Homework Project

MACHINE DESIGN 2020-2021

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PART1

Shaft A2: External forces

- As the power input is **8Kw** on the shaft A1

Shaft rotates with **1000 rpm = 104.7198 rad/s** (Assuming clockwise direction in the whole report)

$$P_i = 8000\ W$$

- Since $.\ \tau_{12} = \frac{Z_2}{Z_1} = \frac{44}{33}$ with $Z_2 = 44$ and $Z_1 = 15$

$.\ \tau_{34} = \frac{Z_4}{Z_3} = \frac{45}{15}$ with $Z_4 = 45$ and $Z_3 = 15$

- Since Radius, $r = \frac{m_n Z}{2cos\psi}$

… Gear 2 and Gear 1 have the same m_n, and likewise for the Gear 4 and Gear 3

$\Psi = 25°$ for all gears

Thus, $r_1 = 28.69\ mm$

$r_2 = 97.10\ mm$

$r_3 = 41.38\ mm$

$r_4 = 124.13\ mm$

- Assuming an efficiency of 1

$C_1\ \omega_1 = C_2\ \omega_2$

$C_1 = \frac{P_{in}}{\omega_1}$

$\frac{\omega 1}{\omega 2} = \frac{Z_2}{Z_1} = \tau_{12}$

$$C_2 = P_{in}\ .\ \tau_{12}\ /\omega_1 = 258.6Nm$$

- …Torque C_2 applied on gear 2 induces force on gear 2 as F_{t2} which must be balanced by the torque C_3 applied by gear 2.

Thus, $C_3 = F_{t3}\ r_3 = F_{t2}\ r_2 = C_2$ and so $C_3 = C_2$

$$F_{t2} = C_2/r_2 = 2663N$$

$$F_{t3} = C_2\ /r_3 = 6249\ N$$

- For force calculations on helical gears, helix angle (ψ) = 25°

Normal pressure angle **(ϕ_n) =20°**

Thus, transverse pressure angle(ϕ_t) = tg^{-1} [(tg ϕ_n)/ (cos ψ)]

ϕ_t=21.88°

- with,

$F_r = F_r \cdot tg\, \phi_t \cdot (\cos \psi)^{-1}$

$F_a = F_t \cdot tg\, \psi$

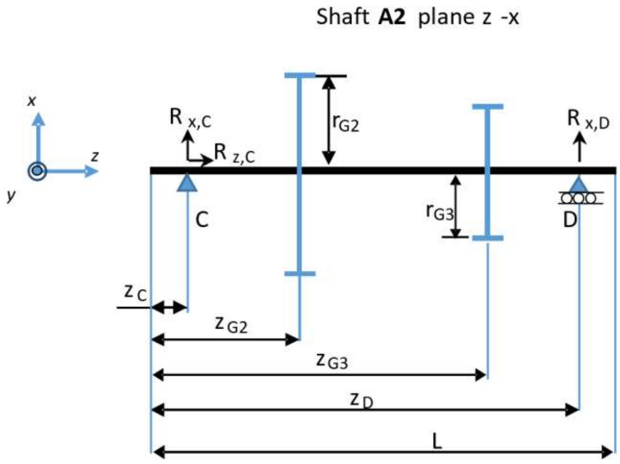
From the above two equation we get forces on shaft as,

F_{r2} =1180N **F_{r3} = 2769N**

F_{a2} =1242N **F_{a3} = 2914 N**

Shaft A2: Reaction forces plane z-x

- taking midpoint as point of application for deep groove ball bearings and same for gears, the midpoints positions of reference geometry from the technical drawing of shaft A2 are:



Z_c = 15.6 mm

Z_{G2} = 55mm

Z_{G3} =119.5mm

Z_D = 172.4mm

L= 188 mm

coordinates of z axis considering that Z=0 is where the shaft start from left to right in the required graphs.

- For reaction forces On Z-X plane,

Static equations:

↑) $R_{xc} + F_{R2} + F_{T3} + R_{XD} = 0$

→) $R_{ZC} + F_{a2} + F_{a3} = 0$

Thus, **R_{ZC}= 1672N**

Moment about C) $F_{a2}\, r_2 - F_{T3}\,(Z_{G3} - Z_C) - R_{XD}(Z_D - Z_C) = 0$ (assuming counter clockwise as positive)

R_{XC} = -1994N

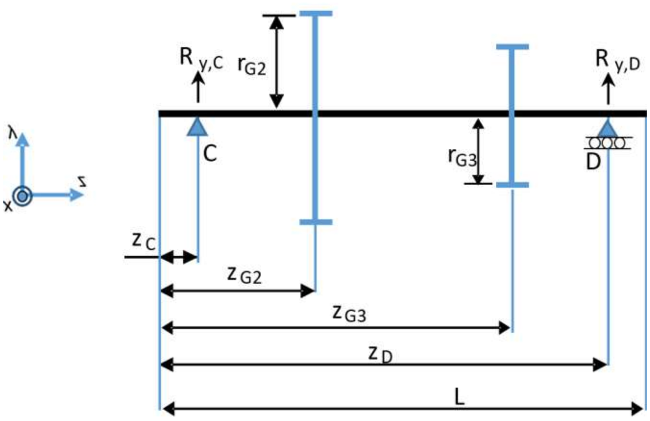
R_{XD} = -3075N

R_{ZC} = 1672N

Shaft A2: Reaction forces plane z-y

- For reaction forces on Z-Y plane,

Shaft **A2** plane z -y



With static equations,

↑) $R_{YC} + F_{t2} - F_{R3} + R_{YD} = 0$

Moment about C) $-F_{t2}(Z_{C2} - Z_C) + F_{R3}(Z_{G3} - Z_C) - R_{YD}(Z_D - Z_C) = 0$ (Assuming counter clockwise as positive)

$\mathbf{R_{YC} = -291N}$

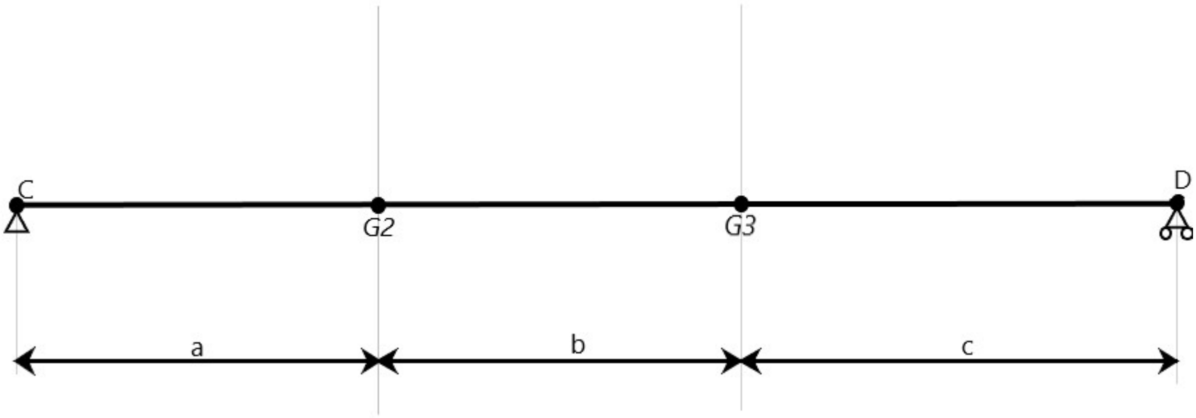
$\mathbf{R_{YD} = 397N}$

- For simplification, taking $Z_{G2} - Z_C = \mathbf{a = 39.4mm}$

$Z_{G3} - Z_{G2} = \mathbf{b = 64.5mm}$

$Z_b - Z_{G3} = \mathbf{c = 52.9mm}$

We get,



Internal loads: plane z-x

Using sectioning and superposition principle for Internal Loads on Z-X plane

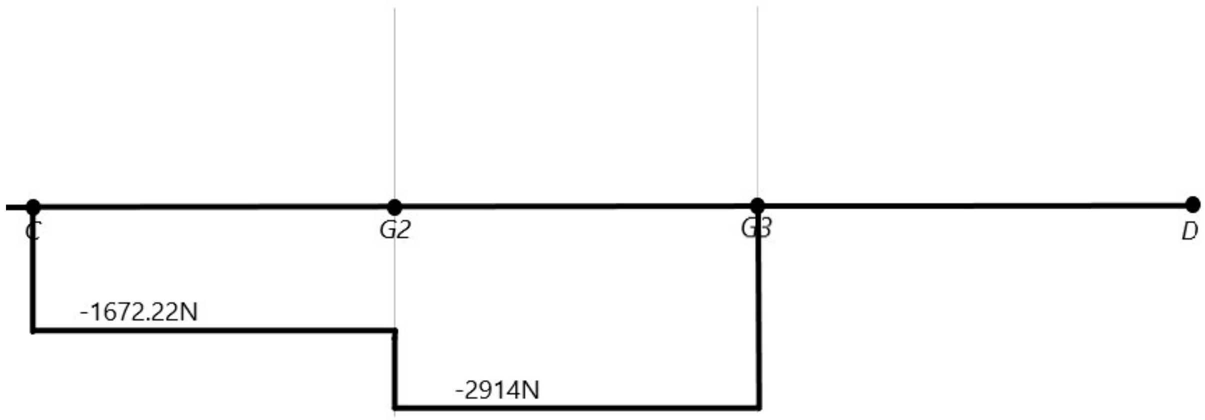
- **N(z)**

in section a : N = -Rzc=**-1672.22N**

in section b: N = -Rzc – F_{a2}=**-2914N**

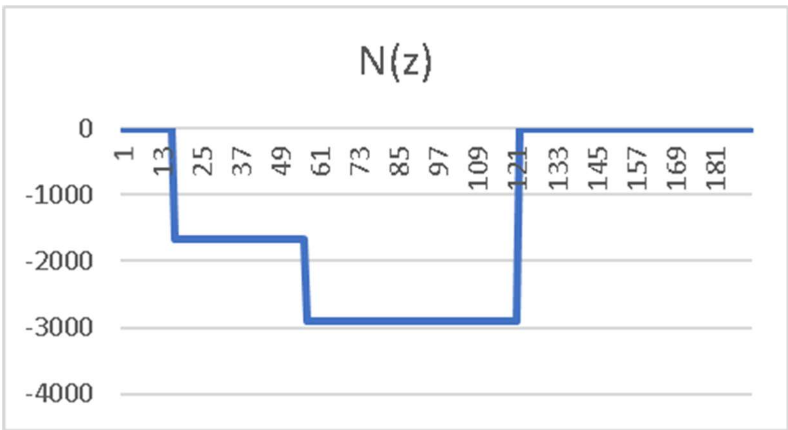
in section c : N = -Rzc-F_{a2}+F_{a3} =**0N**

the graph below depicts this trend of N(Z) along shaft,



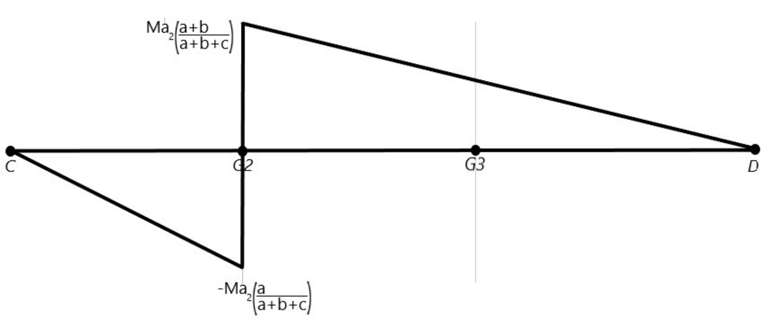
The following graph contains a more accurate representation of N(z) on Y-axis as Force in (N)

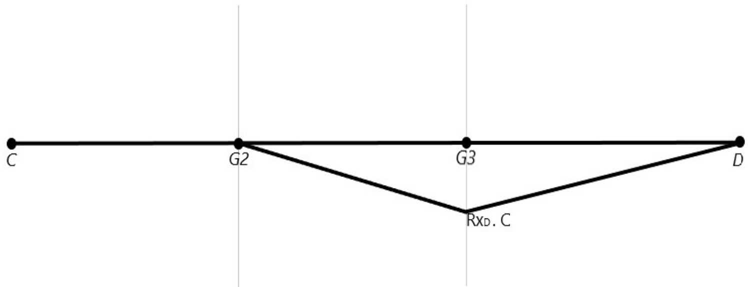
and on X-axis as the z coordinate of the position on the Shaft in (mm).

