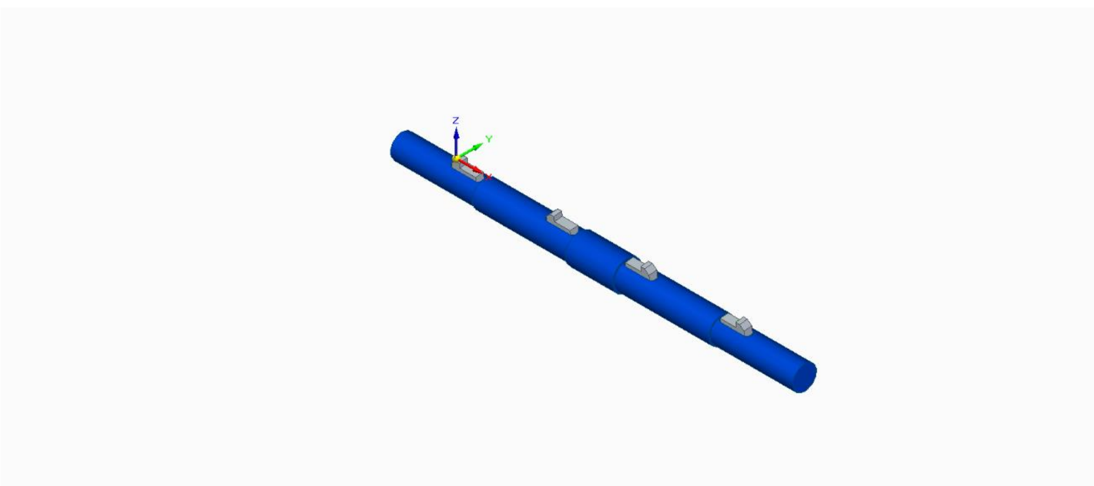


Bottom view

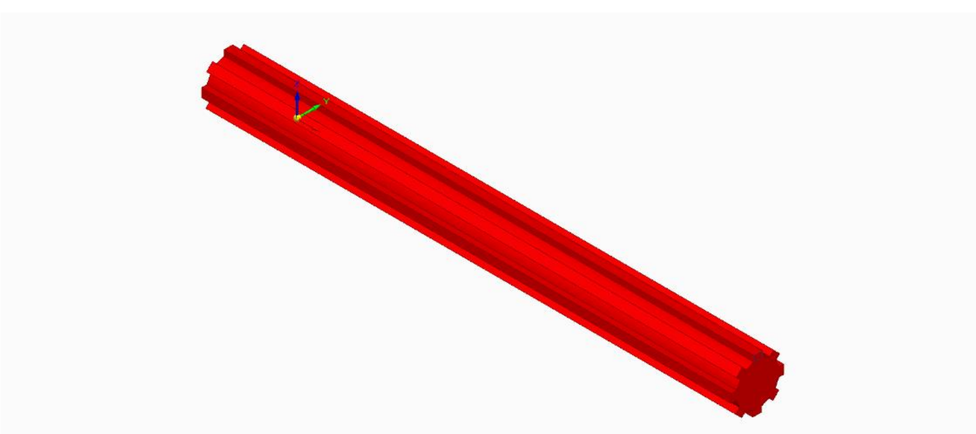
Gear



Shaft

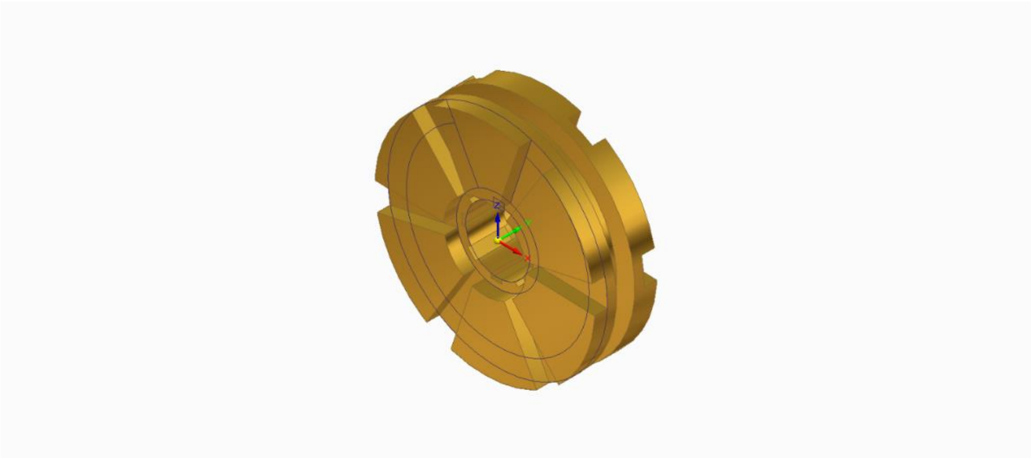


Spline



Dog clutch

As mentioned above this will help for speed variation in output shaft.

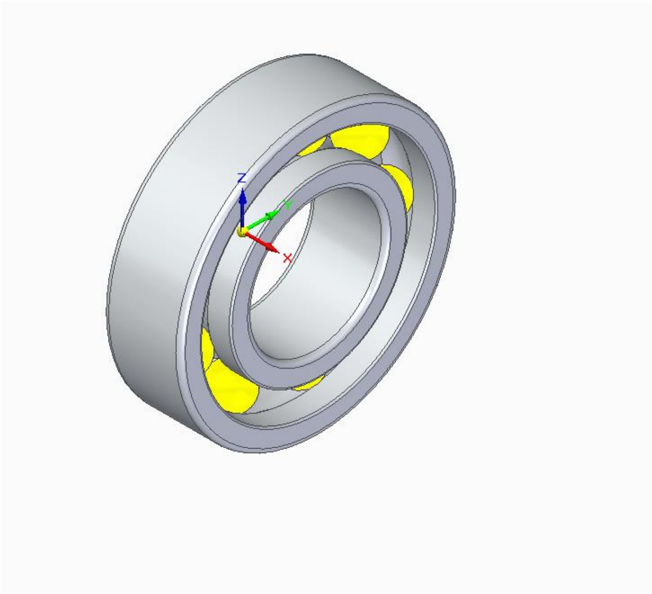


Keys

