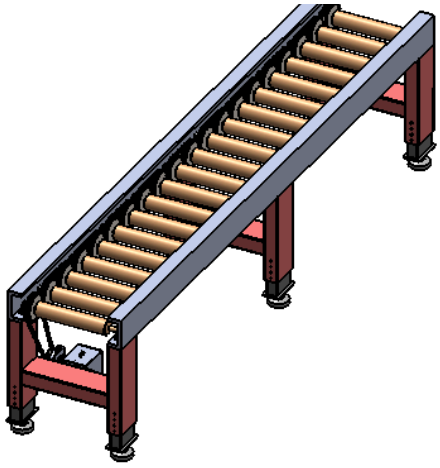
Chain-Roller Conveyor project

Spring 2022

EDPT602



Under supervision of:

Prof.**Hesham Hegazy**

Done by:

Names	ID number	Tutorial
Maryam Sherif	49-4013	T-32
Ayeshah Emad	49-19471	T-32
Salah Eldin Mohamed	49-18052	T-32
Yasmin Gabr	49-3334	T-32
Shehab Ibrahim	49-7639	T-32

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Introduction

Nowadays, almost all industries require the transportation and handling of materials for the packaging, sorting, and distributing phases of production. Consequently, conveyors became a large necessity in any wide range industry. The main purpose of a conveyor system is to move materials from one position to another in a quick but efficient fashion. Conveyors are also used to: create buffers in the delivery of products in a specific sequence that fits the production line, transport products that are considerably dense for team members to carry, prevent workers from getting injured due to repetitive movement, avoid possible damage caused by movement of products, and create a flexible storage system, where the process flow is stabilized by raising or lowering the amount of products.

Based on their functionality and usage mode, there are many types and designs of industrial conveyors. Some types of conveyors that are mounted on the floor are conventional roller, motorized roller, and belt conveyors, while overhead conveyor types include electric track, inclined, and ramped conveyors. Although all types are very important in different industries, roller conveyor systems are the most employed type of conveyor systems. Since conveyors provide a significant service for industries, their applications are limitless: airports, distribution centers, food industry, packaging industries, and postal sectors.

In this project, we will be designing a roller conveyor that is driven by chains for the transportation of boxes. The design considerations of each component of the roller conveyor are clearly documented in this report.

Motor Selection

We first calculated the number of boxes to be held by the conveyor:

$$N_{boxes} = \frac{L_{total}}{L_{box}} = \frac{6000}{500} = 12\ Box$$

Then using the velocity required and the weight given of the boxes we got the power required:

$$H = W * V = (12 * 50 * 9.81) * 0.8 = 4.7\ kW$$

In addition, from the CHIARAVALLI catalog we chose the next higher motor in power.

6 POLI 1000 min^{-1}

P_N [kW]		Motore	Poli	n_N [min^{-1}]	M_N [Nm]	I_N [A]	COS φ	η 100%	M_S/M_N	M_{max}/M_N	I_S/I_N	J_0 [kg m²]	W [kg]
5,5	**CHT**	**132 MB**	**6**	960	54,7	11,9	0,78	85,3	2,0	2,2	6,5	0,046	64
7,5	**CHT**	**160 M**	**6**	940	76,2	16,1	0,78	86,0	2,0	2,0	6,5	0,088	108
11	**CHT**	**160 L**	**6**	940	111,8	23,4	0,78	87,0	2,0	2,0	6,5	0,116	126

Figure 1. CHIARAVALLI Catalog selected motor

The roller diameter required to be within (100<D<150mm), we will choose 125mm

Specifications now goes as follow:

$$\omega = 960\ RPM v = 0.8\ m/s \phi_{roller} = 125mm$$

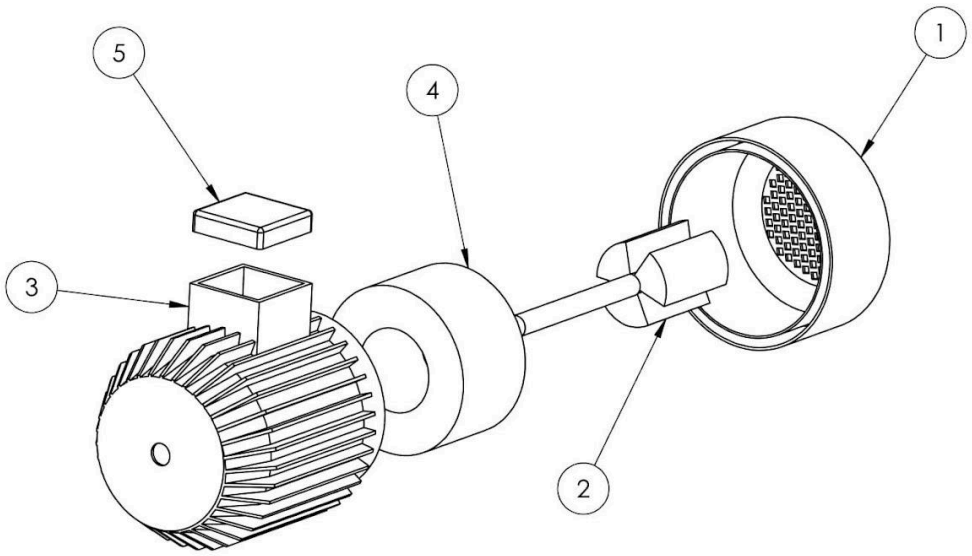
We can then find the required speed in RPM

$$w_{required} = \frac{v}{r} * \frac{60}{2\pi} = \frac{0.8}{0.125} * \frac{60}{\pi} = 122.23 \approx 120$$

Reduction ratio

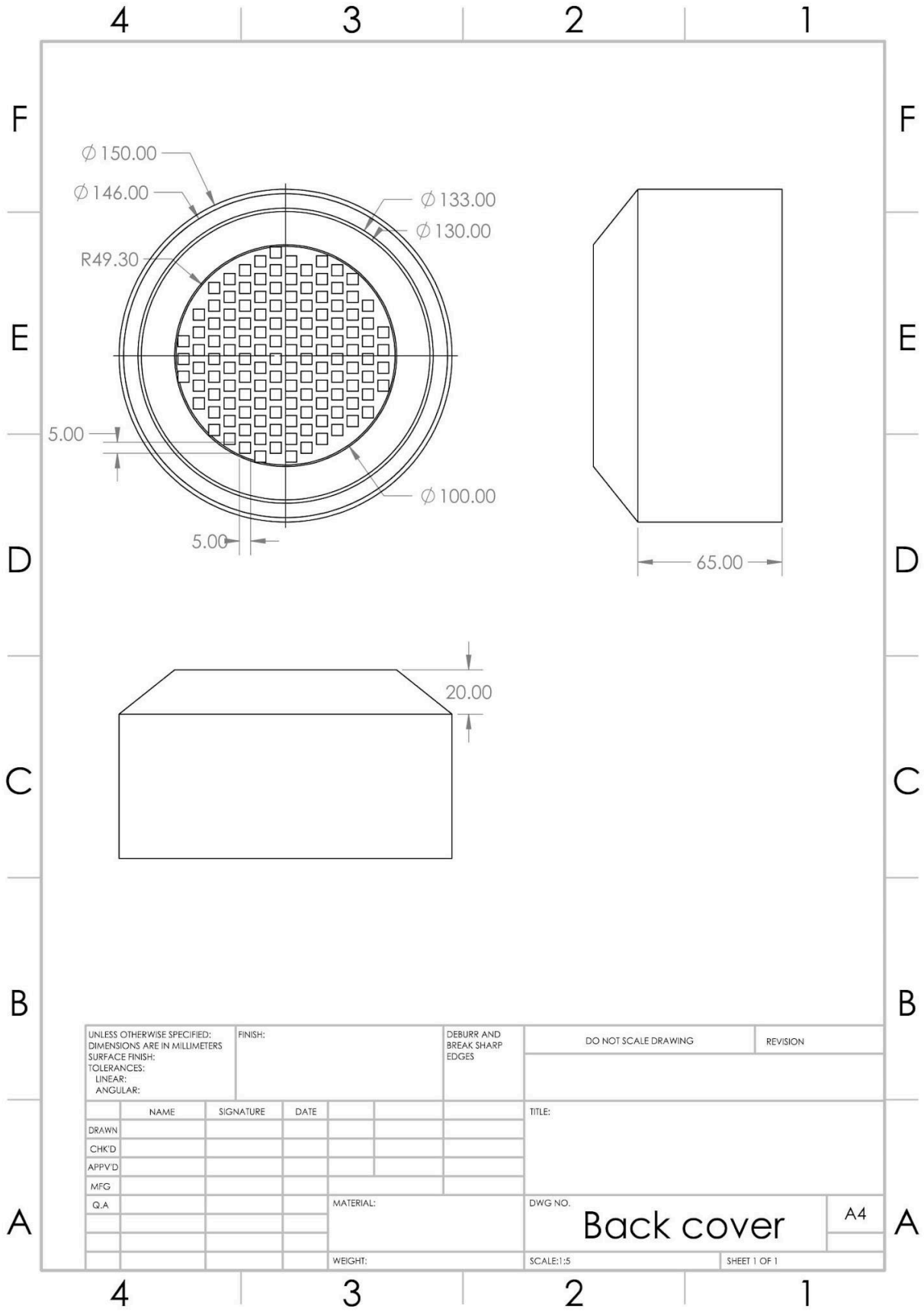
$$i = \frac{\omega_{driver}}{\omega_{driven}} = \frac{960}{120} = 8$$

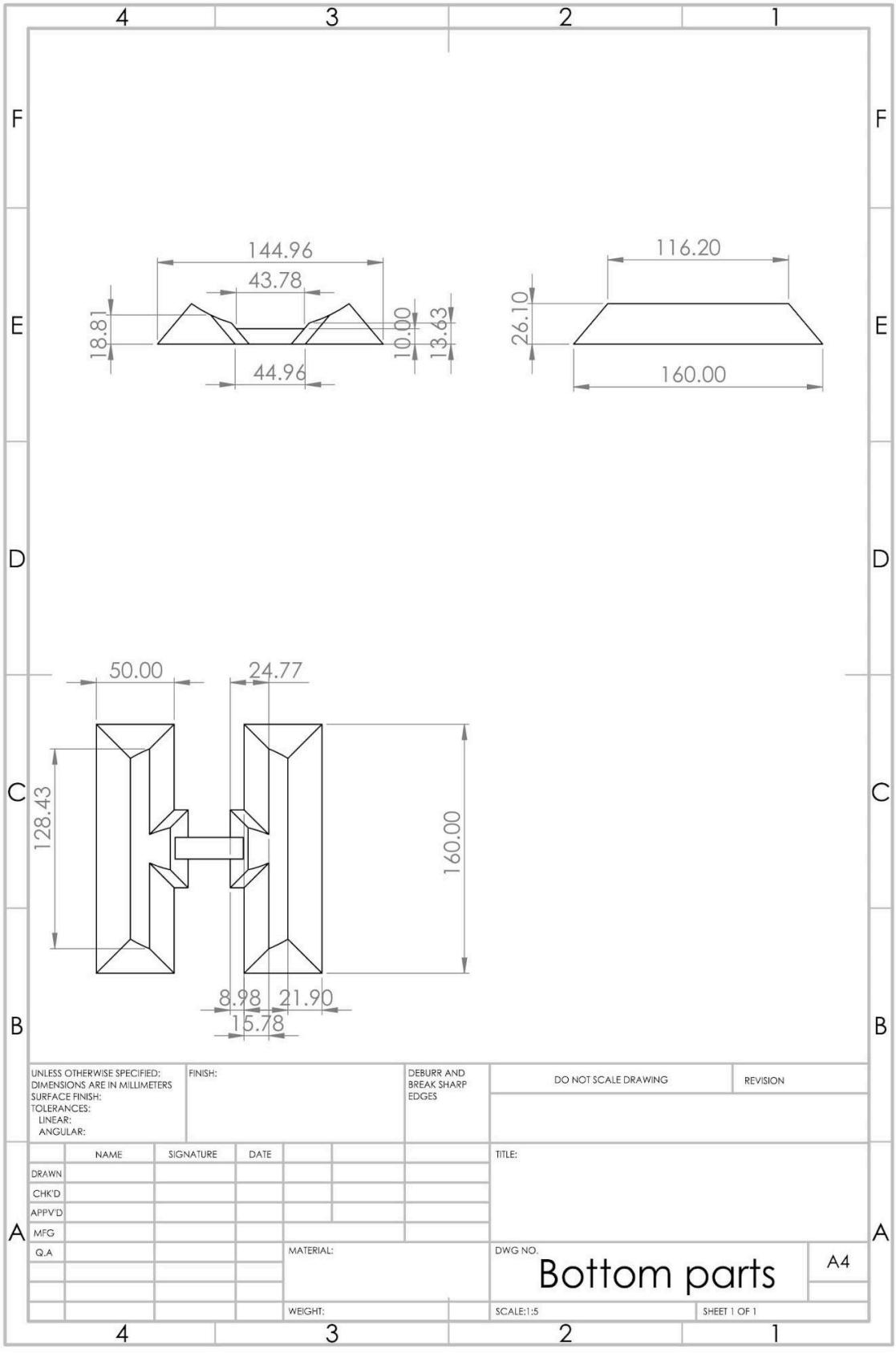
In our design we will divide this reduction into two stages, one in the gearbox with $i = 4$ and another using the chain driver with $i = 2$ so that $i_{total} = 4 * 2 = 8$

