Machine System Design

1.Name of Machine:

"Mechanical Power Press"

2.Purpose of the Machine:

To make perforated metal sheets, by punching holes in metal sheets.

3.Design Objective:

Perforated Metal sheets are used for various applications in various industries like automotive, construction, food and beverage industry just to name a few.

Our machine will be focused on production of perforated baking sheets, which are used domestically and can easily fit inside a regular microwave commonly used in homes.

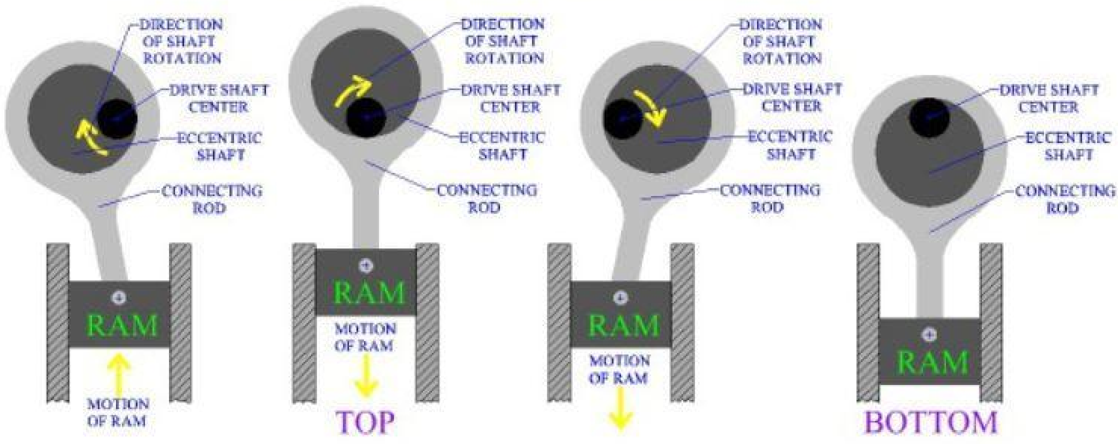
4.Broad Design Specifications:

- Capacity – 80 ton.
- Stroke – 60 mm.
- No. of strokes per min – 30
- Sheet size – 270 mm*400 mm
- Sheet Thickness – 1 mm
- Dimensions of table – 300 mm*450 mm.
- Motor – Induction motor of 1440 rpm.
- Motor Power – 1 HP

5.Selection of Process and Mechanism:

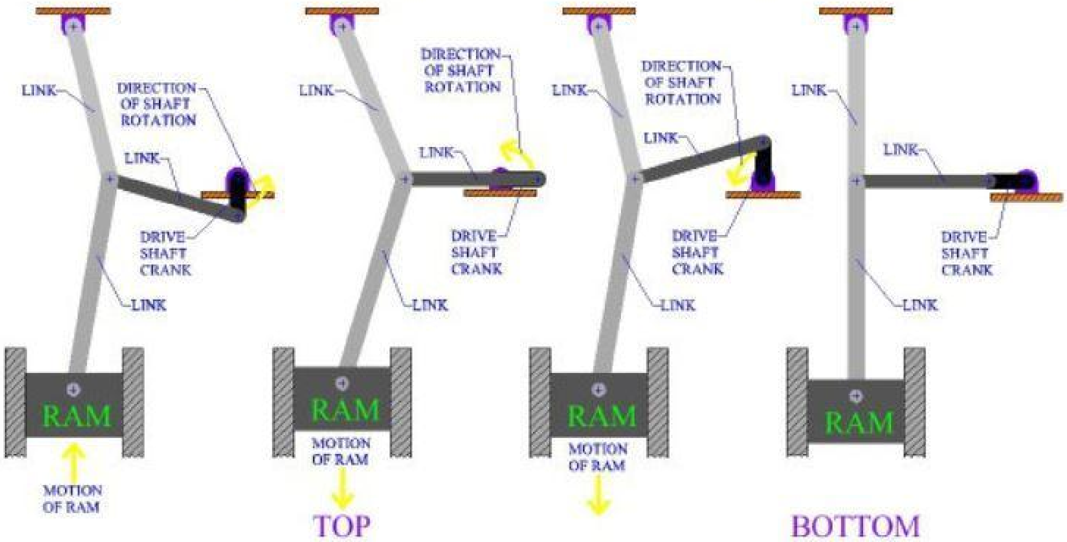
a) Eccentric Press –

The eccentric press uses a motor to drive an eccentric shaft, rotating in a connecting rod. The connecting rod moves a ram in a slider joint one dimensionally. The eccentric shaft itself is round, therefore it may completely rotate within the connecting rod. The centre of the drive is not the centre of the overall shaft. As the motor rotates, the centre of the drive remains stable but the overall centre of the shaft changes. This causes the shaft to change position, providing motion. The actual principle of an eccentric press is very similar to a crank press.



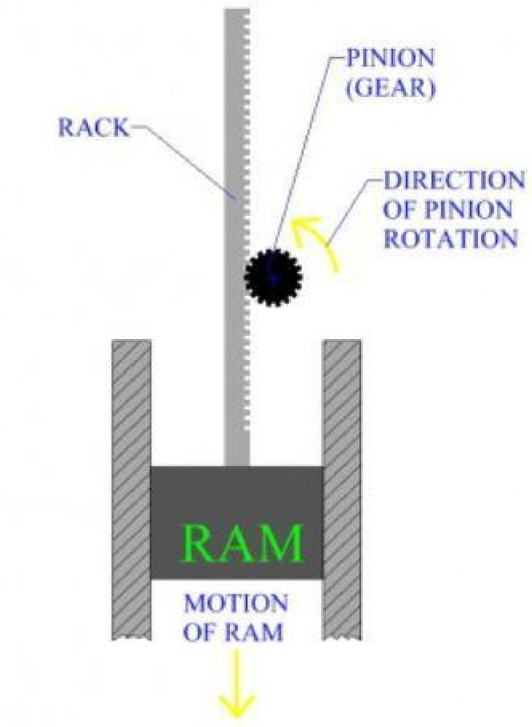
b) Knuckle Joint Press –

The knuckle joint press translates the energy of a motor through a powerful linkage design, and is capable of delivering a tremendous amount of force. The drive shaft crank rotates completely. The links are well grounded to support such pressure.



c) Rack and Pinion Press –

The rack and pinion press delivers the motors energy from a gear directly connected to the drive shaft. The rack is actually a round gear of infinite radius. A rotating gear (pinion), provides force through the rack. This gives the one-dimensional, translational motion desired of press machines.



Advantages of the Eccentric Shaft over other mechanisms:

Eccentric mechanism	**Knuckle Joint mechanism**	**Rack and Pinion mechanism**
Best suited for punch (shear) operations as it suites the high velocity requirement criteria	Speed is low at the bottom dead centre, best suited for stamping, cold forging	Cannot generate enough force to shear the metal.
Relatively compact mechanism	Mechanism itself takes too much space	Occupies large space as well
Upward motion of the ram is easily obtained	Upward motion of mechanism requires displacement of huge mechanism/links.	Direction of rotation of needs to be changed for ram to change its direction

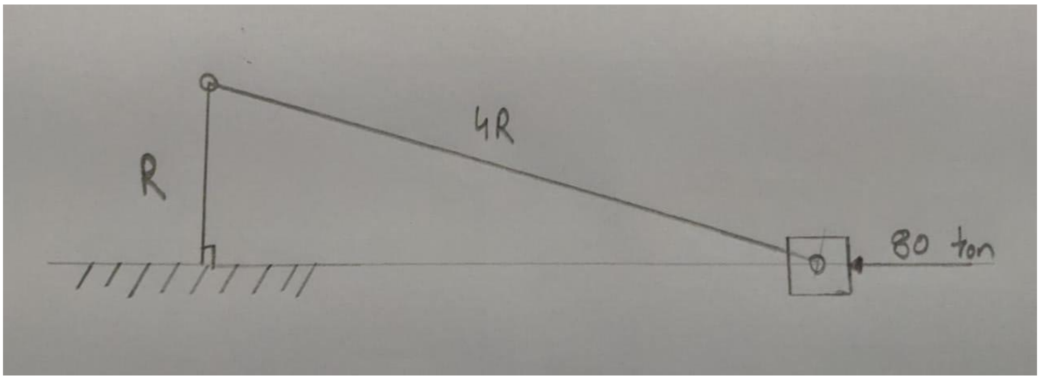
Hence, the eccentric mechanism suits well to the requirement. We proceed the design with an eccentric mechanism.

6.Design of Mechanism:

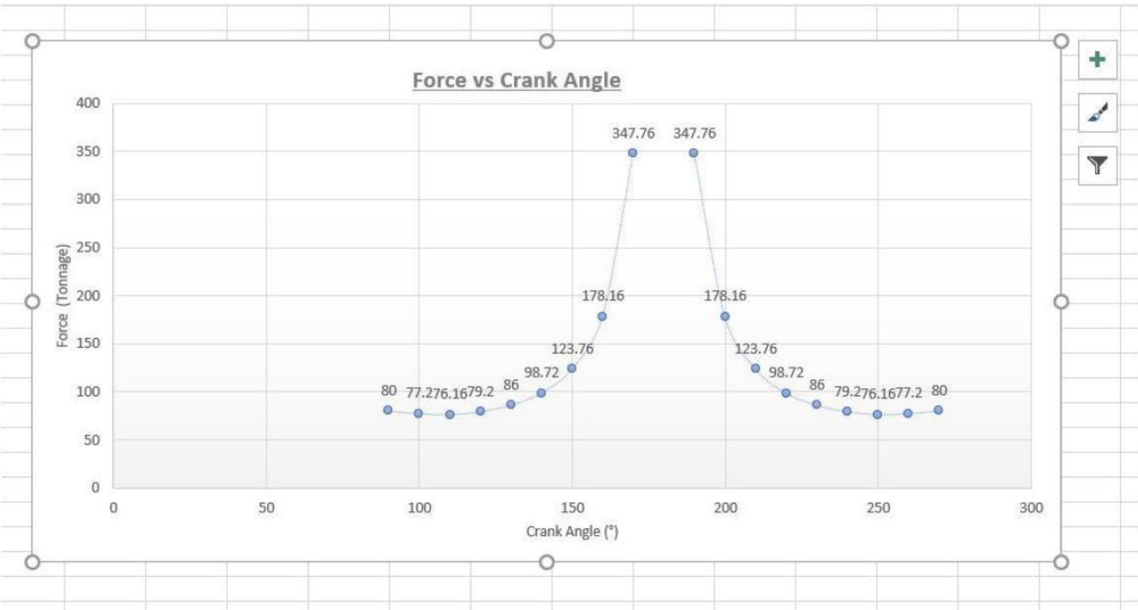
The eccentric press operates on an eccentric mechanism, that is inherently similar to that of a slider crank mechanism.

So, if we analyse the slider crank mechanism, we can analyse our eccentric mechanism.

Talking L/R as 4 ,



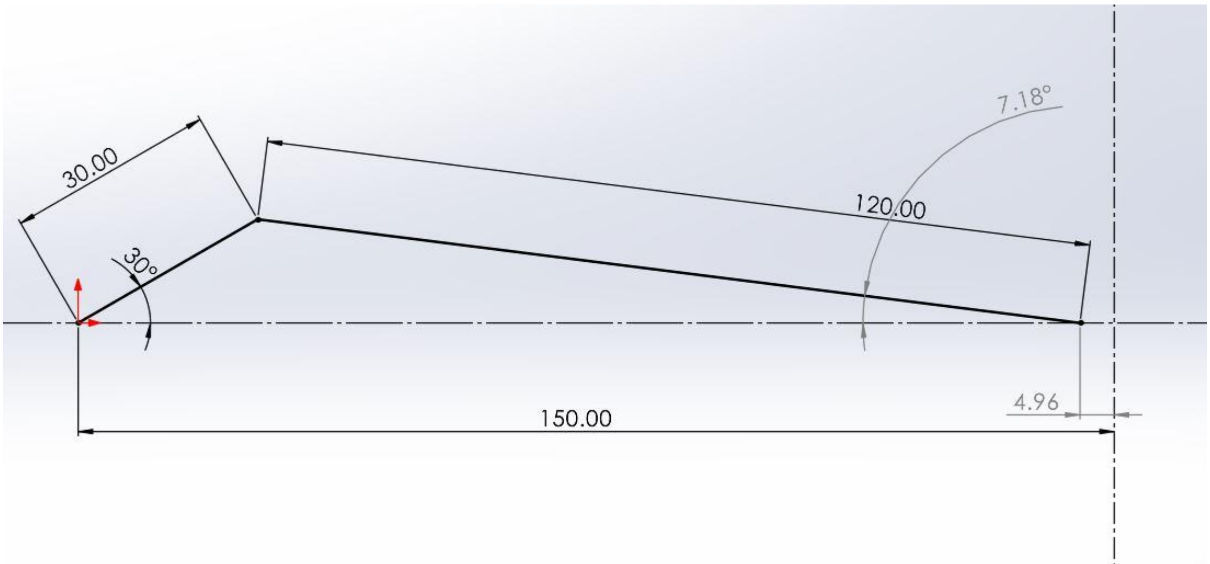
We are taking the above position as reference (as no mechanical advantage (input force = output force)). From this we did force analysis for crank angle from 90° to 270°, and the below graph was plotted.



From the above plot, we can see that in region 150 – 210 we are getting enough mechanical advantage (1.547 times input force). So, we are preferring that range.
For us,
Sheet thickness- **1 mm**.
Clearance – **1mm**.
Punching clearance – **2mm** (penetration after punch).

So, useful stroke is **4mm**.

Schematic of mechanism



For crank angle 150 to 180,

At 150, force on slider is = 1.547*80 ton
= 123.76 ton

Region for mechanical advantage is = 4R-3.82R
= 0.18R

Required region,
0.18R = 5mm
R = 27.7 mm
R ~ 30mm.

Crank radius = 30 mm.
So, Stroke = 2R = 60mm.

Stroke = 60 mm.

7.Calculation Design Load:

We are punching a metal sheet of size (270*400)mm with maximum 450 holes.

Diameter of hole = 3mm.

Shear area = 450*2*π*1.5*1 =4241.15 mm^2.

Material of sheet = Aluminium.

Shear strength of material = 185 MPa.

This implies that force required to shear aluminium is

F= Shear area *shear strength
= 4241.15*185
= 784612.75 N
F= **784.612 kN**.

So tonnage = (784.612)/9.81
= 79.98
≈ 80
So the tonnage of the machine is **80 tons**.

8.Calculation of Power and Speed Requirement:

Since we are using a flywheel. We can use average power for our requirement.

F = Force applied in punching= 80*9.81 kN.
t = thickness of sheet = 1 mm.

As per the production rate of baking sheets, we are choosing the production rate of 30 punches per minute.

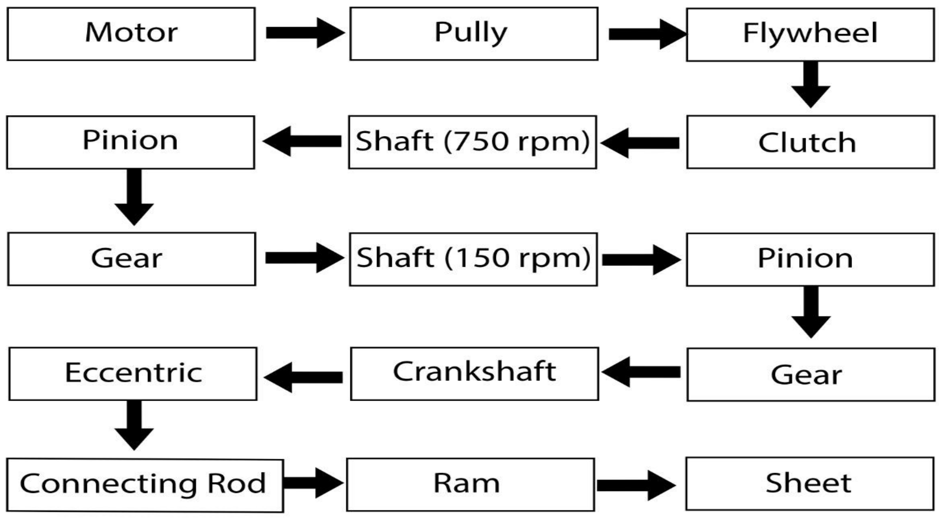
This implies that the speed of crankshaft = **30 rpm**.
So time required per cycle is = 60/30.
Cycle time = **2 sec**.

For power transmission we have 2 gear pair drives and a belt drive system.
Considering approximate efficiency of gear drive as 90% and belt drive as 80%,

Power = (Workdone required for punching)/ (Cycle time*efficiency)
= (F*t)/(2*0.8*0.9*0.9)
= (80*9.81*1)/2 W
= 650 W
So power required is **650 Watts**.

We are choosing a motor of of 1 HP = 745 watts

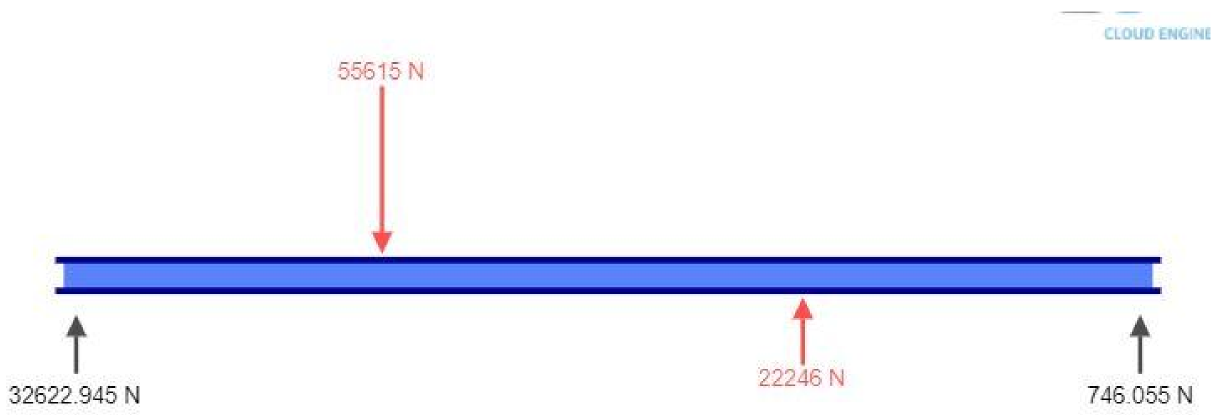
9.Power Transmission Chain:

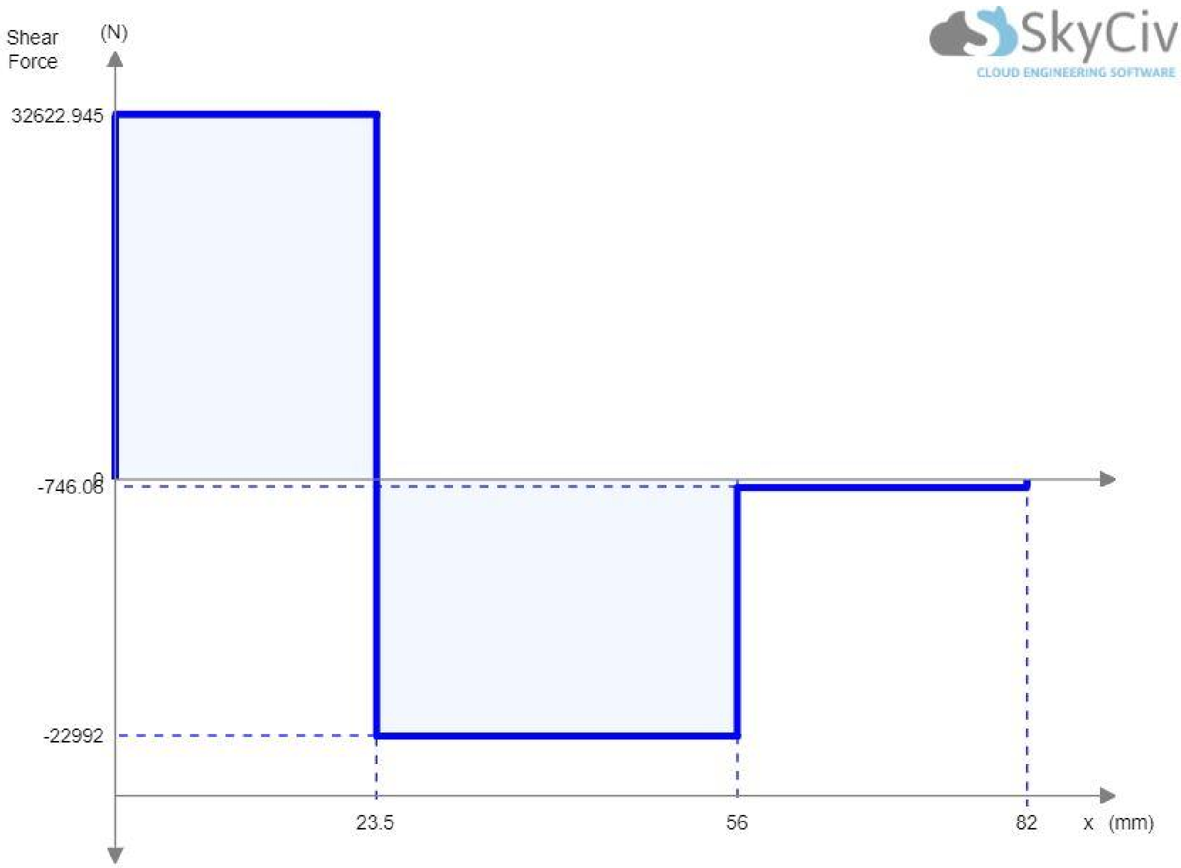


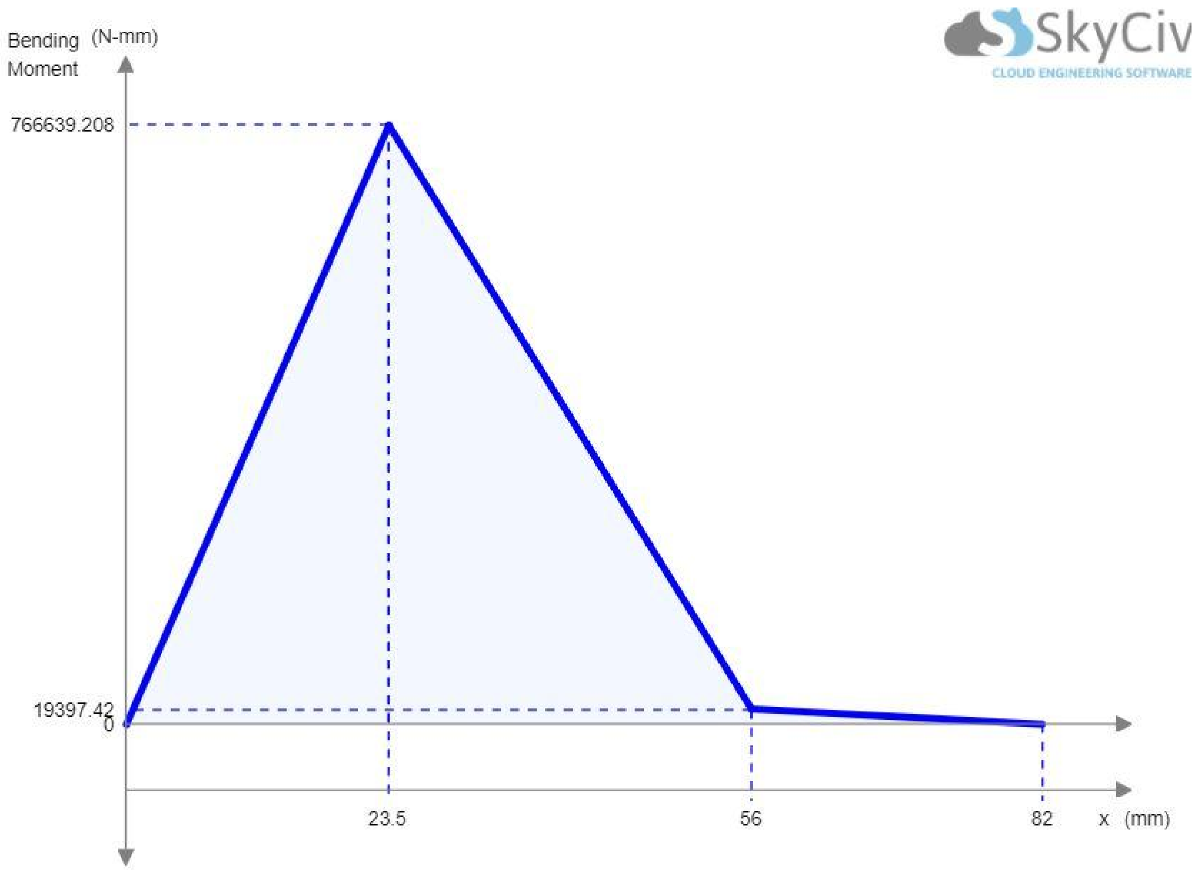
10.Force Analysis:

Analysis of 150 rpm Shaft

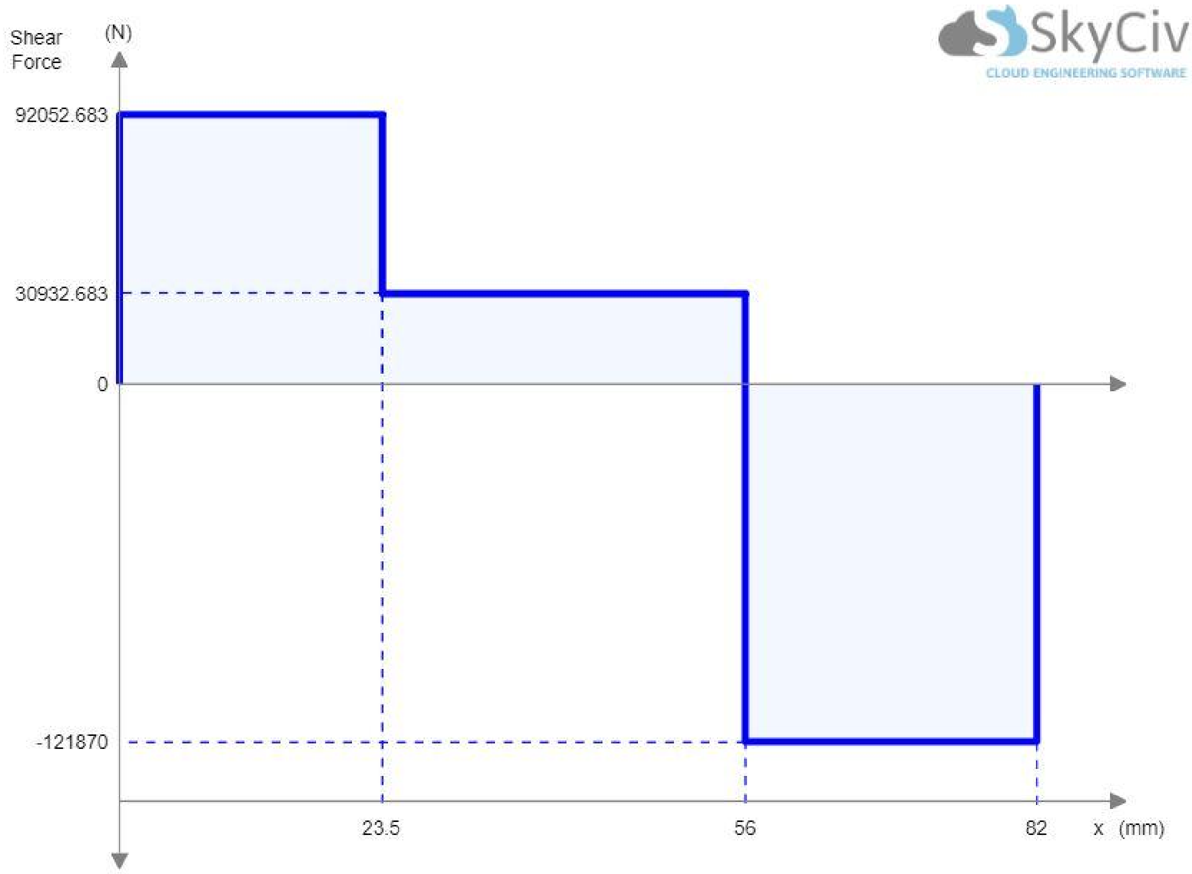
calculation of shear force on shaft on vertical direction

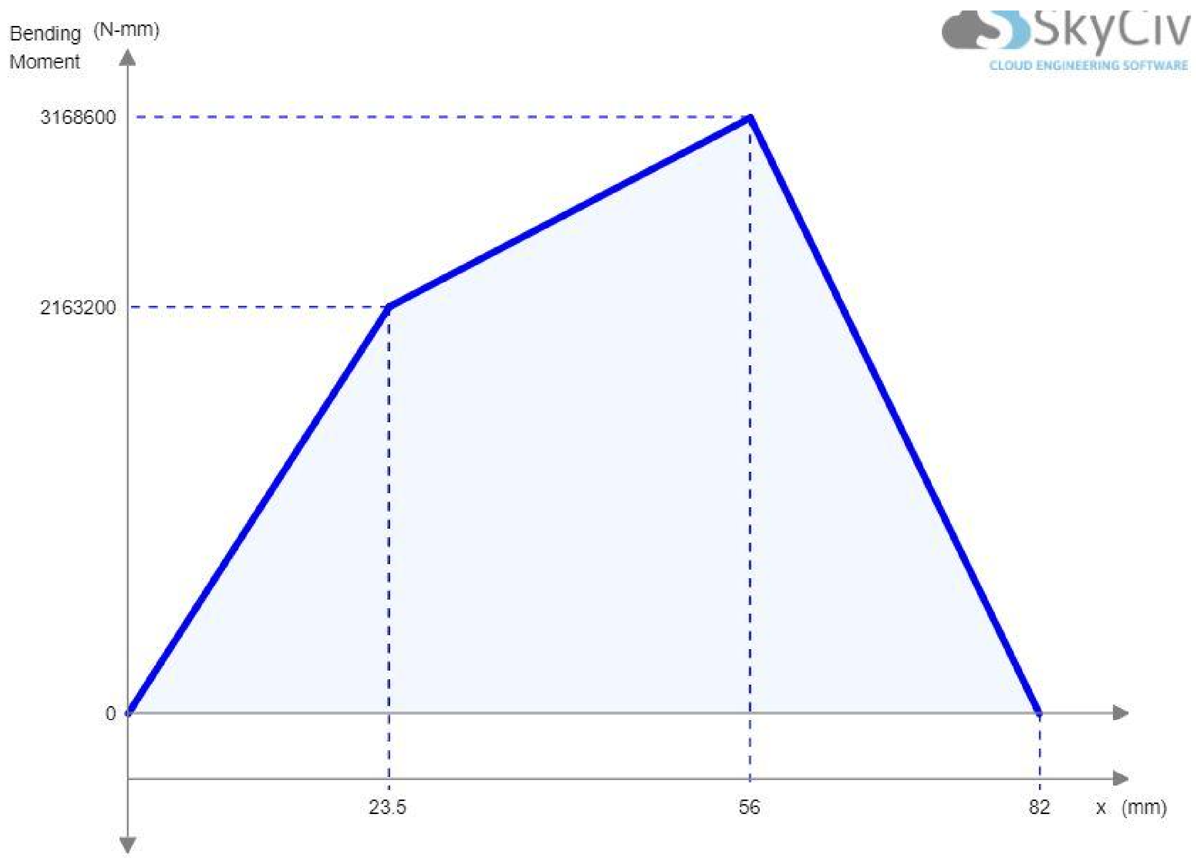



Shear
Force
(N)
32622.945
0
-746.08
-22992
23.5
56
82
x (mm)


Bending
Moment
(N-mm)
766639.208
19397.42
0
23.5
56
82
x (mm)

<u>Calculation of 150 rpm shaft in horizontal direction</u>



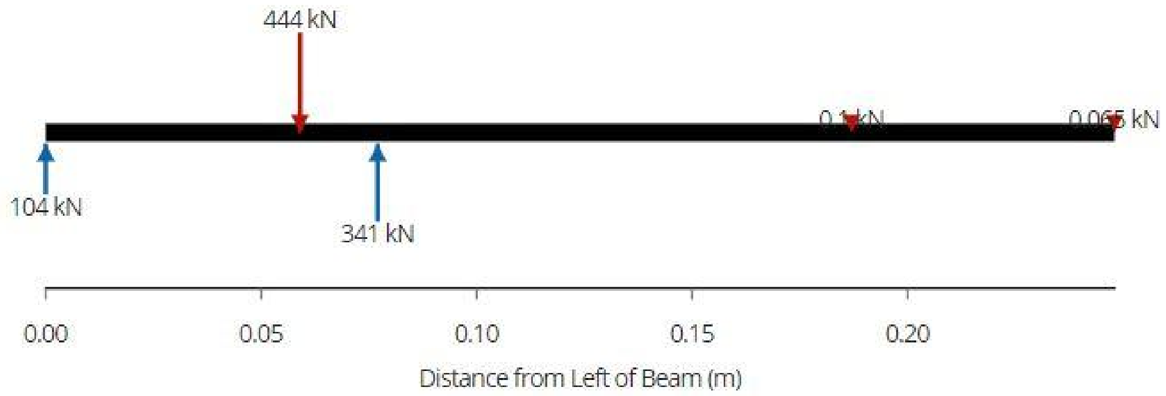


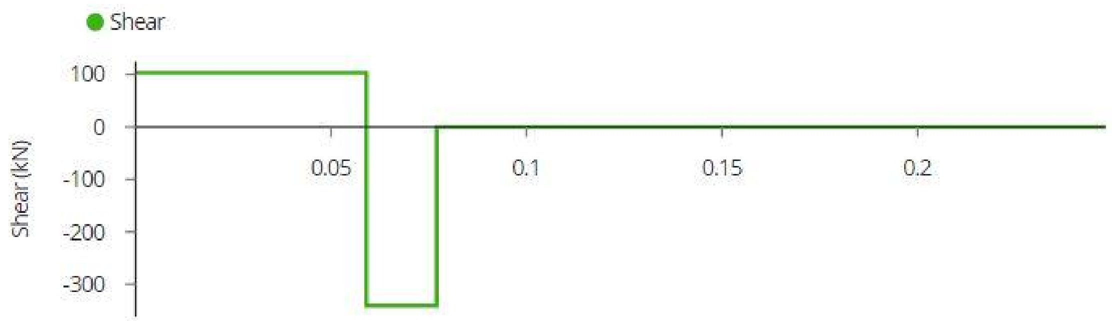
Hence net maximum bending moment on shaft (150 rpm) is 3169 kN-m

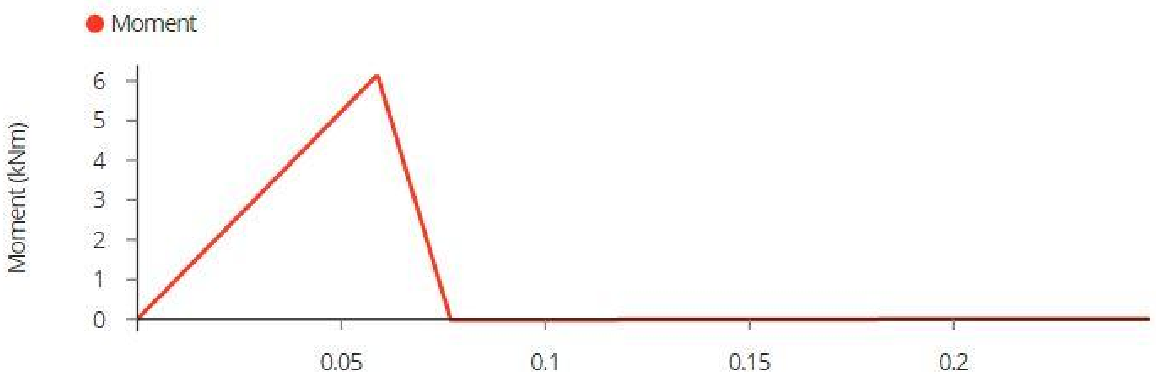
<u>Calculation for 750 rpm Shaft</u>

Forces in vertical directions are much more than in horizontal so we are neglecting horizontal forces

calculations in vertical direction are as follows:



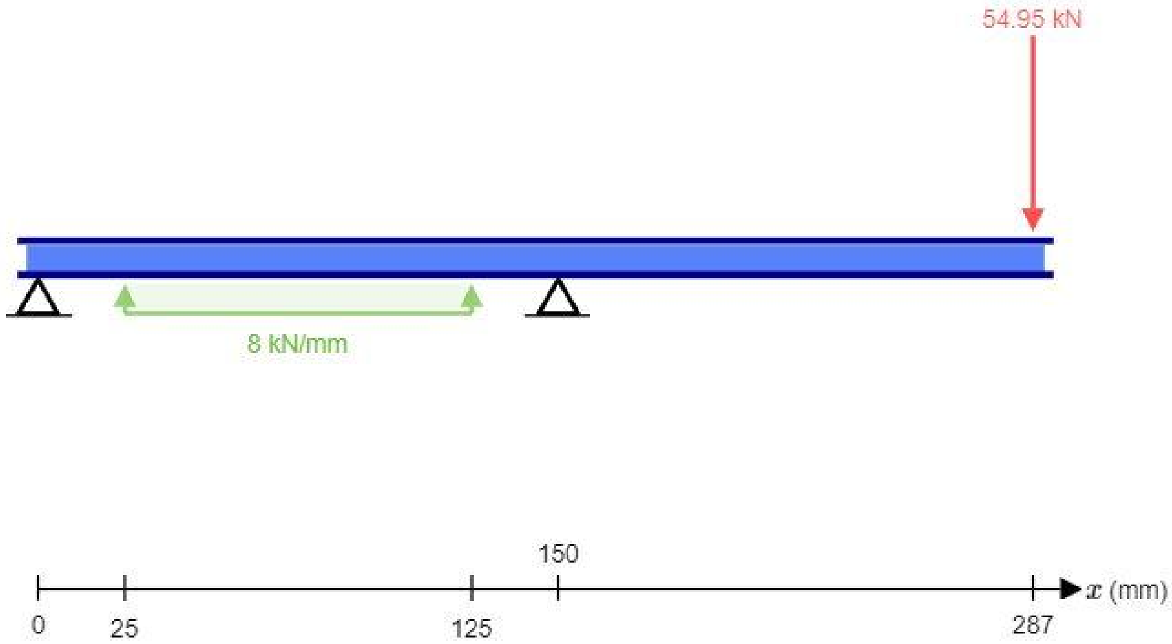


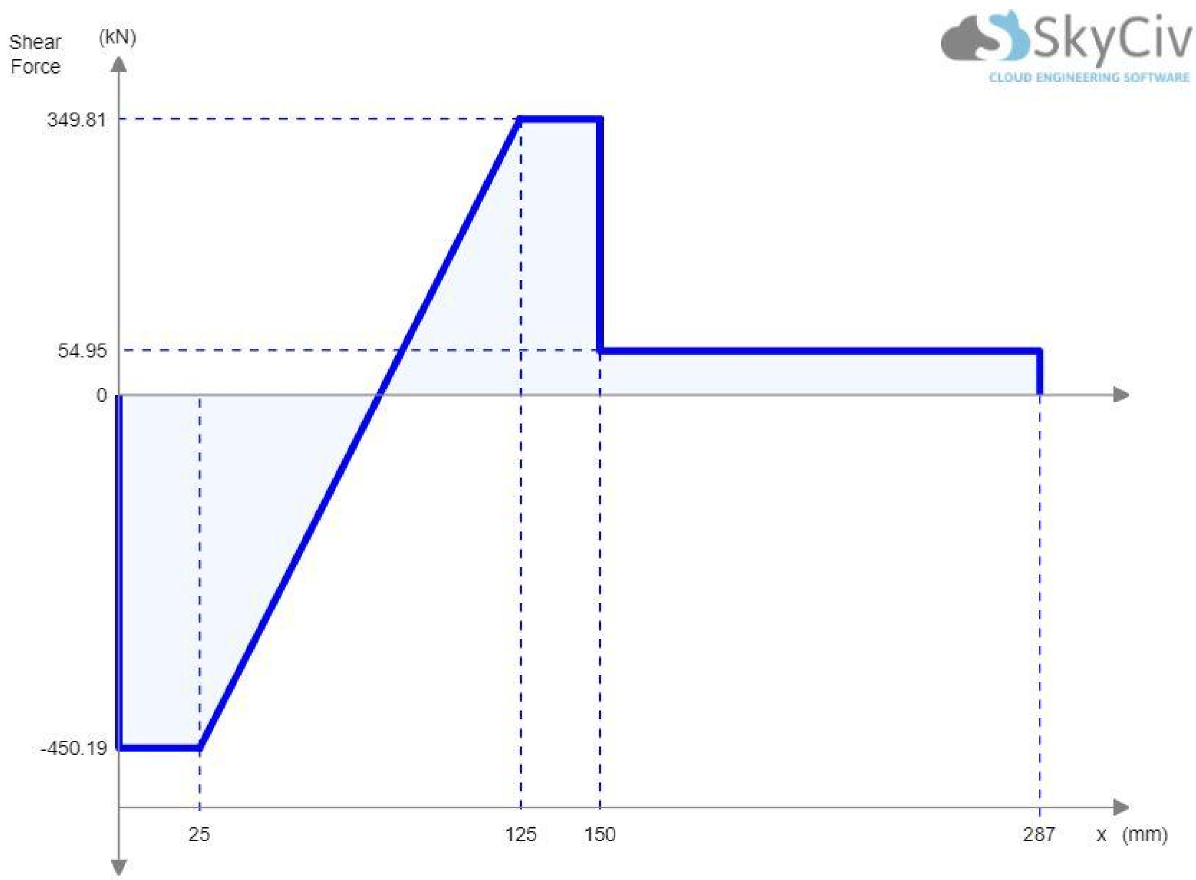


Maximum bending moment on shaft is 6.5 kN-m

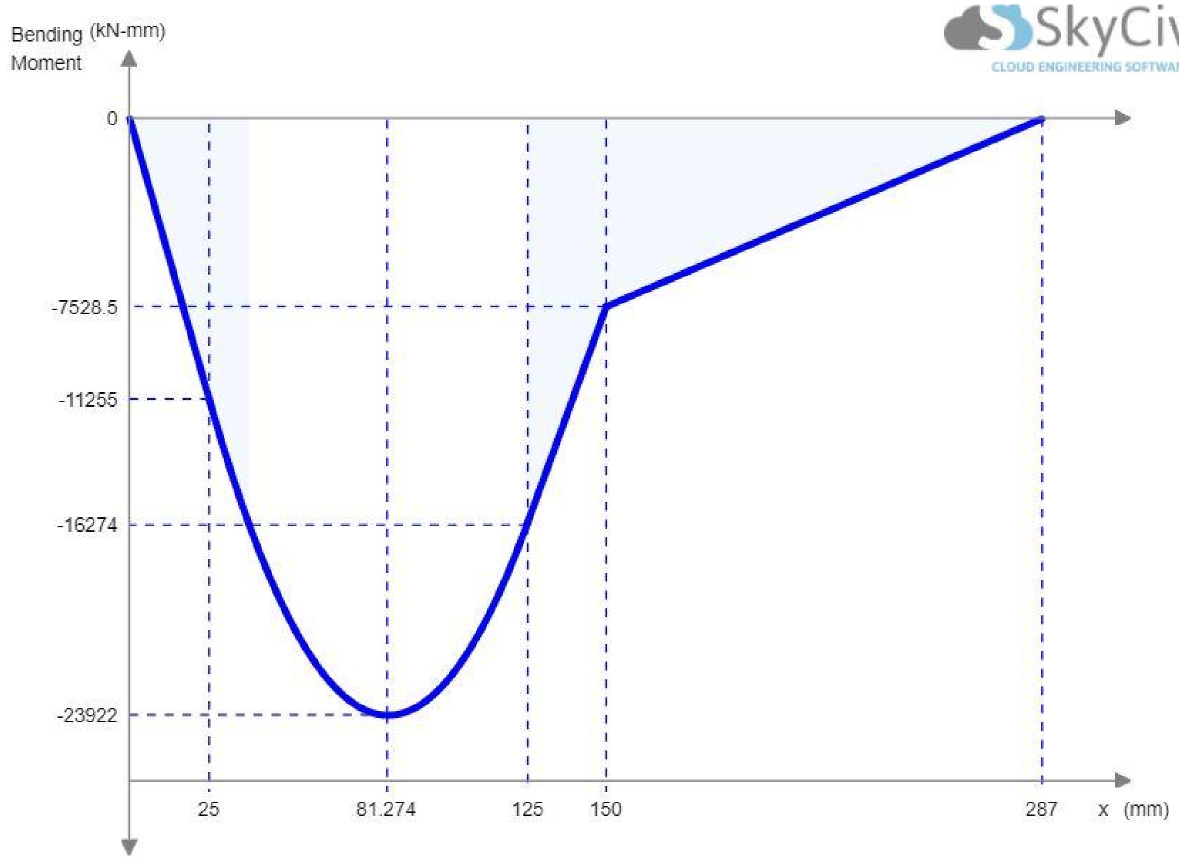
Analysis of 30 Rpm Shaft

Calculation in vertical direction

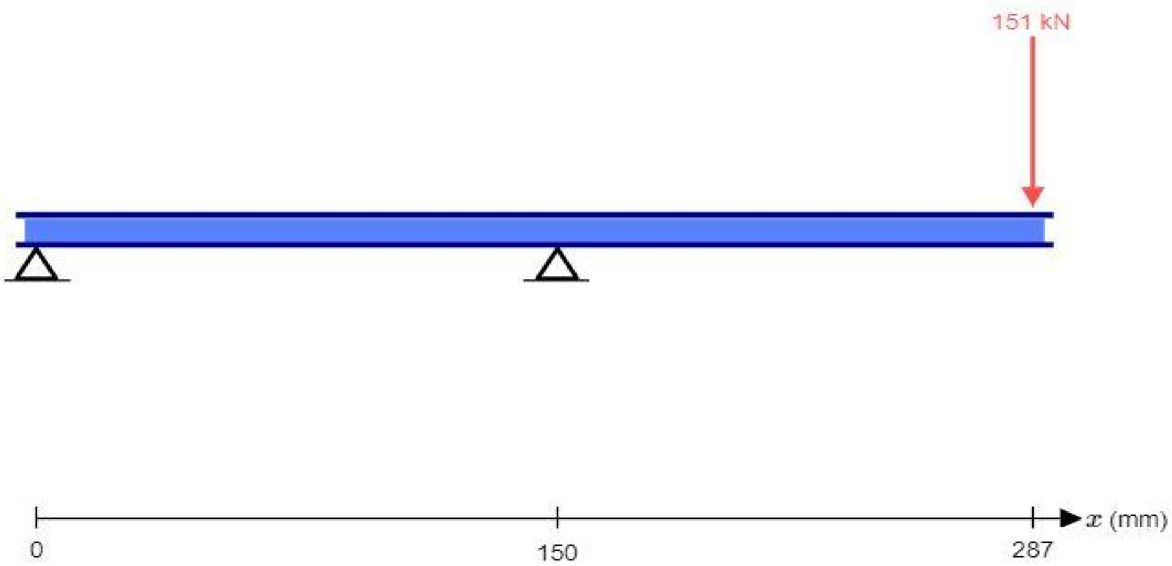


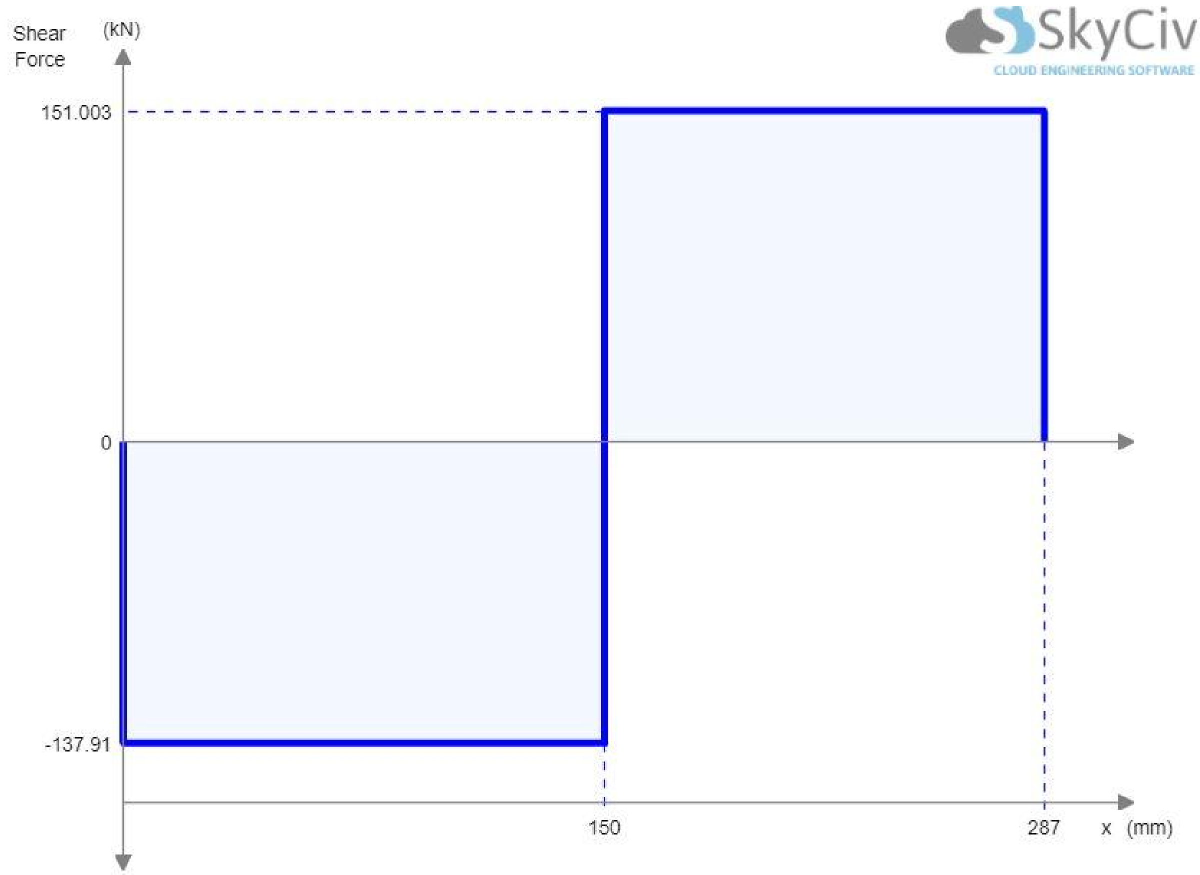


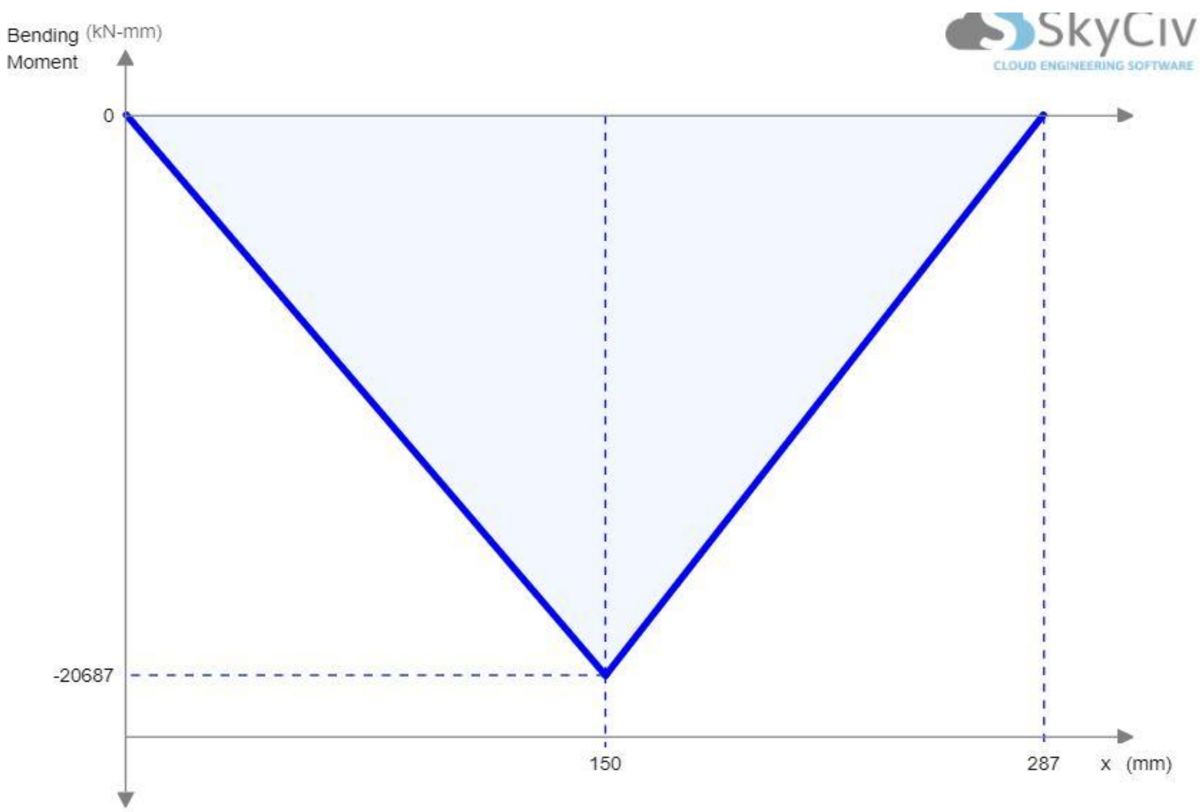
CCCCCal



<u>Calculation in Horizontal direction</u>







Hence Net Bending moment on shaft is 26401 kN-mm

11. Design of machine elements:

This section contains designs of various machine elements used in the machine.

1. Design of 1st Gear Pair
2. Design of 2nd Gear Pair
3. Design of flywheel
4. Selection of belt and design of pulley
5. Design of shafts
6. Design of pin
7. Design of eccentric and connecting rod assembly
8. Selection of T-slot bed

11.1 Design of 1st Gear Pair

Power to be transmitted – 0.745 kW

N1 = Input shaft rpm (pinion) = 750 rpm

N2 = Output shaft rpm (gear) = 150 rpm

Tooth profile – Involute

Pressure Angle (α)- 20°

Type of tooth – Full Depth

i = gear ratio = 750/150 = 5

Z1 = 25

Z2 = Z1*i= 25*5= 125

Y = Lewis form factor

For pinion,

$Y_1 = \pi*(0.154-0.912/25) = 0.369$

For gear,

$Y_2 = \pi*(0.154-0.912/125) = 0.460$

Material Selection –

	Material	Bending strength (MPa)	Crushing strength (MPa)
Pinion	Alloy Steel 40 Ni 2 Cr 1 Mo 28	400	1100
Gear	Alloy Steel 15 Ni2 Cr 1 Mo 15	320	950

For identifying weaker member,

$S_1 = 400^* Y_1 = 400^*0.36 = 144$ N/mm^2

$S_2 = 320^* Y_2 = 320^*0.460 = 147.48$ N/mm^2

This implies that pinion is weaker member. So, we design for pinion.

1/3

For gear,

[Mt] = ((P*60)/(2π*N2))*K = ((745*60)/(2π*750))*1.5 = 14.228 N.m = 14228 N.mm

Putting all the values in formula for module, we get

m = 1.02

increasing by 20%,

m = 1.224

selecting standard value of m from databook,

m = 1.25 mm

Checking for contact stresses,

σ_c = 0.74 * ((i + 1)/a) * (((i+1)/i*b) *E *[Mt])

a = centre distance = m*(Z1+Z2)/2 = 67.5 mm

$b = 20$ mm

$E = 2.15 * 10^5$ N/mm^2

[Mt] = 14228 N.mm

This implies that

$\sigma_c = 891.14$ N/mm^2

$\sigma_c < 1100$ N/mm^2

This means that design is safe.

Checking of dynamic load,

$F_s = S_1 * b * m = 130 * 25 * 2.5 = 2.437$ kN.

$F_t = 2Mt / d1 = 2*12414 /22.5 = 1103.46$ N.

$V = \pi*d1*N1/60 = 0.8835$ m/s

$C_v = (3+V)/3 = 1.2945$

$F_d = F_t * C_v = 1428.42$ N

$F_d < F_s$

This means that design is safe in dynamic loading.

Checking for wear load,

$F_w = d1*Q*b*k$

$Q = (2*i)/(i + 1) = 10/6 = 1.667$

$k = (\sigma_c{}^2*\sin(\alpha) *(2/E))/1.4 = 2.099$

$F_w = 45*1.667*25*1.575 = 1.575$ kN

$F_w > F_d$

Hence design is safe in wear loading.

Summarization

Module	m	1.25 mm
Teeth on pinion	Z1	25
Teeth on gear	Z2	125
Width	b	20 mm
Centre distance	a	67.5 mm

11.2 Design of 2 Gear Pair

Power to be transmitted – 0.745 kW

N1 = Input shaft rpm (pinion) = 150 rpm

N2 = Output shaft rpm (gear) = 30 rpm

Tooth profile – Involute

Pressure Angle (α)- 20°

Type of tooth – Full Depth

i = gear ratio = 150/30 = 5

Z1 = teeth on pinion = $2F_o/\sin^2(\alpha) = 2/\sin^2(20) = 17.097 = 18$

Z2 = Z1*i= 18*5= 90

Y = Lewis form factor

For pinion,

$Y_1 = \pi*(0.154-0.192/18) = 0.3246$

For gear,

$Y_2 = \pi*(0.154-0.192/90) = 0.4519$

Material Selection –

	Material	Bending strength (MPa)	Crushing strength (MPa)
Pinion	Alloy Steel 40 Ni 2 Cr 1 Mo 28	400	1100
Gear	Alloy Steel 15 Ni 2 Cr 1 Mo 15	320	950

For identifying weaker member,

$S_1 = 400^* Y_1 = 400^*0.325 = 130$ N/mm^2

$S_2 = 320^* Y_2 = 320^*0.451 = 144.32$ N/mm^2

This implies that pinion is weaker member. So, we design for pinion.

$$m = 1.26 * \left(\frac{[Mt]}{\text{Y}*\text{bending strength}*\varphi*\text{Z}}\right)^{1/3}$$

For pinion ,

[Mt] = ((P*60)/(2π*N1))*K = ((745*60)/(2π*150))*1.5 = 71.014 N.mm

= 71014 N.mm

φ = 10

Putting all the values in formula for module, we get

m = 1.744

increasing by 20%,

m = 2.09

selecting standard value of m from databook,

m = 2.5

Checking for contact stresses,

$\sigma_c = 0.74 * ((i + 1)/a) * (((i+1)/i*b) *E *[Mt])$

a = centre distance = m*(Z1+Z2)/2 = 135 mm

b = 10*m = 25 mm.

$E = 2.15 * 10^5$ N/mm^2

[Mt] = 71014 N.mm

This implies that

$\sigma_c = 890.4$ N/mm^2

$\sigma_c < 1100$ N/mm^2

This means that design is safe.

Checking of dynamic load,

$F_s = S_1 * b * m = 130 * 25 * 2.5 = 8.125$ kN.

$F_t = 2Mt / d1 = 2*41380 /45 = 1839.11$ N.

$V = \pi*d1*N1/60 = 0.3534$ m/s

$C_v = (3+V)/3 = 1.1178$

$F_d = F_t* C_v = 2055.757$ N

$F_d< F_s$

This means that design is safe in dynamic loading.

Checking for wear load,

$F_w = d1*Q*b*k$

Q = (2*i)/(i + 1) = 10/6 = 1.667

$k = (\sigma_c{}^2*\sin(\alpha) *(2/E))/1.4 = 1.575$

$F_w = 45*1.667*25*1.575 = 2.953$ kN

$F_w > F_d$

Hence design is safe in wear loading.

Summarization

Module	m	2.5 mm
Teeth on pinion	Z1	18
Teeth on gear	Z2	90
Width	b	25 mm
Centre distance	a	135 mm

11.3 Design of Flywheel

$\Delta E = I*K_s*\omega_m{}^2$

Working stroke – 6/360 = 1/60

Energy is stored in flywheel during 59/60 part of stroke.

ΔE = (59/60)* work done = (59/60)*784.5 = 771.425 J

$K_s = 0.2$ $\omega_m = 2\pi N/60 = 78.53$ rad/s

This implies that,

$I = \Delta E/(K_s*\omega_m{}^2) = 0.625$ kg-m^2

$I_{rim} = I*0.9$

So, $I_{rim} = 0.5625$ kg m^2

For power <100 HP, v<1500 m/min

Hence choosing v=1250m/s so as to get optimum value for flywheel diameter.

$D_o = v/\pi*N \Rightarrow D_o$=530.5 mm And assuming D_m = 480mm

Now calculation of mass of flywheel.-

$I=m*Rm^2 \Rightarrow m$=9.76kg.

$m=\rho*\pi*b*h*D_m \Rightarrow b*h$=898.93 mm^2

taking h=b/2 for maximum possible width of the flywheel.

we get b= 42.4 mm and h= 21.2 mm

$D_o = D_m + 2h \Rightarrow D_o = 522.4$mm and consequently v=1230.81m/min

Checking for Centrifugal and Total stress--

$\sigma_c = \rho * v^2 \Rightarrow \sigma_c = 3.12$ MPa (< 8Mpa; hence safe.)