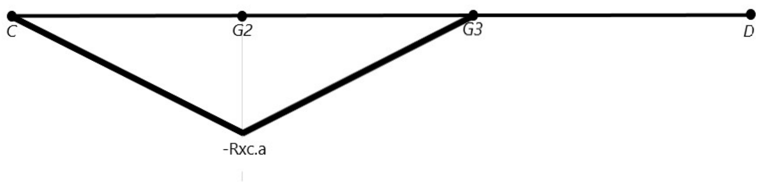


- **$M_Y(z)$**

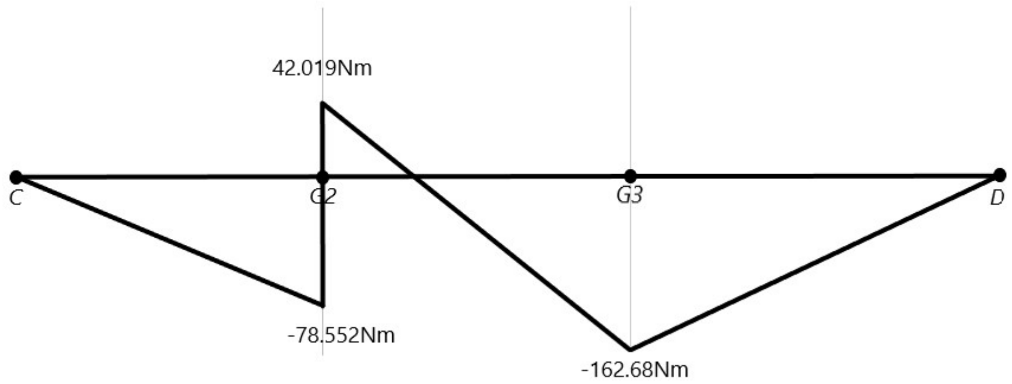
For the trend of $M_y(Z)$ using the superposition principle,



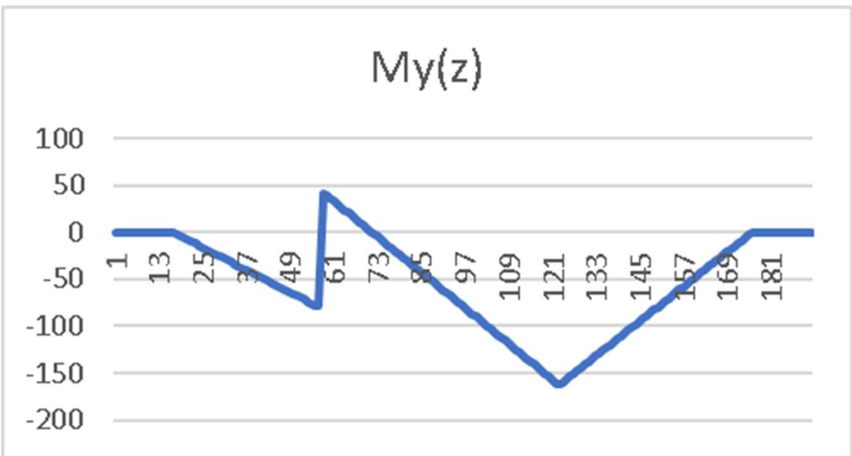




Results in overall trend of moment My,



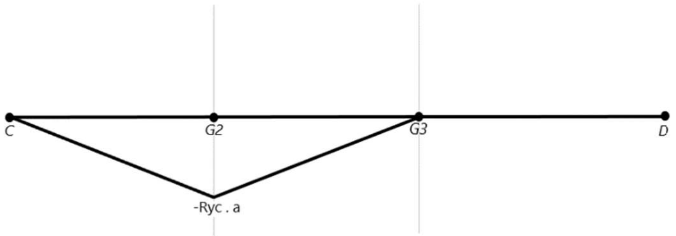
The following graph contains a more accurate representation of $M_y(z)$ with Y-axis as Moment in (Nm) and X-axis as the z coordinate of the position on the Shaft in (mm).

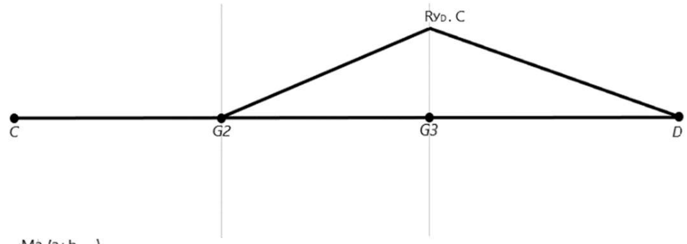


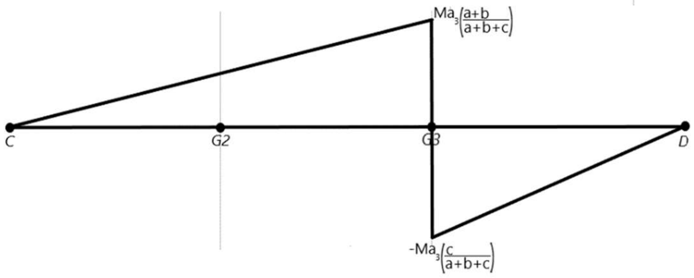
Internal loads: plane z-y

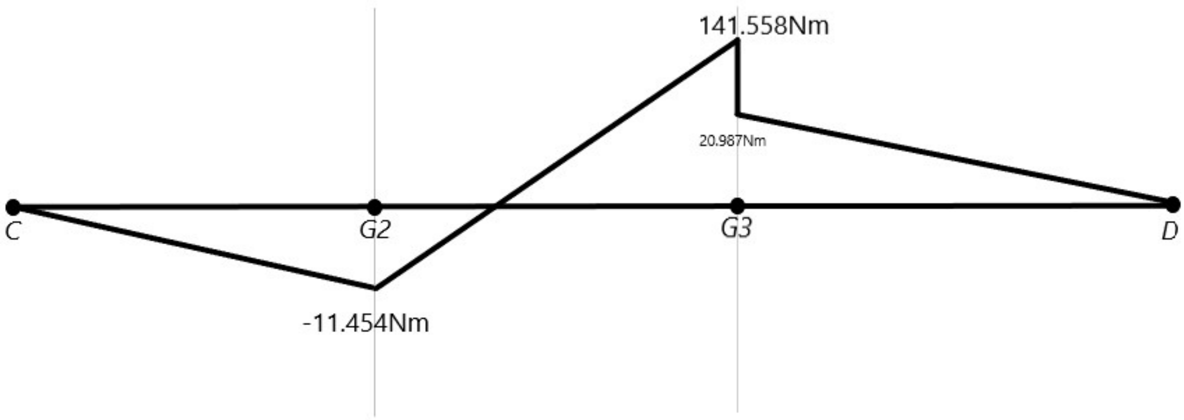
- **$M_x(z)$**

Using superposition principle for trend of bending moment Mx along shaft



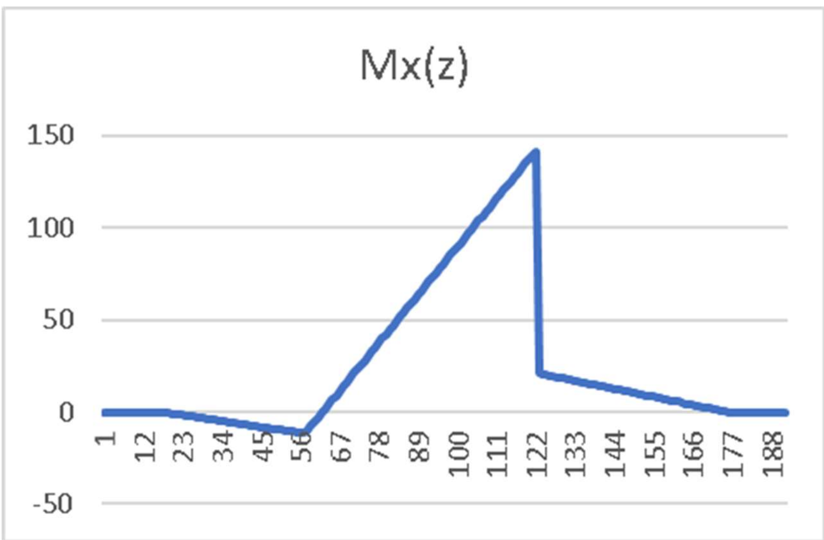






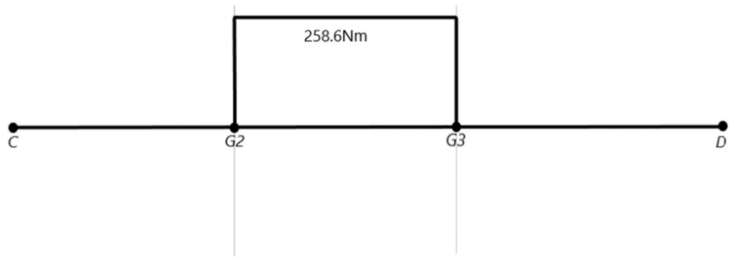
The following graph contains a more accurate representation of $M_x(z)$ with Y-axis as Moment in Nm

and X-axis as the z coordinate of the position on the Shaft in mm.

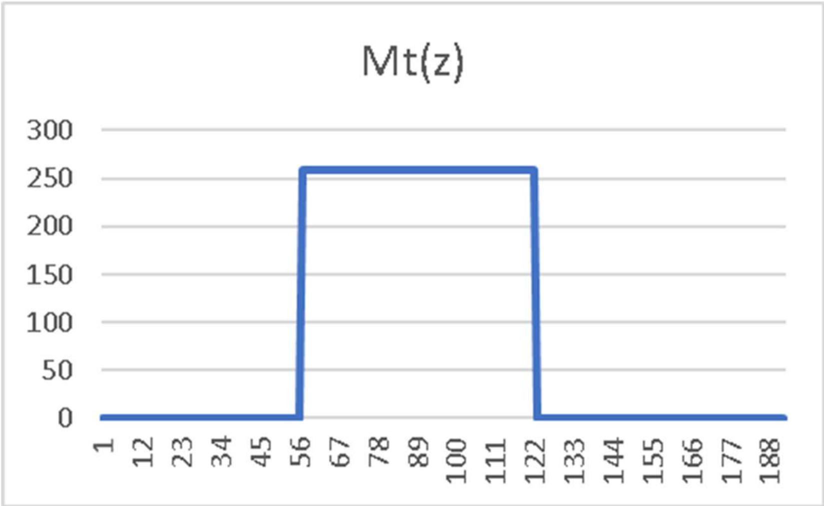


- **M_t(z)** (Torsional Moment) is applied between two gears, and it is equal to C_2=C_3

Thus, the trend is :

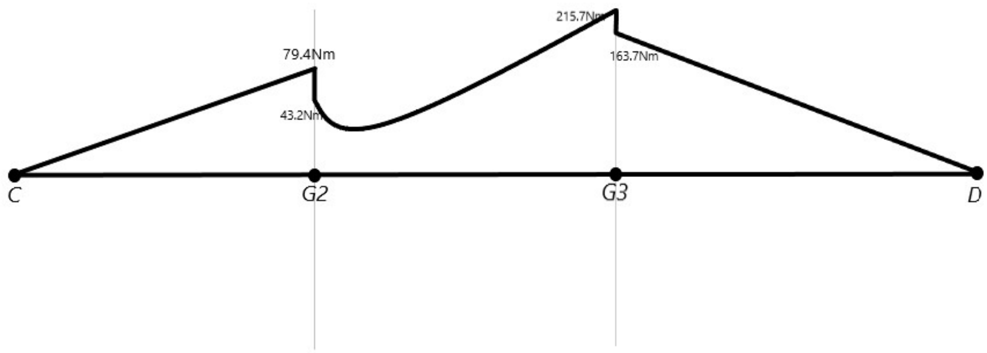


The following graph contains a more accurate representation of M_t(z) with Y-axis as Moment in Nm and X-axis as the z coordinate of the position on the Shaft in mm.