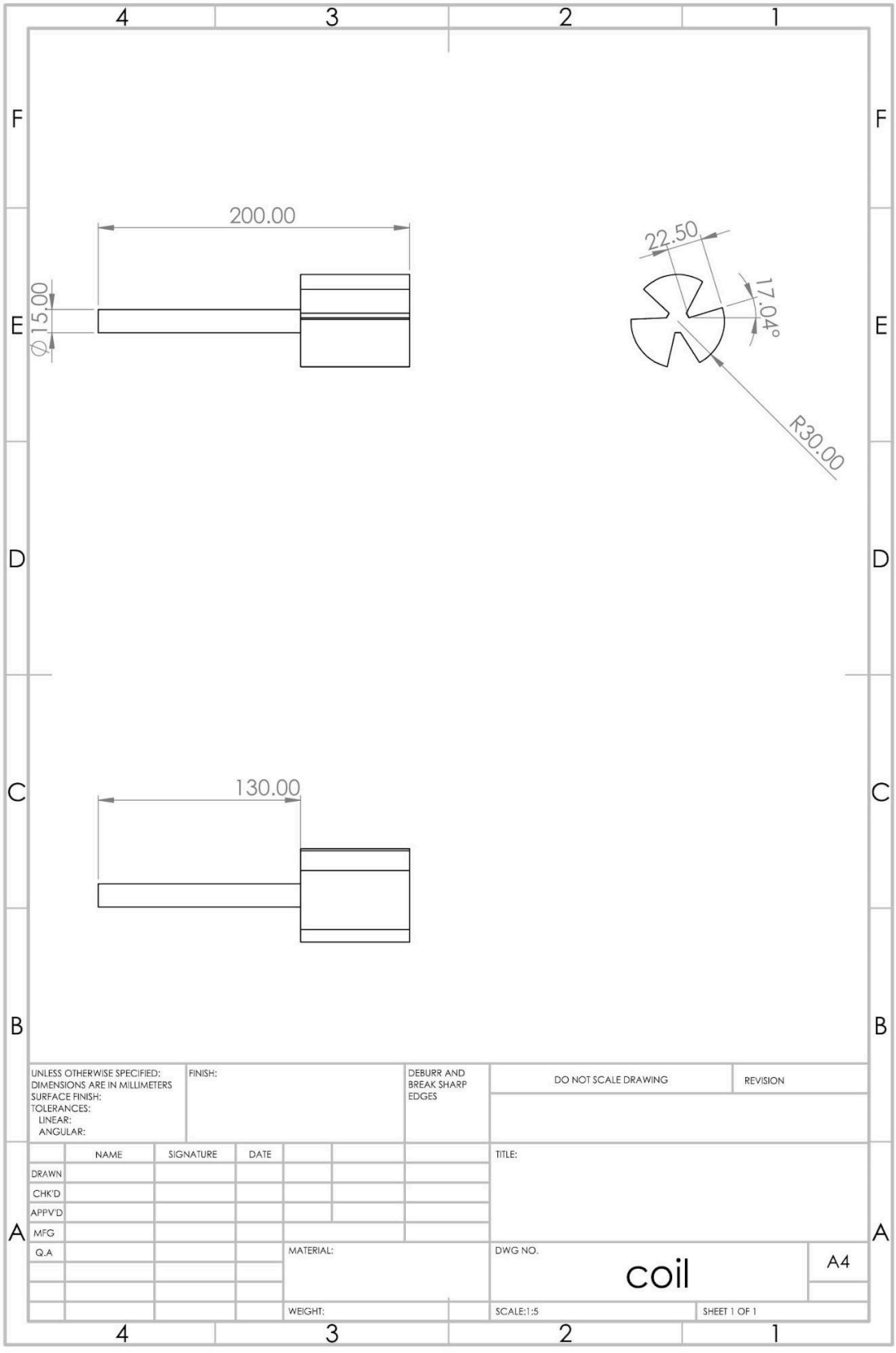


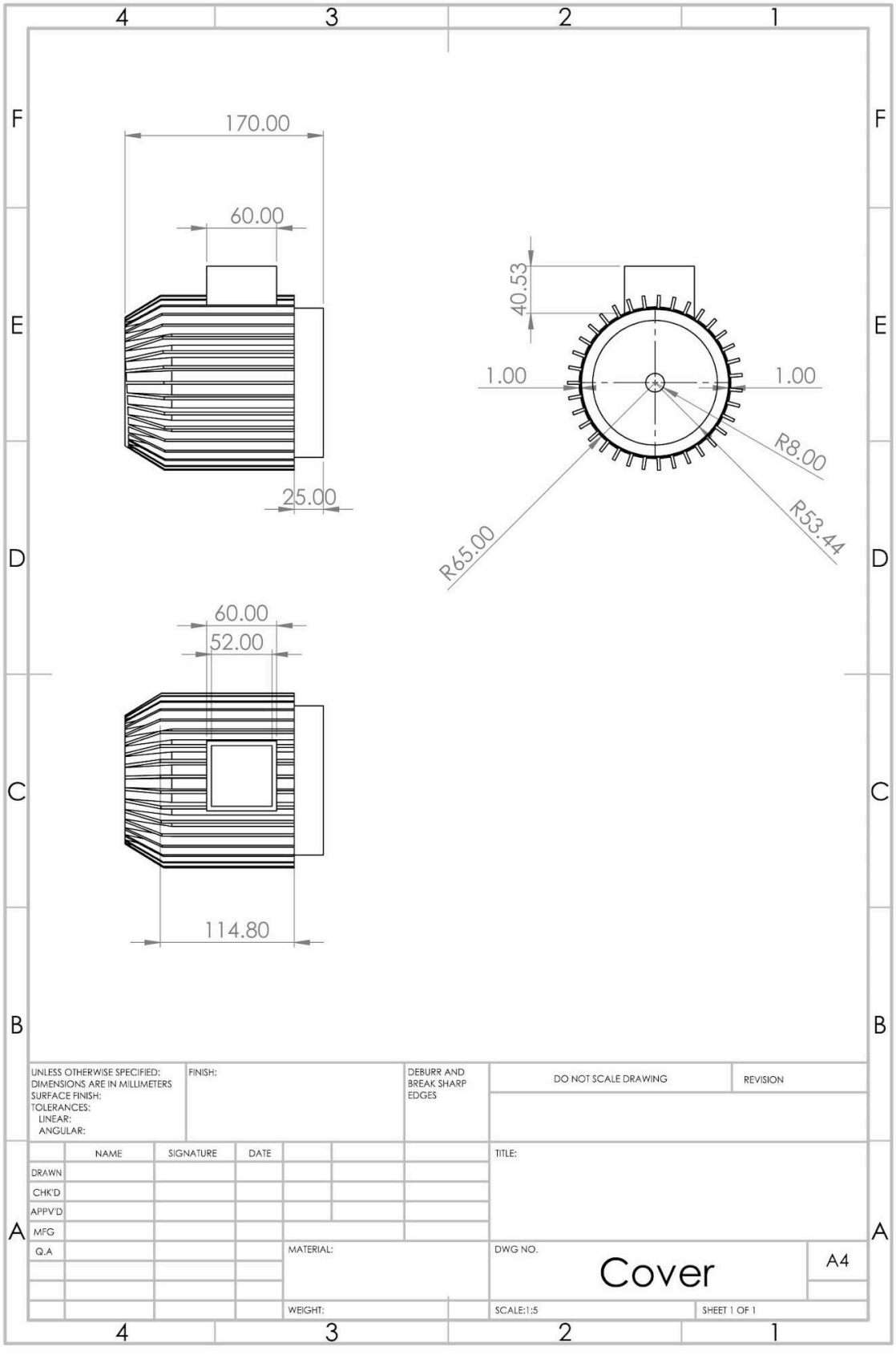
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Back cover		1
2	coil		1
3	Cover		1
4	ring		1
5	Top		1