It will help gears from breaking, it act as sacrifice metal when overload conditions since repairing or change to new gear wheel costly and difficult.

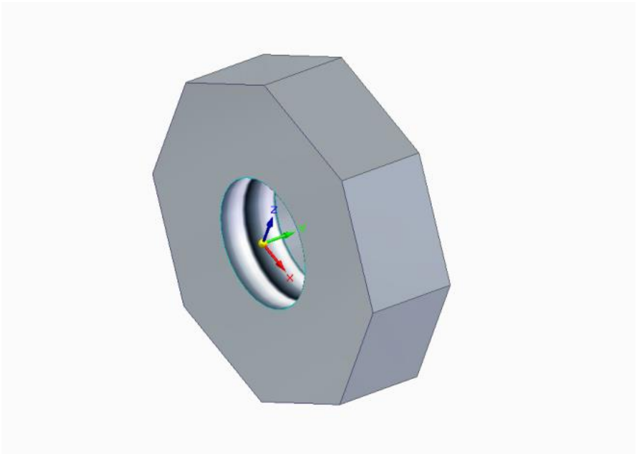
Bearing

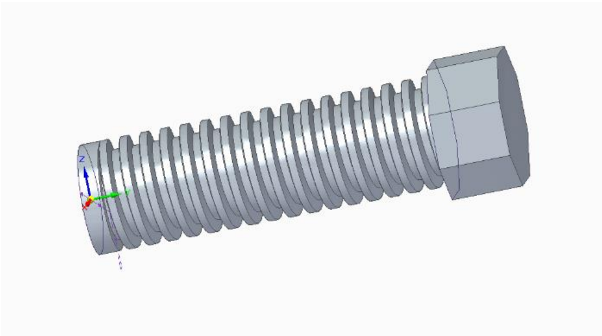
As name mentioned it bear the acting forces on shafts.