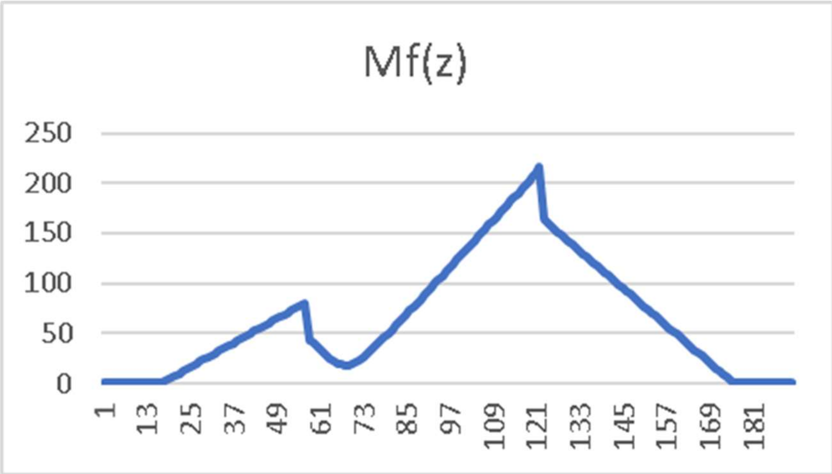


Internal loads: resulting bending moment

- **M_f =** $\sqrt{(M_x^{\,2} + M_y^{\,2})}$ Resulting Bending moment trend in (Nm)



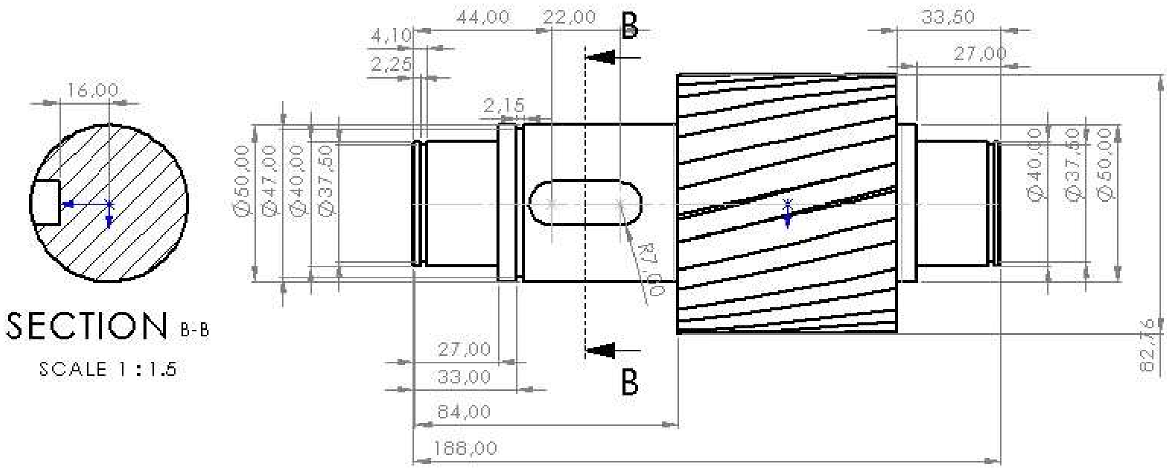
The following graph contains a more accurate representation of $M_f(z)$ with Y-axis as Moment in Nm and X-axis as the z coordinate of the position on the Shaft in mm.



Diagrams of cross-section and section moduli

- **A(z) : trend of cross section area**

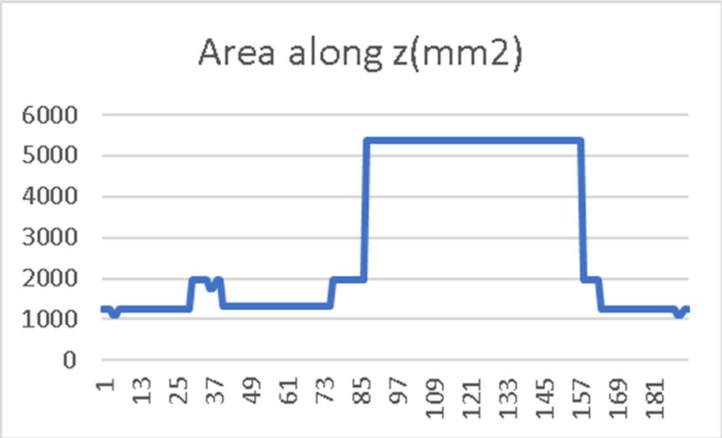
  First, to calculate the area, using technical drawing of shaft A2, taking all the necessary diameter and length dimensions as,



For Gear Area : taken gear pitch circle for diameter

For key seat section: D = surface diameter -Key seat depth = 50-9 = 41mm

the resulting cross section area trend, using diameter obtained with equation $A = \pi \frac{d^2}{4}$



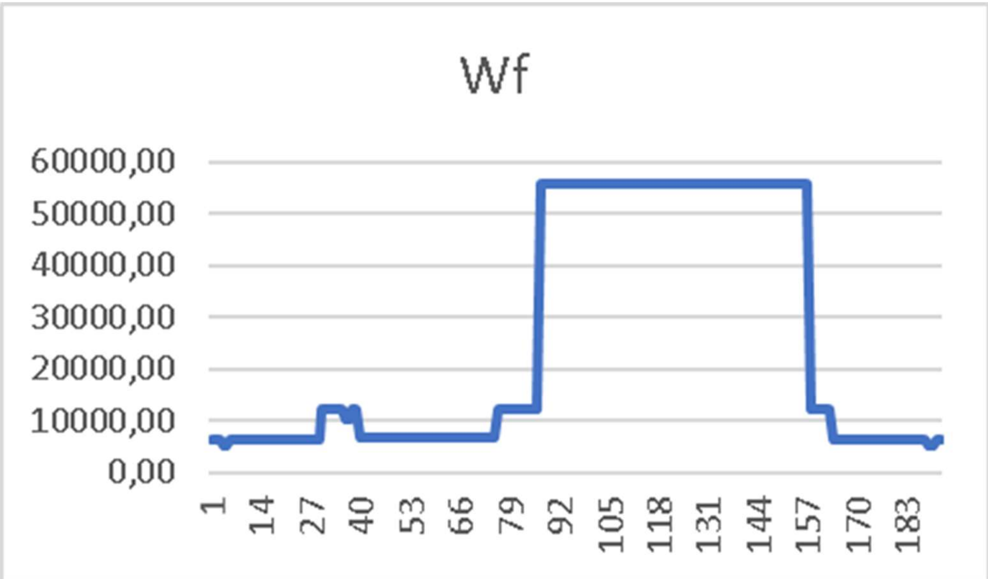
- **W_f(Z) : Bending section Modulus**

Bending section modulus is calculated as $\mathbf{W_f}=\frac{\pi d^3}{32}$ for each section of the shaft,

Thus, giving resulting trend as

The following graph contains a more accurate representation of $W_f(z)$ with Y-axis as W_f in (mm^3)

and X-axis as the z coordinate of the position on the Shaft in (mm).



- **$W_t(z)$: Torsional section Modulus**

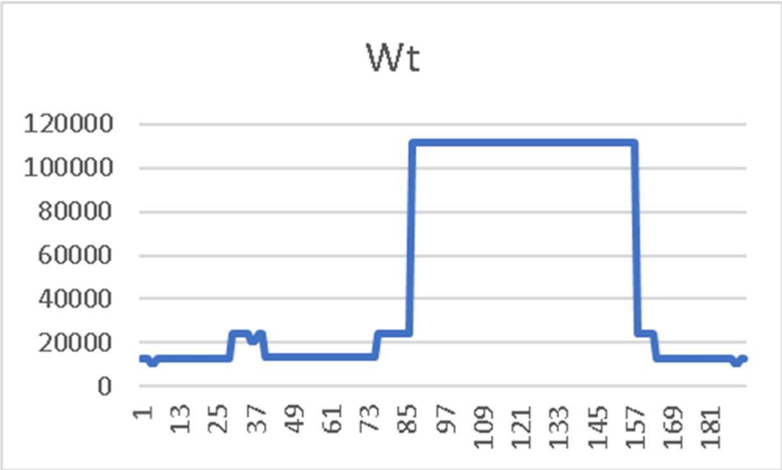
. Torsional sectional modulus is calculated as,

$W_t = \frac{\pi d^3}{16}$ for each section of the shaft,

And thus, the resulting trend is,

The following graph contains a more accurate representation of $W_f(z)$ with Y-axis as W_t in (mm^3)

and X-axis as the z coordinate of the position on the Shaft in (mm).

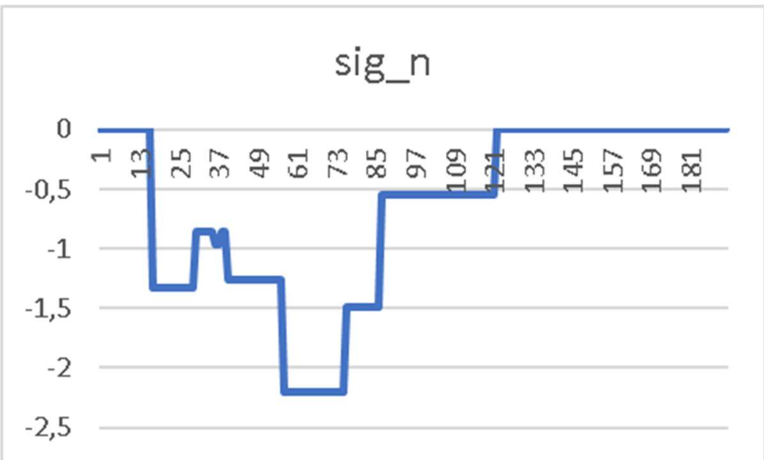


Stress in the shaft: single components

- **$\sigma^N(z)$: Stress due to Normal load**

and σ^N is calculated as $\sigma^N = \frac{N}{A_{section}}$

And thus, trend comes as ,

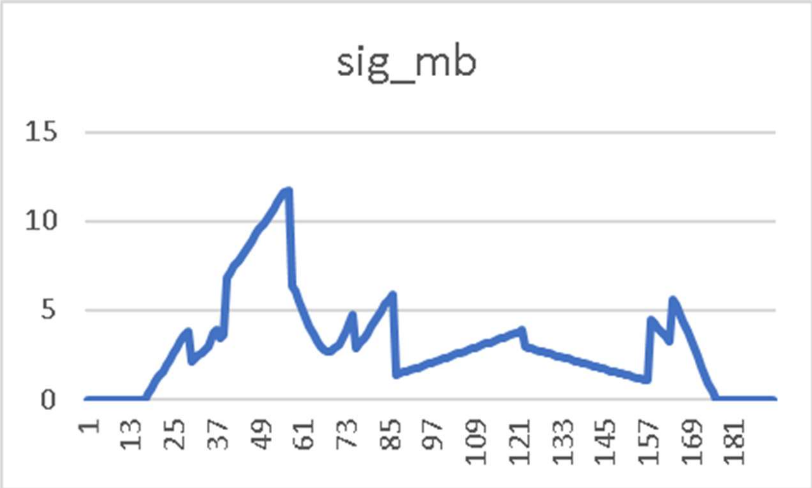


The above graph contains a more accurate representation of $\sigma^N(z)$with Y-axis as σ^N in MPa and X-axis as the z coordinate of the position on the Shaft in mm.

- **$\sigma^M{}_B$ (z) : Stress due to bending moment**

$\sigma^M{}_B$ calculated as $\boldsymbol{\sigma^M{}_B = M_f/W_f}$

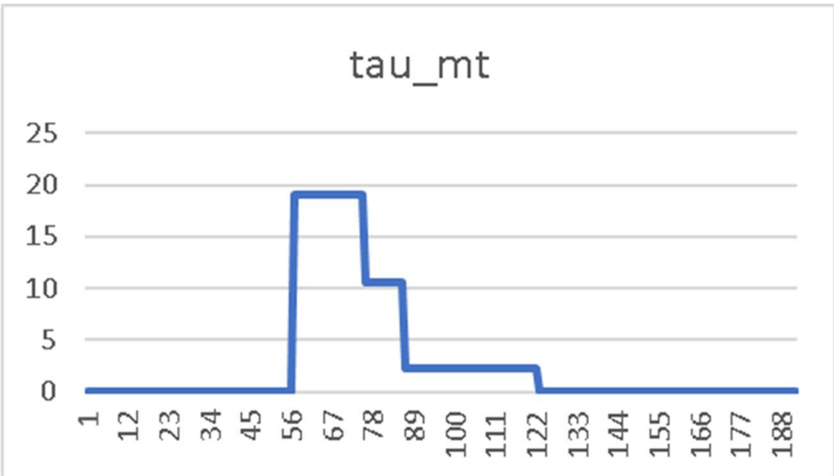
And the trend along z-axis:



The above graph contains a more accurate representation of $\sigma^M{}_B$ (z)with Y-axis as $\sigma^M{}_B$ in MPa and X-axis as the z coordinate of the position on the Shaft in mm.