$\sigma_b = \rho\pi^2 v^2 D_m / n^2 h \Rightarrow \sigma_b = 18.782$ MPa

$\sigma_{total} = 0.75\sigma_c + 0.25\sigma_b \Rightarrow \sigma_{total} = 7.03$MPa (safe)

<u>Design for Flywheel Hub and Arm</u>

Mean Torque- T_m = Power/radial velocity

$\Rightarrow T_m = 745/78.53 = 9.48$ N-m

$T_{max} = 59 * T_{mean} \Rightarrow T_{max} = 560$ N-m

Hub Diameter and Length-

By torsion $T_{max} = \pi d_s^3 \tau_{max}/16 \Rightarrow d_s = 15.2$mm

taking $d_s = 46$mm

$D_h = 2 * d_s \Rightarrow D_h = 92$mm also $L_h = 101.2$mm

Cross Section of Arms -

Ellipse cross section with Major axis = a and Minor axis = c.

Bending moment of arm-

$\sigma_b = T_{max}(D_m - D_h)/nzD_m$ $\sigma_b = 15$MPa

z= 5029.63mm^3

a= 47mm c=23.5mm

$\sigma_b = (T_1 - T_2) * (D_m - D_h)/2nZ \Rightarrow \sigma_b = 0.2$MPa

$\sigma_t = 8 * h * b * \rho * v^2 / n * c * a$

$\sigma_t = 3.28$MPa

σ_{total} = 18.48 MPa Safe Design as it is less than 19.62MPa.

11.4 Selection of belt and pulley design

Belt drive will be used in

Driving pulley – 1440 rpm to Driven pulley(flywheel) – 750 rpm

Power to be transmitted = 0.745 kW

This implies that cross section of belt = A

Time of operation – 12 hrs/day.

Speed ratio = 1440/750 = 1.92

Diameter of flywheel = D2 = 480 mm

Diameter of pulley = 530/1.92 = 250 mm

Centre distance C= 540 mm

For smaller pulley,

$V_p = \pi * D_1 * N_1/60 = 18.85$ m/s

$\Theta = \pi - (D_2 - D_1)/C = 2.716$ rad $= 155.61° = 156°$

Correction factors

(a) Equivalent Diameter

$D_2/D_1 = 1.92$

$K_e = 1.13$ (from databook)

$D_e = 250 * 1.13 = 282.5$ mm.

From databook,

Rating of belt = 3 kW

(b) Arc of Contact Factor

Rating = $3 * F_d = 3 * 0.935 = 2.805$ kW

(c) Industrial Service Factor

Running for 12 hrs/day

K = 1.2

Rating = 2.805/1.2 = 2.338 kW

(d) Length Correction Factor

Length of belt = $L = 2C + \pi*(D1+D2)/2 + (D2-D1)^2/4C$

$$L = 2251.17 \text{ mm} = 2252 \text{ mm}$$

From data book, L is close to pitch length of 2195 mm

This means that Nominal Inside length of belt = 2159 mm

Designation of Belt – **A 2159 / 85 - 27**

Rating = 2.338/ 1.1 = 2.16 kW

No of belts = 0.745/2.16 ≈ 1 belt.

1 Belt of designation A 2159 / 85 - 27 will be used.

No of belts = 1

So no of grooves on pulley = 1

From databook,

For cross section A,

Pulley dimensions are as follows,

Pitch width = l_p = 11 mm.

Distance down to pitch line = b = 6.6 mm

Pulley pitch diameter = d_p = 250 mm

Angle A = 38°

Depth below pitch line = 17.4 mm

11.5 Design of Shaft

30 rpm shaft

$P = 2 \pi N M_t/60$

Mt= 17000 N-m

Max Mb= 26401 N-m

$d^3 = 16(Mb^2+Mt^2)^{1/2}/\pi\ \tau$

d=58 mm

d = 60 mm (FOS=1.14)

<u>150 rpm shaft</u>

P= 2 π N Mt/60

Mt= 3437.74 N-m

Max Mb= 3169 KN-mm

$d^3 = 16(Mb^2+Mt^2)^{1/2}/\pi\ \tau$

d = 18 mm

d =20 mm (FOS=1.33)

<u>750 rpm Shaft</u>

P= 2 π N Mt/60

Mt= 687.4 N-m

Mb= 6500 N-m

$d^3 = 16(Mb^2+Mt^2)^{1/2}/\pi\ \tau$

d = 35 mm

d = 40 mm (FOS= 1.6)

11.6 Design of pin

Material of the pin is AISI 303 stainless steel.

Shear Strength = 260 MPa

This pin will have to take double shear load.

Considering factor of safety as 1.5.

Allowable shear stress on pin = 260/1.3 = 200 MPa = 200 N/mm^2

Force (F) = 785 kN

Average shear stress in case of double shear for 3 rivets is

$\sigma = F/6A$, where A= Shear area = $(\pi/4)*d^2$ where, d= nominal diameter of rivet

$200 = 785000/6*(\pi/4)*d^2$

This implies that

$d = 29.6 \approx 30$ mm

Nominal diameter of the pin is 30 mm.

11.7 Design of Connecting rod

For eccentric,

D_{bore} = 50 mm (Diameter of crankshaft)
Eccentricity = 30 mm
D = 140 mm

Material of connecting rod is 4340 Chromium Alloy Steel

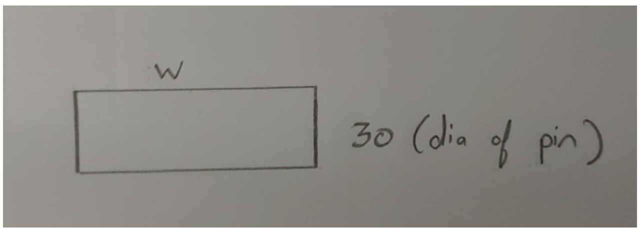
Yield Strength = 470 MPa

Taking Factor of safety as 2,

Allowable Stress = 235 MPa

Force = 785 kN

Projected area of cylindrical area which will take force, is a rectangle of size



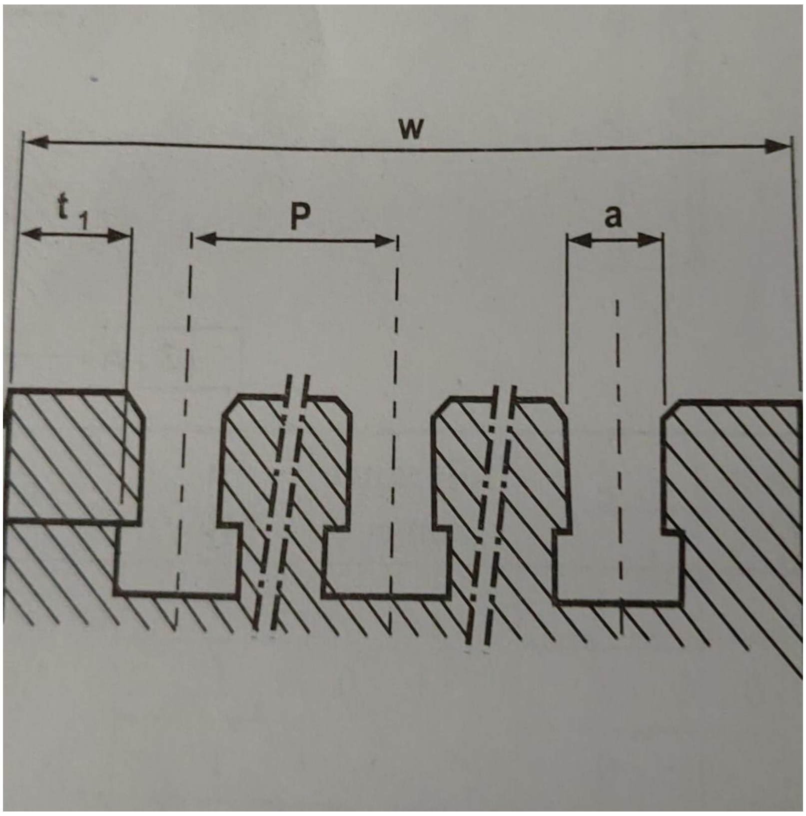
Stress = Force/Area

$235 = 785*10^3/(40*w)$

$w = 83.51$ mm ≈ 85 mm.

Thickness of the connecting rod is 110 mm.

11.8 Selection of T- slot table and T- nut

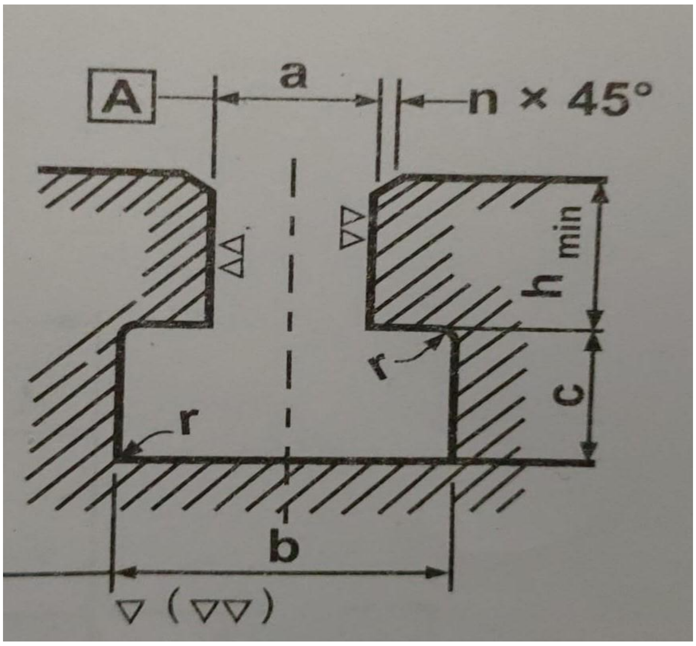


For us , Width of table = w = 300 mm

Using the standard values given in PSG data book,

a = 12 mm   P = 50 m

No of T slots = 6



Further details of selected t slot are

b = 19 mm

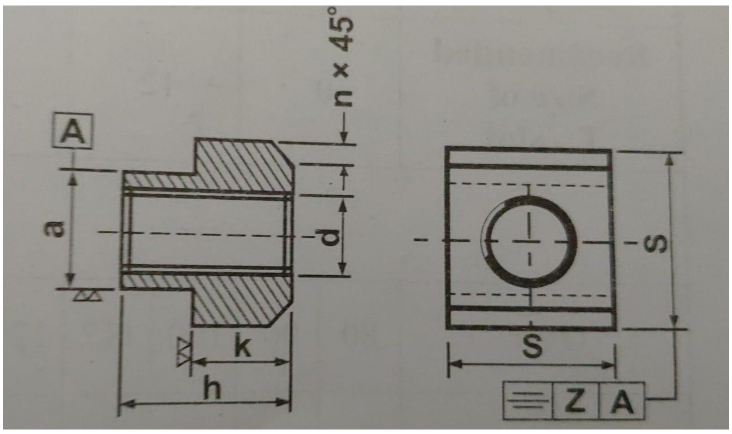
c = 8 mm

h = 12 mm

n = 1.6 mm

r = 0.5 mm

Details of T nut for selected T slot are:



d = M 8   S = 18 mm

h = 14 mm

k = 7 mm   n = 2.5 m

Part List

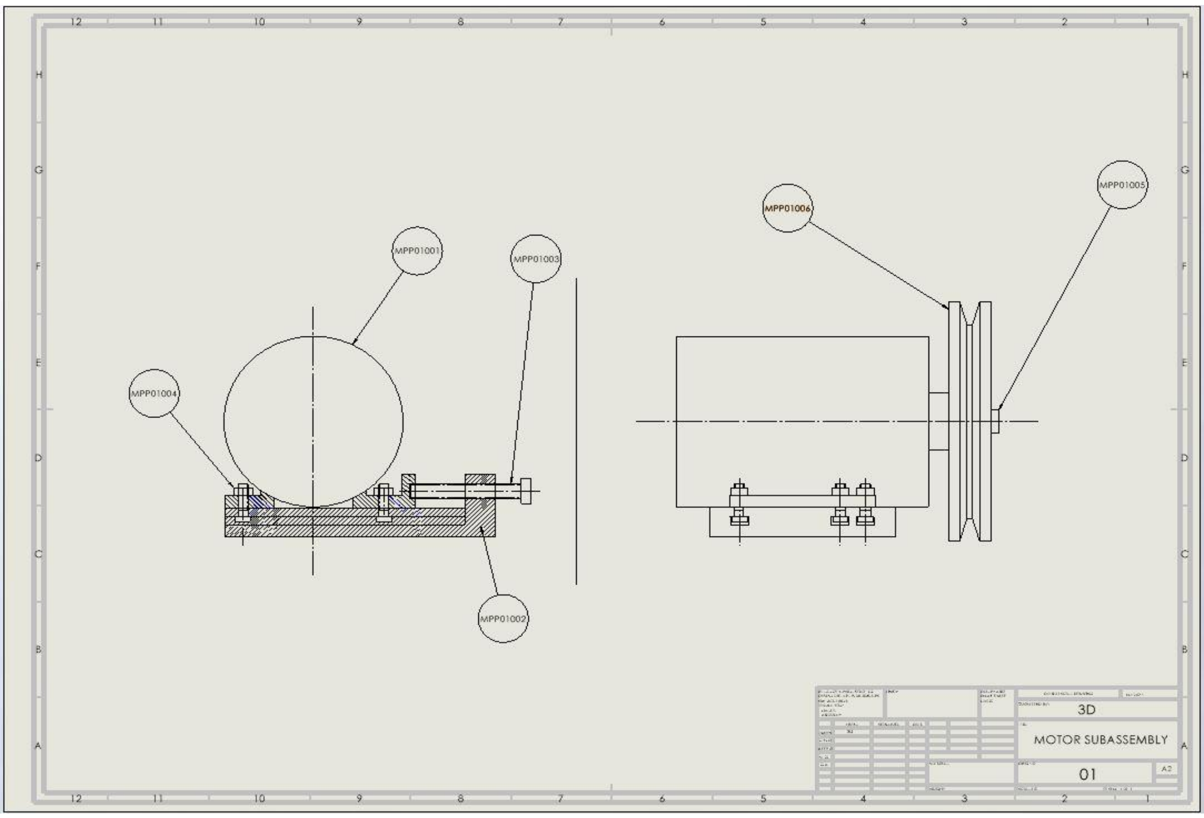
Sr no.	Component Number	Component Name	Material	Quantity	Drawing Number/ Specifications
1	MPP01001	Motor	-	1	-
2	MPP01002	Base Plate for motor	MS - zinc plated	1	-
3	MPP01003	Screw	Carbon Steel C60	1	M16 x 110
4	MPP01004	Taps	-	6	M10 x 28
5	MPP01005	Motor Shaft	AISI 4140 Alloy Steel	1	Φ 24 mm
6	MPP01006	Pulley	CI	1	1
7	MPP02007	Gib Head Key	C50 Carbon steel	1 1 1	8*7*50 14*9*110 14*9*80
8	MPP02008	Flywheel	CI	1	2
9	MPP02009	Clutch	-	1	Matrix Altra industrial motion clutch Model- Tooth clutch Series 5EC S.no: 5EC 055p
10	MPP02010	Pinion 1	AS 40 Ni 2 Cr 1 Mo 28	1	3
11	MPP02011	Ball Bearings	-	2	SKF6008
12	MPP02012	Plug	MS C018	1	4
13	MPP02013	Taps for plug	-	2	M2.5 x 26
14	MPP02014	Step Shaft	AISI 4140	1	5

			Alloy Steel		
15	MPP03015	Gear 1	AS 15 Ni 2 Cr 1 Mo 15	1	6
16	MPP03016	Pinion 2	AS 40 Ni 2 Cr 1 Mo 28	1	7
17	MPP03017	Ball Bearings	-	2	SKF6004
18	MPP03018	Plug	MS C018	1	8
19	MPP03019	Taps for plug	-	2	M2.5 x 18
20	MPP03020	Step Shaft	AISI 4140 Alloy Steel	1	9
21	MPP04021	Gear 2	AS 15 Ni 2 Cr 1 Mo 15	1	10
22	MPP04022	Eccentric	Carbon steel C-14	1	11
23	MPP04023	Bush	Brass	1	12
24	MPP04024	Connecting Rod	AISI 4340 Alloy Steel	1	13
25	MPP04025	Bush Bearings	-	2	PCM606550 E
26	MPP04026	Crank Shaft	AISI 4140 Alloy Steel	1	14
27	MPP04027	Pin	AISI 303 S.S.	1	15
28	MPP04028	Ram	CI	1	16
29	MPP04029	Ram Guides	CI	1	17
30	MPP04030	Bush for Ram guides	PTFE composite strips	1	PCMS 1005002.5 E

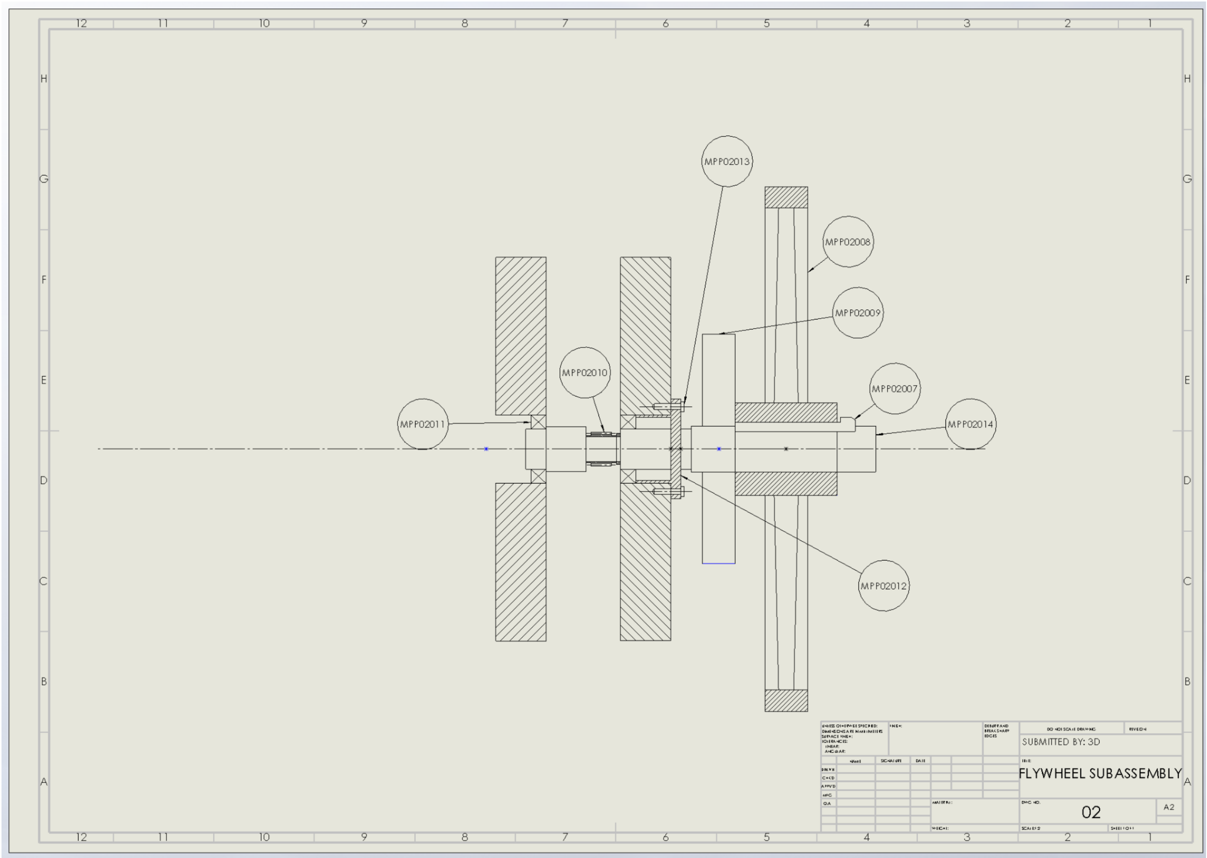
31	MPP04031	Taps for ram guide	-	2	M10 x 45
32	MPP05032	T slot	CI	1	18
33	MPP05033	T Slot Bed	CI	1	19
34	MPP05034	Taps for bed	-	2	M20 x 60
35	MPP04035	Plummer Block	CI	2	20
36	MPP04036	Sleeve	SAE 2100	1	21
37	MPP04037	Parallel Keys	Carbon Steel C50	1 2	18*11*50 for Eccentric. 6*6*14