Nut and Bolt

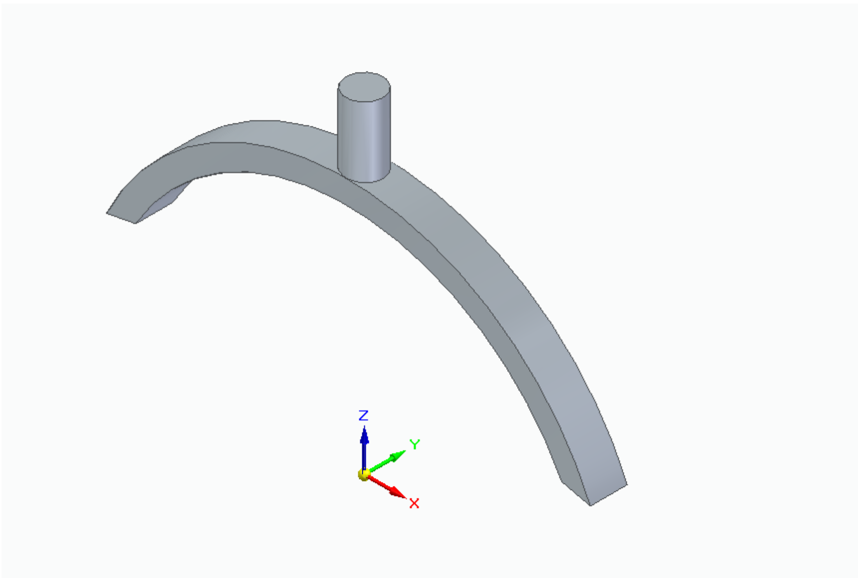
It will used to fit the upper and lower casing of gear box tighly.





Folk

It will help to change gear, when lever move this folk, it will the move the dog clutch so when dog clutch move and clamp another gear wheel then speed changes in main output shaft.

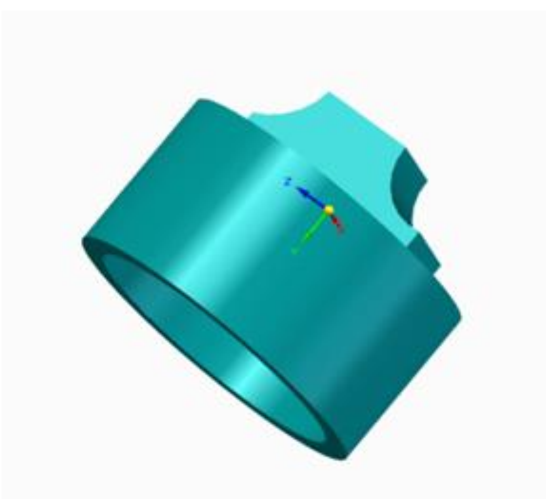


Lever

It will help to change the gear, to make move the folk mate with dog clutch.