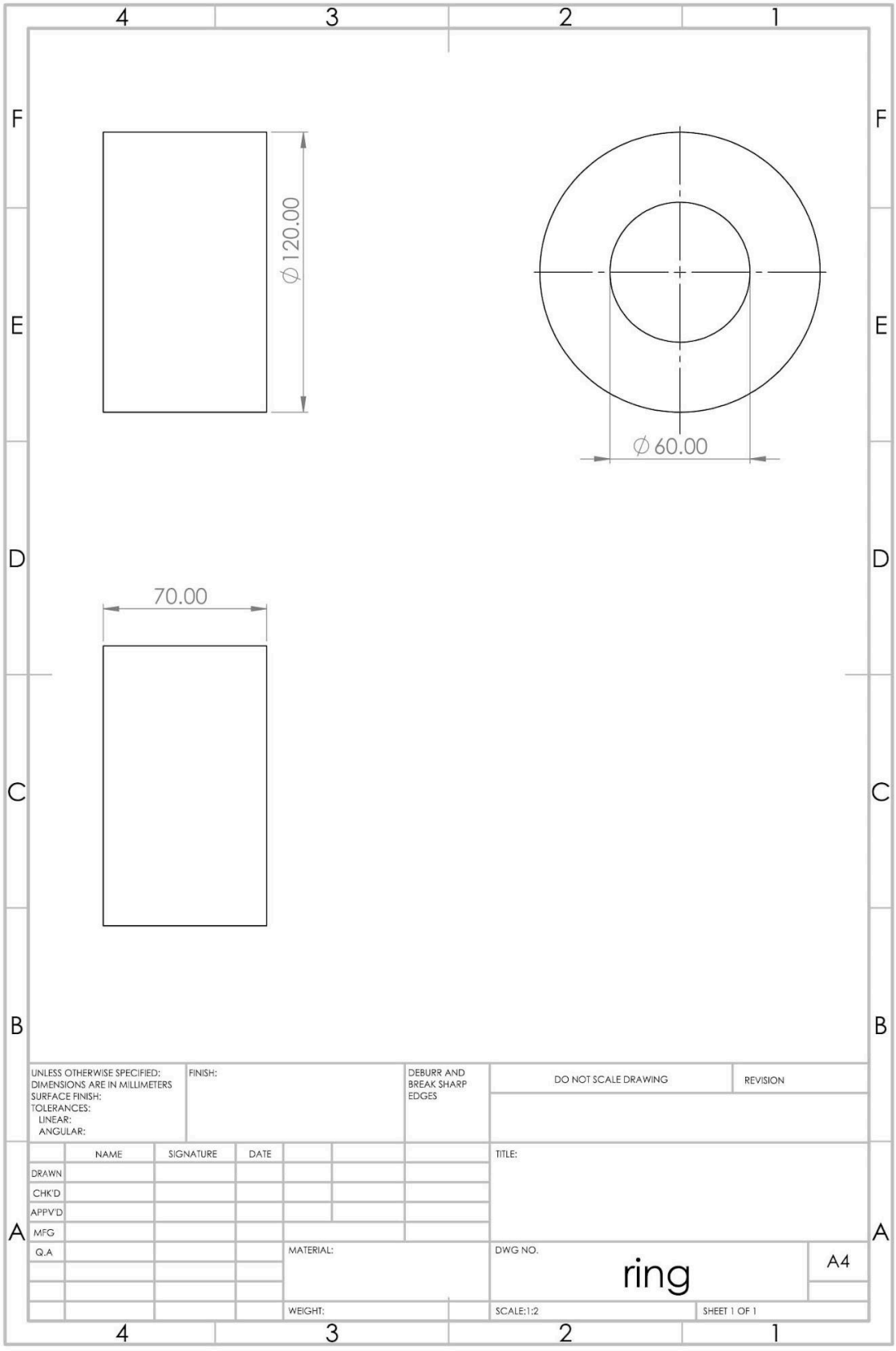
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APPV'D								
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Q.A				MATERIAL:			DWG NO. Motor	A4
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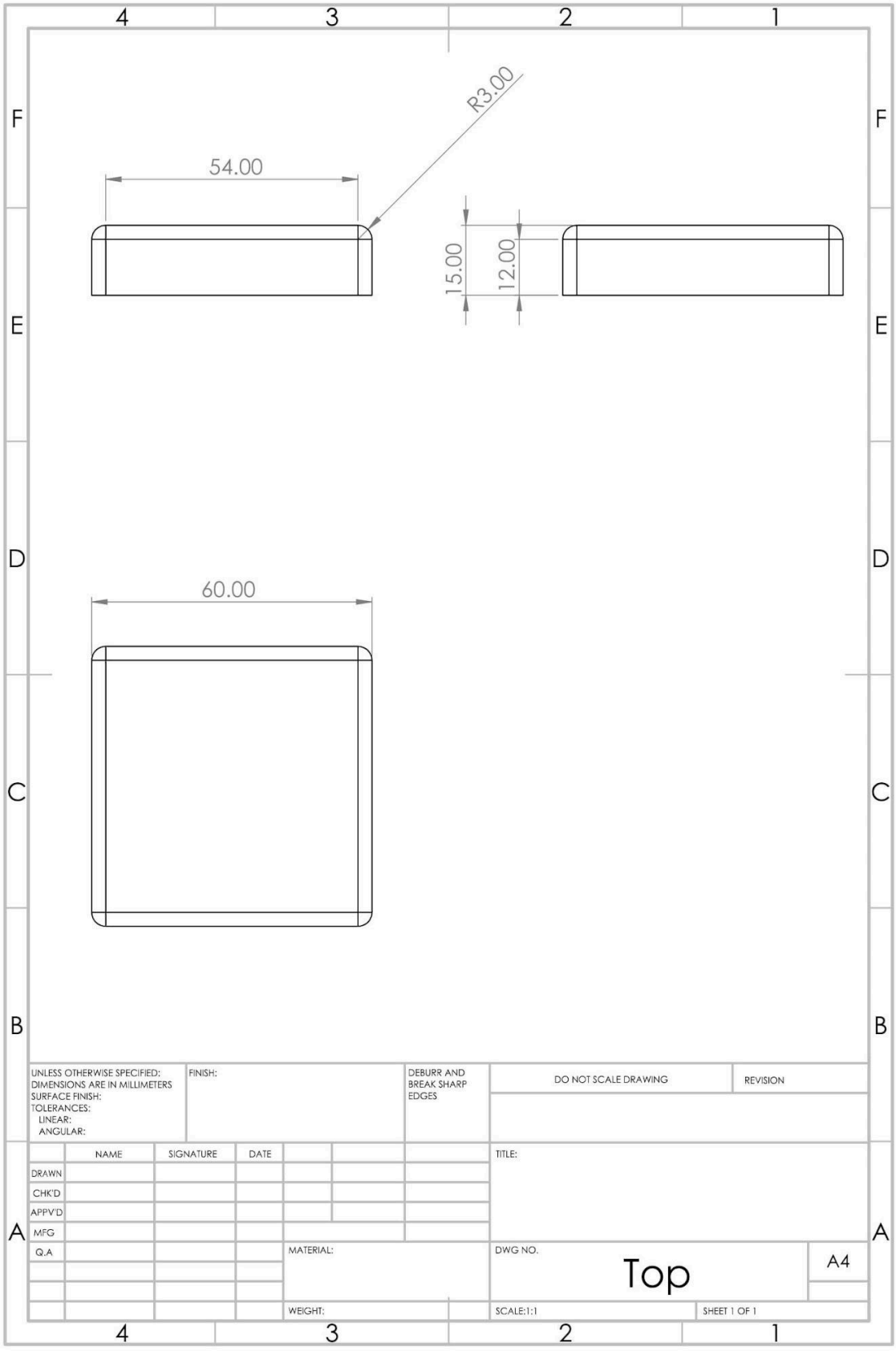

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SHEET 1 OF 1


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SHEET 1 OF 1


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WEIGHT:
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SHEET 1 OF 1


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SHEET 1 OF 1


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MATERIAL:
DWG NO.
ring
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WEIGHT:
SCALE:1:2
SHEET 1 OF 1


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CHK'D
APPV'D
MFG
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TITLE:
MATERIAL:
DWG NO.
Top
A4
WEIGHT:
SCALE:1:1
SHEET 1 OF 1

Gearbox Design

Given that:

$$H = 5.5\ kW$$

$$n_1 = 960\ RPM$$

$$i = 4$$

Assuming $N_1 = 18\ teeth$

Selecting the module $m \approx 2.3$ so standard $m = 2.5$

$$d_1 = mN_1 = 2.5 * 18 = 45\ mm$$

$$d_2 = mN_2 = miN_1 = 180\ mm$$

$$C = \frac{d_1+d_2}{2} = \frac{45+180}{2} = 112.5\ mm$$

$$F = 4\pi m = 10\pi = 31.416\ mm \Rightarrow take\ F = 32mm$$

Bending Stress on Pinion:

$$\sigma = \frac{W_t}{FK_v mY}$$

$$W_t = \frac{H}{V} = \frac{H}{w_1 r_1} = \frac{H*60}{w_1 2\pi r_1} = 2.432\ kN$$

$$v = 2.262 \frac{m}{s}$$

$$K_v = \frac{6}{6+V} = 0.7262$$

$$Y = 0.29327\ from\ table\ 2$$

$$\sigma = 142.741\ MPa$$

$$\sigma_{design} = 4\sigma = 570.965\ MPa$$

Hence we will use **Carbon steel AISI 1040 OQT400** as $S_y = 600 > \sigma_{design}$

Bending Fatigue:

$$S_e^{'} = 0.5 S_{ult} = 0.5 * 779 = 389.5 MPa$$

$$K_a = 0.71$$

$$K_b = 0.974$$

$K_c = 0.814 \quad (99\%\ reliability)$

$$K_d = K_e = K_f = 1$$

$$S_e = 219.255\ MPa$$

$$\sigma_a = \frac{W_t}{K_v F m J}$$

$K_v = 0.82$ as Q = 7 => For a mining conveyor

$$J = 0.317$$

$$\sigma_a = 116.95\ MPa$$

$$n_G = \frac{S_y}{\sigma_a} = 1.87$$

$$K_o = 1.25$$

$$K_m = 1.3$$

$$n = \frac{n_G}{K_O K_m} = 1.15 > 1$$

Surface Durability

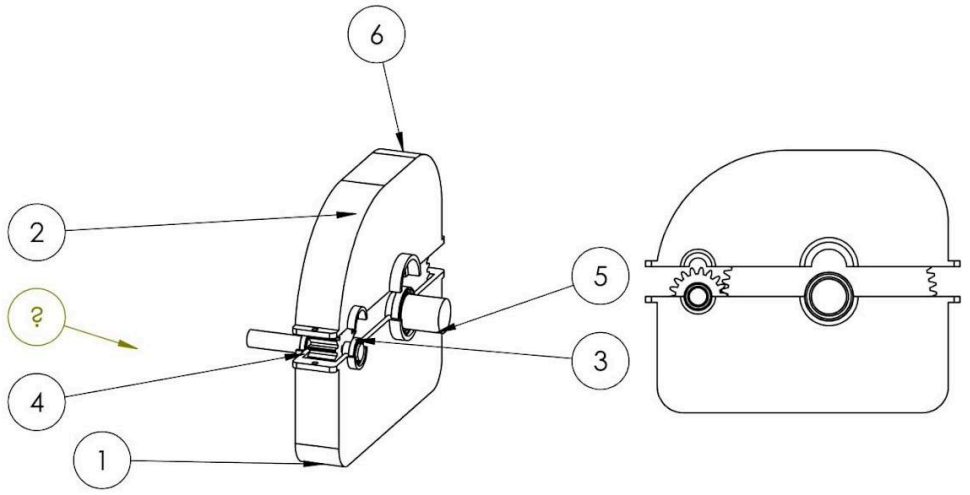
$$S_c = 2.76HB - 70 = 653.12$$

$$S_H = \frac{C_L C_H}{C_T C_R} S_c = \frac{1.1*1}{1*0.8} * 653.12 = 898.04$$

$$\sigma_H = C_P \sqrt{\frac{W_{t,p}}{C_v F d_1 I}} = 191 * \sqrt{\frac{2.432K}{0.82*32*45*0.9}} = 288.939$$

$$S_H = \sqrt{K_o K_m n}\, \sigma_H$$

$$n = 5.944 > 1$$



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	coverBottom		1
2	Metric - Spur gear 2.5M 72T 20PA 25FW ---S72N75H50L30N		1
3	Metric - Spur gear 2.5M 18T 20PA 25FW ---S18N75H50L15S1		1
4	shaft		1
5	shaftDriven		1
6	coverTop		1
7	SKF - 61806 - 26,SI,NC,26_68		2
8	key		1
9	SKF - 61802 - 18,SI,NC,18_68		2