12. Subassembly Drawings:

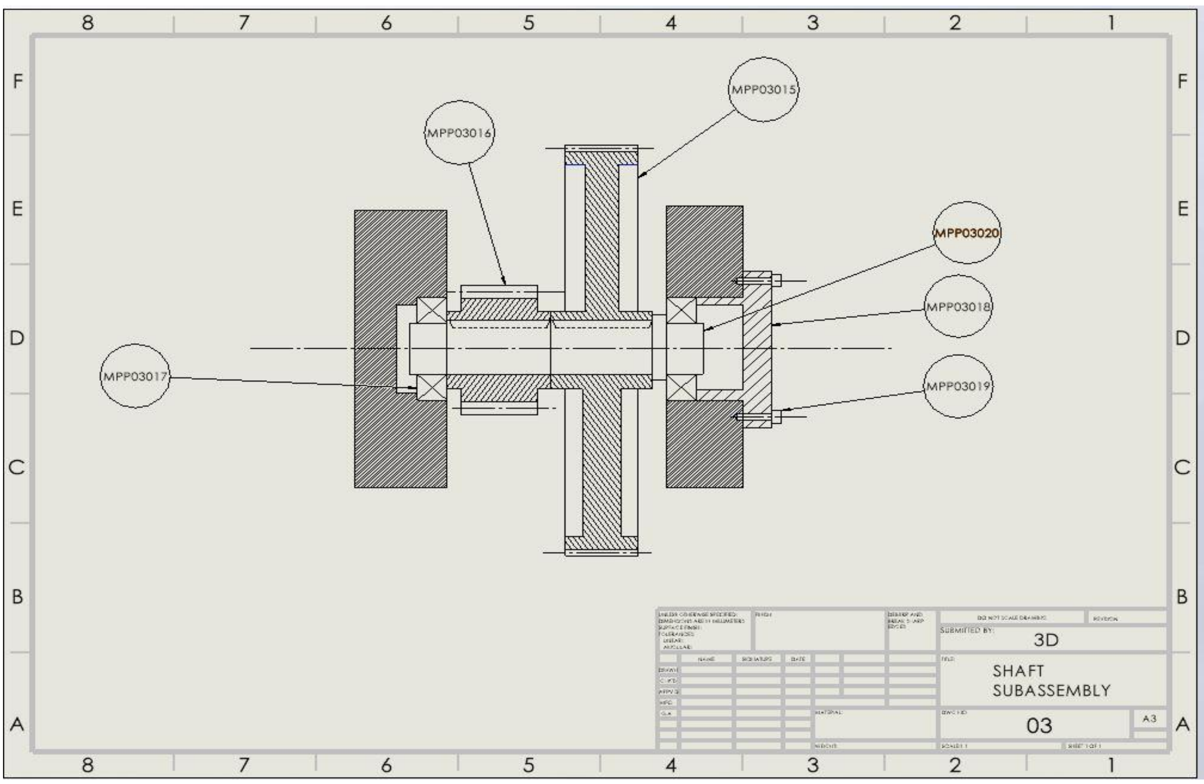
12.1 Sub Assembly of Motor and Pulley



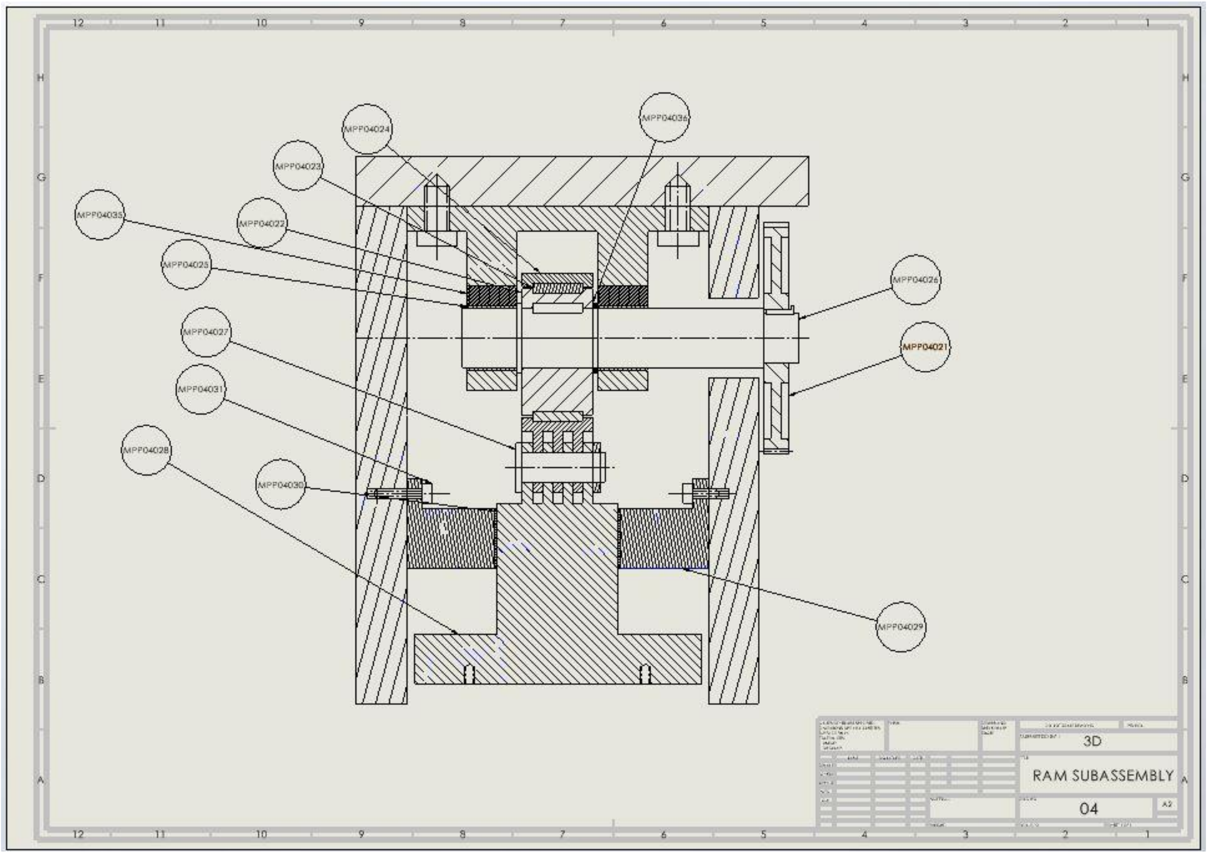
12.2 Sub Assembly of Shaft at 750 rpm



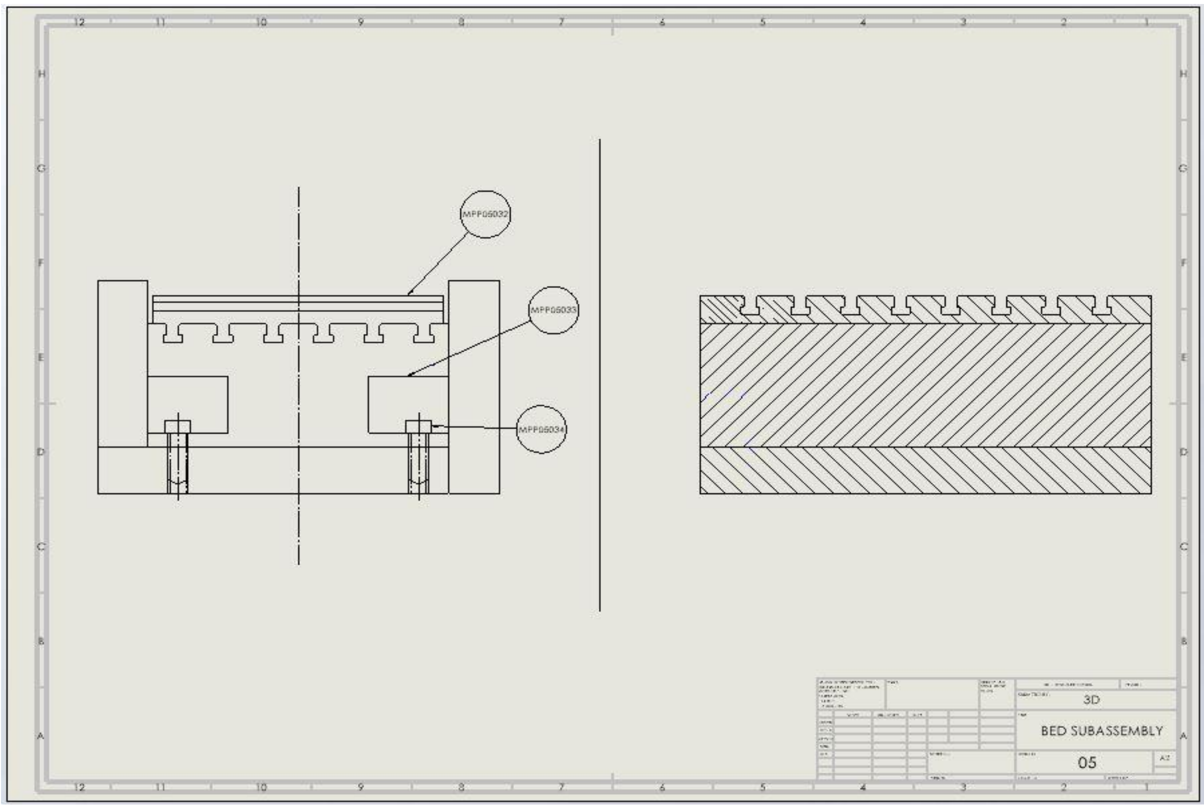
12.3 Sub Assembly of Shaft at 150 rpm



12.4 Crankshaft and Ram Assembly

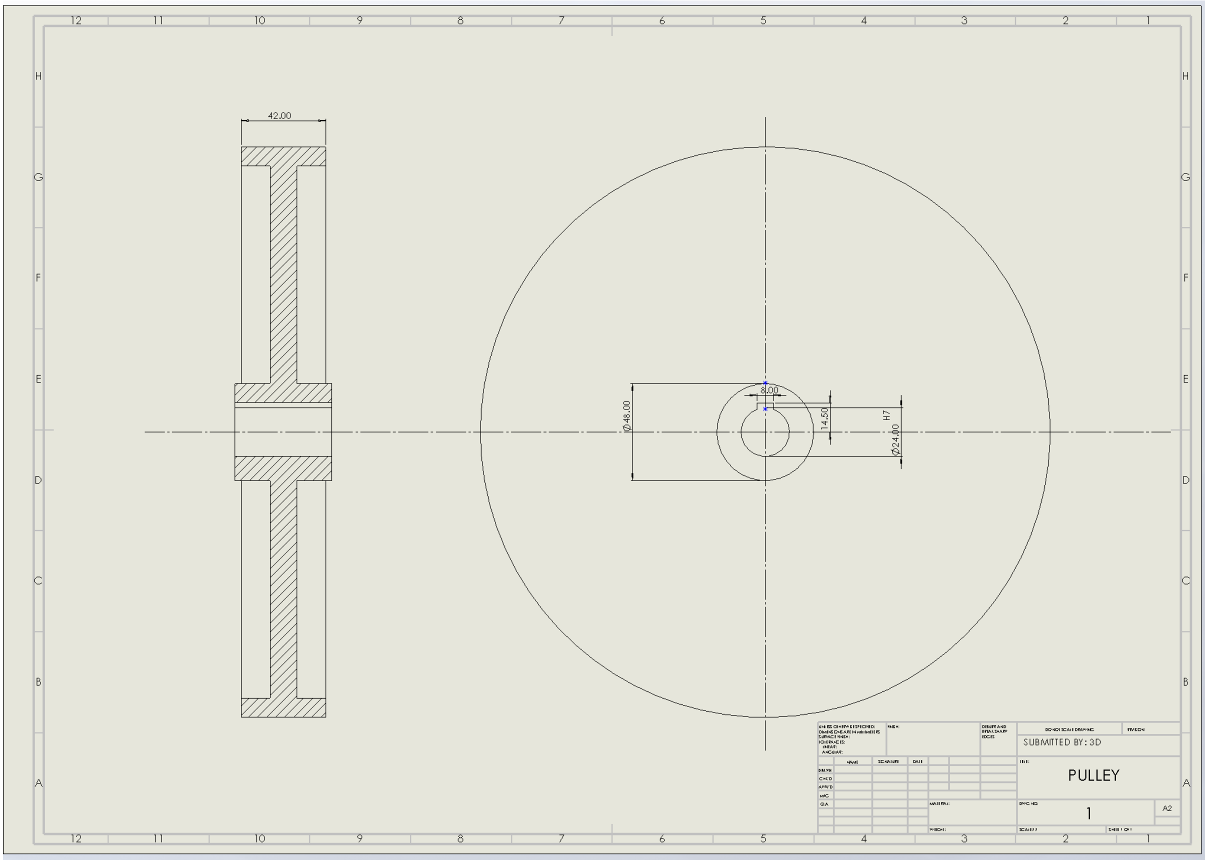


12.5 Bed Assembly

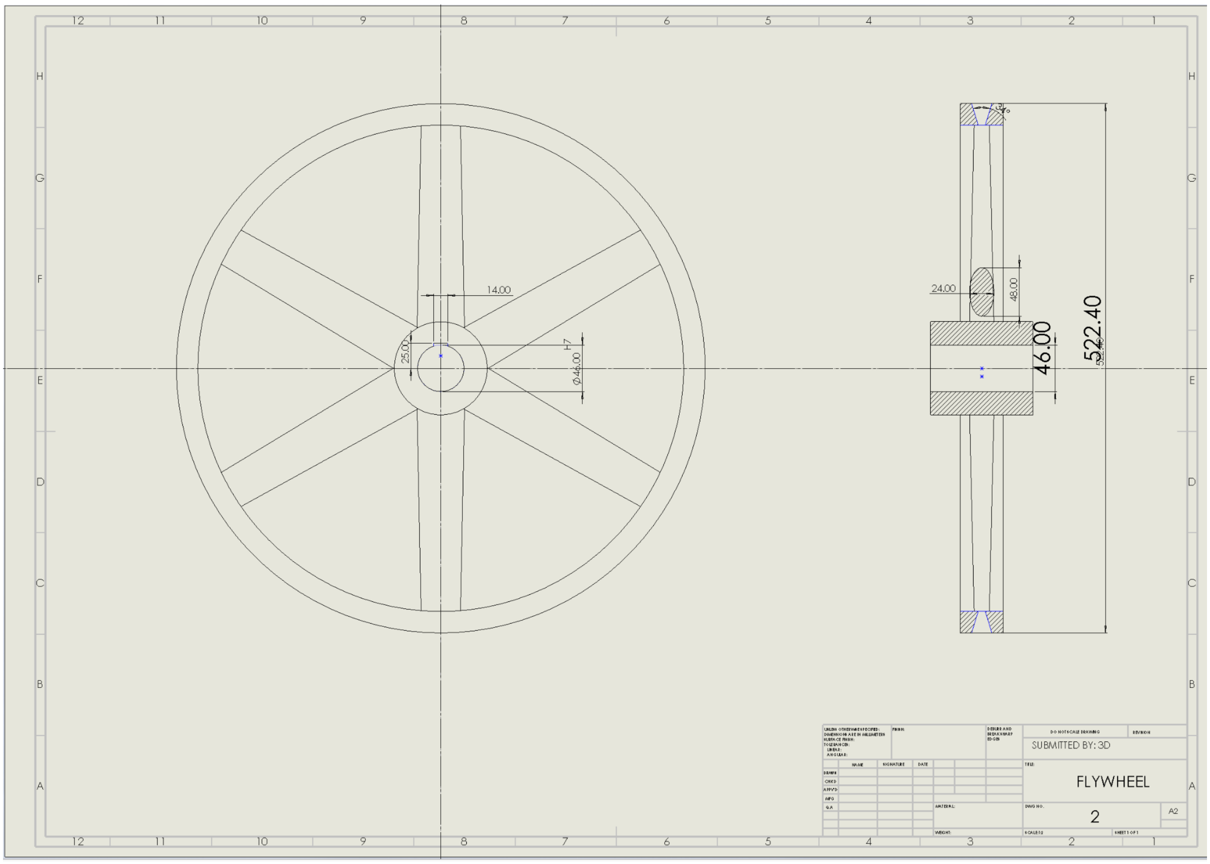


13. Component Drawings:

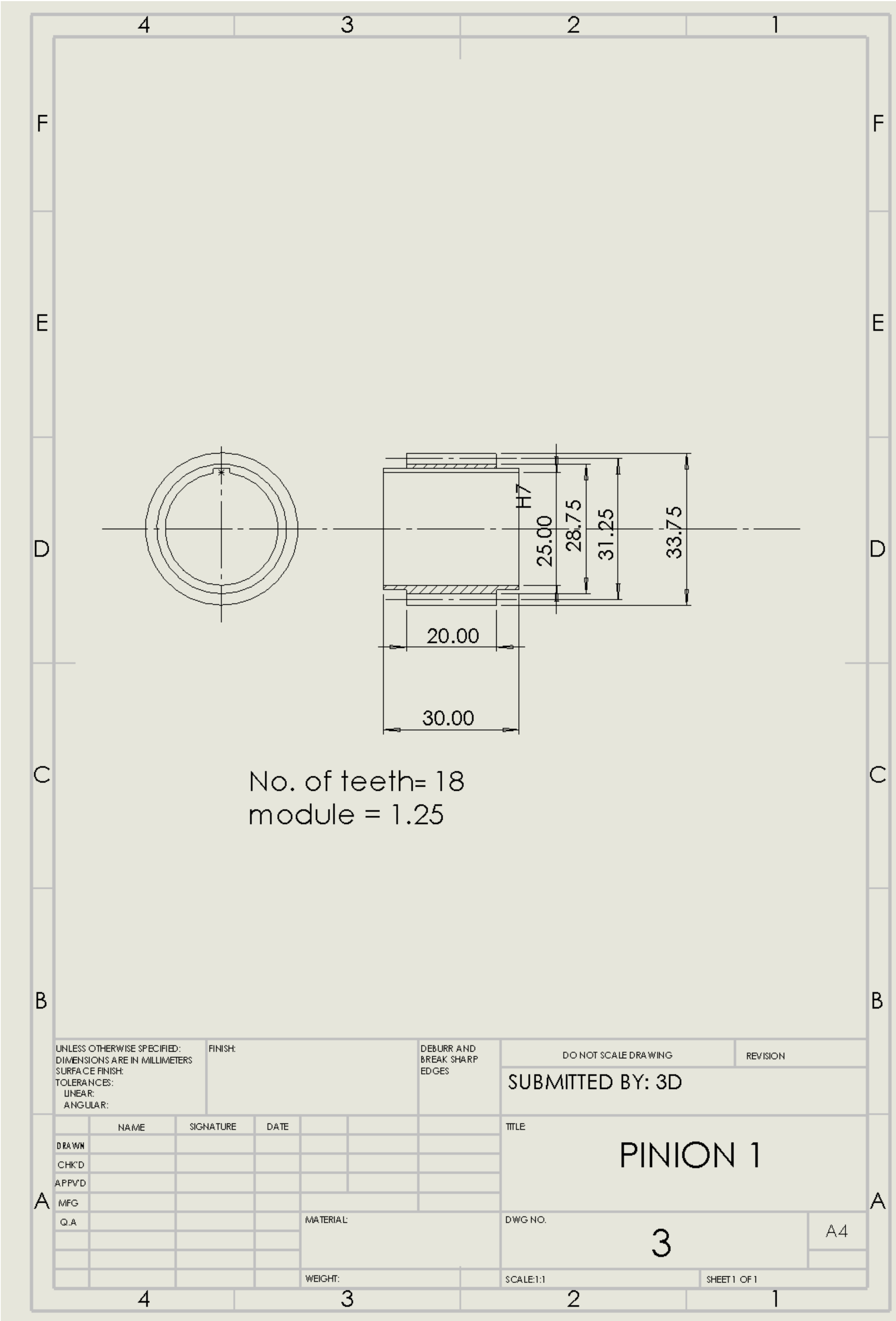
13.1 Pulley



13.2 Flywheel

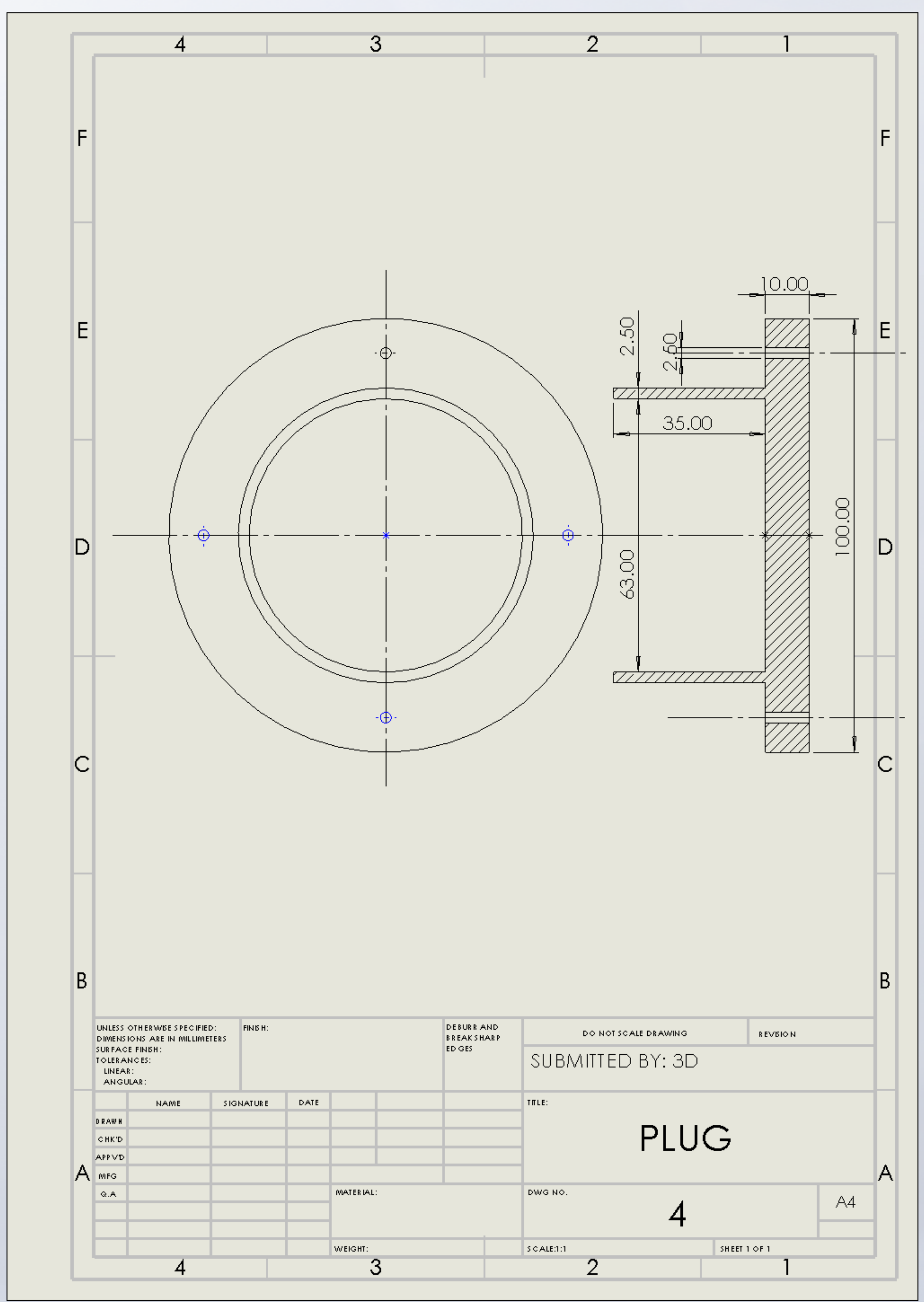


13.3 Pinion 1



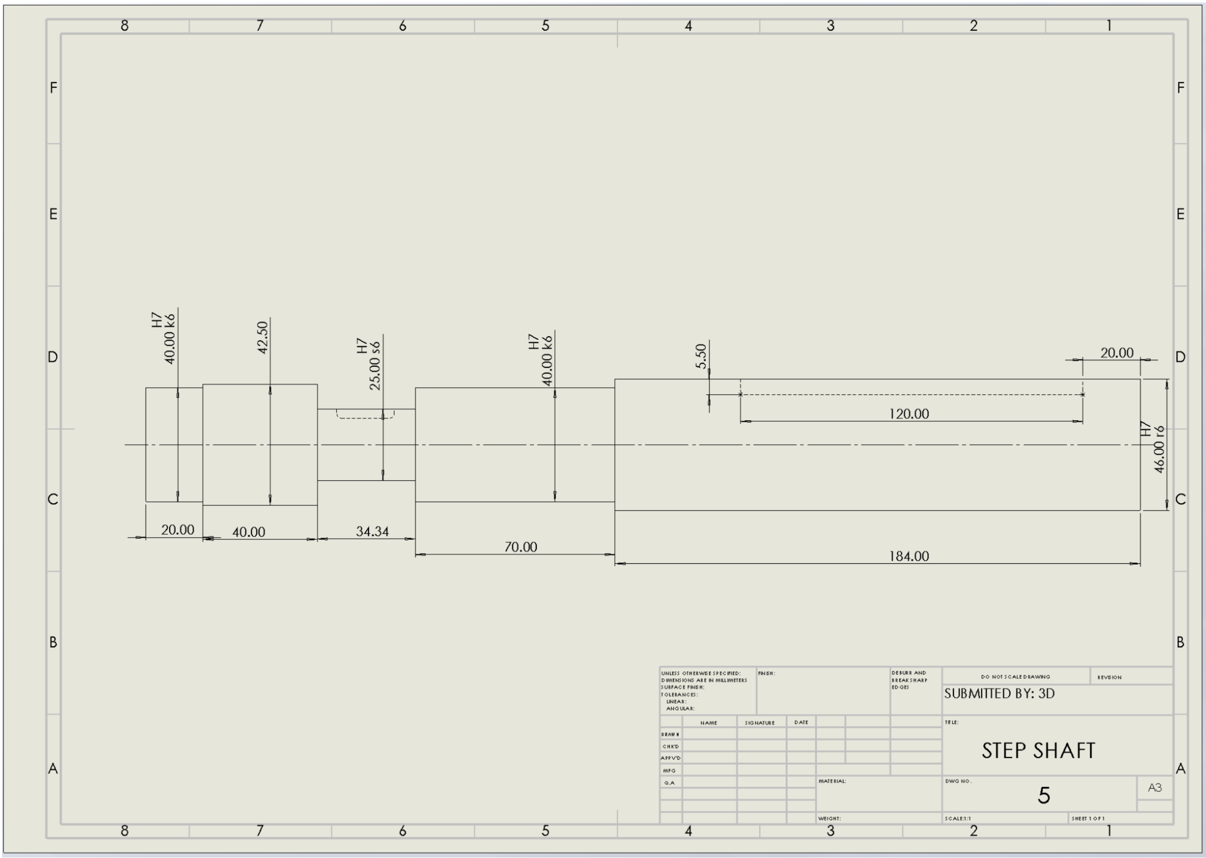
H7
25.00
28.75
31.25
33.75
20.00
30.00
No. of teeth= 18
module = 1.25
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
SUBMITTED BY: 3D