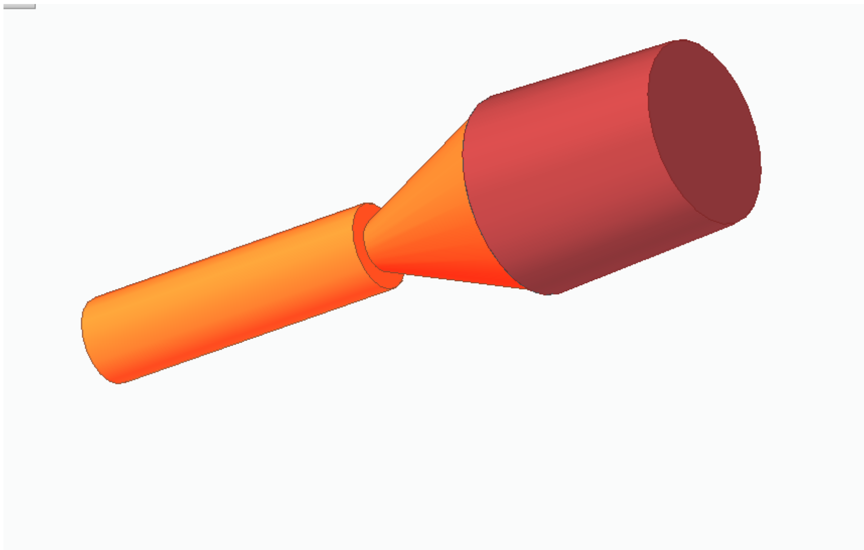
Oil cap

Used to change oil in the system



Gear Manual Changing lever

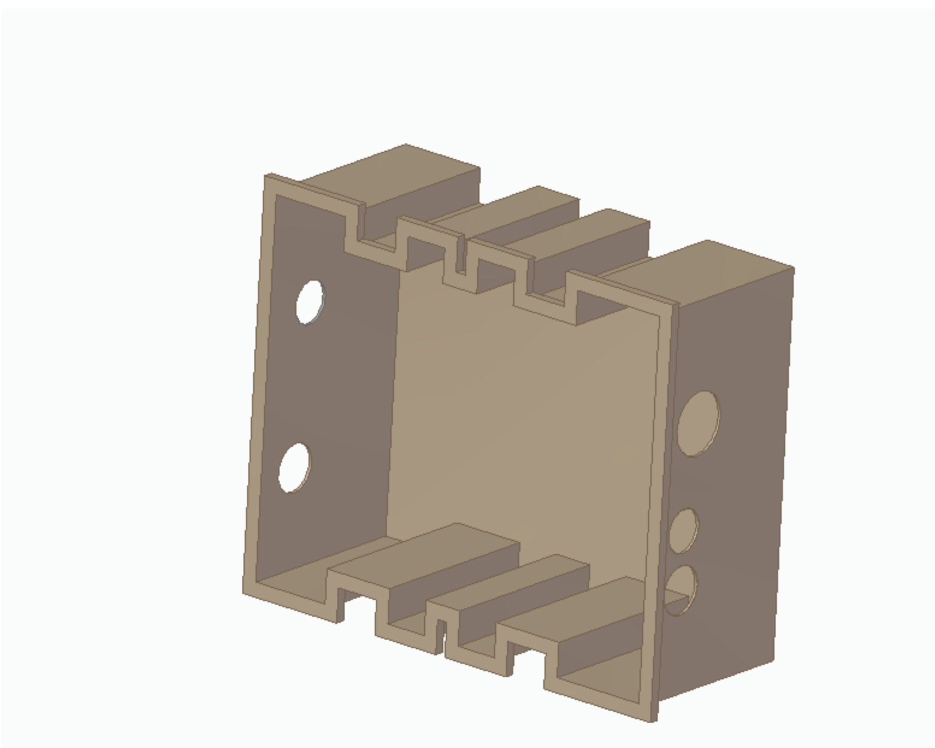
Used to change the gear by operator.



