MIET 2136 Mechanical Design 1
Assignment 1 SEMESTER II 2024
PULLEY REDUCTION DRIVE AND GEARBOX DESIGN

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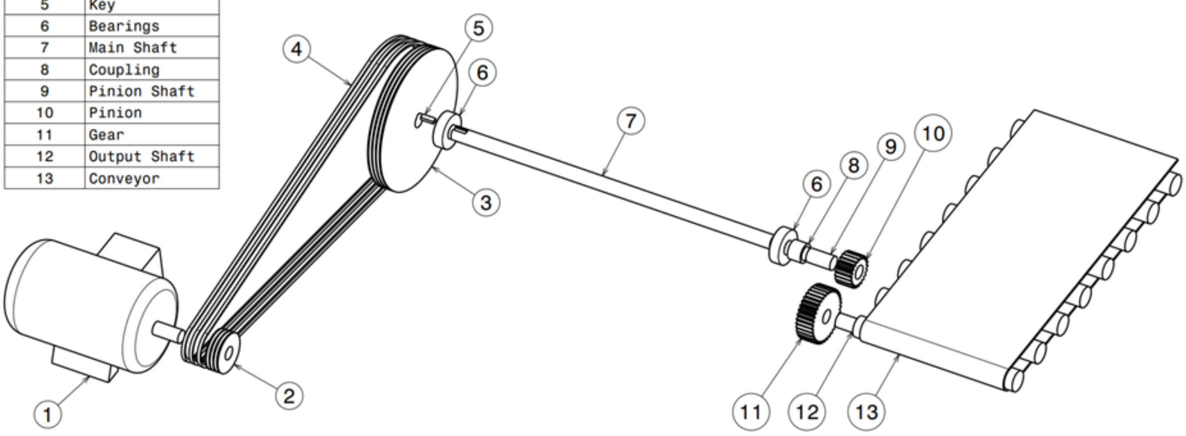
The general concept and layout of a combination: belt driven torque increasing system and torque increasing gearbox has progressed to the stage where the detailed design can now be undertaken. The general system layout is as follows:

Part No.	Part Name
1	Electric Motor
2	Small Pulley
3	Large Pulley
4	V-Belts
5	Key
6	Bearings
7	Main Shaft
8	Coupling
9	Pinion Shaft
10	Pinion
11	Gear
12	Output Shaft
13	Conveyor

Layout Exploded View



The detail design will be undertaken in three phases, firstly the <u>belt drive</u> system, secondly shaft drive system, and thirdly the <u>gearbox</u> system.

An alternating current (A/C) electric motor will be selected for your system. This motor will drive a pulley and belt system, which will alter the power transmission characteristics. The power will then be transmitted through a shaft, mounted to bearings to allow for rotational freedom while constraining the shaft radially. This shaft will then drive a one stage gear reduction.

The electric motor is a 3-phase, 4-pole (per phase) induction motor supplied with 400V and 50 Hz AC. A motor from the WEG catalogue such that the system can transmit a power of at least 12kW is selected.

W21-Cast iron frame motor
GB3(1) - IE2 (2)

Output		Frame	Full Load Torque (kgfm)	Locked Rotor Current Il/In	Locked Rotor Torque Tl/Tn	Break-down Torque Tb/Tn	Inertia J (kgm2)	Allowable locked rotor time (s)		Weight (kg)	Sound dB(A)	400 V							
												Rated speed (rpm)	% of full load						Full load current In (A)
													Efficiency			Power Factor			
kW	HP							Hot	Cold				50	75	100	50	75	100	
4P - 1500 RPM - 50Hz																			
0.12	0.16	63	0.080	4.5	2.4	2.9	0.0004	30	66	5.7	44.0	1435	51.0	57.0	60.0	0.51	0.60	0.70	0.412
0.25	0.33	71	0.170	4.4	2.2	2.3	0.0006	35	77	7.0	43.0	1400	59.0	65.0	68.5	0.55	0.68	0.76	0.693
0.37	0.5	71	0.260	4.3	2.0	2.0	0.0007	48	106	8.0	43.0	1380	63.0	68.0	72.7	0.50	0.62	0.72	1.02
0.55	0.75	80	0.370	5.8	2.1	2.6	0.0022	18	40	15.6	44.0	1440	73.0	76.0	77.1	0.55	0.68	0.75	1.37
0.75	1	80	0.520	6.0	2.6	2.9	0.0029	15	33	16.6	44.0	1410	79.0	79.5	79.6	0.63	0.76	0.83	1.64
1.1	1.5	90S	0.740	6.5	2.1	2.6	0.0049	14	31	20.6	49.0	1440	81.0	81.8	81.8	0.62	0.75	0.81	2.40
1.5	2	90L	1.01	6.5	2.4	2.8	0.0055	10	22	24.4	49.0	1450	81.5	83.0	83.0	0.57	0.70	0.78	3.34
2.2	3	100L	1.49	8.0	3.0	3.2	0.0082	11	24	36.6	53.0	1435	83.0	84.5	84.5	0.60	0.73	0.80	4.70
3	4	100L	2.04	7.8	2.9	3.3	0.0123	12	26	37.6	53.0	1430	83.0	85.5	86.0	0.64	0.76	0.83	6.07
4	5.5	112M	2.71	6.6	2.0	2.6	0.0156	13	29	43.9	56.0	1440	86.0	86.7	86.7	0.64	0.76	0.82	8.12
5.5	7.5	132S	3.67	7.3	1.9	3.0	0.0416	8	18	60.4	56.0	1460	87.5	88.0	88.1	0.68	0.80	0.86	10.5
7.5	10	132M	4.97	7.8	2.1	3.0	0.0528	7	15	70.5	56.0	1470	86.5	88.0	88.7	0.55	0.69	0.80	15.3
9.2	12.5	160M	6.10	7.1	2.6	2.8	0.0803	8	18	92.5	67.0	1470	87.5	89.0	89.5	0.63	0.76	0.82	18.1
11	15	160M	7.29	6.9	2.5	2.7	0.0779	8	18	119	67.0	1470	87.5	89.0	89.8	0.63	0.76	0.82	21.6
15	20	160L	9.94	7.4	2.7	3.0	0.1023	8	18	134	67.0	1470	89.5	90.6	90.6	0.64	0.76	0.82	29.1
18.5	25	180M	12.2	8.1	3.0	3.4	0.1573	9	20	169	64.0	1475	91.0	91.4	91.4	0.65	0.76	0.82	35.6
22	30	180L	14.6	8.0	2.7	3.3	0.2010	8	18	186	64.0	1470	91.0	91.6	91.6	0.68	0.79	0.84	41.3
30	40	200L	19.8	7.0	2.5	2.6	0.2941	10	22	246	69.0	1475	92.2	92.6	92.6	0.67	0.78	0.83	56.3
37	50	225S/M	24.4	7.2	2.2	2.7	0.6145	10	22	330	70.0	1475	92.6	93.0	93.0	0.76	0.84	0.87	66.0
45	60	225S/M	29.7	7.4	2.4	3.0	0.7169	10	22	385	70.0	1475	93.2	93.4	93.4	0.76	0.83	0.87	79.9
55	75	250S/M	36.2	7.2	2.5	3.0	0.8767	10	22	430	70.0	1480	93.5	93.7	93.7	0.74	0.83	0.87	97.4
75	100	280S/M	49.2	7.2	2.2	2.6	1.80	15	33	600	72.0	1485	94.0	94.2	94.2	0.78	0.86	0.87	132
90	125	280S/M	59.0	7.8	2.6	2.8	2.27	20	44	760	72.0	1485	94.0	94.5	94.5	0.79	0.85	0.88	156
110	150	315S/M	72.2	7.9	2.9	3.6	2.82	10	22	830	72.0	1485	94.4	94.5	94.5	0.77	0.85	0.87	193
132	175	315S/M	86.6	7.8	2.4	2.6	3.48	15	33	1050	72.0	1485	94.0	94.5	95.0	0.77	0.84	0.87	231
150	200	315S/M	98.4	7.5	2.4	2.7	3.77	20	44	1005	72.0	1485	94.1	95.1	95.1	0.78	0.84	0.87	262
160	220	315S/M	105	7.6	2.4	2.6	3.79	20	44	1005	72.0	1485	94.1	95.1	95.1	0.76	0.84	0.87	279
185	250	315S/M	121	7.3	2.4	2.9	3.77	19	42	1005	77.0	1485	94.2	95.0	95.1	0.72	0.81	0.85	328
200	270	355M/L	131	6.6	2.1	2.3	6.86	49	108	1525	79.0	1490	94.9	95.4	95.4	0.80	0.86	0.88	342
220	300	355M/L	144	7.0	2.1	2.4	6.86	38	84	1620	79.0	1490	94.4	95.4	95.4	0.79	0.86	0.88	375
250	340	355M/L	163	6.9	2.2	2.5	8.12	36	79	1615	79.0	1490	94.6	95.4	95.4	0.80	0.86	0.88	425
260	350	355M/L	170	6.5	2.2	2.3	8.12	32	70	1615	79.0	1490	94.6	95.4	95.5	0.80	0.86	0.88	445
280	380	355M/L	183	7.1	2.2	2.4	9.02	39	86	1770	79.0	1490	95.3	95.5	95.5	0.81	0.87	0.88	471
300	400	355M/L	196	6.7	2.2	2.4	9.92	47	103	1770	79.0	1490	95.1	95.6	95.6	0.81	0.87	0.89	504
315	430	355M/L	206	7.0	2.2	2.4	9.92	42	92	1770	79.0	1490	95.1	95.4	95.6	0.79	0.86	0.88	535
330	450	355M/L	216	6.5	2.3	2.3	10.8	32	70	1865	79.0	1490	94.7	95.4	95.7	0.81	0.87	0.89	554

For my student number X =2, the required power is **12kW**, and
the SMALLEST motor that can meet the required transmitted power requirement has a rated speed of **1470 RPM**.

Machine class

Using the examples of driven machine type from Table A1 of AS2784, **Class 2** is appropriate for a belt conveyor that is not uniformly loaded.

Service factor

The motor system runs for 20 hours per day. Referring to Appendix A1 with regards to design power,For soft start the service factor is 1.3, meaning my design power is
19.5 kW.