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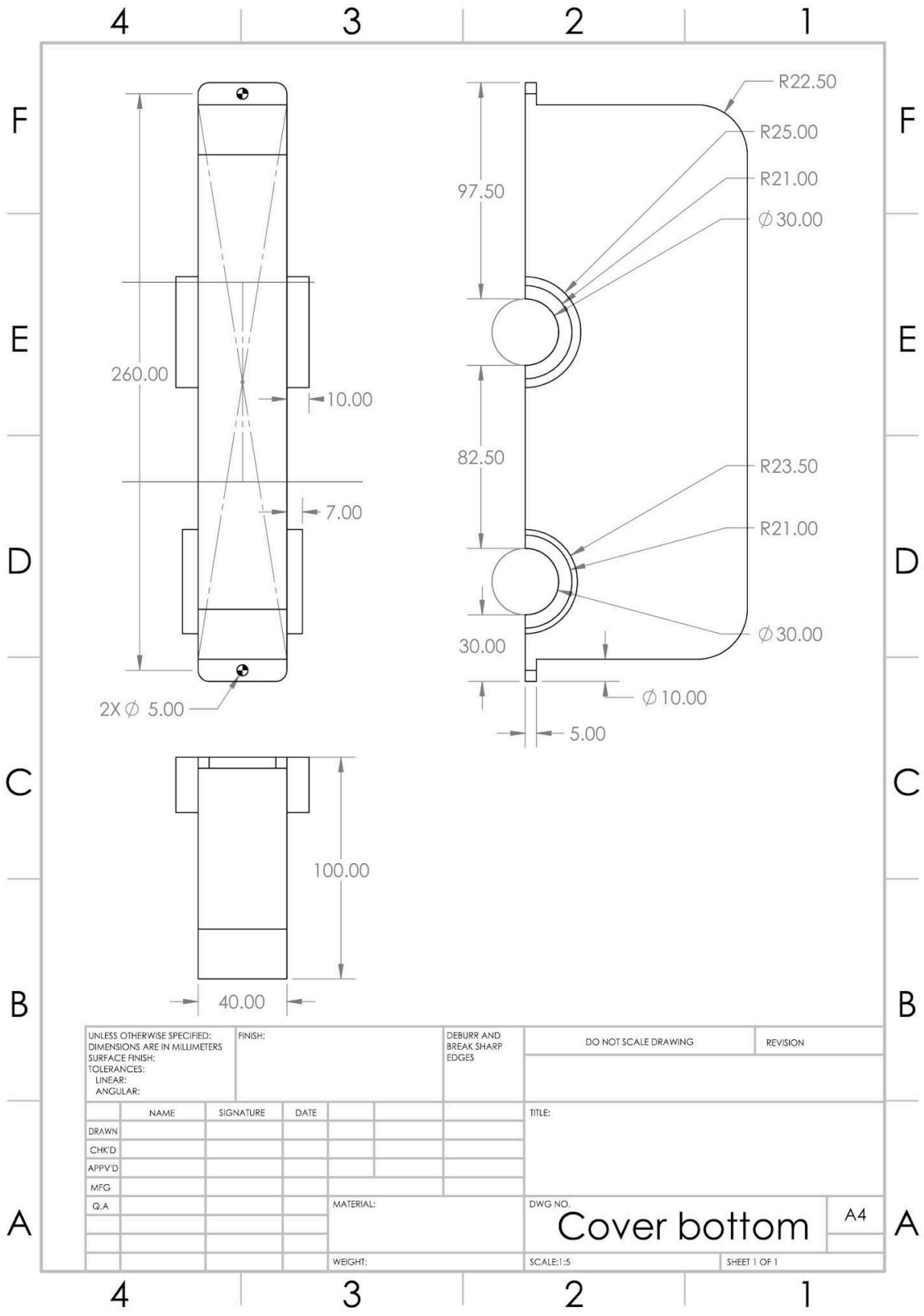
Gearbox Assembly

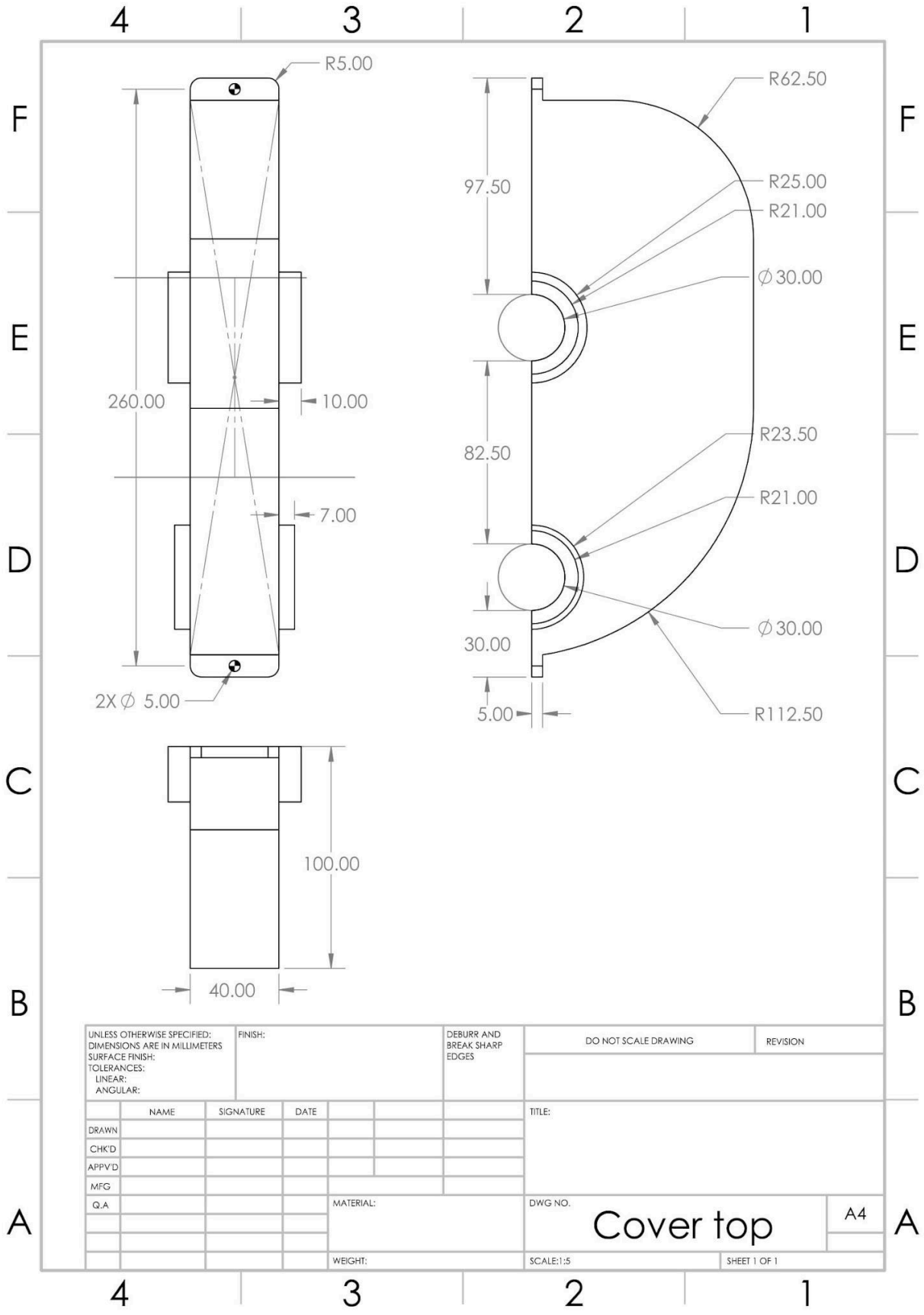
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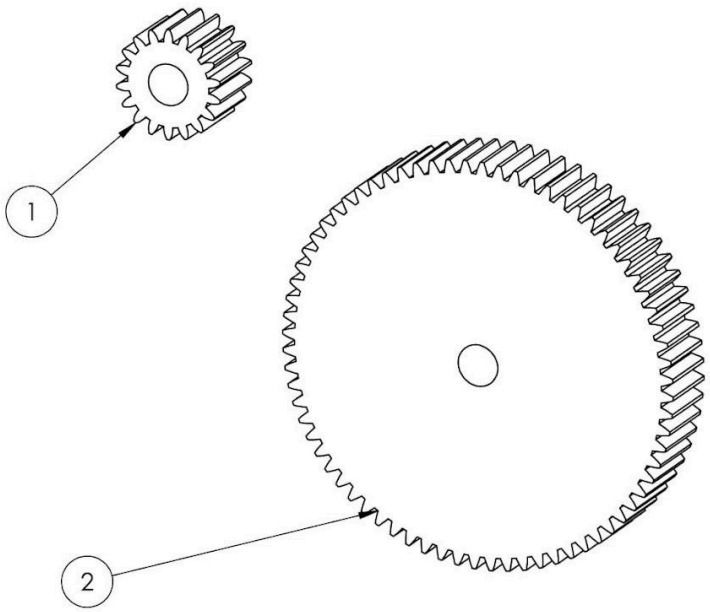
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SHEET 1 OF 1

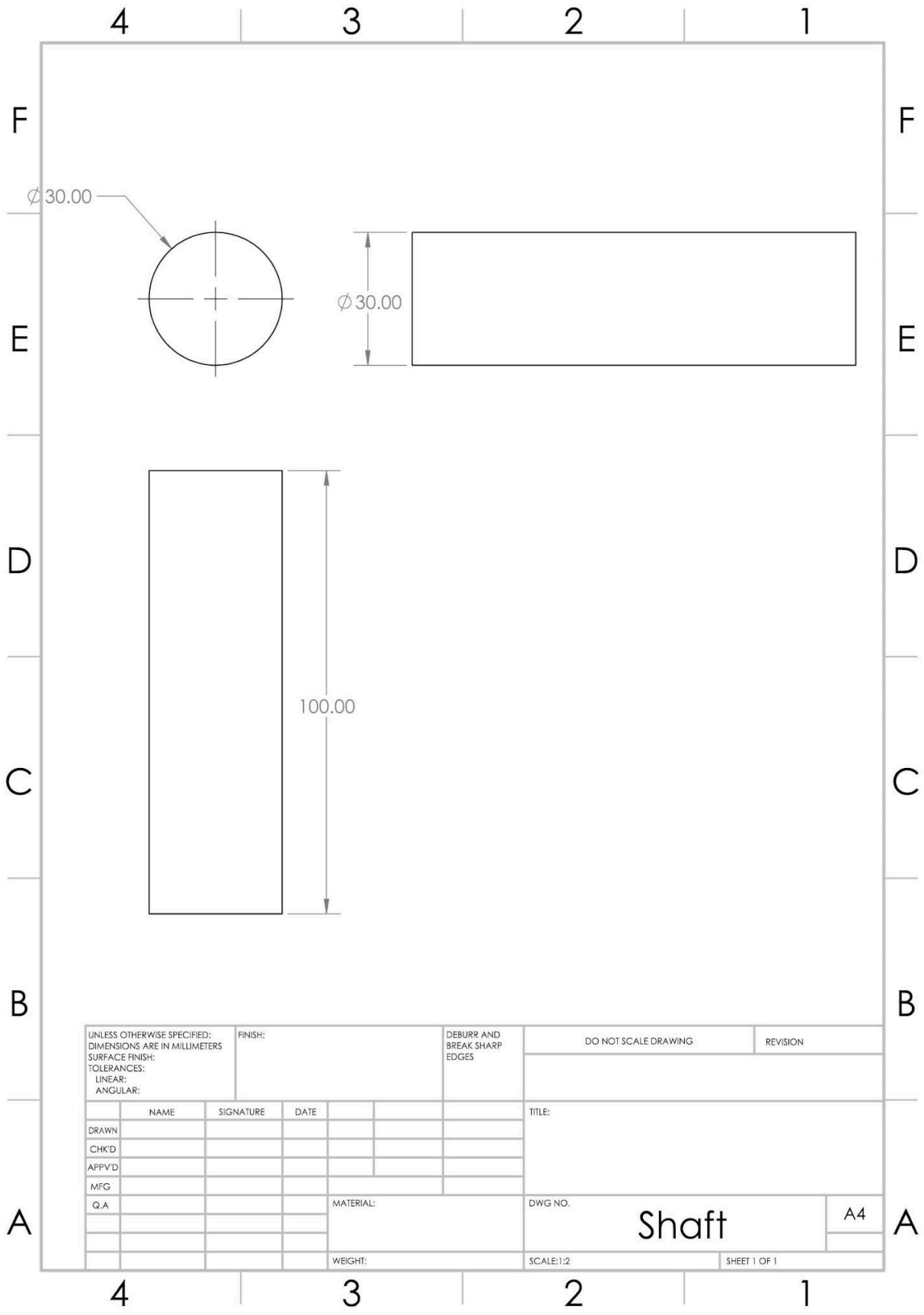

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Cover bottom
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SHEET 1 OF 1