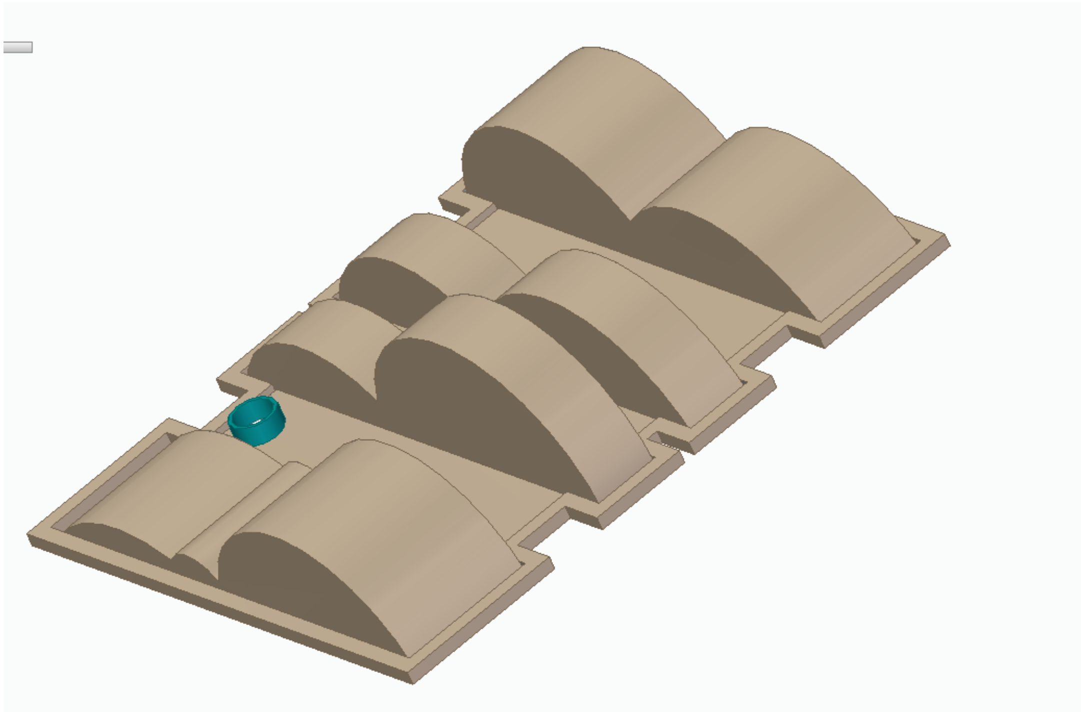
Housing

Bottom housing

This where shafts and bearings lie, protective for gear box from outer area.