TABLE A1

SERVICE FACTORS FOR BELT DRIVES

Type of driven machine		Service factor (See Notes 1, 2 and 6)					
		'Soft' starts (See Note 4)			'Heavy' starts (See Note 5)		
		Hours of duty per day			Hours of duty per day		
Class	Examples	≤10	>10 ≤16	>16	≤10	>10 ≤16	>16
Class 1 (Light duty)	Agitators (uniform density) Blowers, exhausters and fans (up to 7.5 kW) Centrifugal compressors and pumps Belt conveyors (uniformly loaded)	1.0	1.1	1.2	1.1	1.2	1.3
Class 2 (Medium duty)	Agitators and mixers (variable density) Blowers, exhausters and fans (over 7.5 kW) Rotary compressors and pumps (other than centrifugal) Belt conveyors (not uniformly loaded) Generators and exciters Laundry machinery Lineshafts Machine tools Printing machinery Sawmill and woodworking machinery Screens (rotary)	1.2	1.2	1.3	1.2	1.3	1.4
Class 3 (Heavy duty)	Brick machinery Bucket elevators Compressors and pumps (reciprocating) Conveyors (heavy duty) Hoists Mills (hammer) Pulverizers Punches, presses, shears Quarry plant Rubber machinery Screens (vibrating) Textile machinery	1.2	1.3	1.4	1.4	1.5	1.6
Class 4 (Extra heavy duty)	Crushers (gyratory-jaw-roll) Mills (ball-rod-tube)	1.3	1.4	1.5	1.5	1.6	1.8

FIGURE A1 SELECTION OF WEDGE BELT CROSS-SECTION

Wedge belt selection

Using Figure A1 from AS2784, **SPA** is selected as an appropriate Wedge Belt cross-section size for my design power and shaft speed.

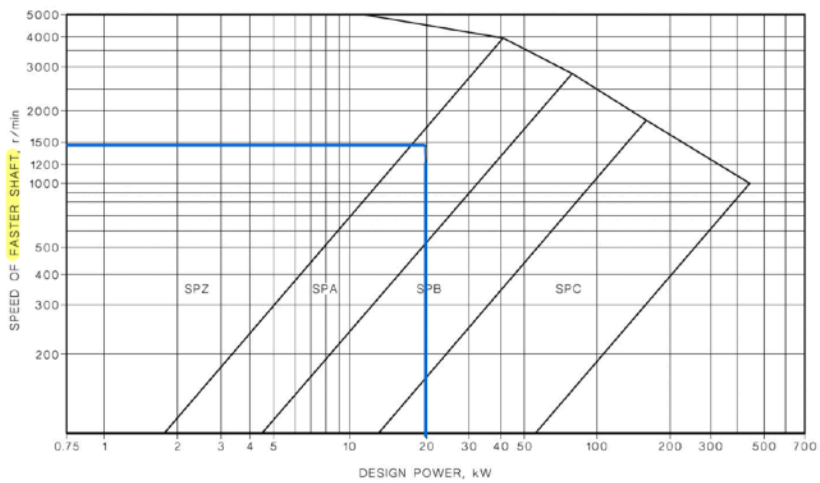


FIGURE A1 SELECTION OF WEDGE BELT CROSS-SECTION

Pulley Selection

The pulley selection should allow the main shaft to run as close as possible to 460 rpm when the motor is running at its rated speed.
Noting that the larger pulley OD must be less than 800 mm, select appropriate pitch diameters for the small and large pulleys from Table 9 of AS2784.
Determine all potential pulley combinations which provide an output rpm with ± 5% error.
I have selected a small pulley pitch diameter of **125 mm,** and a large pulley pitch diameter of **400 mm**, meaning the error in shaft speed (measured in rpm), is **0.625**RPM.

Belt length

AS2784 Table 4 describes commonly available belt lengths for different cross-section sizes, and sections A3 and A4 provide formulae for the calculation of belt length and centre distance, respectively.
The centre distance may not exceed 1000 mm and must be greater than the difference between the two pulley pitch diameters (i.e. (D-d) < C)

I have selected a small pulley pitch diameter of 125 mm and a large pulley pitch diameter of 400 mm. For this pulley combination, the smallest belt pitch length I can use is **1600 mm** and the longest belt pitch length I can use is **2800 mm**.

Belt power rating

Using AS2784 power rating tables (Table A2 for SPZ belts, Table A3 for SPA belts and Table A4 for SPB belts) define the Power Rating of a single belt for each of the potential pulley combinations found in question 5

For my belt system I have selected a small pulley pitch diameter of 125 mm, and the small pulley shaft speed is 1470 RPM, therefore the Power Rating per belt (in kW) from AS2784 is **4.315 kW**.

Power increment per belt

Using AS2784 power rating tables (Table A2 for SPZ belts, Table A3 for SPA belts andTable A4 for SPB belts) I have selected a small pulley pitch diameter of 125 mm, the small pulley shaft speed is 1470 RPM and the **speed ratio is 3.2,** therefore the Power Increment per belt is **0.57 kW**.

TABLE A3 *(continued)*

Speed of faster shaft r/min	Power rating, kW											Power increment per belt, kW									
	Pitch diameter of pulley, mm											Speed ratio									
	100	106	112	118	125	132	140	150	160	180	200	1.00 to 1.01	1.02 to 1.05	1.06 to 1.11	1.12 to 1.18	1.19 to 1.26	1.27 to 1.38	1.39 to 1.57	1.58 to 1.94	1.95 to 3.38	3.39 and over
1300	2.42	2.81	3.20	3.59	4.04	4.48	4.99	5.61	6.23	7.46	8.65	0.00	0.04	0.12	0.21	0.29	0.35	0.41	0.46	0.51	0.54
1400	2.55	2.97	3.39	3.80	4.28	4.76	5.30	5.96	6.62	7.92	9.19	0.00	0.05	0.13	0.23	0.31	0.38	0.44	0.50	0.54	0.58
1500	2.69	3.13	3.57	4.01	4.52	5.03	5.60	6.30	7.00	8.37	9.71	0.00	0.05	0.14	0.25	0.34	0.41	0.48	0.54	0.58	0.62
1600	2.81	3.29	3.75	4.22	4.75	5.29	5.89	6.64	7.37	8.81	10.21	0.00	0.06	0.15	0.26	0.36	0.43	0.51	0.57	0.62	0.66
1700	2.94	3.44	3.93	4.42	4.98	5.54	6.18	6.96	7.73	9.24	10.70	0.00	0.06	0.16	0.28	0.38	0.46	0.54	0.61	0.66	0.70

Power correction for belt pitch length

Using AS2784 Table A13 determine the Power Correction Factor for Belt Pitch Length for all possible belt sizes (i.e., between the smallest and largest length, inclusive).
The SP belt type for my system is SPA, the belt pitch length I have selected is **2000 mm**, and therefore the Power Correction for Belt Pitch Length is **0.98**.

Power correction for angle of wrap

I have selected a small pulley diameter of **125 mm**, and a large pulley diameter of 400 mm, my selected centre distance is **571.11 mm**, and therefore the Power Correction for Arc of Contact is **0.93**.

(D-d)/C= (400-125)/571.11=0.482
θ =180-2*arcsin((D-d)/2c)=152.14 degree

TABLE A12

POWER CORRECTION FACTORS FOR ARC OF CONTACT

$\frac{D-d}{C}$	Arc of contact on smaller pulley, degrees	Correction factor, i.e. proportion of 180-degree rating
0.00	180	1.00
0.05	177	0.99
0.10	174	0.99
0.15	171	0.98
0.20	169	0.97
0.25	166	0.97
0.30	163	0.96
0.35	160	0.95
0.40	157	0.94
0.45	154	0.93
0.50	151	0.93
0.55	148	0.92
0.60	145	0.91
0.65	142	0.90
0.70	139	0.89
0.75	136	0.88
0.80	133	0.87
0.85	130	0.86
0.90	127	0.85
0.95	123	0.83
1.00	120	0.82

Belt Design Summary

From all possible combinations of pulley and belt length, I have select particular combination that I think solves the design problem the best:

Pd(small)mm	Pd(large) to achieve 460 rpm)mm	Pd(standard)mm	speed ratio	error(%)
100	319.6	315	3.15	1.4492754
106	338.776	315	2.971698113	7.5362319
112	357.952	400	3.571428571	-10.521739
118	377.128	400	3.389830508	-5.7282609
125	399.5	400	3.2	-0.1358696
132	421.872	400	3.03030303	5.4565217
140	447.44	400	2.857142857	11.847826
150	479.4	500	3.333333333	-4.1304348
160	511.36	500	3.125	2.2608696

Belt pitch length (mm)	Power correction for belt pitch length	(D-d)/C (125mm/400mm)	Power correction for angle of wrap(125mm/400mm)	A	B	C	N	N
1600	0.94	0.760208571	0.88	193.9375	9453.125	361.74283	4.826	5
1800	0.96	0.588032294	0.91	243.9375	9453.125	467.66139	4.569	5
2000	0.98	0.481333744	0.93	293.9375	9453.125	571.32915	4.38	5
2240	1	0.396105855	0.94	353.9375	9453.125	694.25886	4.247	5
2500	1.02	0.332753904	0.955	418.9375	9453.125	826.43658	4.098	5
2800	1.04	0.281125351	0.965	493.9375	9453.125	978.21132	3.977	4

1. The size of the small pulley pitch diameter is **125 mm**.
2. The size of the large pulley pitch diameter is **400 mm**.
3. The pitch length of the belt is **2000 mm**.
4. The belt Power Rating per belt is **4.315 kW.**
5. The Power Increment per belt is **0.57 kW.**
6. Power Correction for Belt Pitch Length is **0.98**
7. The Power Correction for Arc of Contact is **0.93**
8. Therefore the total Power per Belt is **4.452 kW.**
9. The design power for my system is **19.5 kW.**
10. Therefore the total number of belts required is **5 belts.**

Pulley and Belt selection consideration:
the output rpm is set to be 459.4 rpm which has very small error 0.14% compare to desired output of 460 rpm. The belt length is set to 2000mm which should be easy to source. Compare to longer belt 2240mm or 2500 mm belts which also required 5 total number of belts, 2000 mm sits in a sweet spot where center distance is smaller (571mm) this allows the system be more compact.

Use stronger belt with higher power ratings (eg fiber reinforced belt) could reduce the amount of belt needed to a whole number. Otherwise customize the belt length to get different power correction factor and minimize the wasted power.