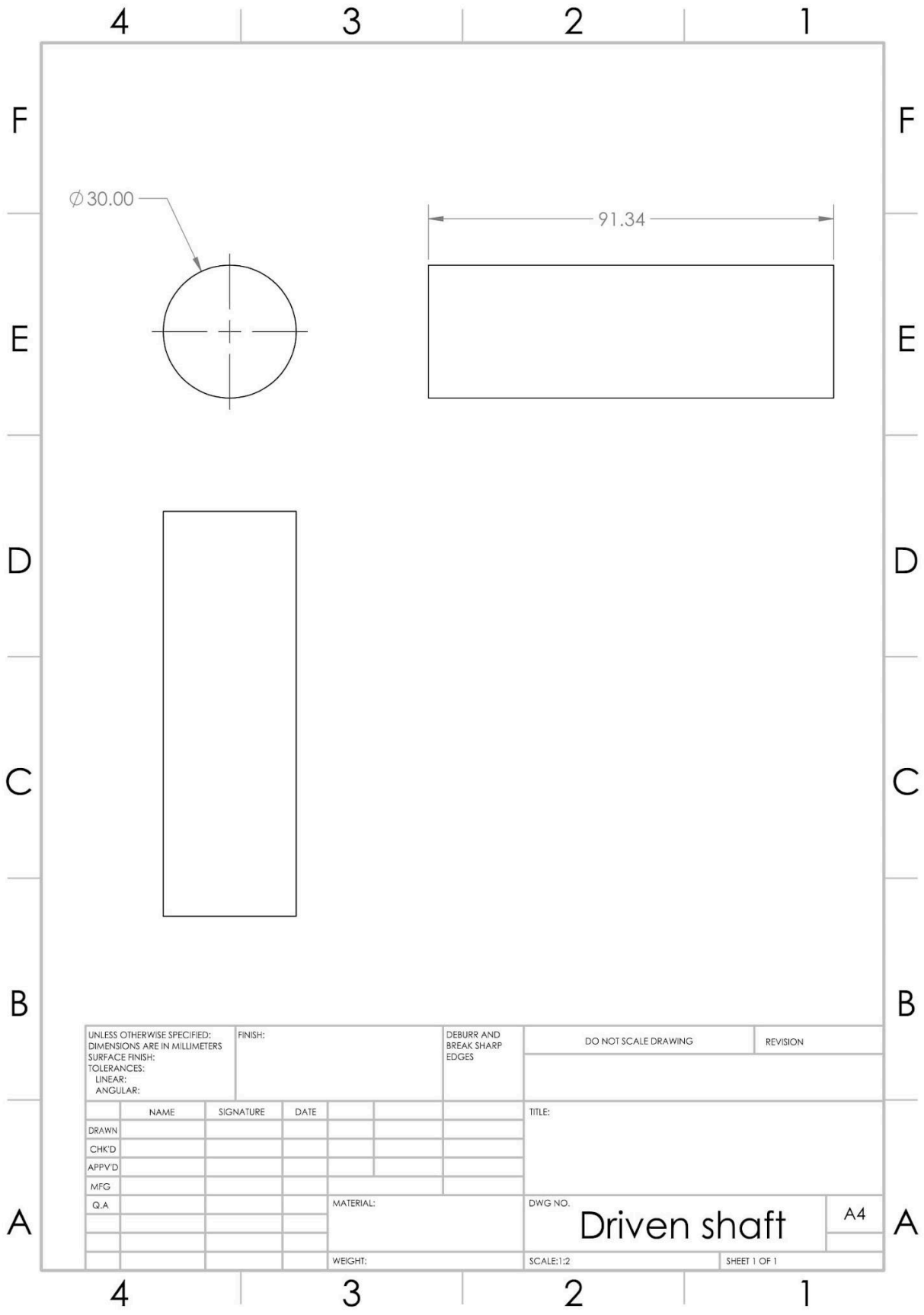

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MATERIAL:
DWG NO.
Cover top
A4
WEIGHT:
SCALE:1:5
SHEET 1 OF 1



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Metric - Spur gear 2M 18T 20PA 25FW --- S18N75H50L15N		1
2	Metric - Spur gear 2M 72T 20PA 25FW --- S72N75H50L15N		1

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MATERIAL:
DWG NO.
Shaft
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SHEET 1 OF 1


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Driven shaft
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SHEET 1 OF 1

Bearings on Gearbox

For the bearings selection we need to calculate the actual loading capacity:

$$C = P'\left(\frac{60 \times L \times n}{10^6}\right)^{\frac{1}{3}}$$

The bearings are used with rollers having the following characteristics:

- $Radial\ load\ (F_r) = \frac{0.88517}{2} = 0.442585\ kN$
- $Axial\ load\ (F_a) = \frac{2.432}{2} = 1.216 kN$
- $Shaft\ diameter\ (d) = 30\ mm$

  Using this, we selected a bearing **6406** from the SKF catalogue as follows:

30	42	7	4,49	2,9	0,146	32 000	20 000	0,025	**61806**
	47	9	7,28	4,55	0,212	30 000	19 000	0,049	**61906**
	55	9	11,9	7,35	0,31	28 000	17 000	0,089	* **16006**
	55	13	13,8	8,3	0,355	28 000	17 000	0,12	* **6006**
	62	16	20,3	11,2	0,475	24 000	15 000	0,2	* **6206**
	62	16	23,4	12,9	0,54	24 000	15 000	0,18	**6206 ETN9**
	72	19	29,6	16	0,67	20 000	13 000	0,35	* **6306**
	72	19	32,5	17,3	0,735	22 000	14 000	0,33	**6306 ETN9**
	90	23	43,6	23,6	1	18 000	11 000	0,75	**6406**

- $Shaft\ rotational\ speed\ (n) = 960\ rpm$

Start of calculations:

1. $C = 43.6\ kN\ and\ C_o = 23.6\ kN$

$$\frac{F_a}{C_o} = 0.0515$$

Using interpolation we find the values of $e,\ x\ and\ y$ as follows

$$\frac{0.0515-0.04}{0.07-0.04} = \frac{e-0.24}{0.27-0.24},\ e = 0.2515$$

$$x = 0.56$$

$$\frac{0.0515-0.04}{0.07-0.04} = \frac{y-1.8}{1.6-1.8},\ y = 1.72$$

$$P = VXF_r + YF_a = 2.34\ kN$$

2. By selecting $K_{sf} = 1.5$ from the table, then :

$$P^{'} = K_{sf} \times P = \ 1.5 \times 122.625 \ = 3.51\ kN$$

3. By choosing L= 20,000 hrs. for machines used 8 hours daily:

$$C = P^{'}\left(\frac{60 \times L \times n}{10^6}\right)^{\frac{1}{3}} = 5.03 \ * \ 1000\left(\frac{60 \times 20000 \times 960}{10^6}\right)^{\frac{1}{3}} = 36.79\ kN$$

Therefore, $C_{calculated} < C_{table}$.

4. Calculating the static loading:

$$P_O = \ 0.6\, F_r + \ 0.5\, F_a = \ 0.873551\ kN.$$

For normal and normal vibrations, $S_O = \ 1$ .Then:

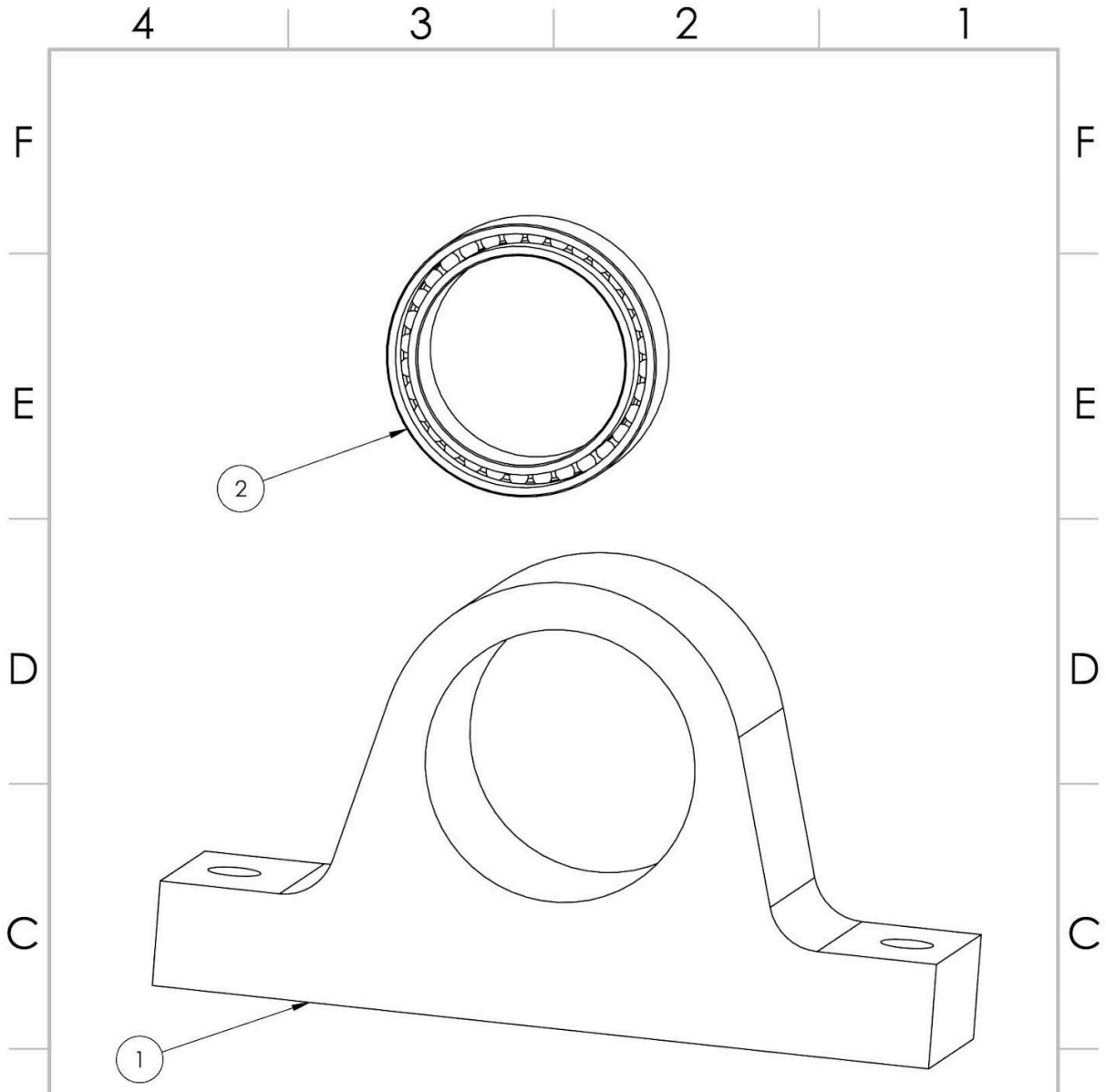
$$C_O = S_O \times P_O = \ 1 \ * P_0 = \ 0.873551 \ \ kN$$

Therefore, $C_O < C_{Otable}$.