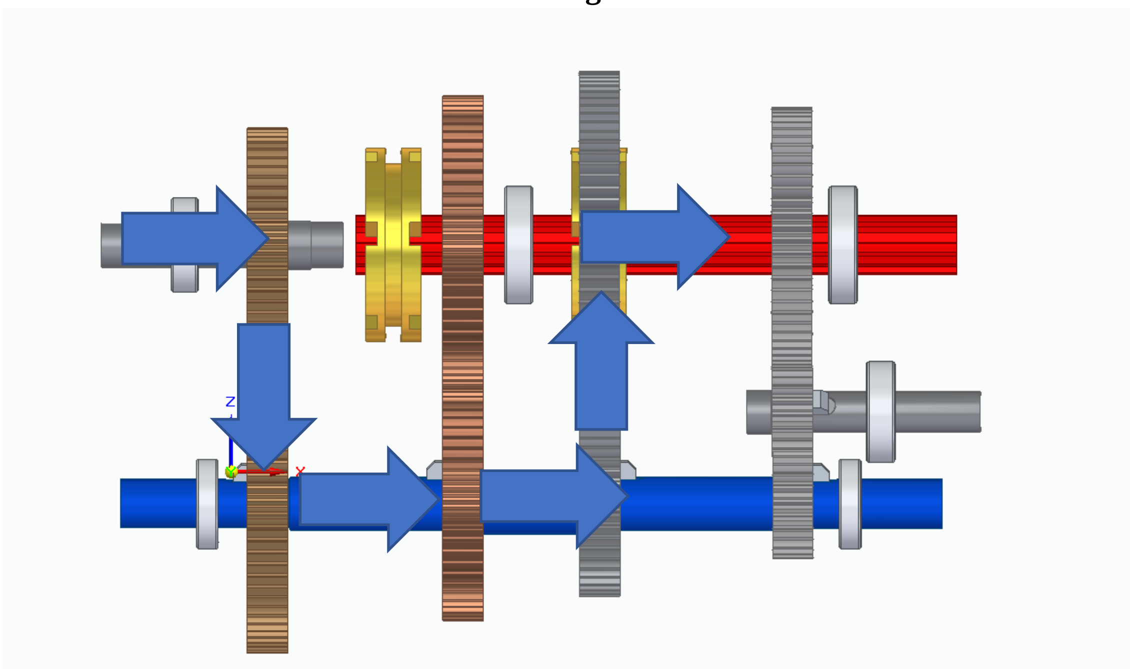
Upper housing



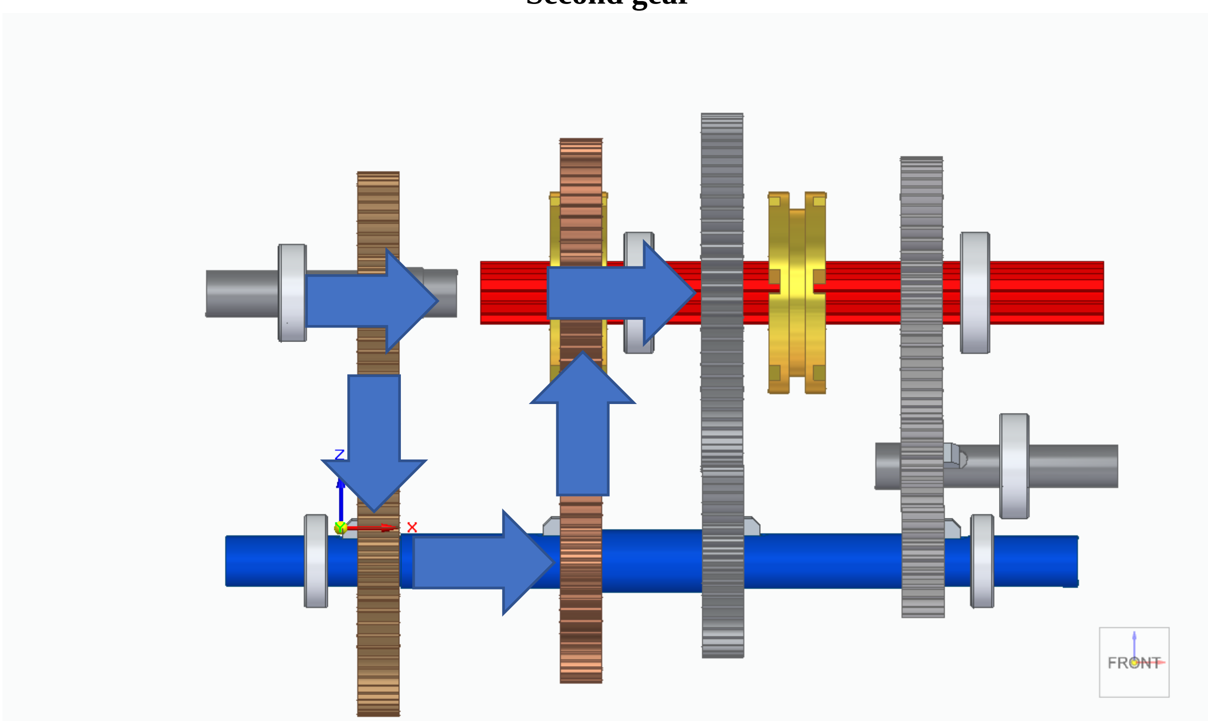
Power Transmission

Power transmission from input shaft to output shaft inside the gear box according to relevant gear changes diagramed below.