- **τ^{Mt} (z) : stresses due to torsional moment**

τ^{Mt} trend was found using $\boldsymbol{\tau^{Mt} = M_t/W_t}$



The above graph contains a more accurate representation of τ^{Mt} (z) with Y-axis as τ^{Mt} in MPa

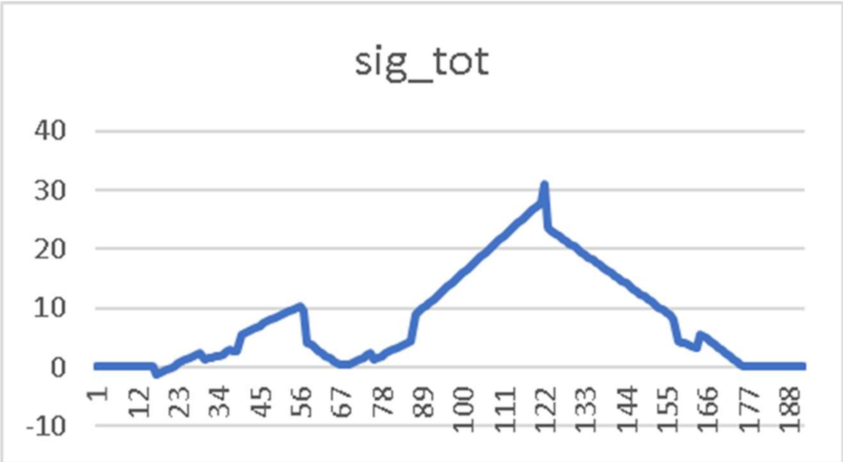
and X-axis as the z coordinate of the position on the Shaft in mm.

Stress on the shaft: resultants

- **$\sigma^{tot}(z)$: total resulting normal stress**

$$\sigma^{tot} = \sigma^{N} + \sigma^{M}{}_{B}$$

using table trend shows:



The above graph contains a more accurate representation of σ^{tot} (z) with Y-axis as σ^{tot} in MPa

and X-axis as the z coordinate of the position on the Shaft in mm.

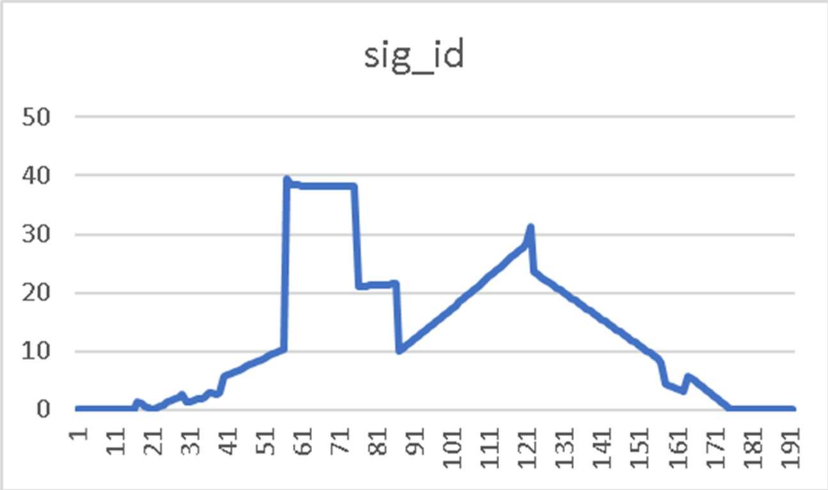
- **$\sigma^{id}(z)$:**

Full yielding criterion, which is when maximum distortion happens, done with Von Mises yield criterion. For design purpose, to be more conservative, we use **Tresca** criterion.

thus ,

$$\sigma^{id} = \sigma^{eq} = \sqrt{(\sigma^{tot^2} + 4\,\tau^2)}$$

And the trend will be ,



The above graph contains a more accurate representation of $\sigma^{id}(z)$with Y-axis as σ^{id} in (MPa) and X-axis as the z coordinate of the position on the Shaft in (mm).

Static safety factor

- From 2D technical drawing of shaft,

Z_{v1}=Z_{G2}=55mm , being mid-point of gear 2

Z_{v2}=Z_{G3}=119.5mm ,being mid-point of gear 3

Z_{v3}=161mm ,being shoulder between gear 3 and Bearing D

- For safety factor calculation for each section,

SF =σ_y / σ_{id}

Where **σ_y is 850 MPa** given.

σ_{id} For V_1=39.4MPa

For V_2=6.1MPa

For V_1=5.6MPa

with the values we get safety for the three sections as,

$\mathbf{SF_{v1}}$= 850/39.4 **=21.6**

$\mathbf{SF_{v2}}$= 850/6.1 **=140.5**

$\mathbf{SF_{v3}}$= 850/6.8 **=151.1**

All three Safety factors are quite high which is very safe for design of the shaft

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- For SF_{min} to find the most critical section of the shaft A2, we find $\boldsymbol{\sigma_{id}^{max}}$ which is **39.4 MPa** at section V_1 Thus

$\mathbf{SF_{min}= SF_{v2}=21.6}$

Shaft is well designed for given working condition of static loading under given working conditions and loads of from the gearbox.

All the above data and diagrams have been obtained through excel calculations shown in following pages:

	Z	dm	area	Wf	Wt	sig_n	sig_mb	tau_mt	sig_tot	sig_id	SF
	mm	mm	mm2	mm3	mm3	Mpa	Mpa	Mpa			
	0	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	1	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	2	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	3	37.5	1104	5177.19	10354.37	0	0	0	0	0	#DIV/0!
	4	37.5	1104	5177.19	10354.37	0	0	0	0	0	#DIV/0!
	5	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	6	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	7	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	8	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	9	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	10	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	11	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	12	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	13	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	14	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	15	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
Zc	15.6	40	1257	6283.19	12566.37	-1.33071	0	0	-1.33071	1.330708	638.7577
	16	40	1257	6283.19	12566.37	-1.33071	0.320853	0	-1.00985	1.009855	841.7051
	17	40	1257	6283.19	12566.37	-1.33071	0.641706	0	-0.689	0.689002	1233.669
	18	40	1257	6283.19	12566.37	-1.33071	0.962559	0	-0.36815	0.368149	2308.85
	19	40	1257	6283.19	12566.37	-1.33071	1.283413	0	-0.0473	0.047295	17972.14
	20	40	1257	6283.19	12566.37	-1.33071	1.604266	0	0.273558	0.273558	3107.205
	21	40	1257	6283.19	12566.37	-1.33071	1.925119	0	0.594411	0.594411	1429.987
	22	40	1257	6283.19	12566.37	-1.33071	2.245972	0	0.915264	0.915264	928.6937
	23	40	1257	6283.19	12566.37	-1.33071	2.566825	0	1.236117	1.236117	687.6371
	24	40	1257	6283.19	12566.37	-1.33071	2.887678	0	1.55697	1.55697	545.932
	25	40	1257	6283.19	12566.37	-1.33071	3.208532	0	1.877824	1.877824	452.6517
	26	40	1257	6283.19	12566.37	-1.33071	3.529385	0	2.198677	2.198677	386.5962
	27	40	1257	6283.19	12566.37	-1.33071	3.850238	0	2.51953	2.51953	337.3645
	28	50	1963	12271.85	24543.69	-0.85165	2.135599	0	1.283945	1.283945	662.0219
	29	50	1963	12271.85	24543.69	-0.85165	2.299875	0	1.448222	1.448222	586.9265
	30	50	1963	12271.85	24543.69	-0.85165	2.464152	0	1.612499	1.612499	527.1321
	31	50	1963	12271.85	24543.69	-0.85165	2.628429	0	1.776776	1.776776	478.3946
	32	50	1963	12271.85	24543.69	-0.85165	2.792706	0	1.941053	1.941053	437.9067
	33	50	1963	12271.85	24543.69	-0.85165	2.956983	0	2.10533	2.10533	403.7373
	34	47	1735	10192.80	20385.6	-0.96384	3.75791	0	2.794065	2.794065	304.2163
	35	47	1735	10192.80	20385.6	-0.96384	3.955694	0	2.99185	2.99185	284.1052
	36	50	1963	12271.85	24543.69	-0.85165	3.449813	0	2.59816	2.59816	327.1546
	37	50	1963	12271.85	24543.69	-0.85165	3.61409	0	2.762437	2.762437	307.6993
	38	41	1320	6766.30	13532.61	-1.26659	6.852713	0	5.586126	5.586126	152.1627
	39	41	1320	6766.30	13532.61	-1.26659	7.150657	0	5.88407	5.88407	144.4578
	40	41	1320	6766.30	13532.61	-1.26659	7.448601	0	6.182014	6.182014	137.4956
	41	41	1320	6766.30	13532.61	-1.26659	7.746546	0	6.479958	6.479958	131.1737
	42	41	1320	6766.30	13532.61	-1.26659	8.04449	0	6.777902	6.777902	125.4075
	43	41	1320	6766.30	13532.61	-1.26659	8.342434	0	7.075847	7.075847	120.127
	44	41	1320	6766.30	13532.61	-1.26659	8.640378	0	7.373791	7.373791	115.2731
	45	41	1320	6766.30	13532.61	-1.26659	8.938322	0	7.671735	7.671735	110.7963
	46	41	1320	6766.30	13532.61	-1.26659	9.236266	0	7.969679	7.969679	106.6542
	47	41	1320	6766.30	13532.61	-1.26659	9.53421	0	8.267623	8.267623	102.8107
	48	41	1320	6766.30	13532.61	-1.26659	9.832154	0	8.565567	8.565567	99.23453
	49	41	1320	6766.30	13532.61	-1.26659	10.1301	0	8.863511	8.863511	95.89879
	50	41	1320	6766.30	13532.61	-1.26659	10.42804	0	9.161455	9.161455	92.78002
	51	41	1320	6766.30	13532.61	-1.26659	10.72599	0	9.459399	9.459399	89.85772
	52	41	1320	6766.30	13532.61	-1.26659	11.02393	0	9.757343	9.757343	87.11388
	53	41	1320	6766.30	13532.61	-1.26659	11.32187	0	10.05529	10.05529	84.53264
	54	41	1320	6766.30	13532.61	-1.26659	11.61982	0	10.35323	10.35323	82.09997
Zg2	55	41	1320	6766.30	13532.61	-2.20712	11.739	19.1094	9.531874	39.38951	21.57935
	56	41	1320	6766.30	13532.61	-2.20712	6.384392	19.1094	4.17727	38.44641	22.1087
	57	41	1320	6766.30	13532.61	-2.20712	6.035555	19.1094	3.828433	38.41007	22.12961
	58	41	1320	6766.30	13532.61	-2.20712	5.515042	19.1094	3.30792	38.36169	22.15752
	59	41	1320	6766.30	13532.61	-2.20712	5.008927	19.1094	2.801805	38.32136	22.18084
	60	41	1320	6766.30	13532.61	-2.20712	4.522048	19.1094	2.314926	38.28884	22.19968
	61	41	1320	6766.30	13532.61	-2.20712	4.061329	19.1094	1.854206	38.26375	22.21423
	62	41	1320	6766.30	13532.61	-2.20712	3.636724	19.1094	1.429602	38.24553	22.22482
	63	41	1320	6766.30	13532.61	-2.20712	3.262367	19.1094	1.055245	38.23337	22.23189
	64	41	1320	6766.30	13532.61	-2.20712	2.9574	19.1094	0.750278	38.22616	22.23608
	65	41	1320	6766.30	13532.61	-2.20712	2.745049	19.1094	0.537927	38.22259	22.23816
	66	41	1320	6766.30	13532.61	-2.20712	2.647693	19.1094	0.440571	38.22134	22.23889
	67	41	1320	6766.30	13532.61	-2.20712	2.677903	19.1094	0.470781	38.2217	22.23868
	68	41	1320	6766.30	13532.61	-2.20712	2.831599	19.1094	0.624477	38.2239	22.23739
	69	41	1320	6766.30	13532.61	-2.20712	3.090412	19.1094	0.88329	38.22901	22.23443
	70	41	1320	6766.30	13532.61	-2.20712	3.430633	19.1094	1.223511	38.23838	22.22898
	71	41	1320	6766.30	13532.61	-2.20712	3.830632	19.1094	1.62351	38.25327	22.22032
	72	41	1320	6766.30	13532.61	-2.20712	4.273656	19.1094	2.066534	38.27463	22.20792
	73	41	1320	6766.30	13532.61	-2.20712	4.747678	19.1094	2.540556	38.30315	22.19139
	74	50	1963	12271.85	24543.69	-1.48407	2.891537	10.53631	1.407469	21.11957	40.24702
	75	50	1963	12271.85	24543.69	-1.48407	3.174595	10.53631	1.690526	21.14033	40.20752
	76	50	1963	12271.85	24543.69	-1.48407	3.464626	10.53631	1.980557	21.16549	40.15971
	77	50	1963	12271.85	24543.69	-1.48407	3.760018	10.53631	2.275949	21.19517	40.10347
	78	50	1963	12271.85	24543.69	-1.48407	4.0596	10.53631	2.575531	21.22943	40.03875
	79	50	1963	12271.85	24543.69	-1.48407	4.36251	10.53631	2.878441	21.26831	39.96557
	80	50	1963	12271.85	24543.69	-1.48407	4.668099	10.53631	3.18403	21.31182	39.88398
	81	50	1963	12271.85	24543.69	-1.48407	4.975875	10.53631	3.491806	21.35997	39.79407
	82	50	1963	12271.85	24543.69	-1.48407	5.285454	10.53631	3.801385	21.41275	39.69597
	83	50	1963	12271.85	24543.69	-1.48407	5.596538	10.53631	4.112469	21.47016	39.58983
	84	50	1963	12271.85	24543.69	-1.48407	5.908889	10.53631	4.42482	21.53217	39.47581
	85	82.76	5379	55649.51	111299	-0.54169	1.372147	2.323471	8.810406	9.960789	85.33461
	86	82.76	5379	55649.51	111299	-0.54169	1.441467	2.323471	9.364968	10.45451	81.30466
	87	82.76	5379	55649.51	111299	-0.54169	1.510963	2.323471	9.920932	10.95532	77.5879
	88	82.76	5379	55649.51	111299	-0.54169	1.580611	2.323471	10.47811	11.46233	74.15598
	89	82.76	5379	55649.51	111299	-0.54169	1.650391	2.323471	11.03636	11.97478	70.98254
	90	82.76	5379	55649.51	111299	-0.54169	1.720288	2.323471	11.59554	12.49202	68.04345
	91	82.76	5379	55649.51	111299	-0.54169	1.790289	2.323471	12.15554	13.0135	65.31679
	92	82.76	5379	55649.51	111299	-0.54169	1.86038	2.323471	12.71627	13.53874	62.78278
	93	82.76	5379	55649.51	111299	-0.54169	1.930553	2.323471	13.27765	14.06734	60.42363