Pulley Catalogue Selection

5 Groove SPA

Pitch Diameter	OD	Pulley Type	Bush No.	Bore		F	G	K	L	M	H	Weight*	Designation
				Min	Max							kg	
100	105,5	2	1615	14	42	80	70	-	38	42	-	1,9	PHP 5SPA100TB
106	111,[illegible]	[illegible]	2012	14	50	80	76	48	32	-	-	2,1	PHP 5SPA106TB
112	117,5	6	2012	14	50	80	80	48	32	-	-	2,4	PHP 5SPA112TB
118	123,5	2	2012	14	50	80	86	-	32	48	-	2,7	PHP 5SPA118TB
125	130,5	3	2012	14	50	80	92	24	32	24	-	3,1	PHP 5SPA125TB
132	137,5	3	2517	16	60	80	98	17,5	45	17,5	-	3,2	PHP 5SPA132TB
140	145,5	3	2517	16	60	80	106	17,5	45	17,5	-	3,9	PHP 5SPA140TB
150	155,5	3	2517	16	60	80	116	17,5	45	17,5	-	4,7	PHP 5SPA150TB
160	165,5	3	2517	16	60	80	126	17,5	45	17,5	-	5,6	PHP 5SPA160TB
170	175,5	3	3020	25	75	80	146	17,5	45	17,5	-	6,2	PHP 5SPA170TB
180	185,5	3	3020	25	75	80	165	14,5	51	14,5	-	6,8	PHP 5SPA180TB
190	195,5	2	3020	25	75	80	189	-	51	29	-	7,4	PHP 5SPA190TB
200	205,5	3	3020	25	75	80	215	14,5	51	14,5	-	9,1	PHP 5SPA200TB
212	217,5	2	3020	25	75	80	245	-	51	29	-	10,7	PHP 5SPA212TB
224	229,5	2	3020	25	75	80	280	-	51	29	-	12,3	PHP 5SPA224TB
250	255,5	7	3020	25	75	80	215	14,5	51	14,5	150	11,7	PHP 5SPA250TB
280	285,5	8	3525	35	90	80	245	4,5	89	4,5	170	17,4	PHP 5SPA280TB
315	320,5	8	3525	35	90	80	280	4,5	89	4,5	170	20,0	PHP 5SPA315TB
355	360,5	4	3525	35	90	80	320	4,5	89	4,5	170	22,8	PHP 5SPA355TB
400	405,5	4	3525	35	90	80	365	4,5	89	4,5	170	24,8	PHP 5SPA400TB
450	455,5	4	3525	35	90	80	415	4,5	89	4,5	170	28,5	PHP 5SPA450TB
500	505,5	4	3525	35	90	80	465	4,5	89	4,5	170	31,7	PHP 5SPA500TB
560	565,5	4	3525	35	90	80	525	4,5	89	4,5	170	35,0	PHP 5SPA560TB
630	635,5	4	3525	35	90	80	595	4,5	89	4,5	170	42,0	PHP 5SPA630TB
800	805,5	4	4030	40	100	80	765	11	102	11	216	60,0	PHP 5SPA800TB
1000	1005,5	4	4545	55	110	80	965	17	114	17	225	70,0	PHP 5SPA1000TB

Belt tension

APPENDIX B

RECOMMENDED PRACTICE FOR TENSIONING BELT DRIVES DURING INSTALLATION AND CALCULATION OF THE RESULTANT FORCE IMPOSED ON THE SHAFT

(Informative)

B1 GENERAL

The high power ratings of wedge belts and V-belts necessitate the measurement of belt tensions with sufficient accuracy to prevent belt slip or overloaded bearings or to meet particularly arduous conditions. The following procedure is recommended for drives coming within the normal range for each belt section as defined in this Standard.

B2 ORIGINAL TENSIONING OF BELT

The length of the span should be measured in millimetres. At the centre of the span a force is to be applied with a spring scale in a direction perpendicular to the span, until the belt is deflected from the normal by an amount equal to—

(a) 0.02 mm for every millimetre of span length if the span length is 500 mm or less (see Figure B1, Condition 1); or

(b) 0.01 mm for every millimetre of span length if the span length exceeds 500 mm (see Figure B1, Condition 2).

For example, the deflection for a span of 1 m would be (1000×0.01) mm, i.e. 10 mm. The force required for this deflection should be noted and compared with the value of P given in Table B1.

In all cases, it is highly recommended that the pulley centres be fixed and that the larger pulley be then rotated at least four times before making the measurement. On a multiple belt drive, it is essential that a matched set of belts be used (see Clause 2.6) and the above procedure be carried out on each belt, the average value of these forces being compared with the specified values of P in Table B1.

The belt system I have designed satisfies Condition **2.** The largest value of "Required deflection force", P from table B1 of Appendix 1 AS2784 is **14 N**. The "Correction for centrifugal tension", K (calculated by the equation give in Appendix 1 of AS2784 is **106.678**.

K=MV^2=0.123*29.45^2=106.678

For the belt system I have designed:

The "Static hub load", Ws, is defined as the magnitude of the force that is exerted on the pulley by the belt system when the pulleys are NOT rotating. For the system I have designed, Ws is equal to **3397.07 Newtons**. The "Dynamic hub load", Wr, is defined as the magnitude of the force that is exerted on the pulley by the belt system when the pulleys ARE rotating. For the system I have designed, Wr is equal to **2361.66 Newtons**.

Ws=2*5*25*14*sin(152.14/2)=3397.07 N

Wr=2*5(25*14-106.68)sin(152.14/2)=2231.66 N

B4 DETERMINATION OF STATIC LOAD ON BEARING DUE TO DRIVE BELT

The total static bearing loading imposed by the belts on the shaft is the vector sum of the tensions in the belts and it can be calculated with sufficient accuracy from the following equation:

$$W_s = 2nT_{stat}\sin\left(\frac{\theta}{2}\right) \quad \text{. . . (B1)}$$

where

W_s = total static bearing loading, in newtons

n = number of belts

θ = arc of contact on smaller pulley, in degrees

T_{stat} = static belt tension, in newtons

for Condition 1,

$T_{stat} = 12.5 \times P$

for Condition 2,

$T_{stat} = 25 \times P$

where

P = the force applied at the centre of the span determined in Paragraph B2, in newtons.

B5 DETERMINATION OF DYNAMIC LOAD ON BEARING DUE TO DRIVE BELT

To determine the dynamic hub loading, a correction has to be made to the static tension to account for the effect of centrifugal force before the vectorial summation i.e.,

$$W_r = 2n\left(T_{stat} - K\right)\sin\left(\frac{\theta}{2}\right) \quad \text{. . . (B2)}$$

where

W_r = dynamic hub loading, in newtons

n = number of belts

θ = arc of contact on small pulley, in degrees

T_{stat} = static belt tension, in newtons

K = correction for centrifugal tension given by—

K = MV^2

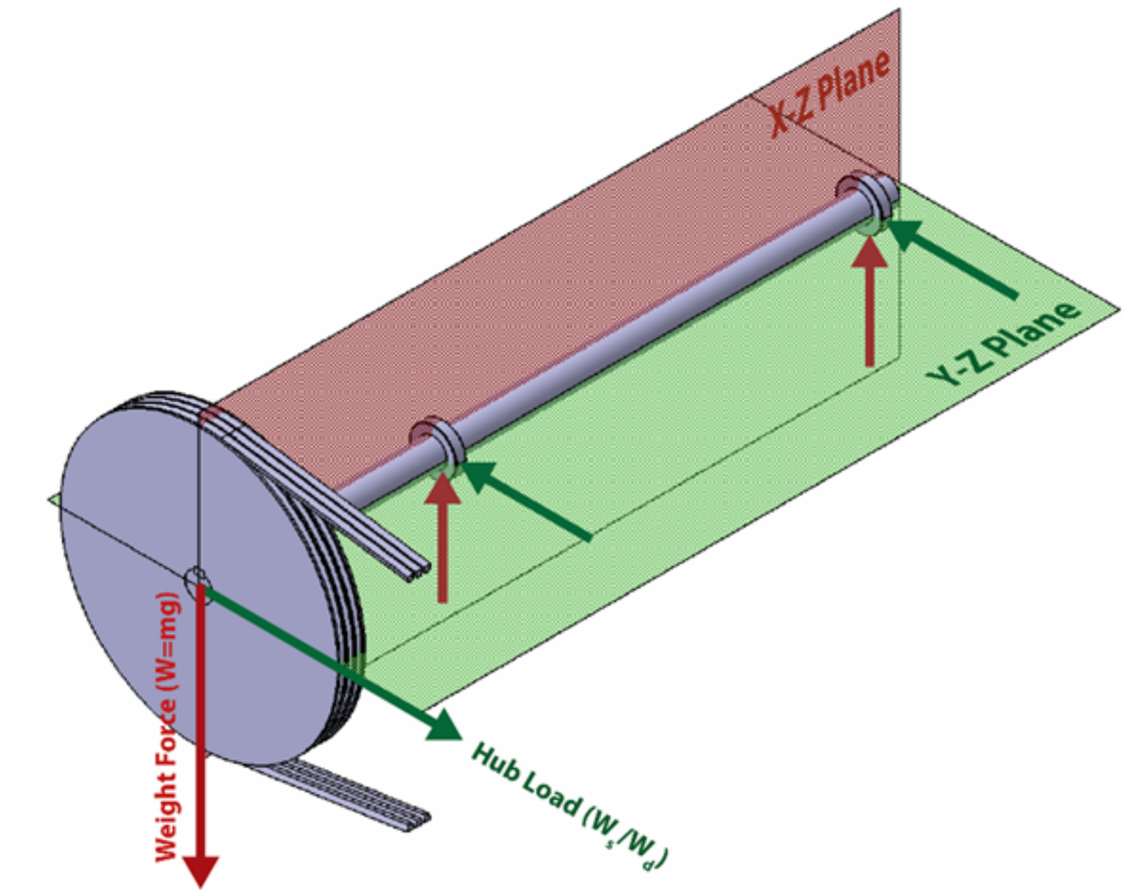
where

M = mass of belt per unit length, in kilograms per metre

V = linear belt speed, in metres per second, given by multiplying the pulley diameter (d_p) by the pulley revolutions per second

Bearing loading

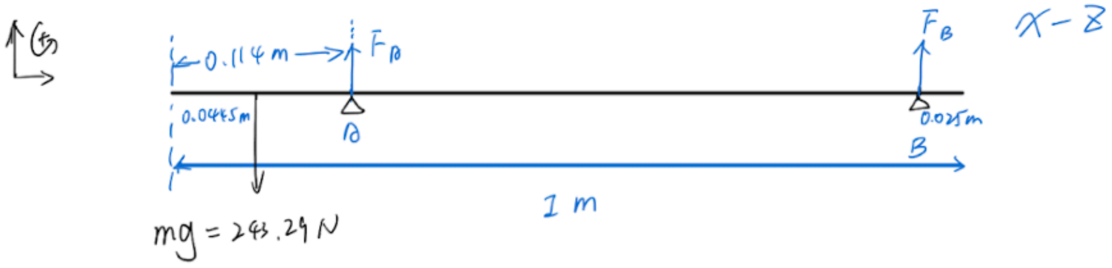
Given that the shaft length is 1 meter, calculate the reaction forces on the bearing near the pulley (bearing 1) and the bearing at the far end of the shaft (bearing 2).