5. Checking for limiting speed: $n < n_{table}$

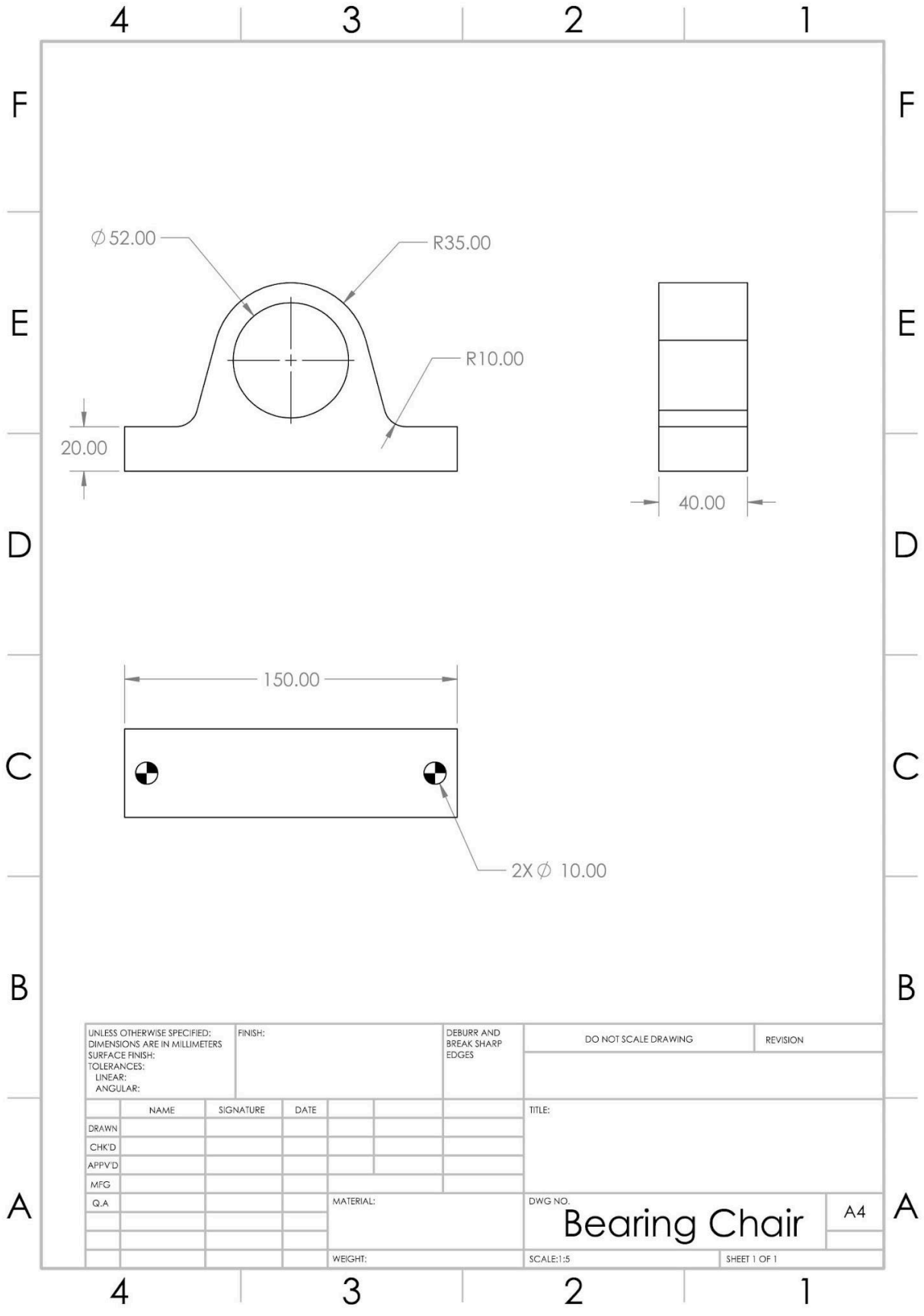
Now, after all these checks, we can consider the bearing designation **6406** as the right bearing selection for our rollers.

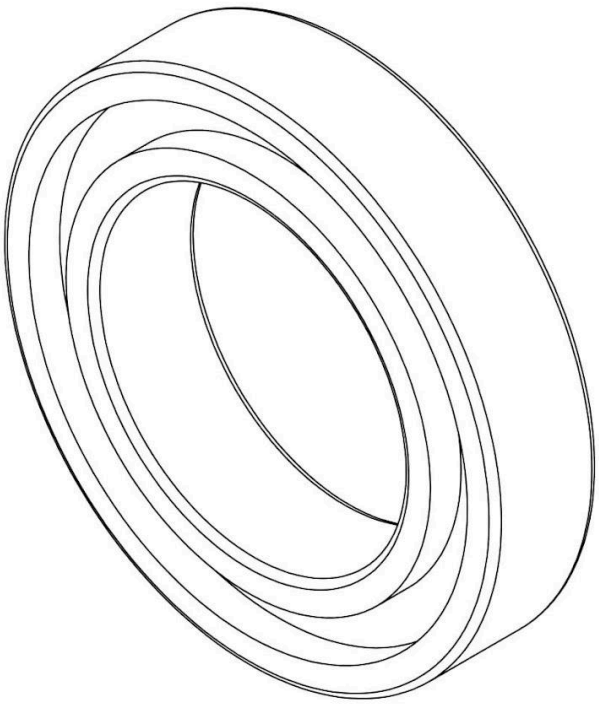
And since the bigger gear is affected by less force we will use the same bearing.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Bearing Chair		1
2	bearingRoller		1

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		TITLE:		
DRAWN							
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL:	DWG NO.	A4	
					Bearing assembly		
				WEIGHT:	SCALE:1:5	SHEET 1 OF 1	


4
3
2
1
F
E
D
C
B
A
∅ 52.00
R35.00
R10.00
20.00
40.00
150.00
2X ∅ 10.00
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
TITLE:
MATERIAL:
DWG NO.
Bearing Chair
A4
WEIGHT:
SCALE:1:5
SHEET 1 OF 1



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		TITLE:		
DRAWN							
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL:	DWG NO.		A4
					radial ball bearing_68_skf		
				WEIGHT:	SCALE:2:1	SHEET 1 OF 1	

Couplings calculations

I. Power=5.5 kw, RPM=960 RPM, Number of bolts=4, Pitch diameter=30mm, Factor of safety=3

$$T = \frac{POWER}{W} = \frac{5.5*1000}{((2\pi*960))/60} = 54.70951169$$

$$\tau_y = 0.5\, S_y = 0.5 * 380 = 190MPa$$

$$\tau_d = \frac{\tau_y}{F.S} = \frac{190}{3} = 63.3MPa$$

$$d^2 = \frac{8T}{\pi DN\tau_d} = \frac{8*54.70951169*(10^3)}{\pi*30*4*63.33} = 18.3311352$$

$$d = 4.281487499mm$$

$$d_o = \frac{4.281487499}{0.85} = 5.037044116mm$$

Outer diameter of hub: $D = 2d = 2(30) = 60mm$

Pitch circle diameter of the bolts: $D_1 = 3d = 3(30) = 90mm$

Outer diameter of flange: $D_2 = 4d = 4(30) = 120mm$

Length of the hub: $L = 1.5d = 1.5(30) = 45mm$

Thickness of flange: $t_f = 0.5d = 0.5(30) = 15mm$

Thickness of protective circumferential flange: $t_p = 0.25d = 0.25(30) = 7.5mm$

Then **M6, 4 Bolts**

II. Power=5.5 kw, RPM=480 RPM, Number of bolts=4, Pitch diameter=30mm, Factor of safety=3

$$T = \frac{POWER}{W} = \frac{5.5*1000}{((2\pi*480))/60} = 109.4190234$$

$$\tau_y = 0.5\, S_y = 0.5 * 380 = 190MPa$$

$$\tau_d = \frac{\tau_y}{F.S} = \frac{190}{3} = 63.3MPa$$

$$d^2 = \frac{8T}{\pi DN\tau_d} = \frac{8*109.4190234*(10^3)}{\pi*30*4*63.33} = 36.6622704$$

$$d = 6.05493768mm$$

$$d_o = \frac{6.05493768}{0.85} = 7.123456mm$$

Outer diameter of hub: $D = 2d = 2(30) = 60mm$

Pitch circle diameter of the bolts: $D_1 = 3d = 3(30) = 90mm$

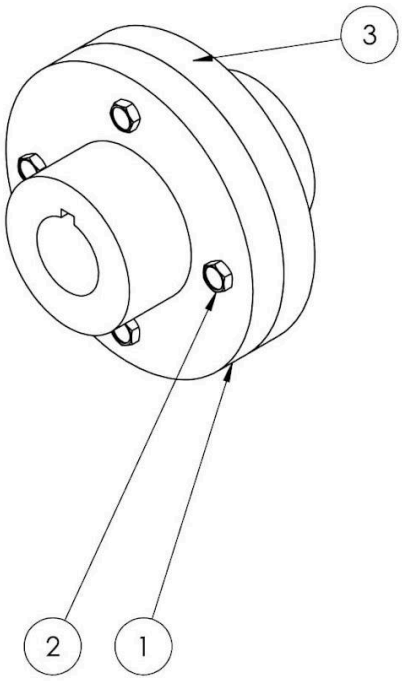
Outer diameter of flange: $D_2 = 4d = 4(30) = 120mm$

Length of the hub: $L = 1.5d = 1.5(30) = 45mm$

Thickness of flange: $t_f = 0.5d = 0.5(30) = 15mm$

Thickness of protective circumferential flange: $t_p = 0.25d = 0.25(30) = 7.5mm$

Then **M8 , 4 BOLTS**



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	coupling, m6		2
2	B18.2.3.2M - Formed hex screw, M6 x 1.0 x 35 --18CN		4
3	B18.2.4.5M - Hex jam nut, M6 x 1 --D-N		4

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			

TITLE:

MATERIAL:

DWG NO.