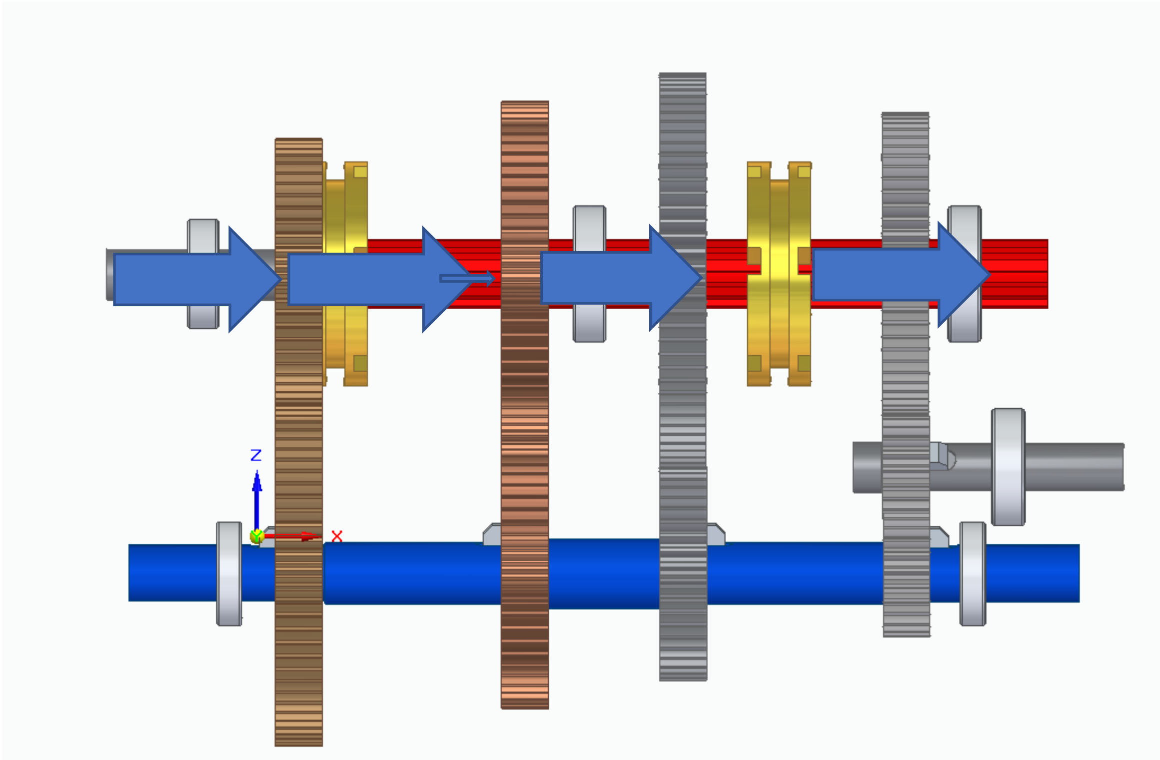
First gear



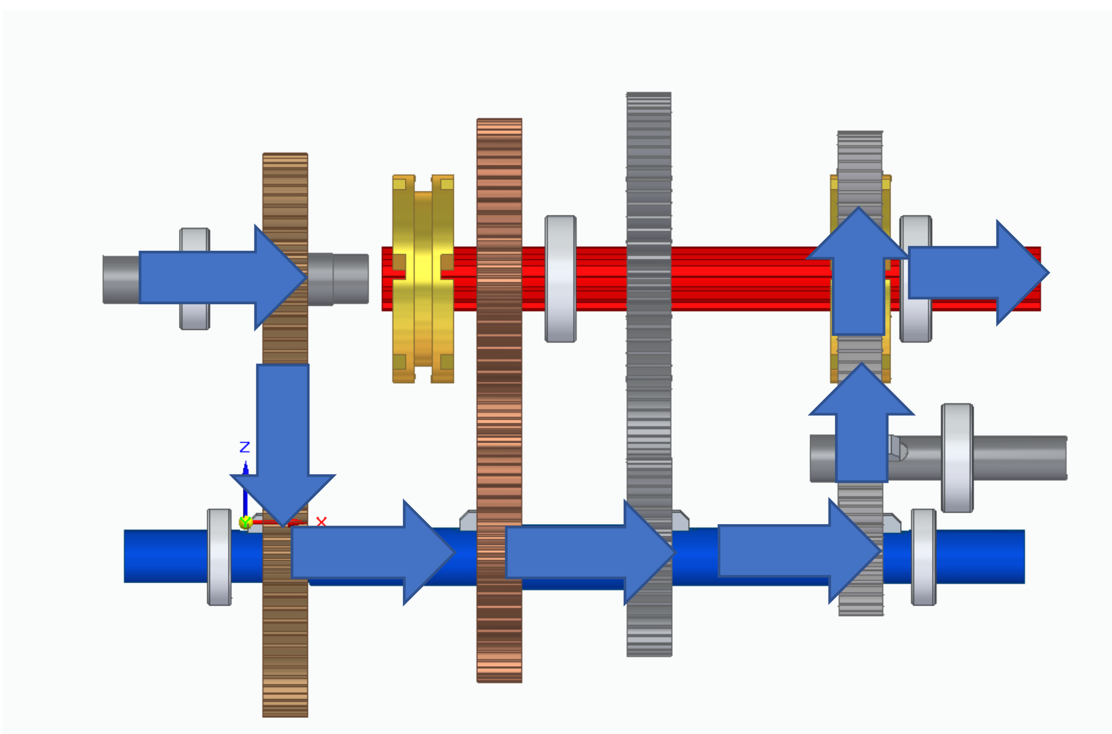
Second gear



Third gear



Reverse gear



Manufacturing Methods used for production process

Gear wheels

First metal piece was picked and then it clamped on lathe machine, gear cutting is done by horizontal milling process in milling machine. now heat treatment and finishing process will end result the gear wheel.

Shafts

Metal rod is taken and clamped on lathe machine, using turning operation we can reduce the rod to our required size and then it was faced and bored by lathe. In the end chamfering, filleting and undercuts were carried out.

Spline shaft

Same starting process like normal shaft then to make spline horizontal milling operation was used.

Keys

They were produced by horizontal milling operation.

Dog clutch

It was produced by milling process.

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