Q 17. i) Static Case

400 mm PD, $L = 89$ mm. $89 + 25 = 114$ mm

$m = 24.8$ kg



$\Sigma F_x = 0 \quad \Sigma F_y = 0, \quad -mg + F_A + F_B = 0$

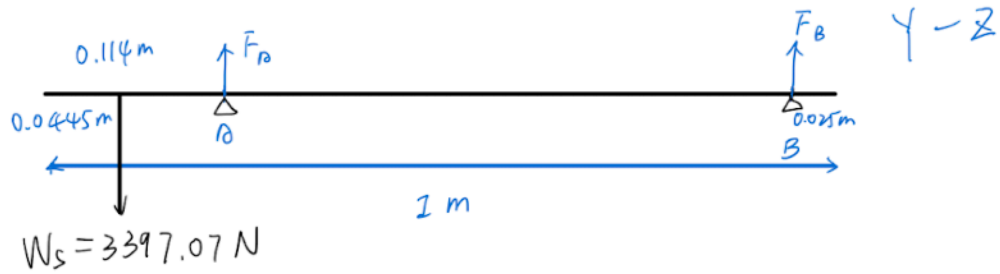
$\Sigma M_A = 0, \quad 243.29 \times 0.0695 + F_B \times (1 - 0.114 - 0.025) = 0$

$16.91 + 0.861 F_2 = 0$

$F_B = -19.64\,N$

$-243.29 + F_A - 19.64 = 0$

$F_A = 262.93\ N$



$\Sigma F_x = 0 \quad \Sigma F_y = 0, \quad -mg + F_A + F_B = 0$

$\Sigma M_A = 0, \quad 3397.07 \times 0.0695 + F_B \times (1 - 0.114 - 0.025) = 0$

$F_B = -274.21\ N$

$-3397.07 + F_A - 274.21 = 0$

$F_A = 3671.28\ N$

$\therefore$ For static case at bearing A, reaction force

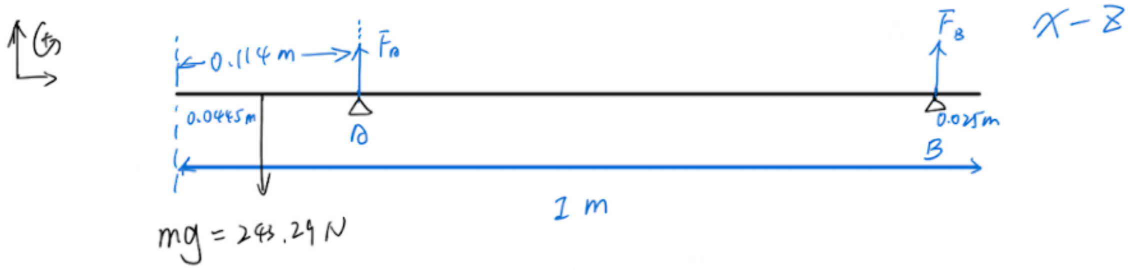
$\Sigma F_A = \sqrt{262.93^2 + 3671.28^2} = 3680.68\ N$

at bearing B, $\Sigma F_B = \sqrt{(-19.64)^2 + (-274.21)^2} = 274.91\ N$

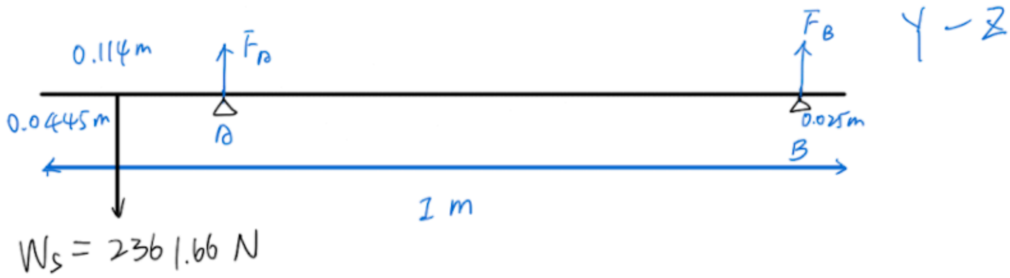
ii) Running Case

400 mm PD , $L = 89$ mm . $89 + 25 = 114$ mm

$m = 24.8$ kg



$F_B = -19.64$ N , $F_A = 262.93$ N .



$\Sigma F_x = 0$ $\quad \Sigma F_y = 0$, $\quad -mg + F_A + F_B = 0$

$\Sigma M_A = 0$, $2361.66 \times 0.0695 + F_B \times (1 - 0.114 - 0.025) = 0$

$F_B = -190.63$ N

$-2361.66 + F_A - 190.63 = 0$

$F_A = 2552.29$ N

$\therefore$ For running case at bearing A , reaction force

$\Sigma F_A = \sqrt{262.93^2 + 2552.29^2} = 2565.80$ N

at bearing B. $\Sigma F_B = \sqrt{(-19.64)^2 + (-190.63)^2} = 191.64$ N

Bearing position

My large pulley has a width of **400 mm**. Bearing 1 is mounted **114 mm** from the start of my shaft. Bearing 2 is mounted **975mm** from the start of my shaft.

Bearing load rating

We will simplify our design by using the same bearing for both bearings at either end of the main shaft. This approach is quite common as it is much easier to implement and costs less to buy two bearings of the same type, rather than two specific bearings. It also helps the engineer to sleep well at night as they don't have to worry about whether the bearings are assembled the right way around.

But using the same bearing type for both ends of the main shaft means that we must design for the bearing that sees the greatest load (we will call this the worst case loaded bearing).

At this stage of the design process, we can calculate the load rating required for our bearings, however we cannot select a bearing until we know the diameter of our shaft. As bearings come in standard bore sizes, we will manufacture our shaft to fit our bearings. Once we know the minimum diameter our shaft requires to resist loading, we will be able to select bearings with an appropriate bore size.

State the Static Load **3.681kN** and Dynamic **2.5658 kN** that you have calculated for the worst case loaded bearing. If we require an L10 design life or 7,000,000 cycles using a rolling element bearing (k = 3.0), state the Static Load Rating 3.681 kN and Dynamic Load Rating **4.908 kN** required for the worst case loaded bearing.

$$C^k = \frac{L_{10}}{E^6} * p^k$$

$$Cdynamic = \sqrt[3]{7} * 2.5658 = 4.908kN$$

Shaft design

The main shaft runs in one direction only (power applied) and is started once per day and used for a total of 20 hours per day. **Equation 2** from Table 2 of AS1403 (Design of Rotating Steel Shafts) is appropriate for this application.

TABLE 2

FORMULAS FOR CALCULATING MINIMUM DIAMETER OF SHAFT D

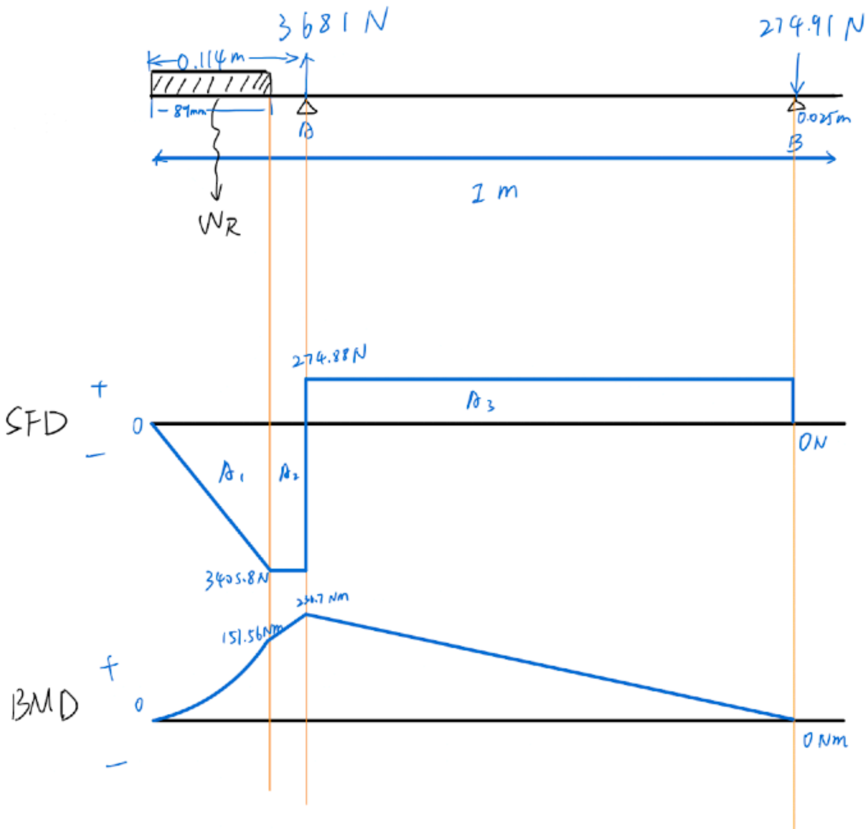
Number of mechanism starts per year	Number of revolutions of shaft per year	Torque application conditions	Formula	Formulas	
≤600	≤900	Manually or power applied	$D^3 = \frac{10^4 F_S}{F_Y}\sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4}T_q^2}$	1	
	>900	Power applied	$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4}T_q^2}$	2	See Notes 2 and 3
>600	>900	Power applied torque reversals	$D^3 = \frac{10^4 F_S}{F_R} K_S K\sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4}T_q^2}$	3	
		Power applied, no torque reversals (see Note 1)	$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{16}\left[(1 + K_S K)T_q\right]^2}$	4	

Bending moment diagram

Free body diagram of the forces acting on the main shaft on the resultant plane and bending moment diagram for this shaft see below

21. Static case. $W_R = \sqrt{W_s^2 + mg^2} = \sqrt{3391.07^2 + 243.29^2} = 3405.8\,N / 0.089\,m$
$= 38267.1\,N/m$

@ resultant plane



$A_1 = 0.089 \times 3405.8 \times \frac{1}{2} = 151.56$

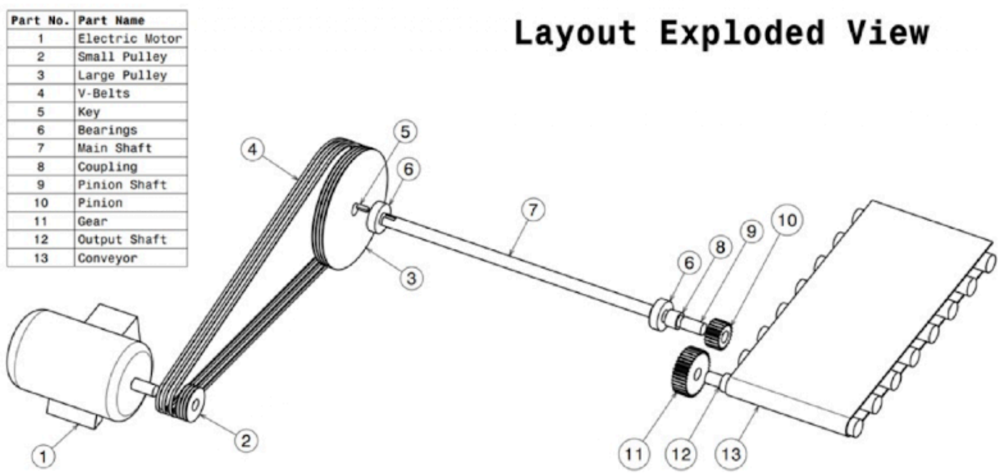
$A_2 = 0.025 \times 3405.8 = 85.145$

$A_3 = 274.88 \times (1 - 0.114 - 0.025) = 236.7$

$M_q = 236.7\,Nm$

Peak shaft torque

To calculate the shaft diameter, we require the maximum shaft torque, Tq. Assume that power is transmitted with 100% efficiency.



① Motor : speed 1470 rpm

output power 15 kW

servic factor 1.3 : (Table A1, AS_2784) Class 2.