	94	82.76	5379	55649.51	111299	-0.54169	2.000799	2.323471	13.83962	14.59895	58.22338
	95	82.76	5379	55649.51	111299	-0.54169	2.071111	2.323471	14.40211	15.13324	56.16775
	96	82.76	5379	55649.51	111299	-0.54169	2.141481	2.323471	14.96508	15.66996	54.24392
	97	82.76	5379	55649.51	111299	-0.54169	2.211905	2.323471	15.52847	16.20887	52.44043
	98	82.76	5379	55649.51	111299	-0.54169	2.282377	2.323471	16.09224	16.74976	50.747
	99	82.76	5379	55649.51	111299	-0.54169	2.352893	2.323471	16.65638	17.29245	49.15439
	100	82.76	5379	55649.51	111299	-0.54169	2.42345	2.323471	17.22083	17.83679	47.65432
	101	82.76	5379	55649.51	111299	-0.54169	2.494043	2.323471	17.78557	18.38262	46.23933
	102	82.76	5379	55649.51	111299	-0.54169	2.56467	2.323471	18.35059	18.92982	44.90269
	103	82.76	5379	55649.51	111299	-0.54169	2.635329	2.323471	18.91586	19.47829	43.63833
	104	82.76	5379	55649.51	111299	-0.54169	2.706015	2.323471	19.48135	20.02791	42.44078
	105	82.76	5379	55649.51	111299	-0.54169	2.776729	2.323471	20.04706	20.5786	41.30505
	106	82.76	5379	55649.51	111299	-0.54169	2.847466	2.323471	20.61296	21.13027	40.22666
	107	82.76	5379	55649.51	111299	-0.54169	2.918227	2.323471	21.17904	21.68285	39.20149
	108	82.76	5379	55649.51	111299	-0.54169	2.989008	2.323471	21.7453	22.23628	38.22583
	109	82.76	5379	55649.51	111299	-0.54169	3.05981	2.323471	22.3117	22.79049	37.29627
	110	82.76	5379	55649.51	111299	-0.54169	3.130629	2.323471	22.87826	23.34542	36.4097
	111	82.76	5379	55649.51	111299	-0.54169	3.201465	2.323471	23.44495	23.90104	35.5633
	112	82.76	5379	55649.51	111299	-0.54169	3.272318	2.323471	24.01177	24.45729	34.75446
	113	82.76	5379	55649.51	111299	-0.54169	3.343185	2.323471	24.57871	25.01414	33.98078
	114	82.76	5379	55649.51	111299	-0.54169	3.414067	2.323471	25.14576	25.57154	33.24008
	115	82.76	5379	55649.51	111299	-0.54169	3.484961	2.323471	25.71292	26.12945	32.53034
	116	82.76	5379	55649.51	111299	-0.54169	3.555868	2.323471	26.28018	26.68786	31.84969
	117	82.76	5379	55649.51	111299	-0.54169	3.626787	2.323471	26.84753	27.24672	31.19642
	118	82.76	5379	55649.51	111299	-0.54169	3.697717	2.323471	27.41496	27.80601	30.56893
	119	82.76	5379	55649.51	111299	-0.54169	3.768657	2.323471	27.98248	28.36571	29.96576
Zg3	119.5	82.76	5379	55649.51	111299	0	3.875356	2.323471	31.00285	31.34917	27.11395
	120	82.76	5379	55649.51	111299	0	2.942197	0	23.53758	23.53758	36.11247
	121	82.76	5379	55649.51	111299	0	2.886385	0	23.09108	23.09108	36.81075
	122	82.76	5379	55649.51	111299	0	2.830663	0	22.64531	22.64531	37.53537
	123	82.76	5379	55649.51	111299	0	2.774942	0	22.19953	22.19953	38.28909
	124	82.76	5379	55649.51	111299	0	2.71922	0	21.75376	21.75376	39.07371
	125	82.76	5379	55649.51	111299	0	2.663498	0	21.30799	21.30799	39.89115
	126	82.76	5379	55649.51	111299	0	2.607776	0	20.86221	20.86221	40.74352
	127	82.76	5379	55649.51	111299	0	2.552055	0	20.41644	20.41644	41.63312
	128	82.76	5379	55649.51	111299	0	2.496333	0	19.97066	19.97066	42.56243
	129	82.76	5379	55649.51	111299	0	2.440611	0	19.52489	19.52489	43.53418
	130	82.76	5379	55649.51	111299	0	2.38489	0	19.07912	19.07912	44.55133
	131	82.76	5379	55649.51	111299	0	2.329168	0	18.63334	18.63334	45.61715
	132	82.76	5379	55649.51	111299	0	2.273446	0	18.18757	18.18757	46.73522
	133	82.76	5379	55649.51	111299	0	2.217724	0	17.7418	17.7418	47.90947
	134	82.76	5379	55649.51	111299	0	2.162003	0	17.29602	17.29602	49.14425
	135	82.76	5379	55649.51	111299	0	2.106281	0	16.85025	16.85025	50.44436
	136	82.76	5379	55649.51	111299	0	2.050559	0	16.40447	16.40447	51.81513
	137	82.76	5379	55649.51	111299	0	1.994838	0	15.9587	15.9587	53.26248
	138	82.76	5379	55649.51	111299	0	1.939116	0	15.51293	15.51293	54.79301
	139	82.76	5379	55649.51	111299	0	1.883394	0	15.06715	15.06715	56.41411
	140	82.76	5379	55649.51	111299	0	1.827672	0	14.62138	14.62138	58.13405
	141	82.76	5379	55649.51	111299	0	1.771951	0	14.17561	14.17561	59.96217
	142	82.76	5379	55649.51	111299	0	1.716229	0	13.72983	13.72983	61.90899
	143	82.76	5379	55649.51	111299	0	1.660507	0	13.28406	13.28406	63.98647
	144	82.76	5379	55649.51	111299	0	1.604786	0	12.83828	12.83828	66.20823
	145	82.76	5379	55649.51	111299	0	1.549064	0	12.39251	12.39251	68.58982
	146	82.76	5379	55649.51	111299	0	1.493342	0	11.94674	11.94674	71.14914
	147	82.76	5379	55649.51	111299	0	1.43762	0	11.50096	11.50096	73.90686
	148	82.76	5379	55649.51	111299	0	1.381899	0	11.05519	11.05519	76.88697
	149	82.76	5379	55649.51	111299	0	1.326177	0	10.60942	10.60942	80.11752
	150	82.76	5379	55649.51	111299	0	1.270455	0	10.16364	10.16364	83.63144
	151	82.76	5379	55649.51	111299	0	1.214733	0	9.717868	9.717868	87.46775
	152	82.76	5379	55649.51	111299	0	1.159012	0	9.272094	9.272094	91.67293
	153	82.76	5379	55649.51	111299	0	1.10329	0	8.82632	8.82632	96.30287
	154	82.76	5379	55649.51	111299	0	1.047568	0	8.02632	8.02632	105.9016
	155	50	1963	12271.85	24543.69	0	4.497756	0	4.497756	4.497756	188.9831
	156	50	1963	12271.85	24543.69	0	4.245074	0	4.245074	4.245074	200.2321
	157	50	1963	12271.85	24543.69	0	3.992391	0	3.992391	3.992391	212.905
	158	50	1963	12271.85	24543.69	0	3.739708	0	3.739708	3.739708	227.2905
	159	50	1963	12271.85	24543.69	0	3.487025	0	3.487025	3.487025	243.7608
	160	50	1963	12271.85	24543.69	0	3.234342	0	3.234342	3.234342	262.8046
V3	161	40	1257	6283.19	12566.37	0	5.626144	0	5.626144	5.626144	151.0804
	162	40	1257	6283.19	12566.37	0	5.330031	0	5.330031	5.330031	159.4737
	163	40	1257	6283.19	12566.37	0	4.83651	0	4.83651	4.83651	175.7466
	164	40	1257	6283.19	12566.37	0	4.342988	0	4.342988	4.342988	195.7178
	165	40	1257	6283.19	12566.37	0	3.849467	0	3.849467	3.849467	220.8098
	166	40	1257	6283.19	12566.37	0	3.355945	0	3.355945	3.355945	253.2818
	167	40	1257	6283.19	12566.37	0	2.862424	0	2.862424	2.862424	296.9511
	168	40	1257	6283.19	12566.37	0	2.368903	0	2.368903	2.368903	358.8159
	169	40	1257	6283.19	12566.37	0	1.875381	0	1.875381	1.875381	453.2412
	170	40	1257	6283.19	12566.37	0	1.38186	0	1.38186	1.38186	615.113
	171	40	1257	6283.19	12566.37	0	0.888338	0	0.888338	0.888338	956.8425
	172	40	1257	6283.19	12566.37	0	0.394817	0	0.394817	0.394817	2152.896
Zd	172.4	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	173	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	174	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	175	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	176	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	177	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	178	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	179	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	180	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	181	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	182	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	183	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	184	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	185	37.5	1104	5177.19	10354.37	0	0	0	0	0	#DIV/0!
	186	37.5	1104	5177.19	10354.37	0	0	0	0	0	#DIV/0!
	187	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!
	188	40	1257	6283.19	12566.37	0	0	0	0	0	#DIV/0!