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
WEIGHT:
TITLE:
PINION 1
DWG NO.
3
A4
SCALE:1:1
SHEET 1 OF 1

13.4 Plug

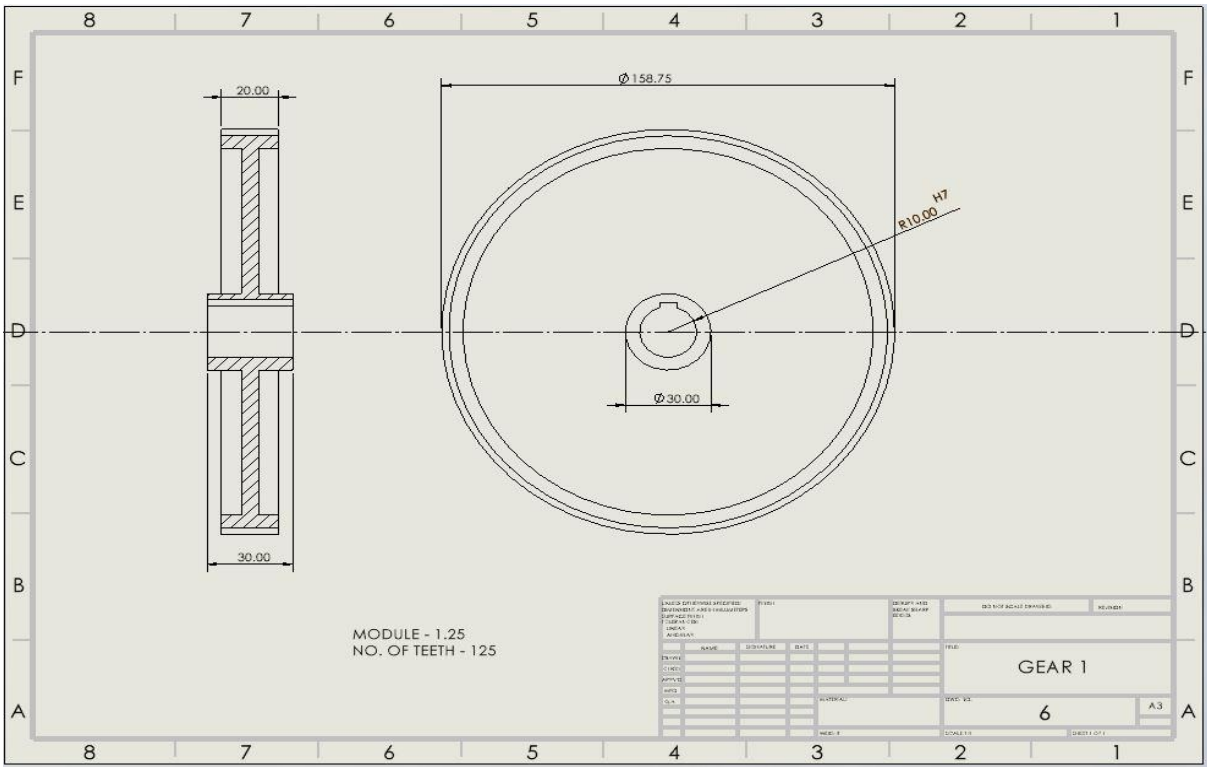


10.00
2.50
2.50
35.00
63.00
100.00
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
SUBMITTED BY: 3D
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
TITLE:
PLUG
MATERIAL:
DWG NO.
4
A4
WEIGHT:
SCALE:1:1
SHEET 1 OF 1

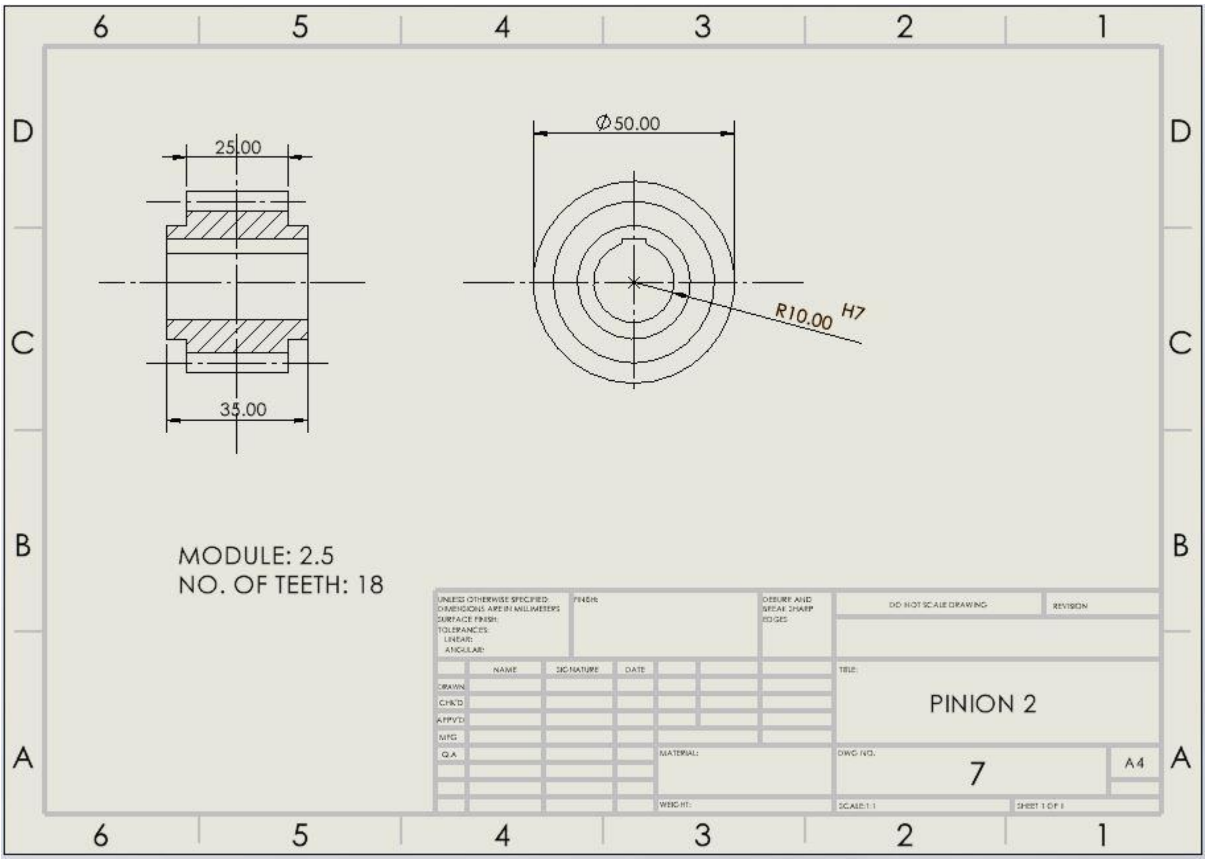
13.5 Step Shaft @750



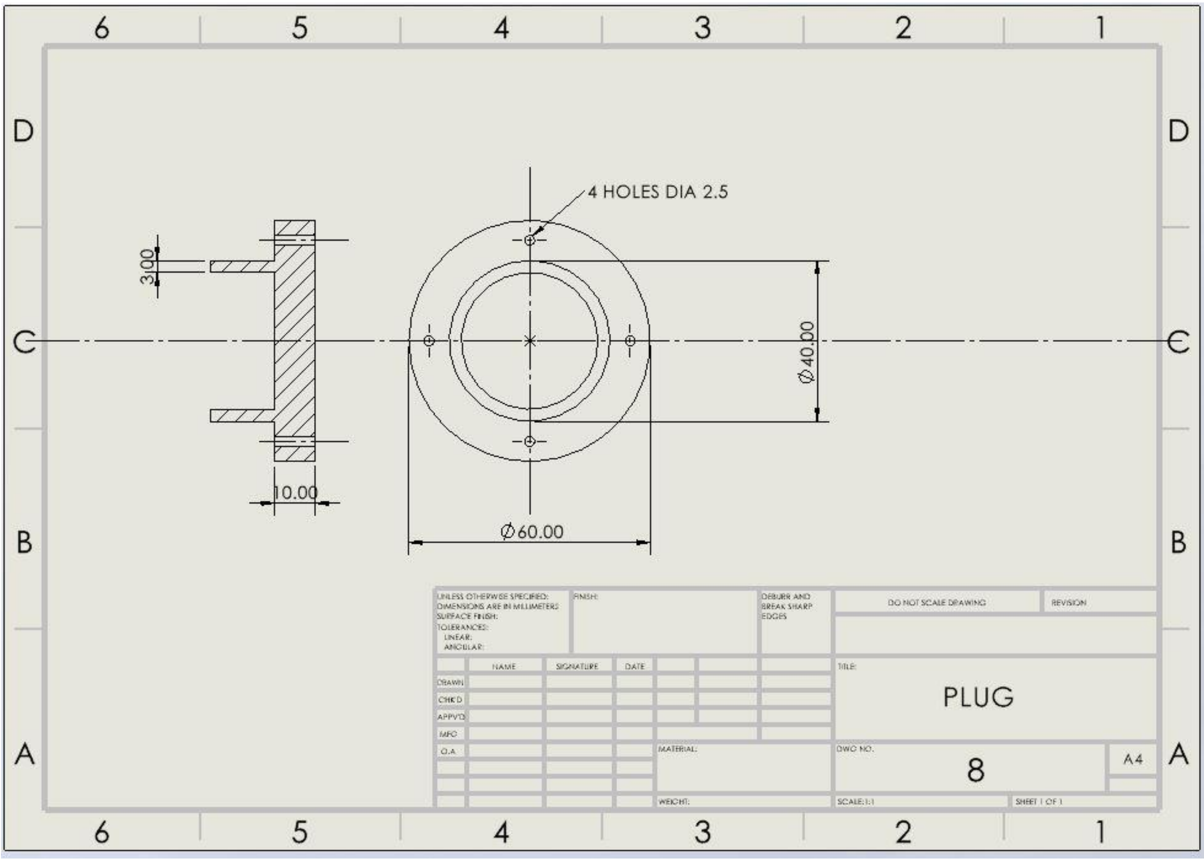
13.6 Gear 1



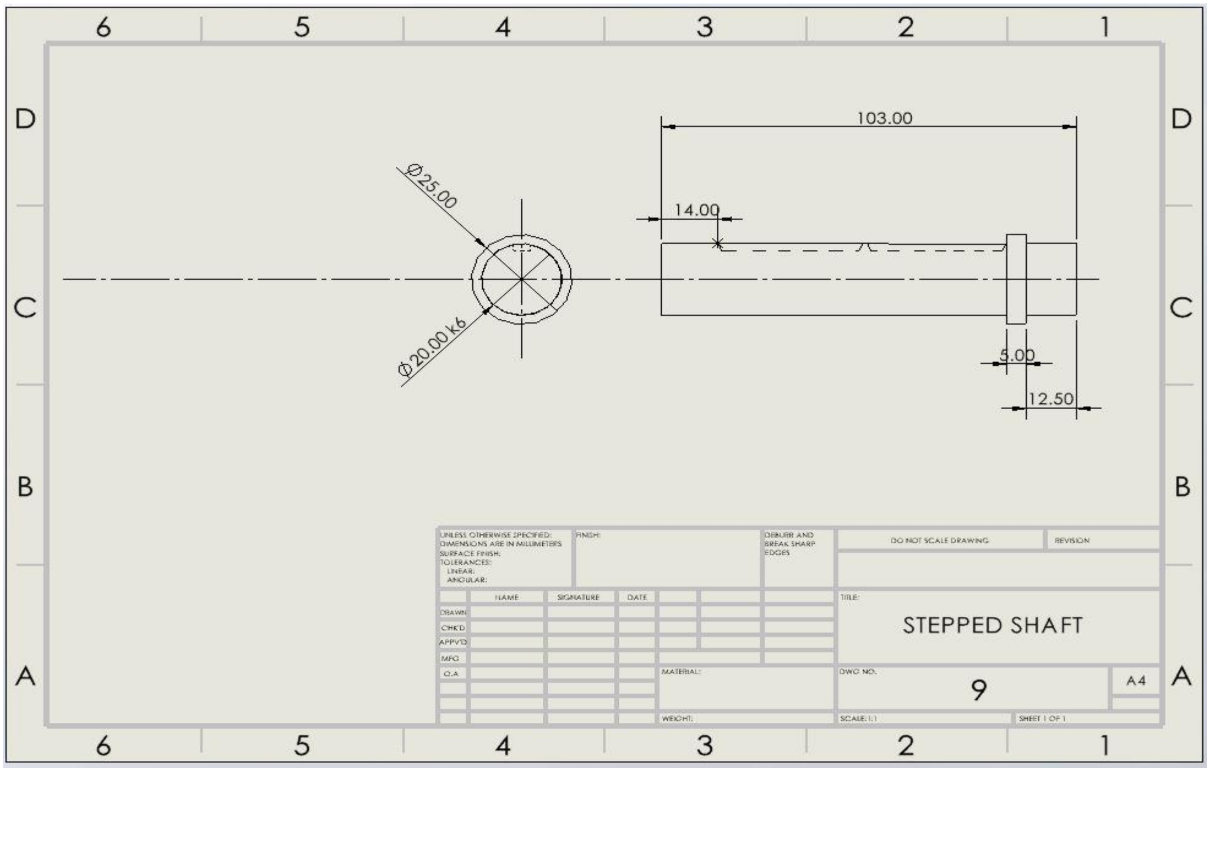
13.7 Pinion 2


25.00
35.00
⌀50.00
R10.00 H7
MODULE: 2.5
NO. OF TEETH: 18
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
WEIGHT:
TITLE:
PINION 2
DWG NO.
7
A4
SCALE:1:1
SHEET 1 OF 1