> 16h/day soft start/no reversing

design power 19.5 kW

② small pulley 125 mm dp

③ Large pulley 400 mm dp

Assuming 100% power transmitting effiency. For shaft ⑦ Max torque

$$T_q = \frac{P_{design}}{\omega_{Large}}, \quad \text{speed } R_{ratio} = \frac{dp_{Large}}{dp_{small}} = \frac{400}{125} = 3.2$$

$$= \frac{P_{design}}{2\pi n} \quad \omega_{Large} = \frac{\omega_{small}}{3.2} = \frac{1470}{3.2} = 459.375 \text{ rpm}$$

$$n_{Large} = 7.656 \text{ rps}$$

$$\therefore T_q = \frac{P_{motor} \times \text{Service Factor} \times \text{speed ratio} \times 60}{2\pi\, \omega_{small}} = \frac{P_{design}}{2\pi\, n_{Large}}$$

$$= 405.36 \text{ Nm}$$

Trial shaft diameter:

From Appendix A from AS1403 (Design of Rotating Steel Shafts)the maximum bending moment, Mq is **236.7 Nm** and the maximum shaft torque, Tq is **405.36 Nm**. Therefore from Appendix A of AS1403, the estimated shaft diameter using low strength steel is **32 mm**.

APPENDIX A

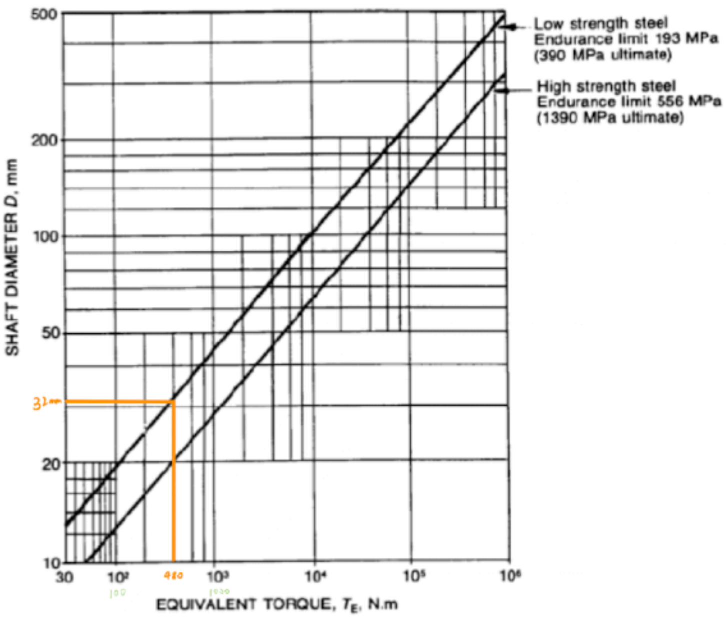
'TRIAL' SHAFT DIAMETER

(Informative)

A 'trial' diameter may need to be assumed in Formulas 1 to 4 given in Table 2.

The 'trial' shaft diameter is to be read directly from Figure A1, which plots shaft diameter against equivalent torque (T_E), where

$$T_E = 1.15\sqrt{M_q^2 + 0.75T_q^2}$$



NOTE: The curve is based on the following formula:

$$D^3 = \frac{12000\,K_s}{F_R}T_E$$

FIGURE A1 'TRIAL' SHAFT DIAMETER

StressRaising Factors

The shaft we are designing will have stress raising factors associated with the components we are mounting to it. We will be fitting a rolling element bearing to the shaft, and we will also design the shaft to have the pulley mounted using a keyway.

Determine the following stress raising factors for your trial diameter using low strength steel:

- Size Factor
- Fitted Rolling Element Bearing with K8/k6 Transition fit.
- Keyed Component with a side milled keyway with a H7/k6 transition fit.

The size factor for my trial diameter is **1.27**. The stress raising factor for a fitted rolling element bearing is **1.45**. The stress raising factor for a keyway is **1.4**.

Resolving Stress Raising Factors

According to section 8.2 of AS1403, the stress raising factors (K) must be resolved in to one value. The keyway will be side-milled into the end of the shaft, and run the width of the pulley. Based off the location of the bearing and the keyway on the shaft determine the resolved stress raising factor.

The width of my pulley is **89mm**, and the bearing is located **25 mm** from the pulley end of my shaft. Therefore, the two stress raising factors are separated by an axial distance of **25 D**. Therefore clause **b** applies, and the resolved stress raising factor is **1.45**.

Q24 /25/26

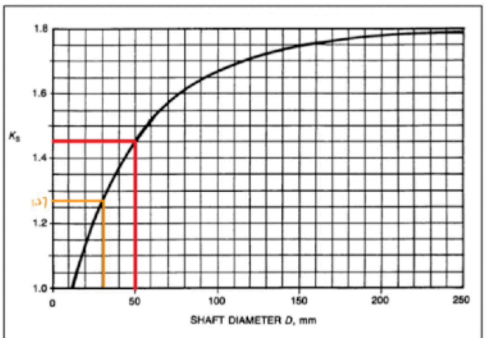


FIGURE 1 SIZE FACTOR K_s

$ks = 1.27$

$0.16D = 0.16 \times 32 = 5.12$ mm $\quad 0.25D = 8$mm

bearing & key apart at least 25 mm $> 0.25D$

8.2 Stress-raising factor (*K*)

The value of the stress-raising factor (*K*) shall be as follows:

(a) Where there is only one stress-raising characteristic .... the value read from Figures 4 to 10, as appropriate.

(b) Where two stress-raising characteristics are separated by an axial distance greater than 0.25*D* .... the greater of the two values read from Figures 4 to 10, as appropriate (see Figure 2).

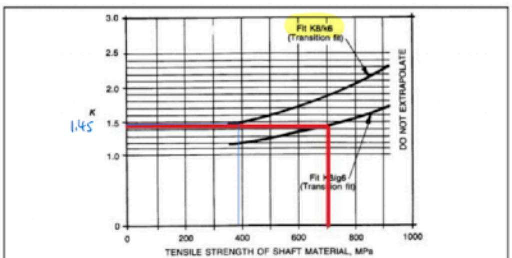
(c) Where two stress-raising characteristics are separated by an axial distance between 0.16*D* and 0.25*D* .... the sum of the greater value and 0.1 times the lesser value (see Figure 2), both values being those read from Figures 4 to 10, as appropriate.

(d) Where two stress-raising characteristics are coincident or separated by an axial distance not greater than 0.16*D* .... the sum of the greater value and 0.2 times the lesser value (see Figure 2) both values being those read from Figures 4 to 10, as appropriate.

NOTE: When the shaft diameter is not known, a 'trial' diameter (see Appendices A and B) should be calculated and the factor *K* selected accordingly.

Stress raising factor due to the bearing:

The bearing fit on shaft is K8/g6, so from Figure 5:



NOTES:

1 Values may be interpolated for fits between K8/k6 and K8/g6 which are recommended by the bearing manufacturers.

2 For tolerances, see AS 1654.

FIGURE 5 STRESS-RAISING FACTOR *K* FOR SHAFT FITTED WITH ROLLING ELEMENT BEARING

Low strength steel

390 MPa UTS

$K_{bearing} = 1.45$

K value due to keyway:

The keyway is side-milled with a component fit H7/s6:

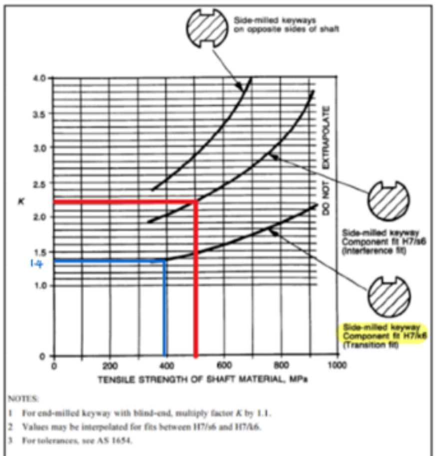


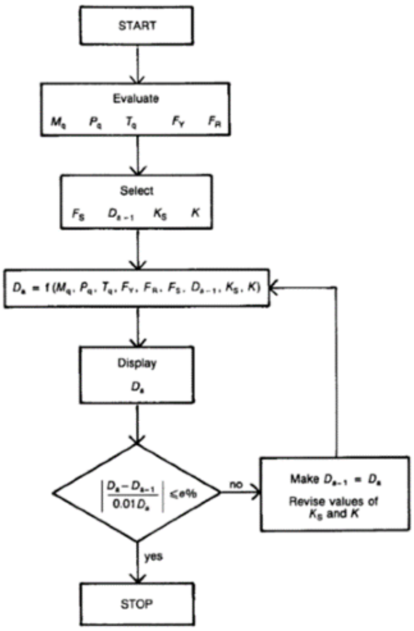
FIGURE 7 STRESS-RAISING FACTOR K FOR KEYED COMPONENT

$K_{keyway} = 1.4$

Minimum shaft diameter

Using the equation found in question 19, and the loads applied to the shaft, as well as an initial trial diameter, we can determine the minimum possible diameter for the shaft use following process:

Q26. $$D^3 = \frac{10^4 F_S}{F_R}\sqrt{\left[K_S K\left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4}T_q^2}$$



The value of Fs is 2.0 for Formula 1 and 1.2 for Formulas 2, 3 and 4. Where severe injury, death or extensive equipment damage is likely to occur because of the failure of the shaft, higher factors of safety may be used.

$F_S = 1.2$, $F_R = 193\,MPa$, $P_q = 0$, $M_q = 236.7\,Nm$. $T_q = 405.36\,Nm$

$$D^3 = \frac{10^4 \times 1.2}{193}\sqrt{\left[1.27 \times 1.4\,(236.7)\right]^2 + \frac{3}{4} \times (405.36)^2}$$

$$D = 32.42\ mm$$

$$\left|\frac{32.42 - 32}{0.01 \times 32.42}\right| = 1.295 > 1$$

Trial	Trial diameter (mm)	Ks	shaft diameter (mm)
1	33	1.28	32.47
2	34	1.3	32.5

$$\left|\frac{32.47 - 33}{0.01 \times 33}\right| = 1.61 > 1 \qquad 1.61 > 1.295$$

∴ Trial 0 is closet to calculated diameter (32.42 mm)

As a result the minimum possible shaft diameter is **32.42mm**

Bearing Catalogue Selection

Using the Timken deep groove ball bearing catalogue provided in the reference material, select a bearing which can service the worst case static and dynamic load rating calculated within question 17.

Ensure that the internal diameter of the bearing is greater than the minimum required shaft diameter calculated in question 22 and specify the final shaft diameter to the internal diameter of the bearing.