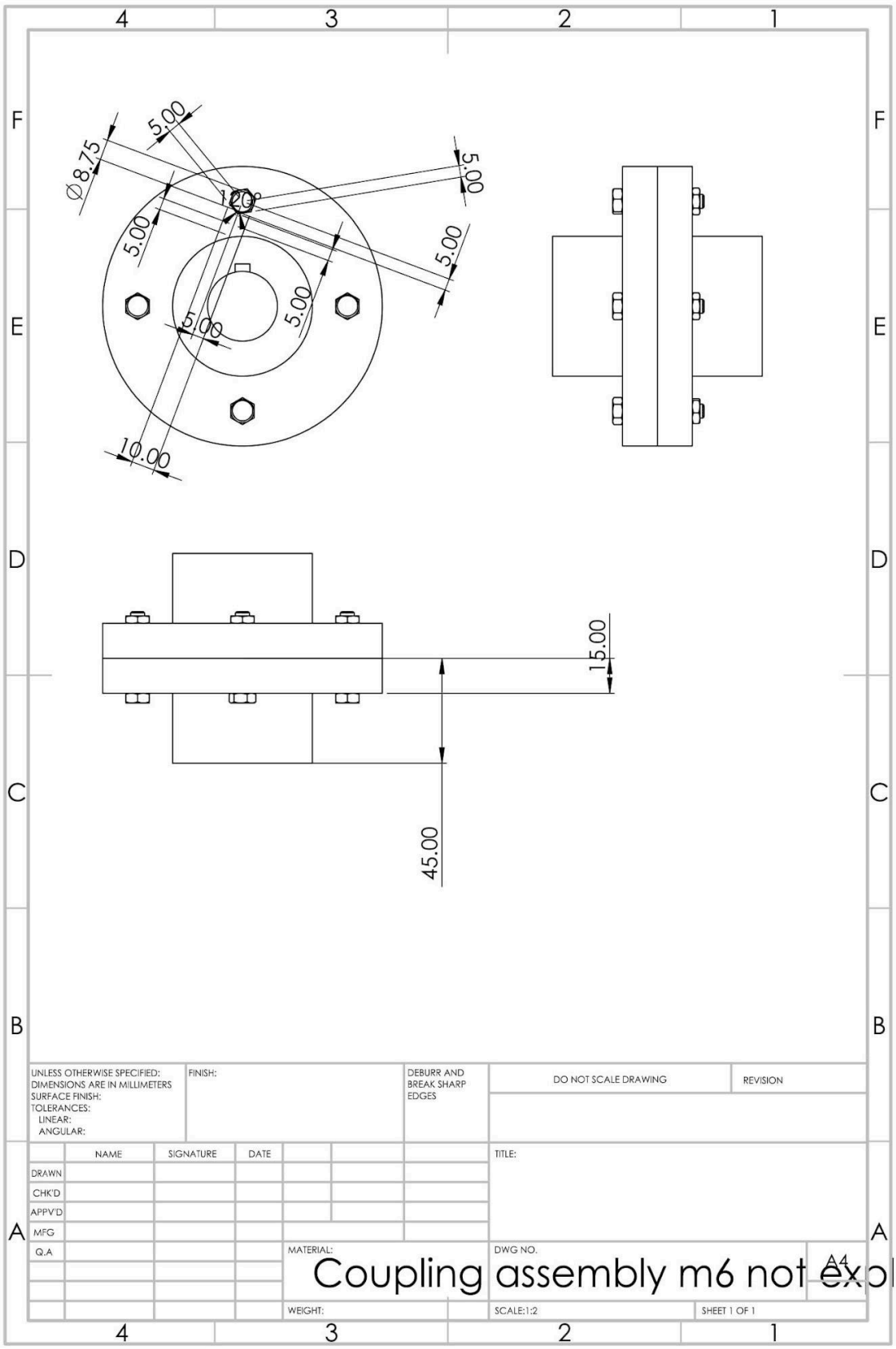
Coupling assembly m6

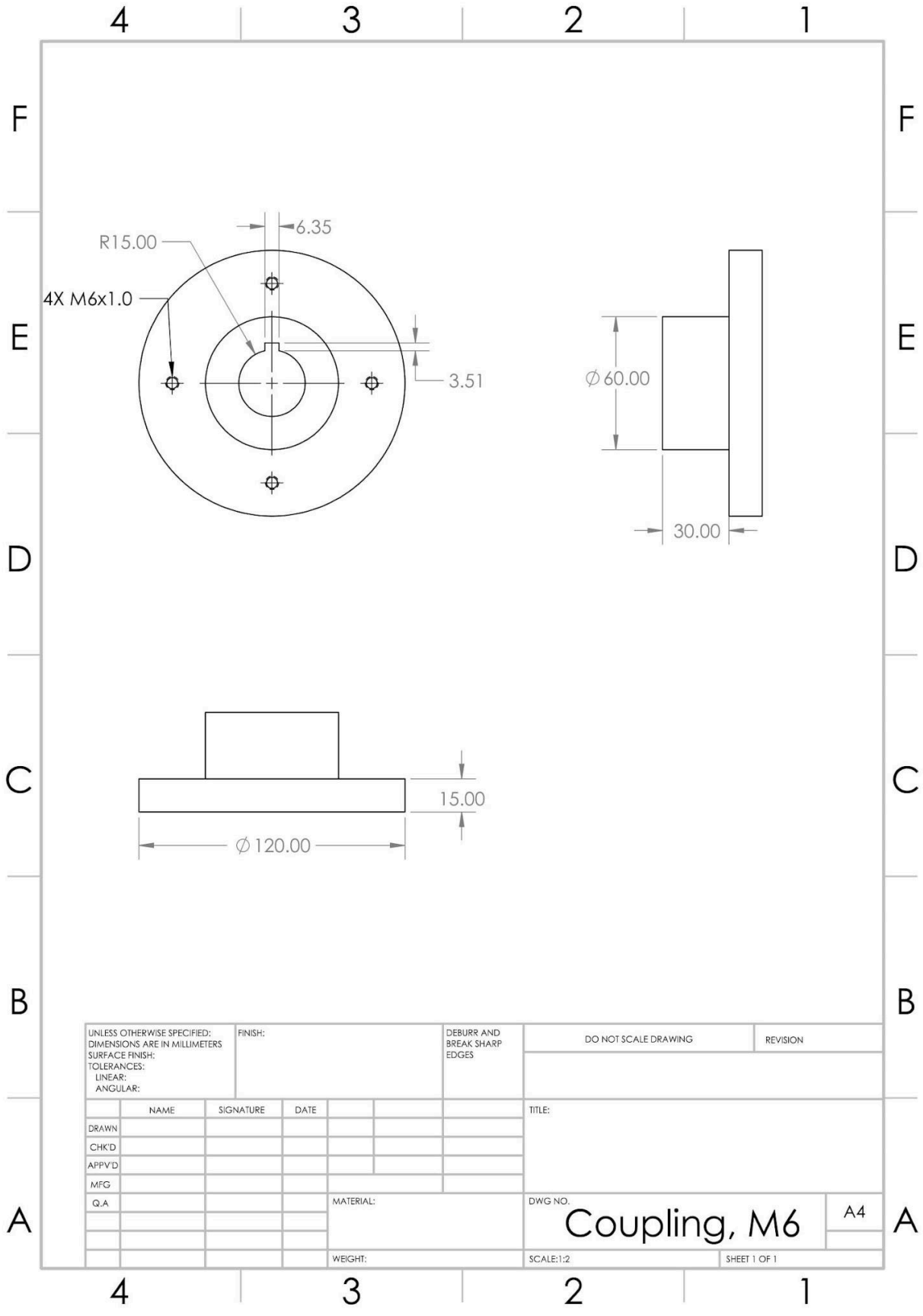
A4

WEIGHT:

SCALE:1:5

SHEET 1 OF 1


4
3
2
1
F
E
D
C
B
A
Ø8.75
5.00
5.00
120°
5.00
5.00
5.00
5.00
10.00
15.00
45.00
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
TITLE:
MATERIAL:
DWG NO.
A4
Coupling assembly m6 not explo
WEIGHT:
SCALE:1:2
SHEET 1 OF 1


4
3
2
1
F
E
D
C
B
A
R15.00
6.35
4X M6x1.0
3.51
∅60.00
30.00
15.00
∅120.00
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
TITLE:
MATERIAL:
DWG NO.
Coupling, M6
A4
WEIGHT:
SCALE:1:2
SHEET 1 OF 1

Choosing the proper chain for rollers

1. Calculating the power rating:

$$Power\ rating\ = (kW\ to\ be\ transmitted) * \frac{Ks}{K1*K2}$$

$$= 5.5\ * \frac{1}{1.26*1} = 4.365\ kW$$

Taking Ks=1 (smooth light conveyor)

In addition, K2=1.26 (the number of teeth of the smaller Sprocket (pinion) to be 21).

And K1 = 1 (only one strand)

2. Selection of chain:

Using 12B chain we will interpolate with pinion speed n=240 rpm:

$$Power(kW) = \frac{200-240}{200-300} = \frac{3.75-x}{3.75-5.43}$$

$$Power\ = 4.42\ kW$$

Which is greater than the Power rating (4.365 kW), then this chain is suitable.

3. Pitch circle diameter of driving and driven pulleys:

From the tables:

Pitch $p\ = 19.05\ mm$

$$D1\ = \frac{p}{\sin sin\left(\frac{180}{z1}\right)} = \frac{19.05}{\sin sin\left(\frac{180}{21}\right)} = 127.816mm$$

$$\frac{N_{(driven)}}{N_{(driver)}} = \frac{n_{(driver)}}{n_{(driven)}}$$

$$i = \frac{n_{(driver)}}{n_{(driven)}} = \frac{240}{n_{(driven)}} = 2$$

$$n_{driven} = 120\ rpm$$

$$z2(No\ of\ teeth\ of\ the\ bigger\ Sprocket) = 30\ * \left(\frac{240}{120}\right) = 60\ teeth$$

$$D2\ = \frac{19.05}{\sin sin\left(\frac{180}{42}\right)} = 254.9\ mm$$

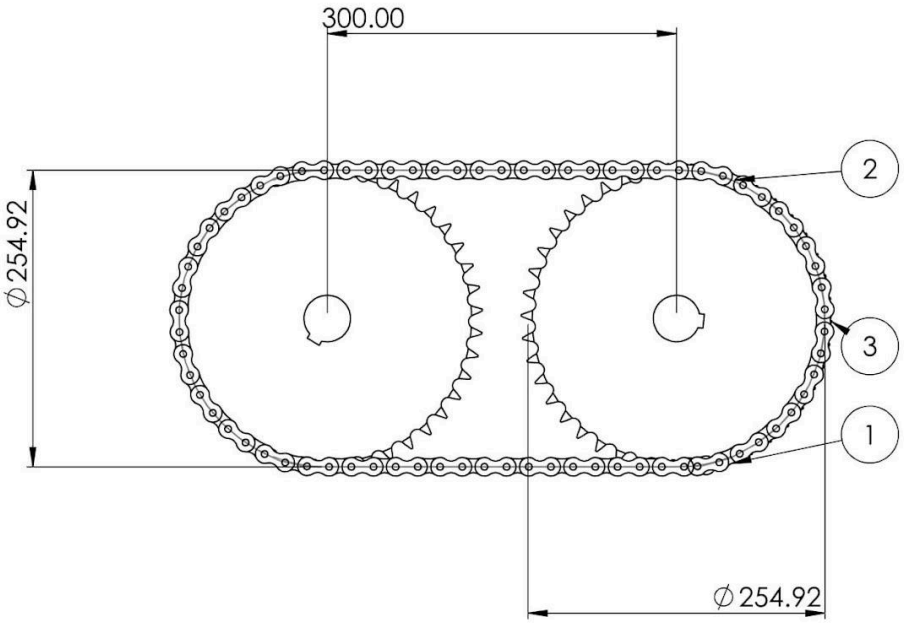
4. Number of chain links:

$$a\ = 40\ *\ p\ = 762\ mm$$

$$Ln(No\ of\ links) = 2 * \left(\frac{a}{p}\right) + \frac{z1+z2}{2} + \left(\frac{z2-z1}{2*\pi}\right)^2 * \frac{p}{a} = 111.77\ or\ 112\ links$$