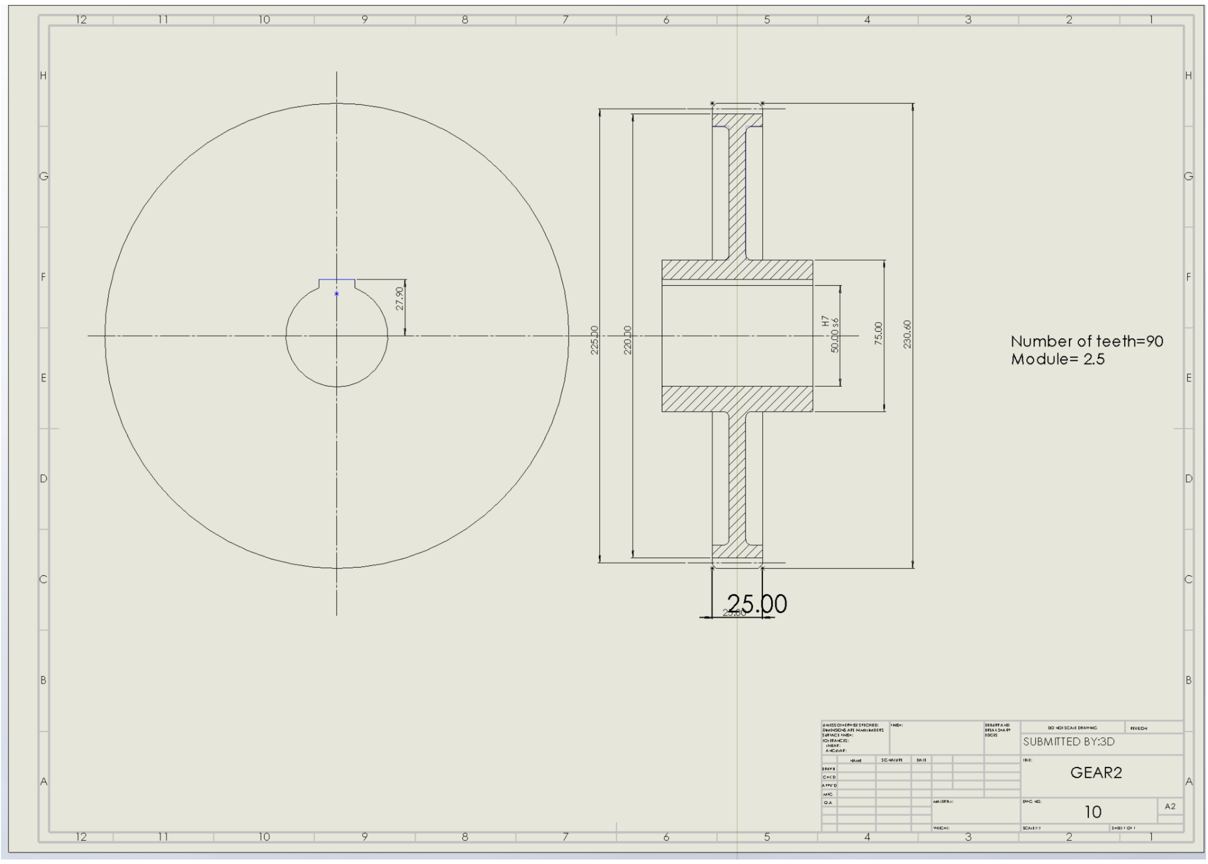
13.8 Plug



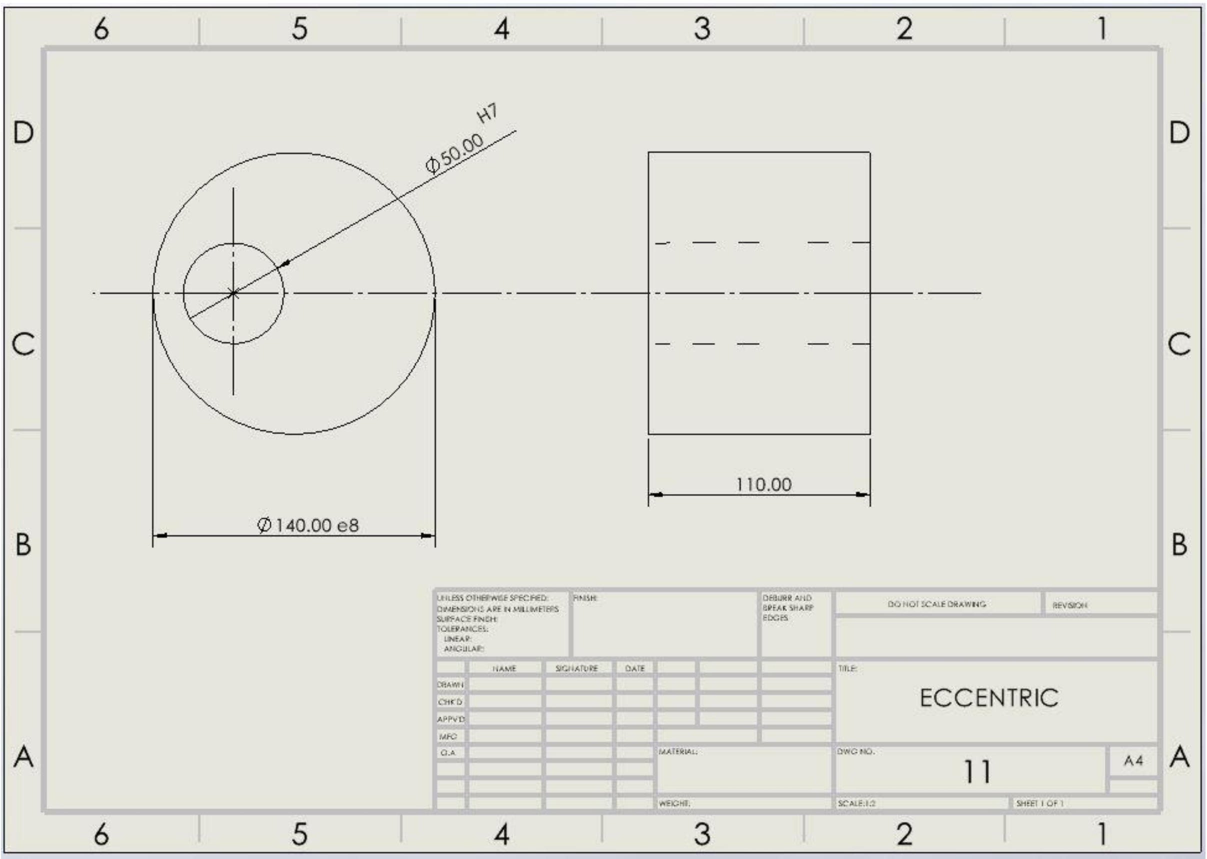
13.9 Step Shaft @150



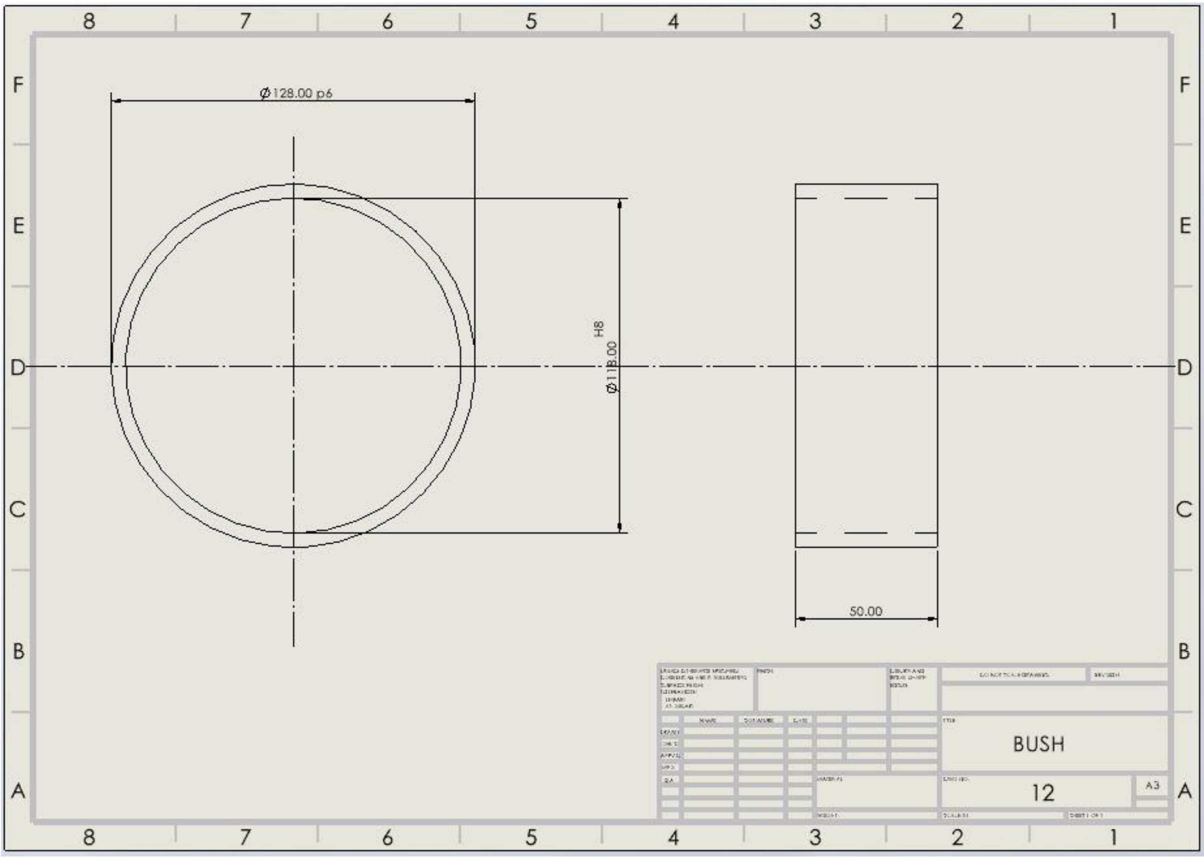
13.10 Gear 2



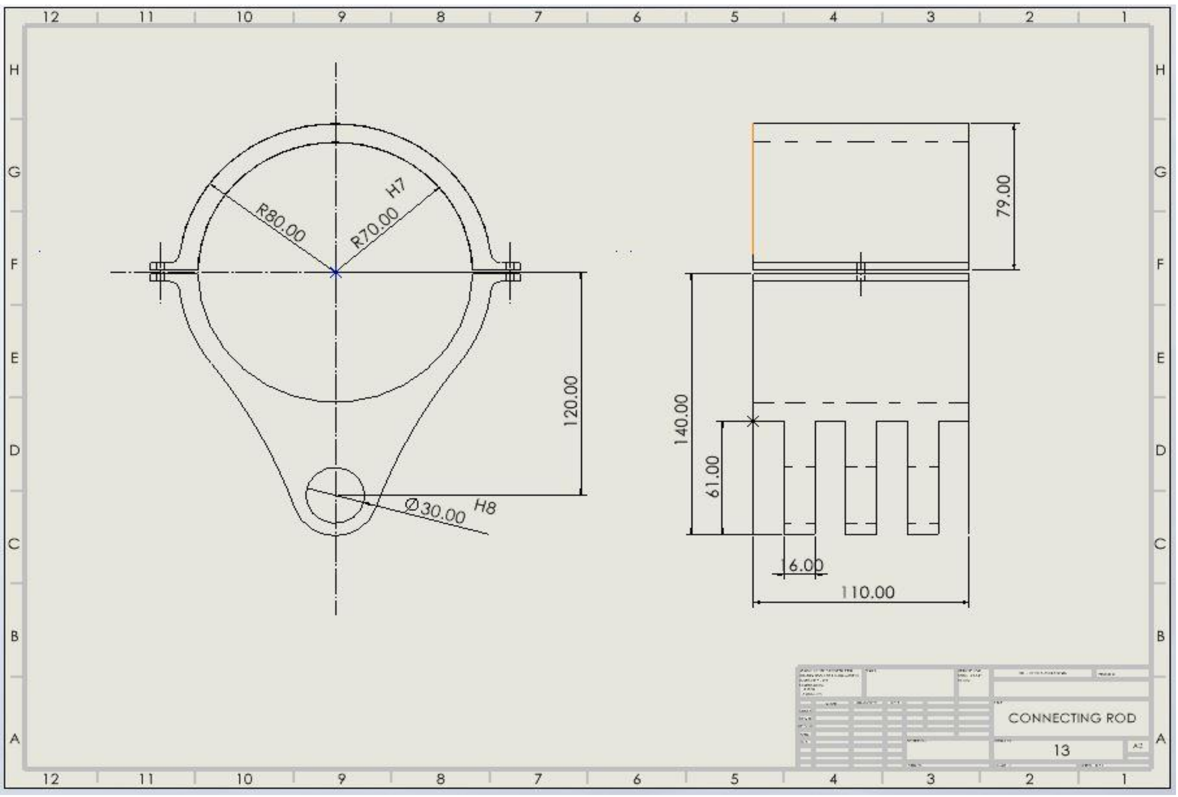
13.11 Eccentric



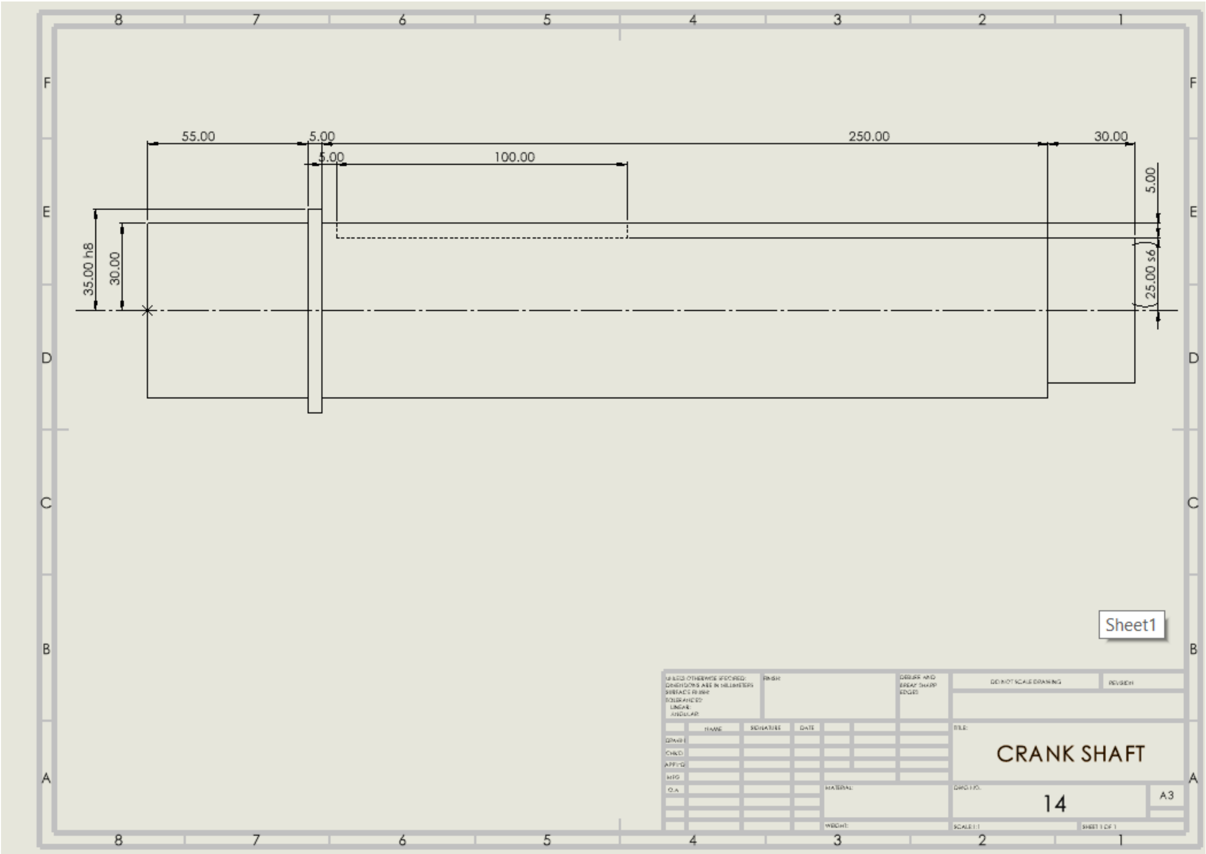
13.12 Bush



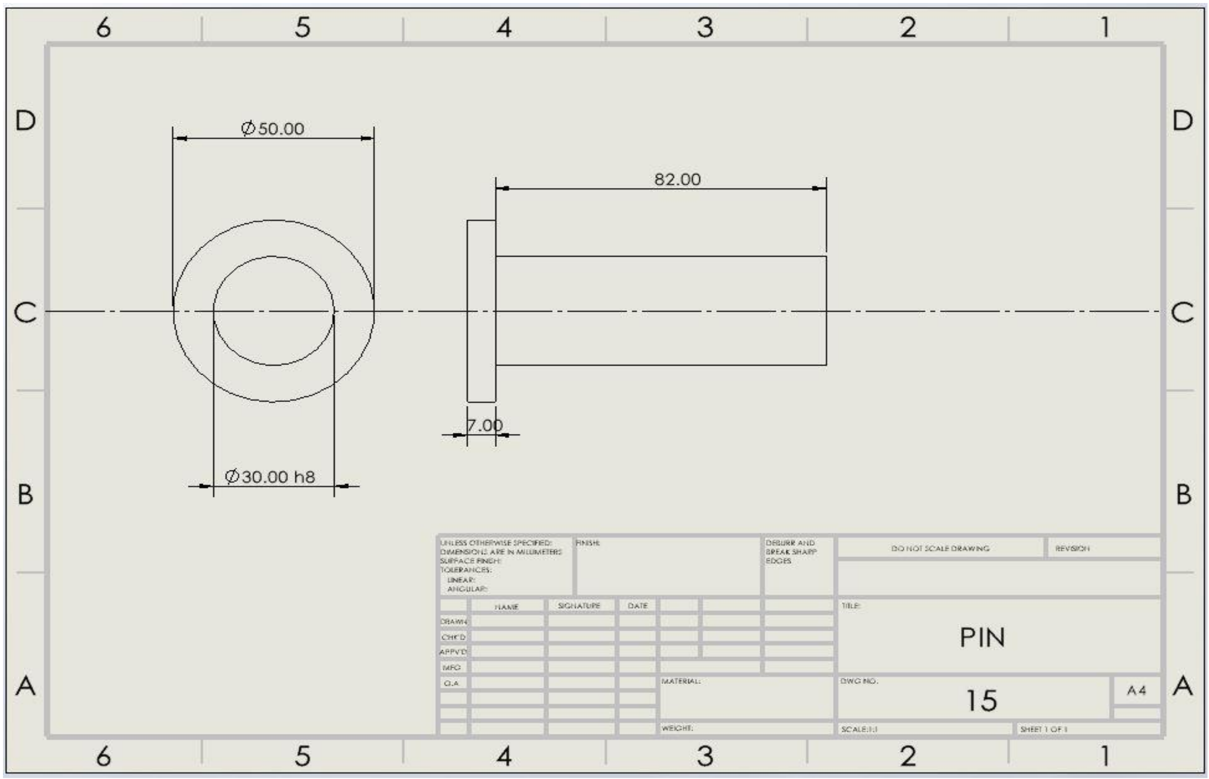
13.13 Connecting Rod



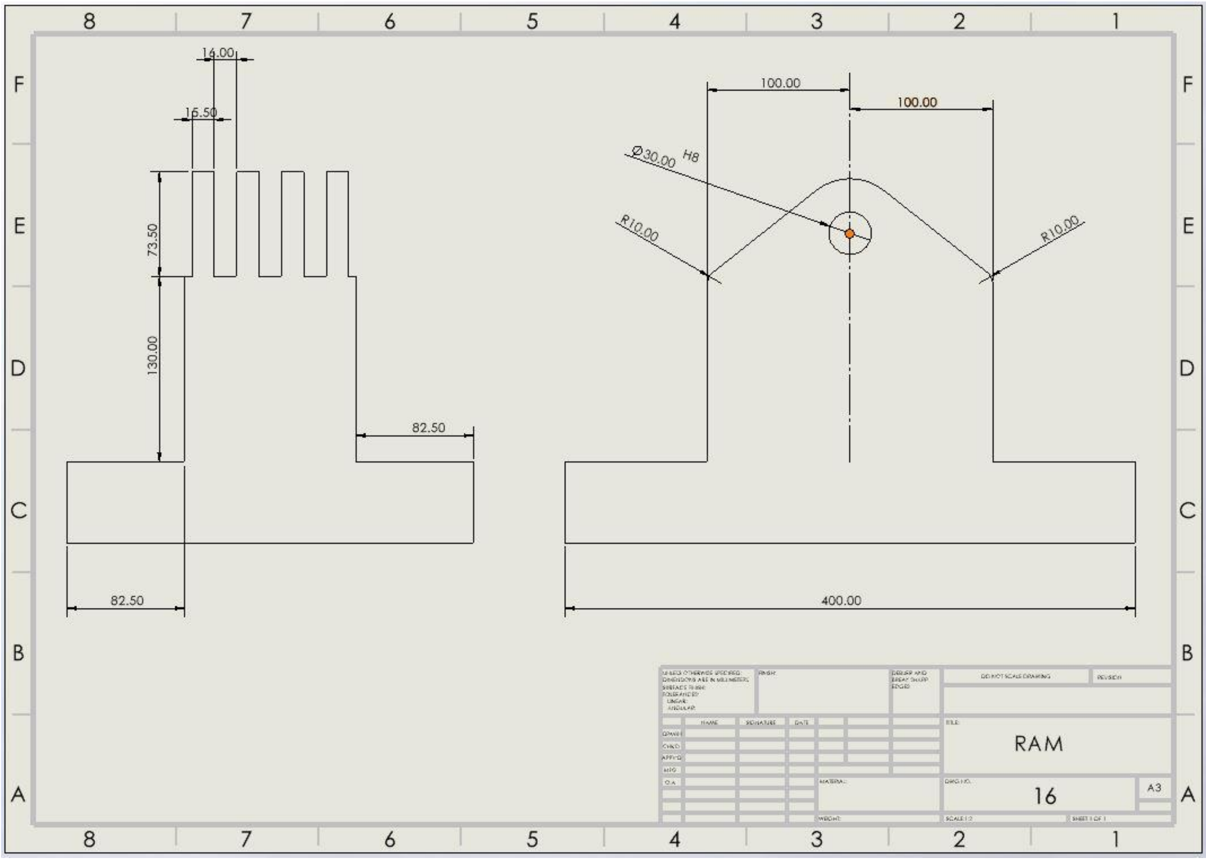
13.14 Crankshaft



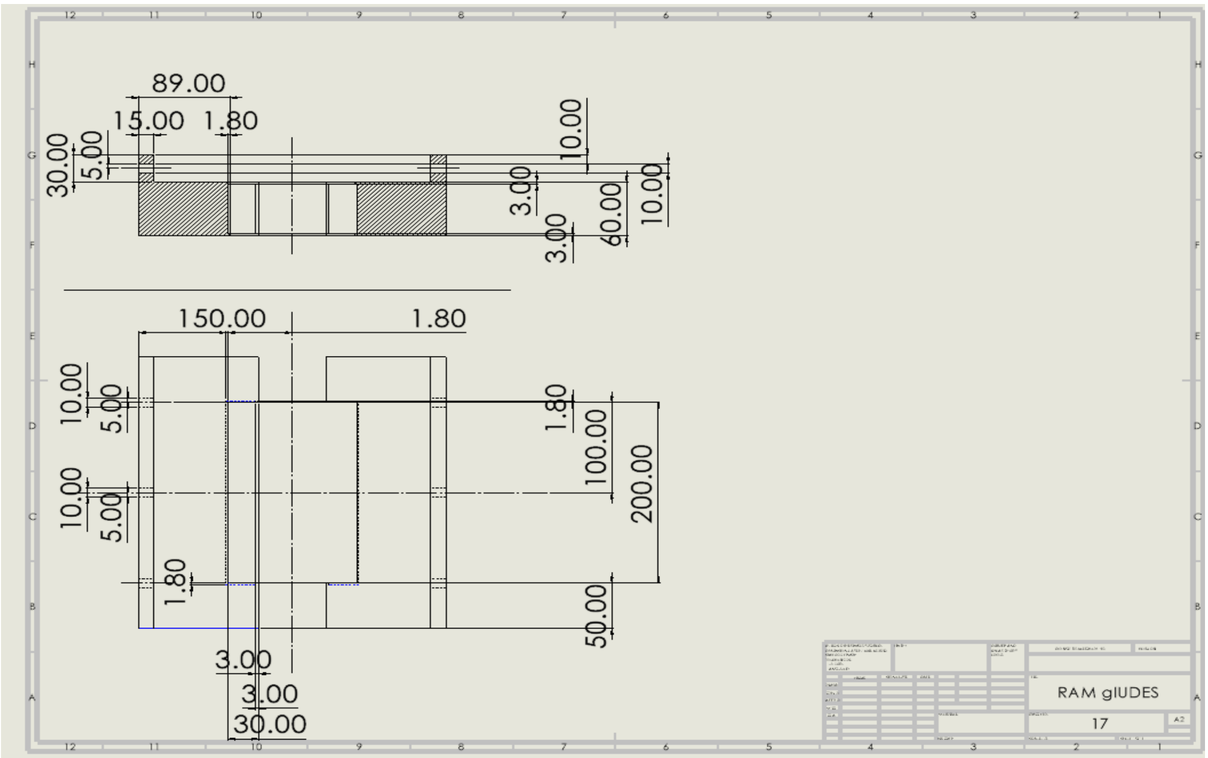
13.15 Pin



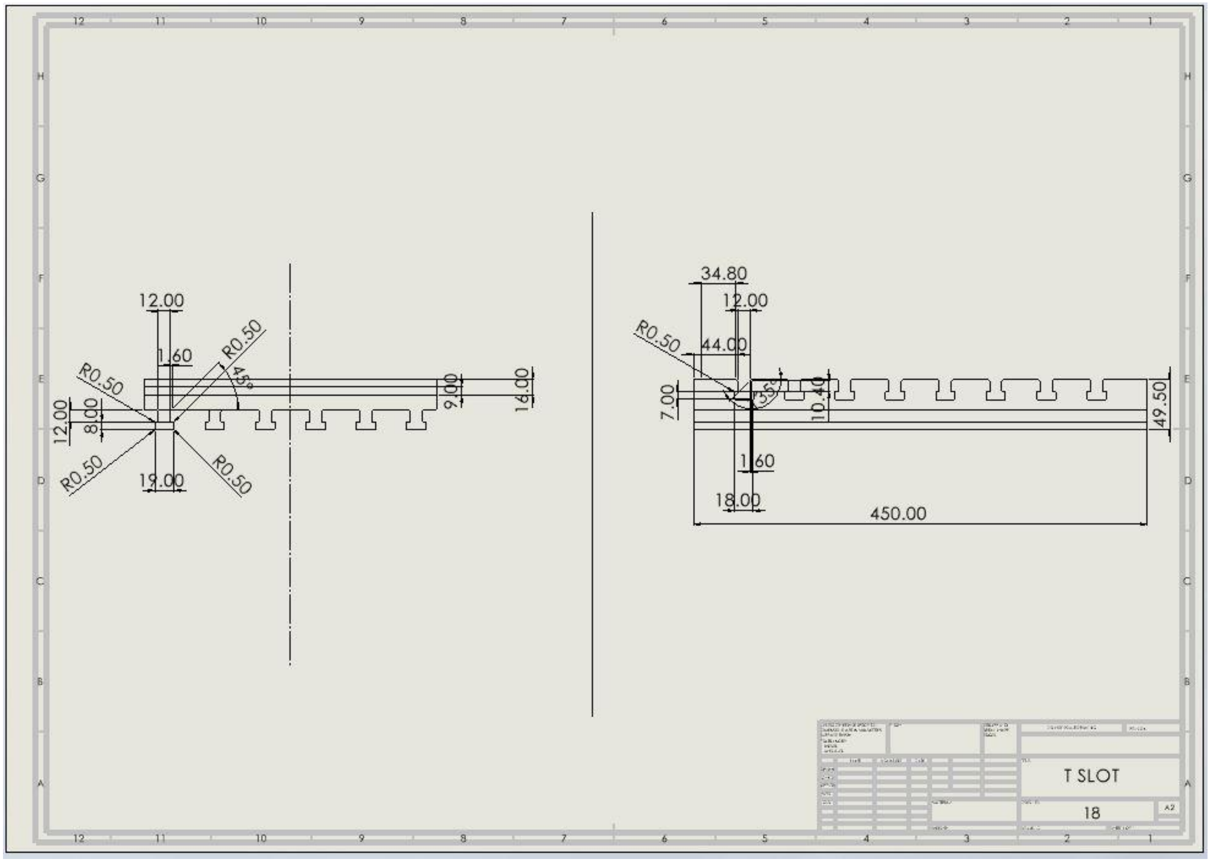