Bearing No.							Boundary Dimensions						Load Ratings		Thermal Reference Speed		
Description	Features						Bore	O.D.	Width	Radius			Dynamic	Static	Grease	Oil	Weight
							d	D	B	$R_{s\,min}$	$D2_{max}$	f_{max}	C_r	C_{0r}			
	Z	ZZ	RS	2RS	2RZ	NR	mm	mm	mm	mm	mm	mm	kN	kN	RPM	RPM	kg
6000	•	•	•	•	•	•	10	26	8	0.3	29.2	0.70	4.60	2.00	26000	38000	0.020
6200	•	•	•	•		•	10	30	9	0.6	34.7	1.12	5.10	2.40	22000	32000	0.030
6300	•	•	•	•		•	10	35	11	0.6	39.7	1.12	8.10	3.50	20000	29000	0.050
6001	•	•	•	•		•	12	28	8	0.3	30.8	0.85	5.10	2.40	23000	33000	0.020
6201	•	•	•	•	•	•	12	32	10	0.6	36.7	1.12	6.80	3.00	21000	30000	0.040
6301	•	•	•	•		•	12	37	12	1.0	41.3	1.12	9.70	4.20	19000	27000	0.060
6002	•	•	•	•		•	15	32	9	0.3	36.7	1.12	5.60	2.80	20000	30000	0.030
6202	•	•	•	•	•	•	15	35	11	0.6	39.7	1.12	7.60	3.70	19000	28000	0.050
6302	•	•	•	•	•	•	15	42	13	1.0	46.3	1.12	11.40	5.40	16000	24000	0.080
6003	•	•	•	•		•	17	35	10	0.3	39.7	1.12	6.00	3.30	19000	28000	0.040
6203	•	•	•	•	•	•	17	40	12	0.6	44.6	1.12	9.60	4.80	17000	25000	0.070
6303	•	•	•	•	•	•	17	47	14	1.0	52.7	1.12	13.60	6.60	15000	22000	0.120
6004	•	•	•	•	•	•	20	42	12	0.6	46.3	1.12	9.40	5.00	17000	25000	0.070
6204	•	•	•	•	•	•	20	47	14	1.0	52.7	1.12	12.80	6.60	15000	22000	0.100
6304	•	•	•	•	•	•	20	52	15	1.1	57.9	1.12	15.90	7.80	13000	20000	0.140
6005	•	•	•	•	•	•	25	47	12	0.6	52.7	1.12	10.10	5.80	14000	21000	0.080
6205	•	•	•	•	•	•	25	52	15	1.0	57.9	1.12	14.00	7.90	14000	20000	0.130
6305	•	•	•	•		•	25	62	17	1.1	67.7	1.70	20.60	11.20	12000	17000	0.220
6405						•	25	80	21	1.5	86.6	1.70	36.10	18.80	10000	15000	0.530
6006	•	•	•	•	•	•	30	55	13	1.0	60.7	1.12	13.20	8.30	12000	18000	0.110
6206	•	•	•	•	•	•	30	62	16	1.0	67.7	1.70	19.50	11.30	11000	16000	0.200
6306	•	•	•	•	•	•	30	72	19	1.1	78.6	1.70	26.60	15.00	10000	15000	0.350
6406						•	30	90	23	1.5	96.5	2.46	47.30	24.50	9300	13000	0.740
6007	•	•	•	•	•	•	35	62	14	1.0	67.7	1.70	15.90	10.30	11000	16000	0.150
6207	•	•	•	•	•	•	35	72	17	1.1	78.6	1.70	25.70	15.30	10000	14000	0.290
6307	•	•	•	•	•	•	35	80	21	1.5	86.6	1.70	33.40	19.20	9300	13000	0.450
6307MB							35	80	21	1.5	-	-	33.40	19.20	9300	13000	0.550
6407							35	100	25	1.5	-	-	55.50	29.40	8500	12000	0.950

Bearing Catalogue Selection

For my design scenario, the selected bearing has a dynamic load rating of **15.9 kN** and a static load rating of **10.30 kN**. The final shaft diameter has been specified to **35mm**, the internal diameter of the bearings selected.

Keyway Cross-Section

Now that we know what the diameter of our shaft will be, we can design the keyway. Using material from SAA HB6 determine the nominal dimensions b and h for a normal fitting metric keyway for your diameter shaft.

The shaft I have designed is **35 mm** in diameter. Therefore, the nominal dimension b for the keyway is **10 mm**, and h is **8 mm**.

Keyway Tolerancing

The tolerance required for the width of the keyway in the shaft is **+0.000 mm** and – **0.036mm.** For the depth the tolerance is + **0.2mm** and – **0.0 mm.**

All dimensions in millimetres

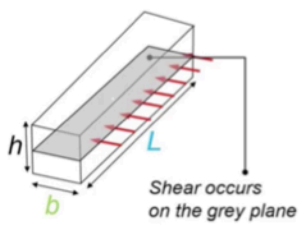
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Shaft		Key (see Note)	Keyway											
nominal diameter (see Note) d		section $b \times h$ width × thickness	width b						depth				radius r	
				tolerance for class of fit					shaft t_1		hub t_2			
				free		normal		close and interference						
over	incl.		nom.	shaft (H9)	hub (D10)	shaft (N9)	hub (J_s9)*	shaft and hub (P9)	nom.	tol.	nom.	tol.	max.	min.
22	30	8 × 7	8	+ 0.036	+ 0.098	0	+ 0.018	– 0.015	4		3.3		0.25	0.16
30	38	10 × 8	10	0	+ 0.040	– 0.036	– 0.018	– 0.051	5		3.3		0.40	0.25
38	44	12 × 8	12						5		3.3		0.40	0.25
44	50	14 × 9	14	+ 0.043	+ 0.120	0	+ 0.021	– 0.018	5.5		3.8		0.40	0.25
50	58	16 × 10	16	0	+ 0.050	– 0.043	– 0.021	– 0.061	6	+ 0.2	4.3	+ 0.2	0.40	0.25
58	65	18 × 11	18						7	0	4.4	0	0.40	0.25
65	75	20 × 12	20						7.5		4.9		0.60	0.40
75	85	22 × 14	22	+ 0.052	+0.149	0	+ 0.026	– 0.022	9		5.4		0.60	0.40
85	95	25 × 14	25	0	+ 0.065	– 0.052	– 0.026	– 0.074	9		5.4		0.60	0.40
95	110	28 × 16	28						10		6.4		0.60	0.40
110	130	32 × 18	32						11		7.4		0.60	0.40
130	150	36 × 20	36	+ 0.062	+ 0.180	0	+ 0.031	– 0.026	12		8.4		1.00	0.70
150	170	40 × 22	40	0	+ 0.080	– 0.062	– 0.031	– 0.088	13		9.4		1.00	0.70
170	200	45 × 25	45						15		10.4		1.00	0.70
200	230	50 × 28	50						17		11.4		1.00	0.70
230	260	56 × 32	56						20	+ 0.3	12.4	+ 0.3	1.60	1.20
260	290	63 × 32	63	+ 0.074	+ 0.220	0	+ 0.037	– 0.032	20	0	12.4	0	1.60	1.20
290	330	70 × 36	70	0	+ 0.100	– 0.074	– 0.037	– 0.106	22		14.4		1.60	1.20
330	380	80 × 40	80						25		15.4		2.50	2.00
380	440	90 × 45	90	+ 0.087	+ 0.250	0	+ 0.043	– 0.037	28		17.4		2.50	2.00
440	500	100 × 50	100	0	+ 0.120	– 0.087	– 0.043	– 0.124	31		19.5		2.50	2.00

Key Length

Given the diameter of the shaft, and the torque being transmitted, determine the length the key must be if the key is made from A151 1020 cold drawn steel (Syt = 352MPa) and has a factor of safety of 2 with regards to shear, and 1.5 with regards to crushing.

$\Phi = 405.36$ Nm , For shear,

$$P_{key} = \frac{\Phi}{r} = \frac{2\Phi}{D} = \frac{2 \times 405.36}{0.035} = 23163.43 \text{ N}$$



$$F_{OS} = \frac{S_{material}}{S_{design}} = \frac{\frac{S_{yt}}{2}}{\tau} = 2$$

$$\tau_{max} = \frac{\frac{S_{yt}}{2}}{F_{OS}} = \frac{352 \times \frac{1}{2}}{2} = 88 \text{ MPa}$$

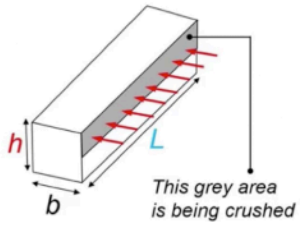
$$\tau_{max} = \frac{P_{key}}{bL}, \quad b = 0.01 \text{ m}$$

$$L = \frac{P_{key}}{b\,\tau_{max}} = \frac{23163.43}{0.01 \times 88 \times 10^6} = 0.02632 \text{ m} = 26.32 \text{ mm}$$

For crush

$S_{key} = S_{yt} = 352\,MPa$, $\quad S_{shaft} = 390 \times 0.65 = 253.5\,MPa$

$S_{shaft} < S_{key}$, $\quad \tau_{max} = \frac{S_{shaft}}{FoS} = \frac{253.5}{1.5} = 169\,MPa$



$$\tau_{max} = \frac{P_{key}}{\frac{h}{2}L}, \quad L = \frac{2P_{key}}{\tau_{max}\,h} = \frac{2 \times 23163.43}{169 \times 10^6 \times 8 \times 10^{-3}}$$

$$= 0.03427\ m$$
$$= 34.27\ mm$$

$$\therefore L \geq 34.27\ mm$$

The keyway I have designed must have a length of **34.27 mm** to resist both shearing and crushing.

Gear design torque increase

Sometime after the pulley and belt system has been commissioned the end-user decides to re-purpose the conveyor for another application that requires the output torque to be increased by 60%.

Spur gear minimum module

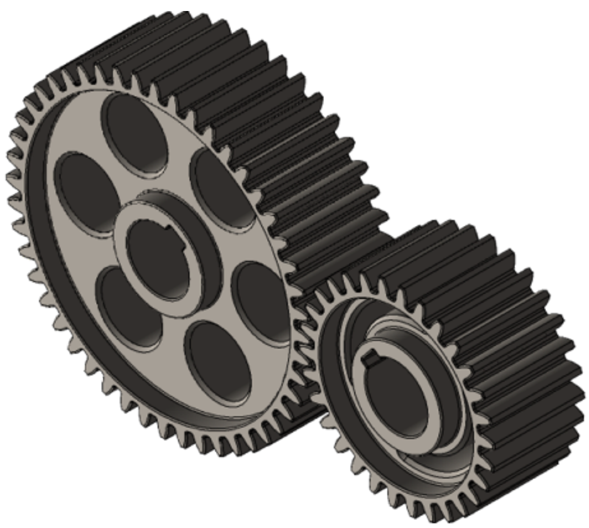
A single-stage spur gear reduction could be used to achieve this torque increase.
I am advised to begin my design with 21 teeth on the pinion, but with a pinion pitch diameter of no less than 2.5 times your shaft diameter.

Pd.pinion no less than 2.5*35=87.5mm
m.pinion should be greater than 87.5/21=4.1667mm

TABLE 1
NORMAL MODULES
millimetres

1st choice	2nd choice*
1	1.125
1.25	1.375
1.5	1.75
2	2.25
2.5	2.75
3	3.5
4	4.5
5	5.5
6	(6.5)
	7
8	9
10	11
12	14
16	18
20	22
25	28
32	36
40	45
50	

* The value in parenthesis is not recommended.



Given that my shaft diameter is 35 mm, according to AS2938, the smallest 1st choice normal module that can accommodate 21 teeth on the pinion with a pinion pitch diameter of no less than 2.5 times the shaft diameter is **5 mm**.