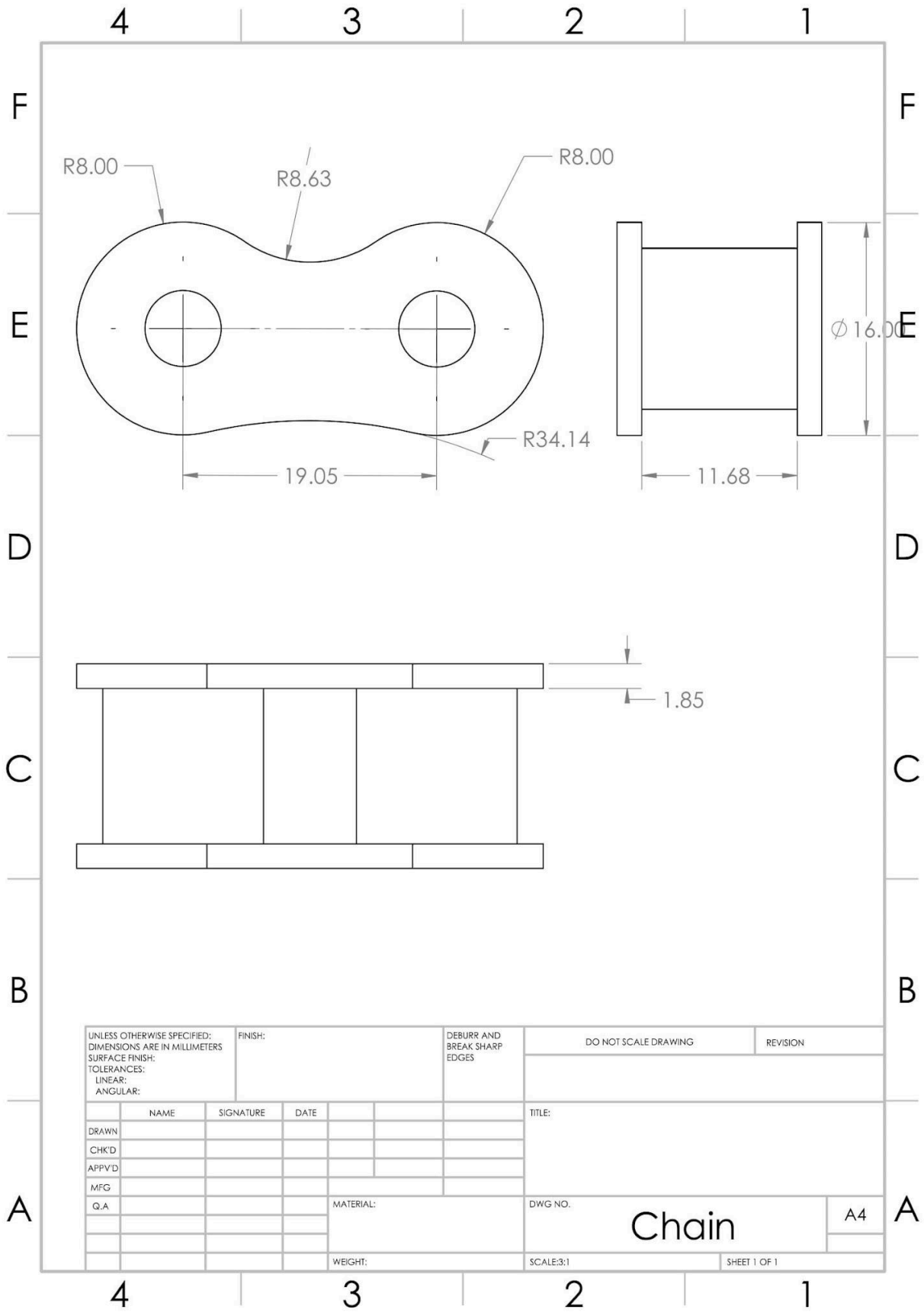
Last step we will substitute in the same equation to calculate the new center to center distance according to our approximation for Ln

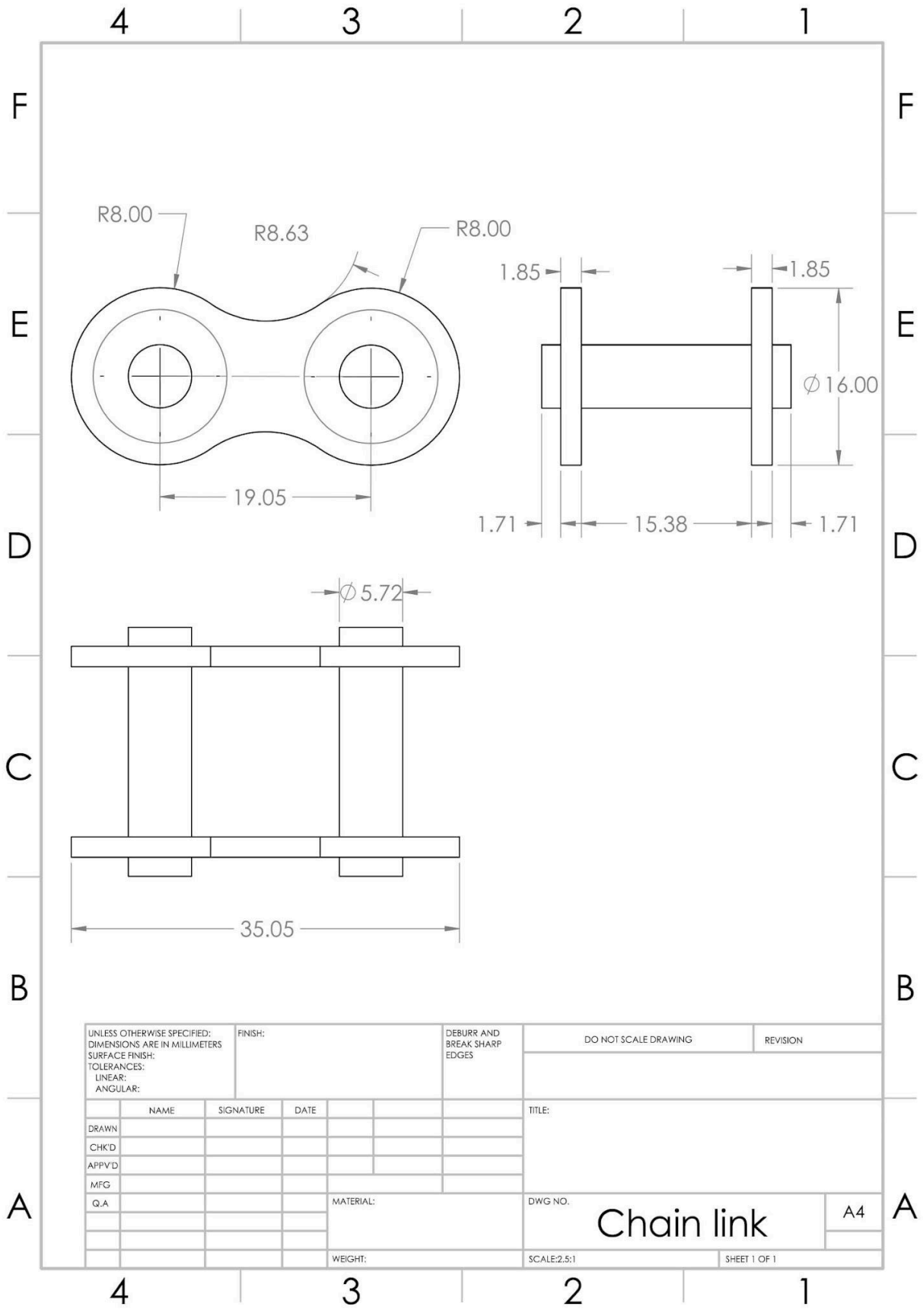
$$a = 764\ mm$$



ITEM NO.	PART NUMBER	QTY.
1	Chain	37
2	chainLink	37
3	Chain wheel ISO - 42Z 12B-1 --42SA40.0R1	2

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				TITLE:	
DRAWN								
CHK'D								
APPV'D								
MFG								
Q.A				MATERIAL:			DWG NO. chainSprocketsRollers	A4
				WEIGHT:			SCALE:1:10	SHEET 1 OF 1


4
3
2
1
F
E
D
C
B
A
R8.00
R8.63
R8.00
R34.14
19.05
∅16.00
11.68
1.85
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
TITLE:
MATERIAL:
DWG NO.
Chain
A4
WEIGHT:
SCALE:3:1
SHEET 1 OF 1


4
3
2
1
F
E
D
C
B
A
R8.00
R8.63
R8.00
19.05
1.85
1.85
∅16.00
1.71
15.38
1.71
∅5.72
35.05
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
TITLE:
MATERIAL:
DWG NO.
Chain link
A4
WEIGHT:
SCALE:2.5:1
SHEET 1 OF 1

Bearings Selection

For the bearings selection we need to calculate the actual loading capacity:

$$C = P'\left(\frac{60 \times L \times n}{10^6}\right)^{\frac{1}{3}}$$

The bearings are used with rollers having the following characteristics:

- $Radial\ load\ (F_r) = \frac{Box\ weight}{4\ bearings} + \frac{Shaft\ weight}{2\ bearings} + \frac{Roller\ weight}{2\ bearings}$

$$= \left(\frac{50}{4} + \frac{10}{2} + \frac{15.232}{2}\right) \times 9.81 = 246.4\ N$$

- $Axial\ load = 0$
- $Shaft\ diameter\ (d) = 40\ mm$

Using this, we selected a bearing from the SKF catalogue as follows:

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
40	52	7	4,49	3,75	0,16	26 000	16 000	0,032	**61808**
	62	12	13,8	10	0,425	24 000	14 000	0,12	**61908**
	68	9	13,8	10,2	0,44	22 000	14 000	0,13	* **16008**
	68	15	17,8	11	0,49	22 000	14 000	0,19	* **6008**
	80	18	32,5	19	0,8	18 000	11 000	0,37	* **6208**
	80	18	35,8	20,8	0,88	18 000	11 000	0,34	**6208 ETN9**
	90	23	42,3	24	1,02	17 000	11 000	0,63	* **6308**
	110	27	63,7	36,5	1,53	14 000	9 000	1,25	**6408**

- $Shaft\ rotational\ speed\ (n) = 120\ rpm$

Start of calculations:

1. Since we have no axial load, then:

$$P = F_r = 246.4\ N$$

2. By selecting $K_{sf} = 1.5$ from the table, then :

$$P' = K_{sf} \times P = 1.5 \times 246.4 = 369.6\ N$$

3. By choosing L= 20,000 hrs. for machines used 8 hours daily:

$$C = P'\left(\frac{60 \times L \times n}{10^6}\right)^{\frac{1}{3}} = 369.6\left(\frac{60 \times 20000 \times 120}{10^6}\right)^{\frac{1}{3}} = 1937.25\ N$$

Therefore, $C_{calculated} < C_{table}$.

4. Calculating the static loading:

$$P_O = 0.6\,F_r + 0.5\,F_a = 0.6 \times 246.4 + 0 = 147.84\ N$$

but since $P_0 < F_r$ we take $P_O = F_r = 246.4\ N$.

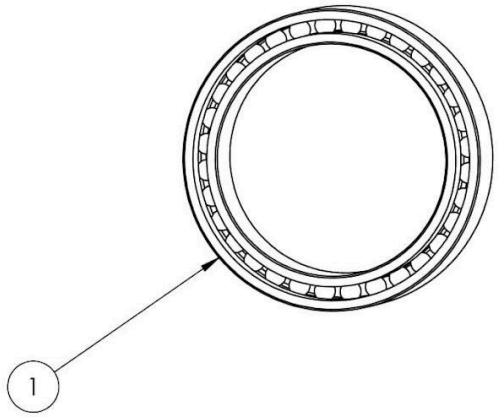
For normal and normal vibrations, $S_O = 1$ .Then:

$$C_O = S_O \times P_O = 1 \times 246.4 = 246.4\ N$$

Therefore, $C_O < C_{Otable}$.

5. Checking for limiting speed: $n < n_{table}$

Now, after all these checks, we can consider the bearing designation **61808** as the right bearing selection for our rollers.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	AFBMA 20.1 - 48-40 - 32,DE,NC,32_68		1

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				TITLE:
DRAWN							
CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL:			DWG NO. Bearing Roller / A4
				WEIGHT:			SCALE:1:1 / SHEET 1 OF 1