13.16 Ram



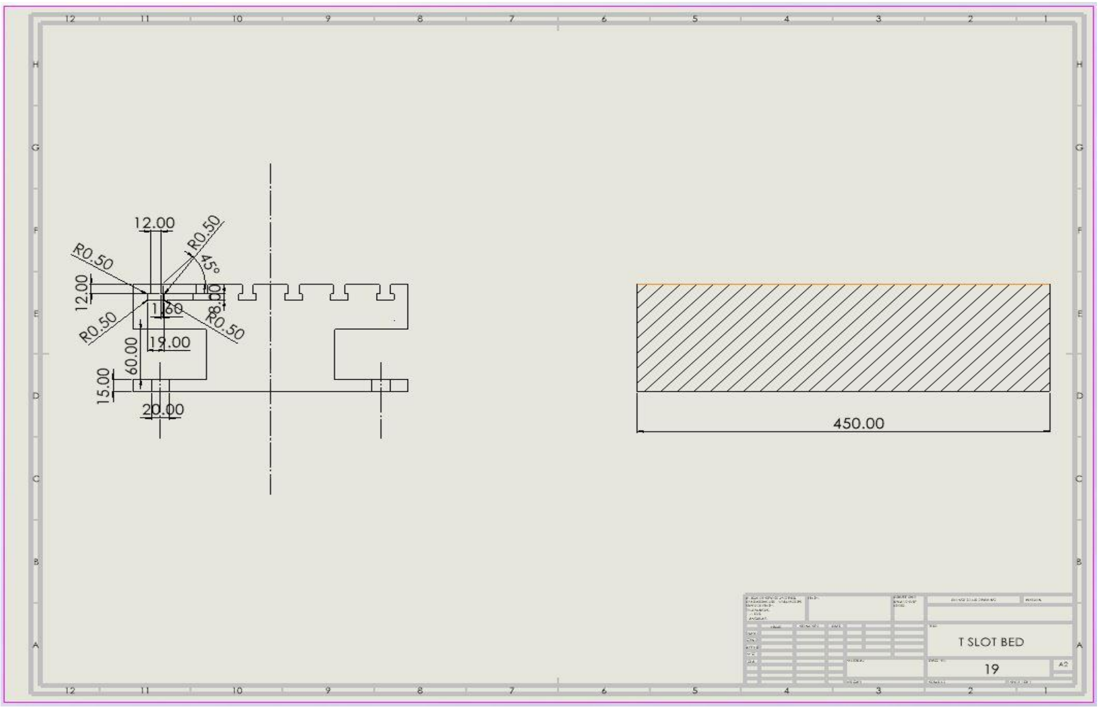
13.17 Ram Guide



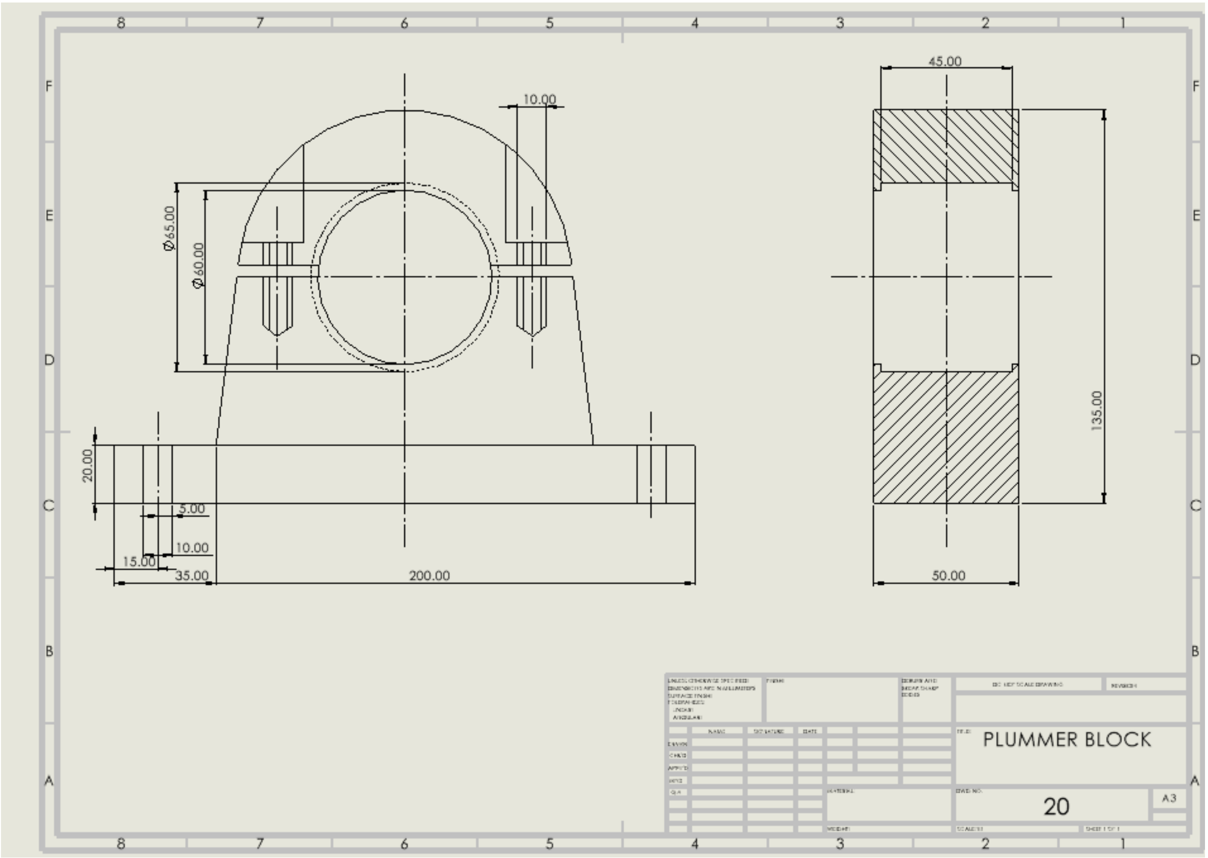
13.18 T Slot



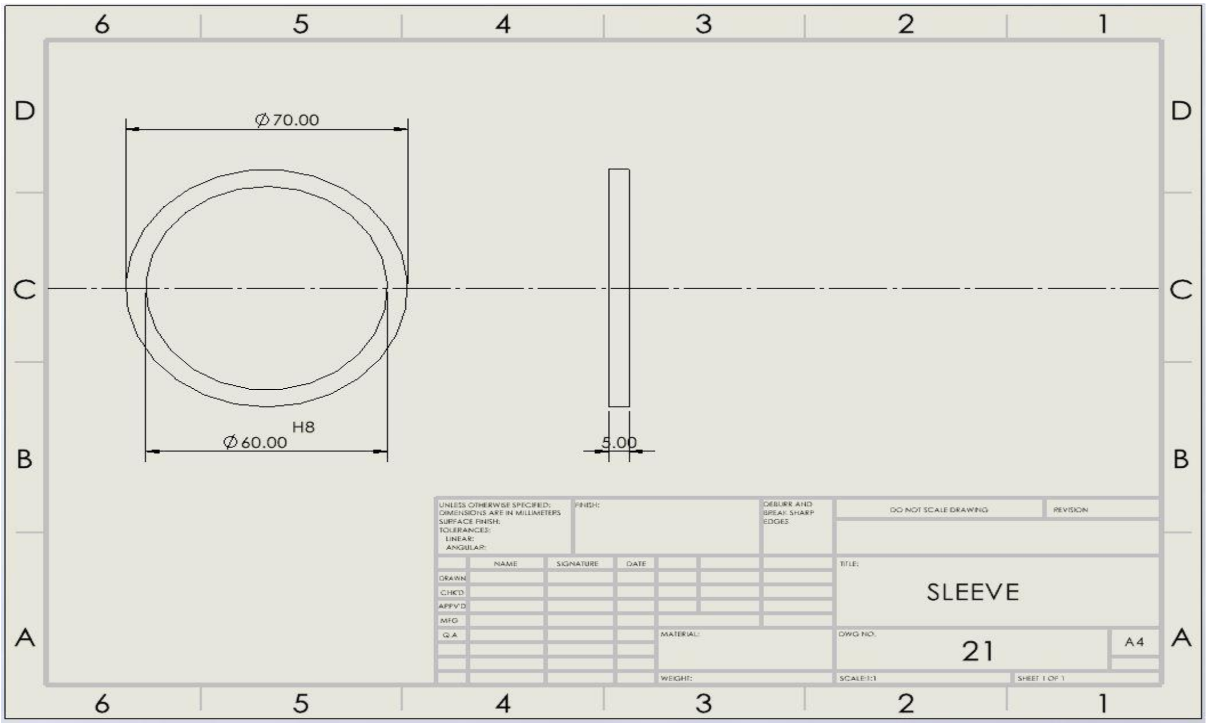
13.19 T Slot Bed



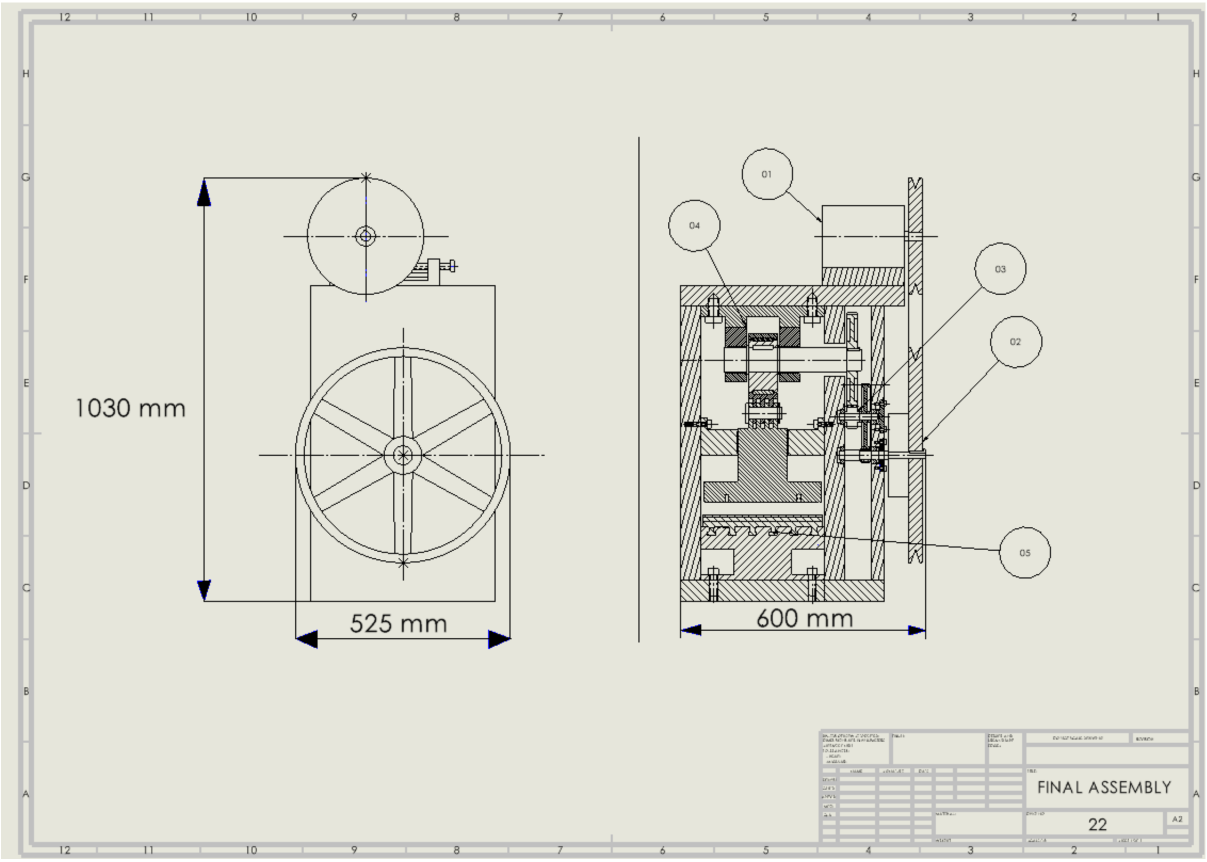
13.20 Plummer Block



13.21 Sleeve



14.Final Assembly Drawing:



15.Specifications of Machine:

a. Overall Dimensions

Length: 600 mm

Breadth: 525 mm

Height: 1030 mm

b. Power Requirement

Power Required for operation = 650 W
Power of Motor = 745 W = 1 H.P.