Gear specification

A torque increase of 60 % is required and I have selected a module of 5mm for my design. For 21 teeth on the pinion, the number of teeth on the mating gear that most closely achieves this required torque increase, while maintaining a hunting tooth ratio, is **34 teeth.**
For this number of teeth on the mating gear, and the module I have selected, the gear pitch dimeter will be **170 mm** and according to the standard tooth profile of AS2938, the addendum diameter of the mating gear will be **180 mm** and the dedendum diameter of the mating gear will be **157.5 mm.**

Torque increase 60%

$$\frac{\Phi\ \text{Output}}{\Phi\ \text{input}} = \frac{N\ \text{gear}}{N\ \text{pinion}} = 1+0.6$$

$N_{gear} = 1.6 \times 21 = 33.6$, Take 34 for hunting tooth ratio

$Pd.gear = 5 \times 34 = 170\ mm$

$\text{Addendum} = 1 \times m = 5mm$

$\text{Dedendum} = 1.25 \times m = 6.25\ mm$

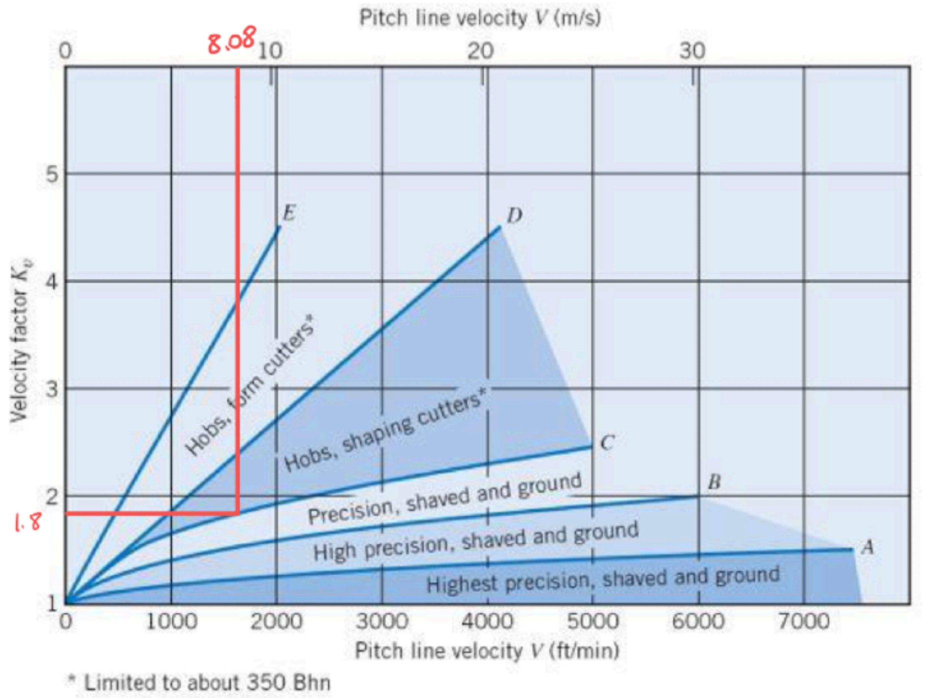
Addendum diameter (outside Diameter) $d.add = 170 + 2 \times 5 = 180\ mm$

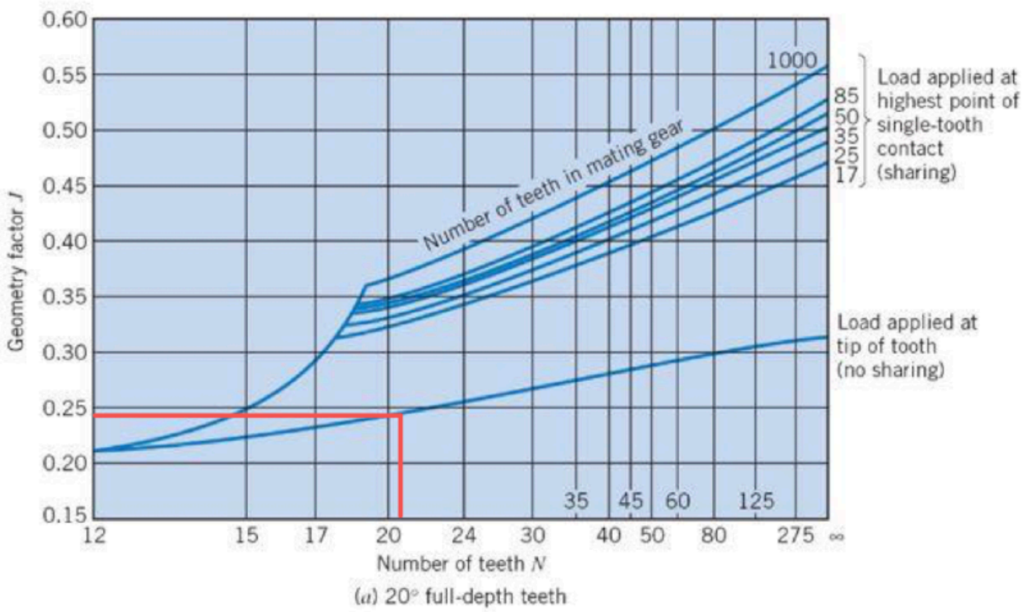
Dedendum diameter (Root Diameter) $d.ded = 170 - 2 \times 6.25 = 157.5\ mm$

Gear design

Assumptions:

- Geometry factor J is calculated from the following chart assuming a 20 degree pressure angle and "load applied at tip of tooth, (no sharing)".
- The velocity factor (Kv) is calculated from the following chart assuming that our design is based on line "C" for "precision shaved and ground" teeth.





Vt=2*pi*(1470/60)*(0.105/2)=8.082 m/s

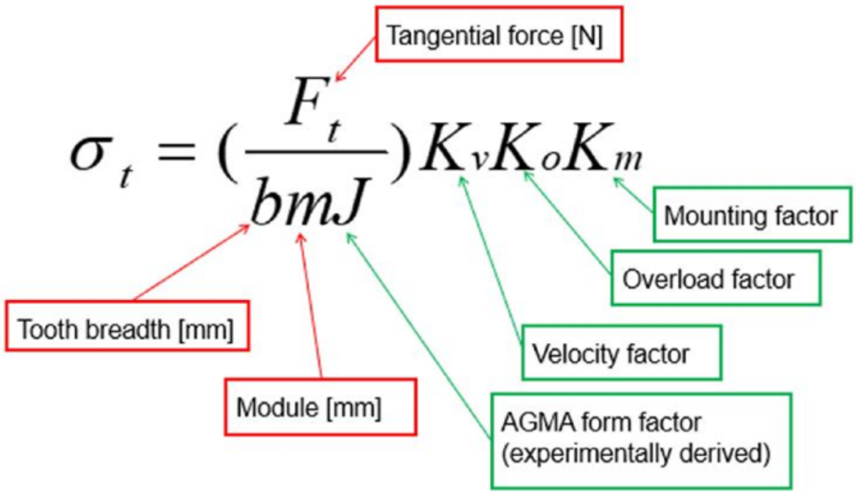
Based on the number of teeth on my pinion 21, the Geometry Factor (found from the table below, based on the above assumptions) is **0.24**. For a pinion shaft speed of 1470 RPM, and pinion pitch circle diameter 105 mm, the pitch line velocity is **8.082** meters per second, and the Velocity factor, Kv, (found from the table below based on the above assumptions) is **1.8**.

Gear face width

Summaries of the gear selection
The Geometry factor, J, found in the previous question is 0.24.
The Velocity factor, Kv, found in the previous question is 1.8.
The module for my pinion is 5 mm.

For the following assumptions:

• The mounting factor (Km) is assumed to be 1.4 (this a common assumption for initial design purposes)
• The overload factor (Ko) is assumed to be 1.5 (again this is a common assumption)
• The gears are made of 4340 normalised steel with bending strength (St) of 474 MPa



Motor power 15 kW

Design power 19.5 kW

$$F_t = \frac{W}{V} = \frac{19.5 \times 10^3}{8.082} = 2412.8\,N$$

$$\sigma_t = S_t = 474\ MPa$$

$$b = \frac{F_t \cdot k_v k_o k_m}{\sigma_t\, m J} = \frac{2412.8 \times 1.8 \times 1.5 \times 1.4}{474 \times 10^6 \times 0.005 \times 0.24} = 0.01603\ m$$

$$= 16.03\ mm$$

Using the AGMA method and the assumptions above, the minimum required face width, b, for the pinion is: **16.03 mm.**

Module Selection

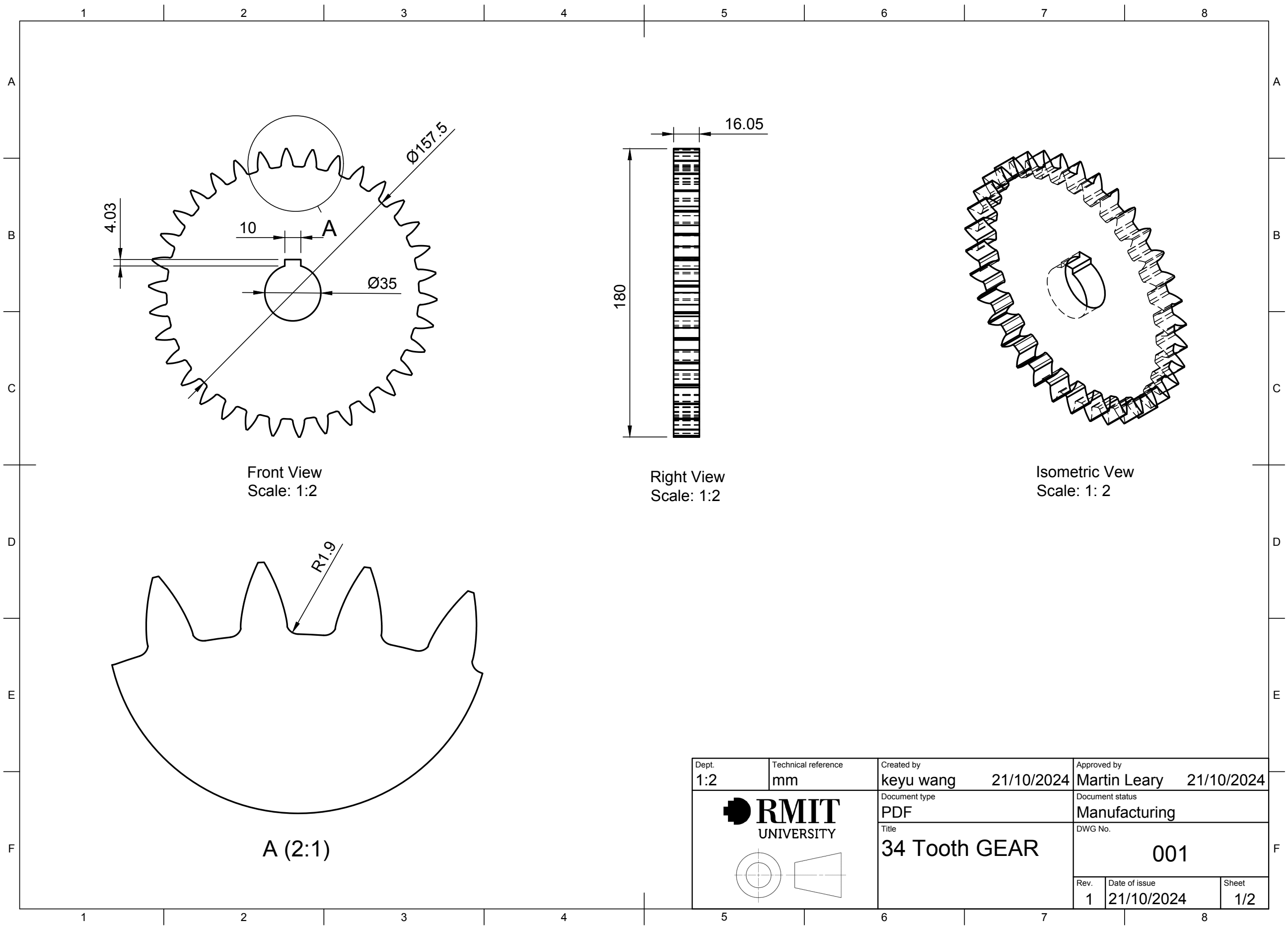
The module should allow pinion pitch diameter greater than 2.5 times my shaft diameter(35mm). The 1st choice common module (5mm) is chosen for easier sourcing or manufacturing of parts. The torque increase should be just greater than 60% and the tooth breadth should kept small to reduce the gear weight thus reduce the cost.