Fatigue Verification

- **Mean and alternate stresses due to normal load, bending moment and torsional moment**

For rotating shafts under load, $\sigma_a^N = \sigma_m^{MB} = \sigma_{a=0}$

While $\sigma_m^N = \sigma^N$

$\sigma_a^{MB} = \sigma^{MB}$ $\tau_m = \tau^{mt}$, for each section

From this we got,

	σ_m^N	σ_a^N	σ_m^{MB}	σ_a^{MB}	τ_m	T_a
V1	**-2.21 Mpa**	**0 Mpa**	**0 MPa**	**11.74 MPa**	**19.1094 MPa**	**0 MPa**
V2	**0 MPa**	**0 Mpa**	**0 MPa**	**3.88 MPa**	**2.323471 MPa**	**0,00 MPa**
V3	**0 MPa**	**0 MPa**	**0 MPa**	**5.63 MPa**	**0 MPa**	**0 MPa**

- **Stress concentration factors**

V1 is a keyseat section

V2 is placed at gear 3

V3 is holder between gear 3 and gearing

So, from table given for keyseat and gear teeth,

	Bending	Torsion
Kf for keyseat	**1.6**	**2.0**
Kf for gear teeth	**1.2**	**1.5**

We get fatigue stress concertation factors Kf as,

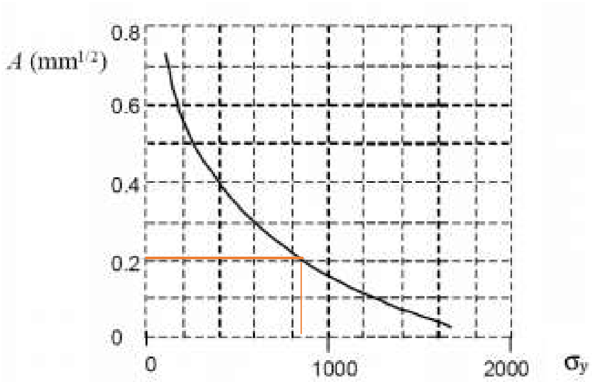
For V1 $Kf^b=1.6$, $Kf^t=2.0$

For V2 $Kf^b=1.2$, $Kf^t=1.5$

- As for V3, being a shoulder, we use different approach

taking notch radius is **r=5 mm**

σy=850 Mpa, using reference diagram



A = 0.2 mm$^{1/2}$

- Now, using these values , notch sesitivity q , from formula

$$q = \frac{1}{1+\frac{A}{\sqrt{r}}}$$

We get, **q =0.92**

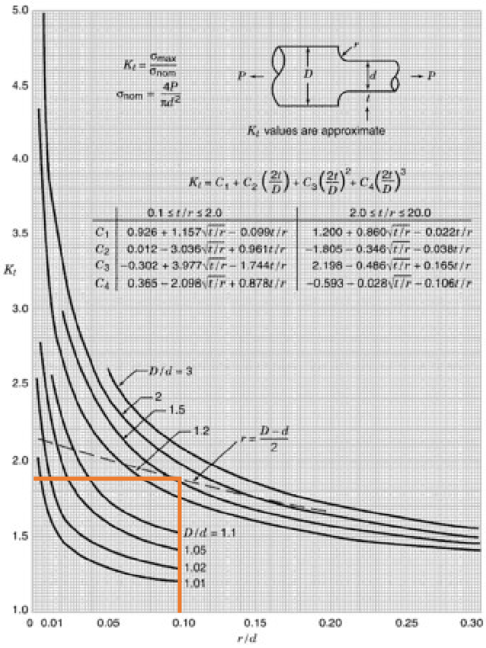
At shoulder, upper diameter **D=50 mm**, smaller diameter **d=40 mm**,

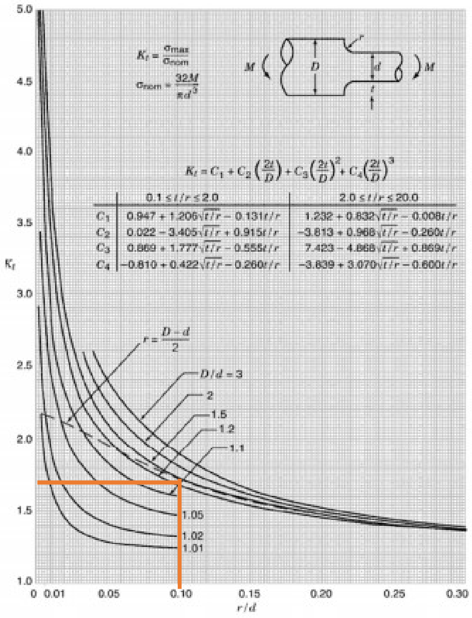
Thus,

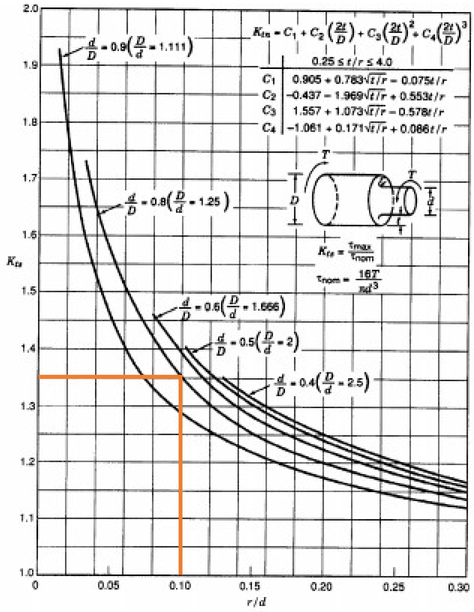
$$\frac{D}{d} = \mathbf{1.25} \qquad \frac{r}{D} = \mathbf{0.1}$$

Using these values and reference diagrams given

For U3, K_t^N because $r = \frac{D-d}{2}$,







For V3, geometric stress raiser notch factors:

K_t^N=1.88

K_t^B=1.65

K_t^τ= 1.35

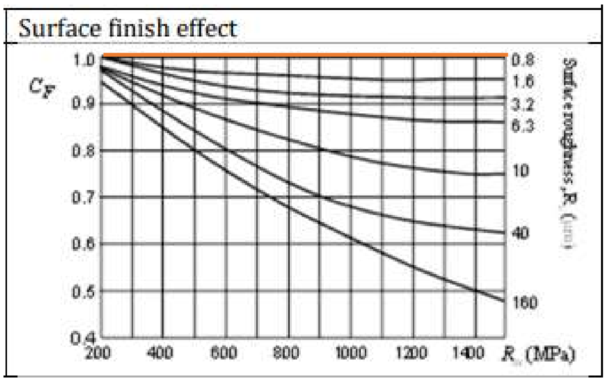
- And, for Fatigue stress concentration factor K_f,

With formula **$K_f=1+q(K_t-1)$**

K_f^N=1.81 **K_f^B=1.60** **K_f^t=1.32**

- **Fatigue limit correction factors**

obtained from the graphs for rotating bending
With Ra=0.8 on all three sections,



C_f for V1, V2, V3

C_f=1 for all three sections

- Diameter at,

dV1=**41mm**

dV2=**41.38mm**

dV3=**40mm**

So, with reference daigram for size effect we get ,

C_s=0.83 for all three sections

- **Fatigue limit of the component**

For fatigue limit of the component, $\boldsymbol{\sigma_{D-1}^{C}}$

$$\boldsymbol{\sigma_{D-1}^{C} = C_s C_f \sigma_{D-1}}$$

Where, σ_{D-1}=**550 MPa**

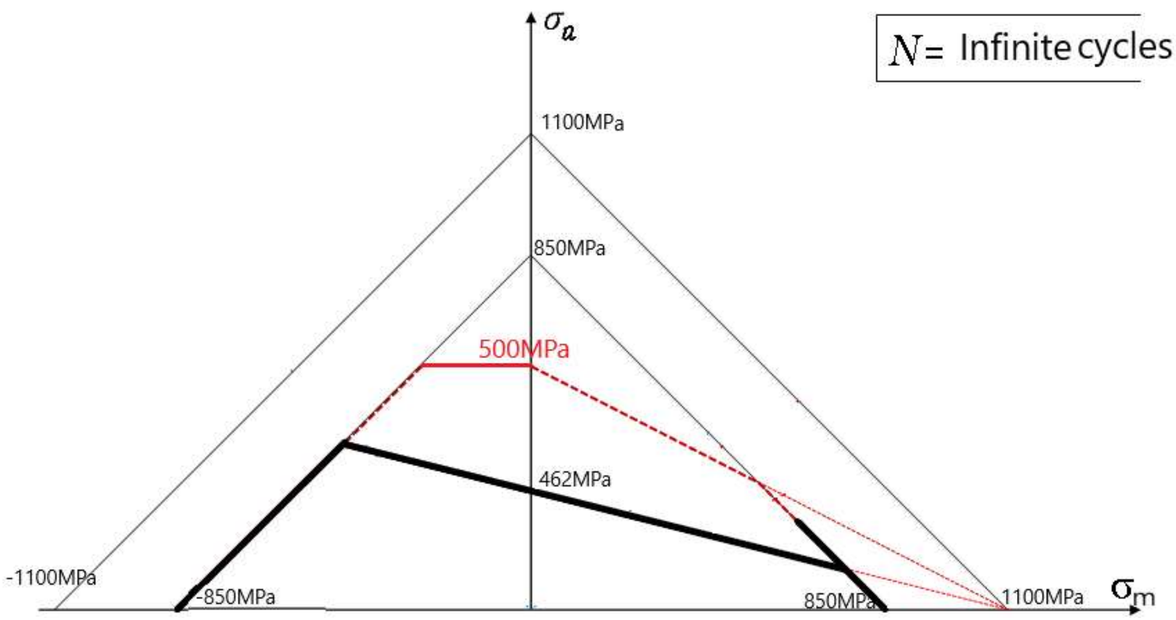
So, $\boldsymbol{\sigma_{D-1}^{c}}$ for V1 is **462 MPa**

For V2 is **462 MPa**

For V3 is **462 MPa**

- **Haigh Diagrams for infinite life cycles**

All three sections have similar $\boldsymbol{\sigma_{D-1}^{c}}$, thus all three sections will have the same Haigh diagrams for infinite life



- **Fatigue safety factor**

For coordinates of working point P,

Using formula for the equivalent stresses,

$$\sigma_{a,eq}^{P} = \sqrt{\left(K_f^B \sigma_a^B + K_f^N \frac{\sigma_a^N}{0.85}\right)^2 + 3\left(K_f^T \tau_a^T\right)^2}$$

$$\sigma_{m,eq}^{P} = \sqrt{\left(K_f^B \sigma_m^B + K_f^N \sigma_m^N\right)^2 + 3\left(K_f^T \tau_m^T\right)^2} =$$

We obtain for all three sections,

	$\sigma_{a,\ eqP}$	$\sigma_{m,\ eqP}$
V1	**18.78MPa**	**66.23MPa**
V2	**4.65MPa**	**6.036554MPa**
V3	**8.98MPa**	**0MPa**

- To calculate fatigue safety factors using Haigh Diagrams for all three sections

To calculate SF_f (fatigue safety factor)

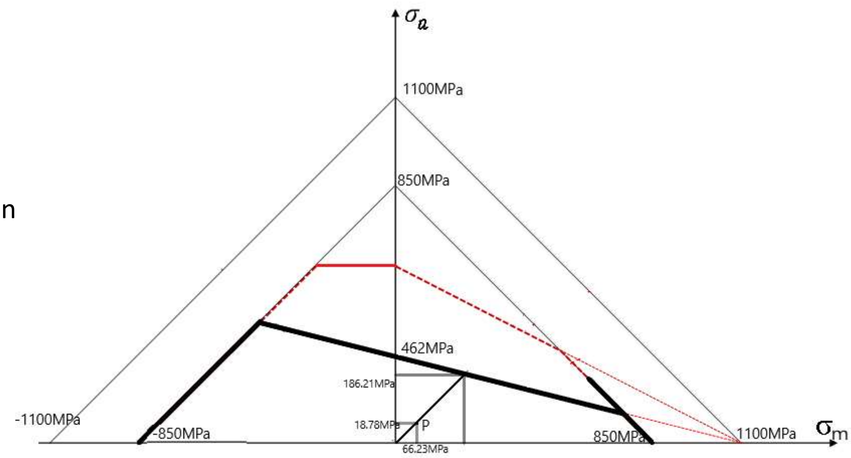
For sections V1 and V2, calcualting σ_D^{lim} as,

$$\sigma_{D,B}{}^{c} = \frac{\sigma_{D-1}^{C}}{1 + \frac{\sigma_m^P}{\sigma_a^P} \cdot \frac{\sigma_{D-1}^{C}}{\sigma_u}}$$

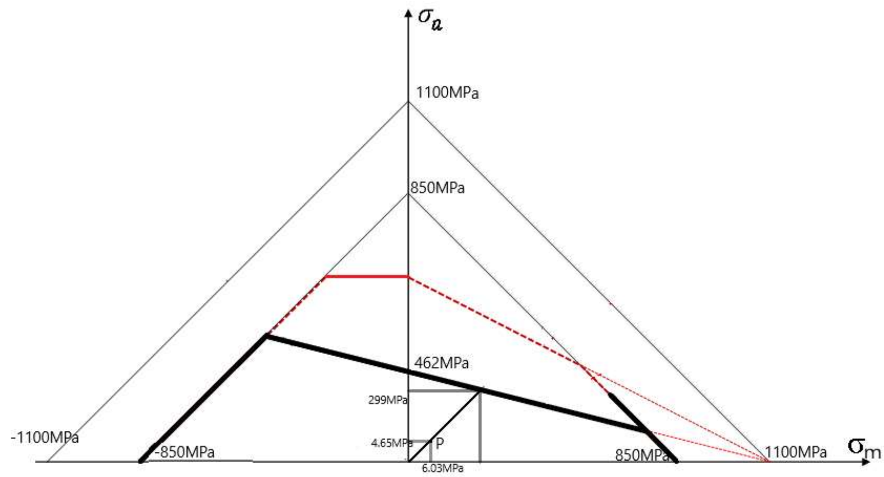
Thus ,

	σ_D^{lim}
V1	**186.21Mpa**
V2	**298.99Mpa**

Results Haigh diagram for V1 for SF calculation



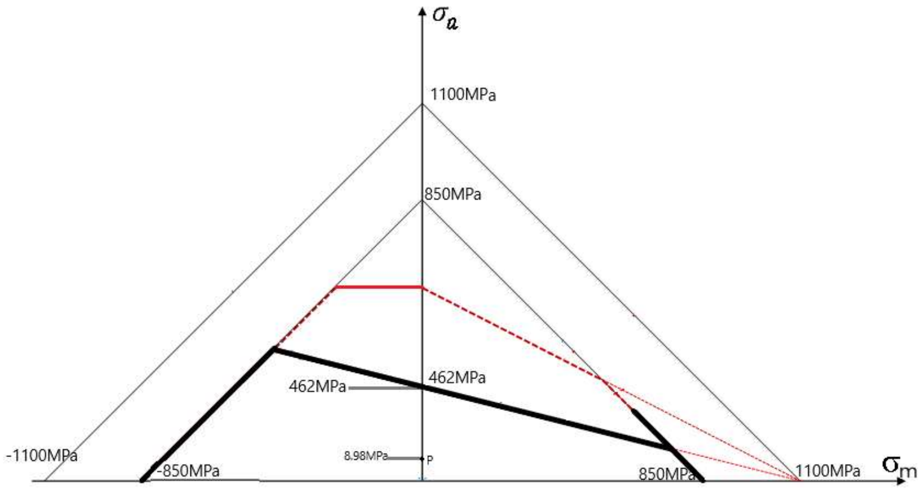
And for V2,



As for V3, since there is no mean stress component,

No need to calculate $\sigma_D^{lim\ for\ V1}$

Haigh diagram for V3



For Fatigue safety factor from Haigh diagrams above,

For V1,

$$\mathbf{SF_f = \sigma_{D,\ lim}/\sigma_{a,\ eq} = 9.9}$$

For V2,

$$\mathbf{SF_f = \sigma_{D,\ lim}/\ \sigma_{a,\ eq} = 64.3}$$

For V3,

$$\mathbf{SF_f = \sigma_{D-1}^{c}/\ \sigma_{a,\ eq} = 51.4}$$

All three values of safety factors are reasonably high for fatigue verification, this implies that, under given working conditions, with given loads and factors, shaft is very well design to meet its fatigue requirements.

GEAR TOOTH VERIFICATION

Maximum tooth gear bending stress

- **Face width b=min [b_r, bp]**

b=70mm = 2.76 inches

- Given uniform power source and driven machine gearbox

Overload Factor **K_o=1.00**

- **Rim thickness Factor 'K_B'**

For **Tr** are consider cases of shaft as tooth base

t_R= r3 - 1.25mm = 35.13 mm

Tooth height =2.25mm=11.25mm

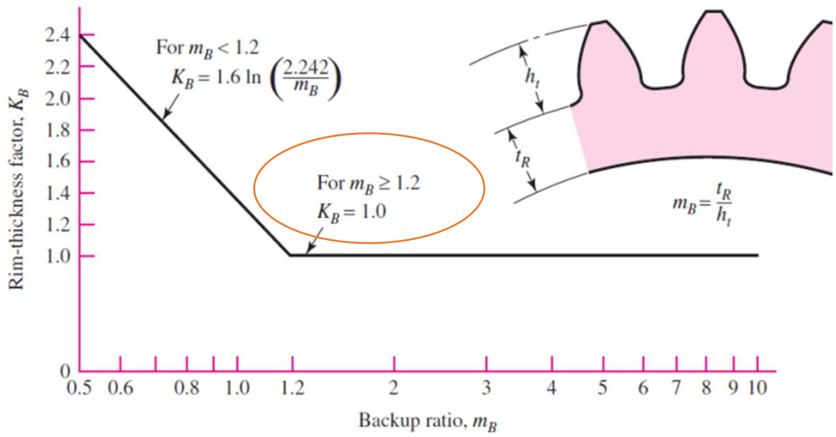
h_t=11.25mm

Backup ratio (mb)= Tr/Ht =3.12

M_B=3.12 > 1.2

Using Rim thickness factor diagram below,

We get, **K_B=1.0**



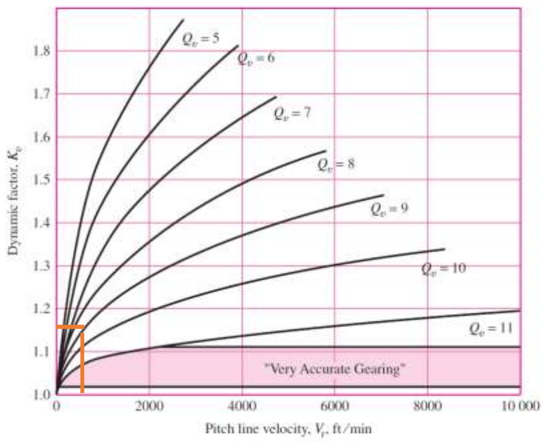
- **Dynamic Factor K_v**

Velocity of tooth is **$V=r_3w_3$ = 1.28 m/s,** **1.28 m/s = 251.97 ft/min**

While transmission accuracy level of 8 is given,

$$Q_v = 8$$

Thus, using reference diagram, we get,



$$K_v = 1.16$$

- **Load distribution factor K_h**

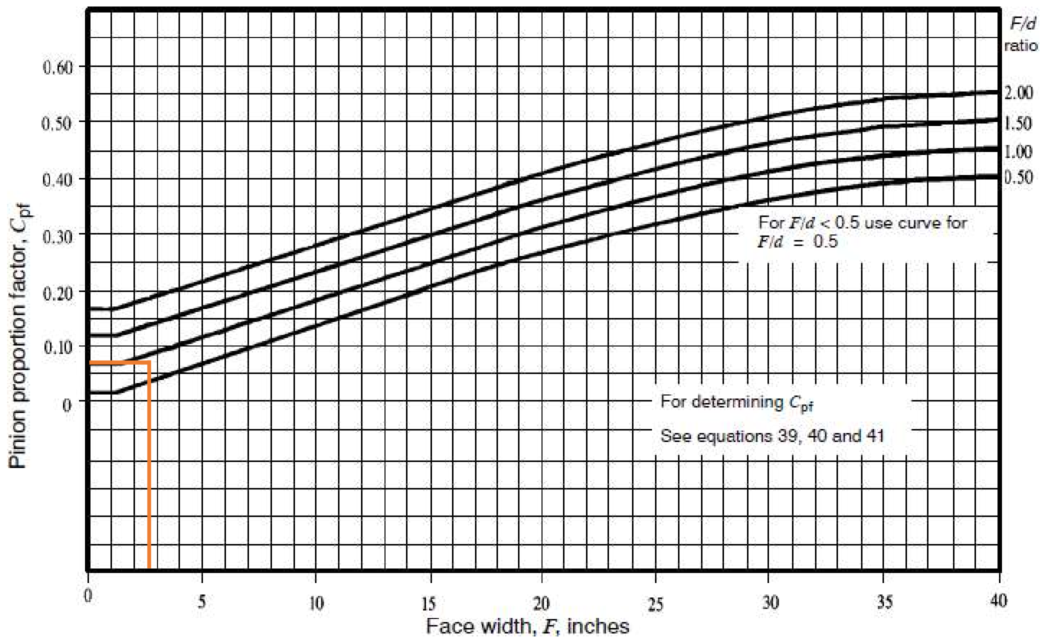
$$K_h = 1 + C_{mc}(C_{pt}C_{pm} + C_{ma}C_e)$$

For this Formula,

$C_e = 1$ is given

Gear has uncrowned teeth, thus **$C_{mc} = 1$**

For **C_{pt}**, using reference diagram

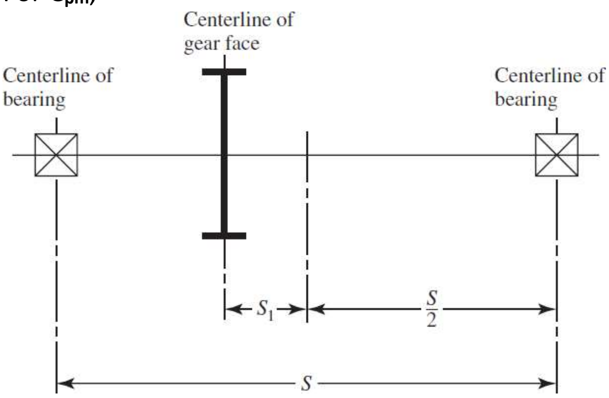


Face width =70mm =2.76 inches =F

∴F/d= 0.8

We get **C_{pt}=0.07**

For **C_{pm},**



$$C_{pm} = \begin{cases} 1 & S_1/S < 0{,}175 \\ 1{,}1 & S_1/S \geq 0{,}175 \text{ or cantilever shaft} \end{cases}$$

S1=25.5mm

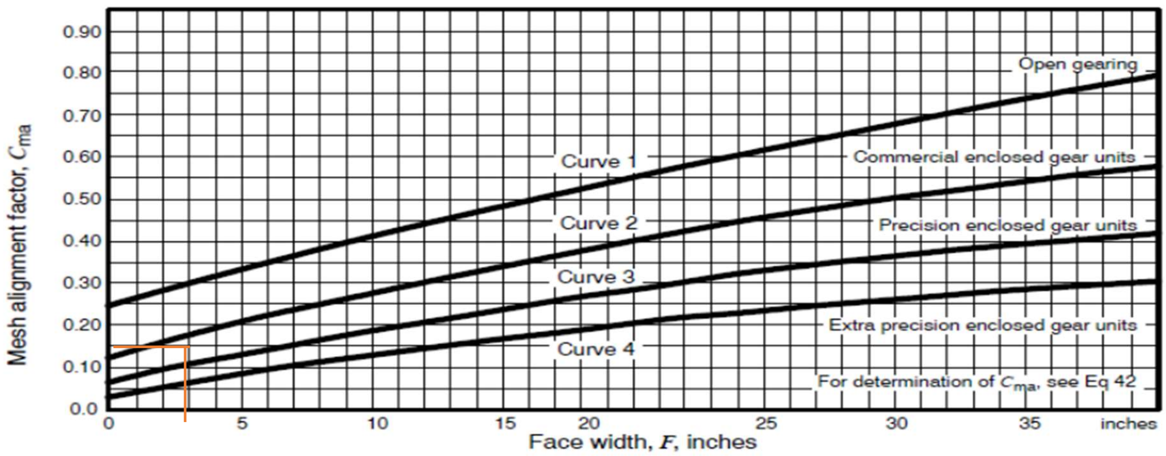
S=156.8mm

Then, S_1/S= 0.162

0.162<0.175

C_{pm}=1

For C_{ma}, using reference diagram



For commercial enclosed gear,

Using curve 2, with F=2.76 inches
We get, **C_{ma}=0.17**
Using **$K_h=1+C_{mc}\left((C_{pt}C_{pm} + C_{ma}C_c\right)$**

K_h=1.24

- **Size Factor K_s**

K_s **= 0.843(b. M_n $\sqrt{y}$)^20.0535**

Where **b= 70mm**

m_n=5

To find y, using table from y

Size factor K_s

Number of Teeth	Y	Number of Teeth	Y
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

Where for helical gears $z' = z/Cos^3\Psi$ [Z is 15 and Ψ = 25˚]