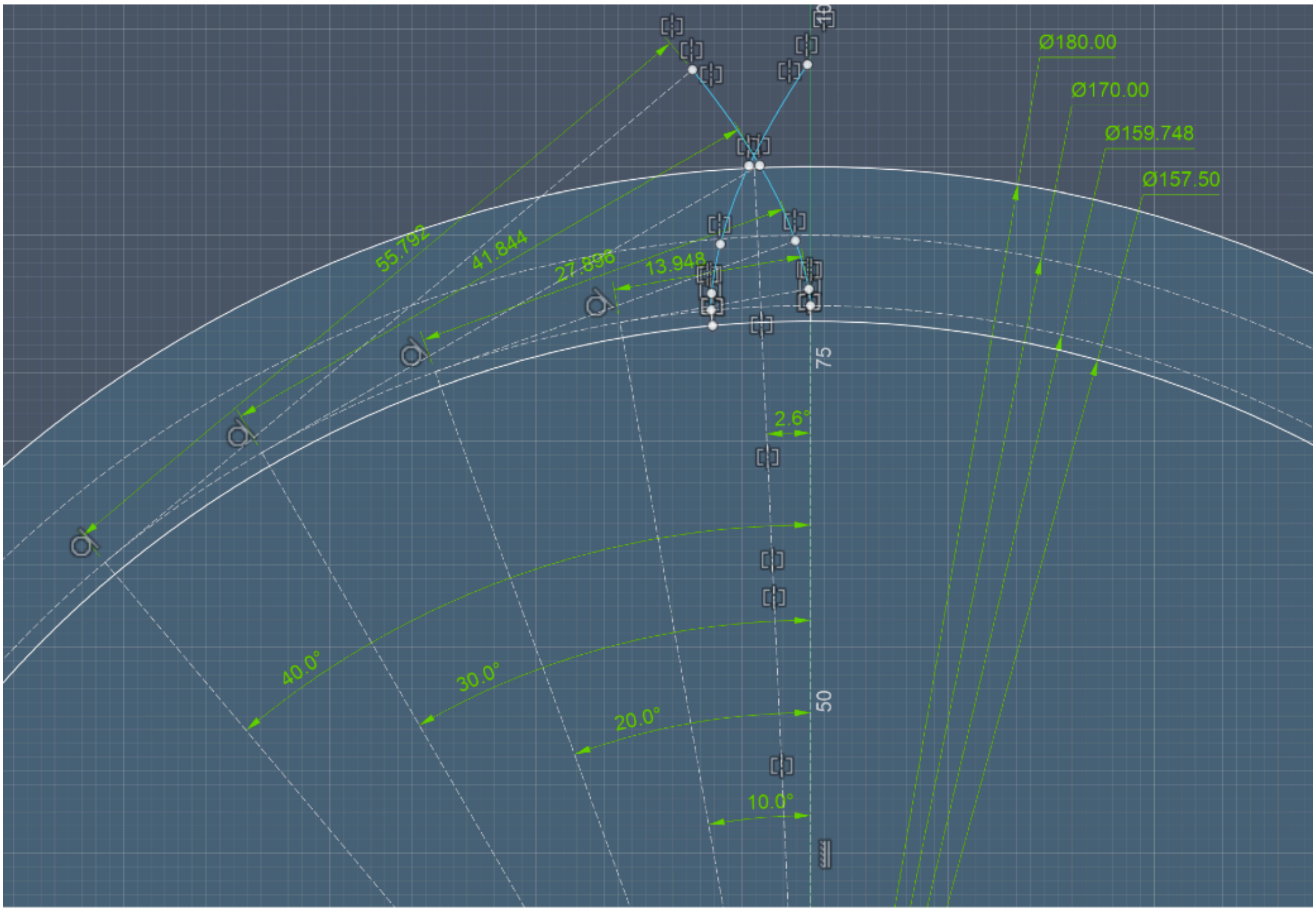
Gear design worksheet

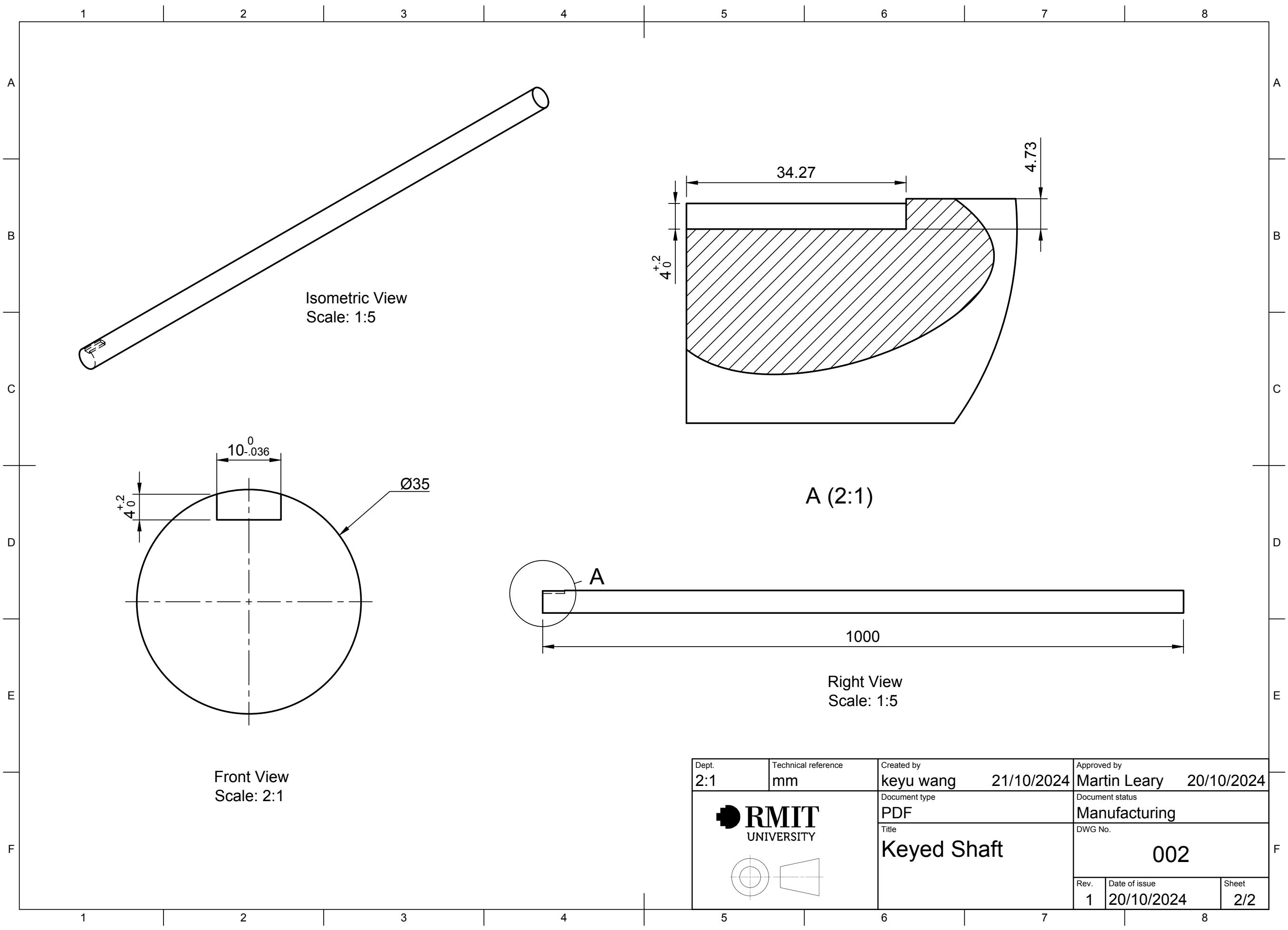
shaft diameter:	35mm
minium moudle	4.1667mm
minium torque increase	60%
pinion N	21
gear N	34
actual torque increase	62%
motor design power	19.5 kW

moudle available(mm)	Pd (pinion)mm	Pd(gear) mm	Vt(m/s)	Kv	Ft(N)	tooth breadth(mm)	estimated volume (pinion+gear)mm^3
4.5	94.5	153	7.269885	1.7	2682.2983	18.7056666	25386.31125
5	105	170	8.07765	1.8	2414.0684	16.04285994	31341.125
5.5	115.5	187	8.885415	1.85	2194.6077	13.62685532	37922.76125
6	126	204	9.69318	1.9	2011.7237	11.75981246	45131.22

Detailed Drawings for Gear and Shaft




Ø180.00
Ø170.00
Ø159.748
Ø157.50
55.792
41.844
27.896
13.948
75
2.6°
40.0°
30.0°
20.0°
50
10.0°


Isometric View
Scale: 1:5
34.27
4.73
4 +2 0
A (2:1)
10 0 -.036
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Right View
Scale: 1:5
Front View
Scale: 2:1
Dept.
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Approved by
Martin Leary 20/10/2024
RMIT UNIVERSITY
Document type
PDF
Document status
Manufacturing
Title
Keyed Shaft
DWG No.
002
Rev.
1
Date of issue
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Sheet
2/2

Reference List

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2.Standards Australia. (2004). Design of Rotating Steel Shafts (AS 1403–2004). Sydney, Australia: Standards Australia.
3.Standards Australia. (1999). SAA HB6–1999: Standards Australia Handbook—Design of Machine Elements. Sydney, Australia: Standards Australia.