∴ Z'=20.15 **taking** **z=21**

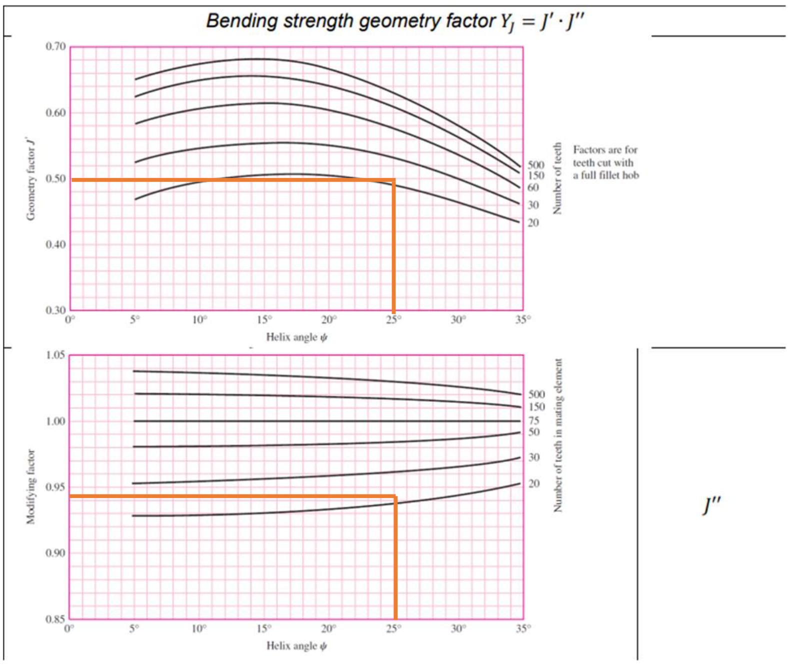
Thus, using z1 teeth to find y, we get **y=0.328**

K_s =1.12

- **Bending strength geometry factor**

Y_j=J'. J'

Using some 21 teeth and 25 helix angle on reference diagram of geometry factor J'



J''

J'=0.5

And with modifying factor ref diagram

J''=0.94

Y_J=J'J''=0.47

Finally,

Maximum tooth gear bending stress we find by,

$$\sigma_{max}, \text{ fatigue} = F_t K_D K_B K_D K_H K_S \frac{1}{B \cdot M_{ty_5}}$$

F_t=6249 N,
m_t= $m_n/\cos\phi$=**5.52 mm** (transverse module)

$$\sigma_{max}, \text{ fatigue} = 55.43 \text{ MPa}$$

- **Bending safety factor**

Now to find bending safety factor for gear teeth

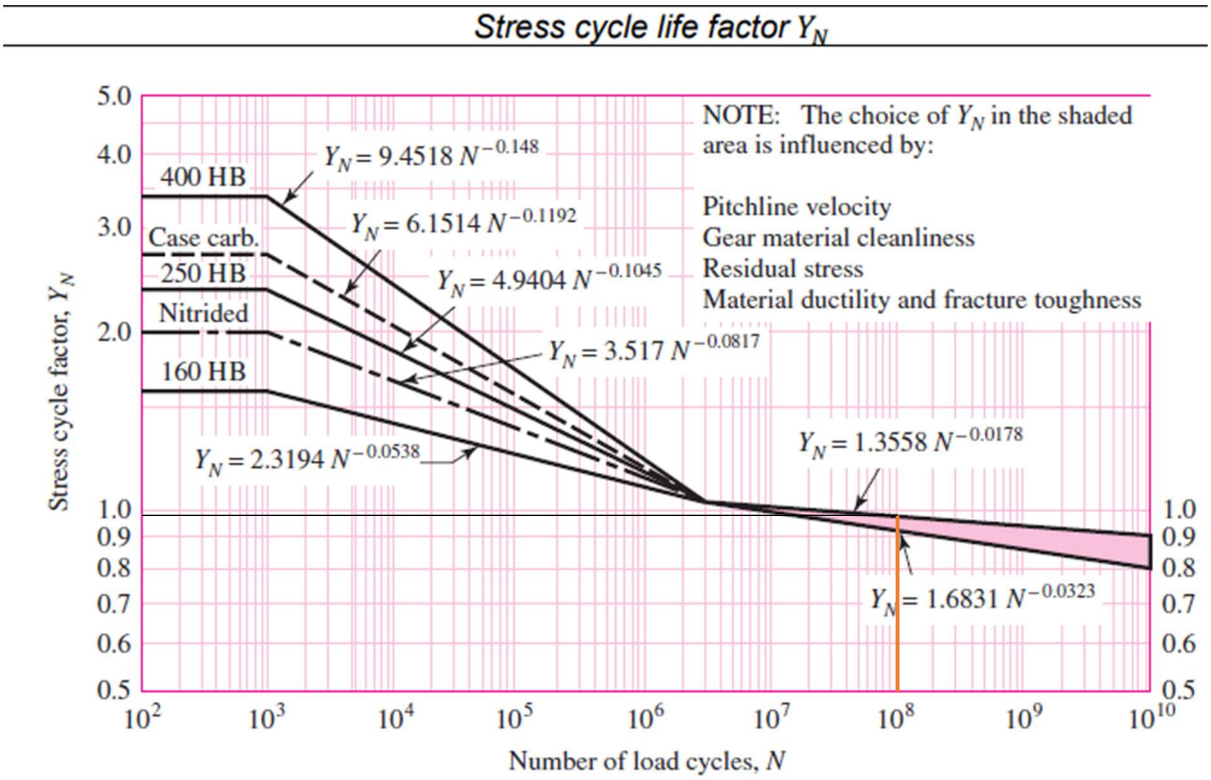
We have,

$$S_f = \frac{\sigma_{FP}}{\sigma_{max,fatigue}} * \frac{Y_N}{Y_\theta} \frac{1}{y_z}$$

σ_{fp}=860 Mpa (given)

$\sigma_{max,\ fatigue}$ we calculated to **55.43 MPa**

For stress life cycle factor Y_N,



We must consider 10^8 loading cycles,

Thus, **$Y_N = 1.3558\ N^{-0.0178} = 0.98$**

Our working condition is **less than 50* C**,

For temperature less than 120*C temperature factor **$Y_\theta = 1$** is given.

Reliability of 0.99 is given, which gives according to table,

Temperature factor Y_θ

$Y_\theta = 1$, for temperature up to 120 °C

Reliability factor Y_Z

Reliability	K_R (Y_Z)
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

Reliability Factor, **$Y_z = 1.00$**

with all the values obtained,

$$S_F = \frac{\sigma_{FP}}{\sigma_{max,figure}} * \frac{Y_N}{Y_{\theta} * Y_Z} = 15.19$$

Safety Factor of 15.19 is quite high for the design of the gear teeth for fatigue loadings, we can conclude that the gear teeth are well designed for given working condition of the gearbox.

Maximum gear contact stress

$$\sigma_{max,pitting} = Z_E \sqrt{F_t K_0 K_v K_s \cdot \frac{K_H}{b \cdot d_P} \cdot \frac{Z_R}{Z_I}}$$

- Where, $\mathbf{F_t=6249\ N}$

b=70mm

$\mathbf{K_o=1}$ $\mathbf{d_p=82.76\ mm}$

$\mathbf{K_v=1.16}$

$\mathbf{K_s=1.12}$

$\mathbf{K_h=1.24}$

we have from previous findings

$\mathbf{Z_R=1}$ **is given**

For Elastic coefficient Z_E, using reference table Z_E

Table 14–8

Elastic Coefficient C_p (Z_E), $\sqrt{\text{psi}}$ ($\sqrt{\text{MPa}}$) *Source:* AGMA 218.01

		Gear Material and Modulus of Elasticity E_G, lbf/in^2 (MPa)*					
Pinion Material	**Pinion Modulus of Elasticity E_p psi (MPa)***	**Steel 30×10^6 (2×10^5)**	**Malleable Iron 25×10^6 (1.7×10^5)**	**Nodular Iron 24×10^6 (1.7×10^5)**	**Cast Iron 22×10^6 (1.5×10^5)**	**Aluminum Bronze 17.5×10^6 (1.2×10^5)**	**Tin Bronze 16×10^6 (1.1×10^5)**
Steel	30×10^6 (2×10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162)	1900 (158)
Malleable iron	25×10^6 (1.7×10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158)	1850 (154)
Nodular iron	24×10^6 (1.7×10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156)	1830 (152)
Cast iron	22×10^6 (1.5×10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154)	1800 (149)
Aluminum bronze	17.5×10^6 (1.2×10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145)	1700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141)	1650 (137)

We find elastic modulus of steel as $2\text{x}10^5$ Mpa and Z_E as 191

$\mathbf{Z_E=191}$

To Find surface strength geometry Z_I

$R_{bp} = r_p\cos\phi_t = 13.92$ mm

$R_{bg} = r_g\cos\phi_t = 41.76$ mm

$$Z_A = \min\left[\sqrt{(r_P + a)^2 - r_{bP}^2}, (r_P + r_G)\mathrm{sen}[\phi_t]\right]$$ = **44.29 mm**

$$Z_B = \min\left[\sqrt{(r_G + a)^2 - r_{bG}^2}, (r_P + r_G)\mathrm{sen}[\phi_t]\right]$$ = **122.19 mm**

$$\bar{Z} = Z_A + Z_B - (r_P + r_G)\mathrm{sen}[\phi_t]$$ = **104.70 mm**

$$m_N = \frac{p_n \cos[\phi_n]}{0{,}95 \cdot Z}$$

Gear 3 and gear 4 are meshing externally, $\mathbf{M_G=3}$

Thus , **Z_I=** $\dfrac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N} \cdot \dfrac{m_G}{m_G + 1}$ **= 0.026**

After obtaining all the required factors,

With,

$$\sigma_{\max,pitting} = Z_E\sqrt{F_t K_0 K_v K_s \cdot \frac{K_H}{b \cdot d_P} \cdot \frac{Z_R}{Z_I}}$$

We get,

$\sigma_{\text{max, pitting}}$ = 269.20 MPa

that is maximum contact stress for the given gear 3 teeth

Wear safety factor

To check the gear design against contact wear, safety factor is calculated as,

$$S_H = \frac{\sigma_{HP}}{\sigma_{\max,pitting}} \frac{Z_N Z_W}{Y_\theta Y_Z}$$

Y_Θ =1, Y_z=1 we already obtained

$\sigma_{max, pitting}$=269.20 MPa we calculated above

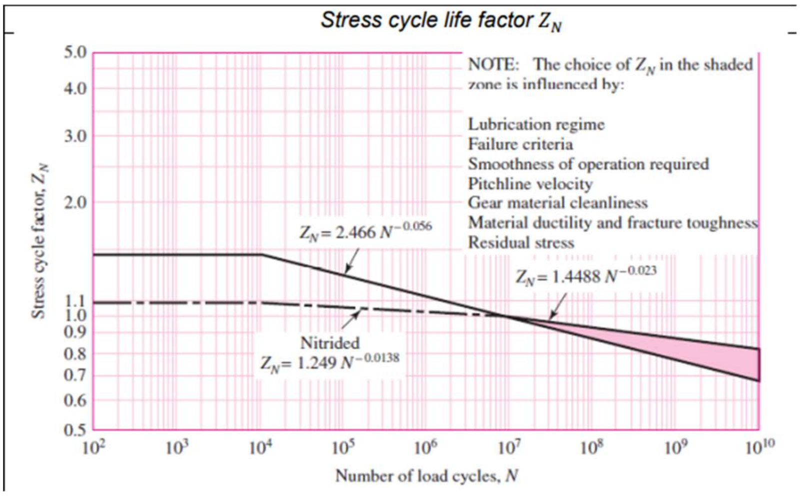
Now for σ_{HP} (contact strength), we use table for σ_{HP} with tempered steel data , of grade 2 steel ,

We obtained 190000 psi which is [1 MPa =145 psi]

σ_{HP} = 1310.34 MPa

Again for 10^8 cycles, H= 10^8 , for stress cycle life factor Z_N

Z_N from reference diagram



$Z_N = 1.4488\,N^{-0.023}$=0.95

For hardness ratio factor Z_N,

Assuming both gears ore made from same material

Tus **HB_P/HB_G =1 <1.2**

A'=0

Now, **Z_w=1+A'(m_G -1)** is given

With **m_G=3**

Z_w =1

As all the relevant data is obtained, we calculate safety factor

$$S_H = \frac{\sigma_{HP}}{\sigma_{\max,pitting}} \frac{Z_N Z_W}{Y_\theta Y_Z} = \mathbf{4.62}$$

Gear's wear safety factor is less than its fatigue safety factor, the gear teeth are more critical to wear.

But still the safety factor is reasonably high, taking minimum limit as 3.

PART : 2

Bearing Life Investigation

- Shaft is rotating on 2 bearings:

Bearing C : Deep groove ball bearing 6308

Bearing D : Deep groove ball bearing 6308

Forces on Bearing C :

$\uparrow$: $\sqrt{R_{XC}{}^2 + R_{YC}{}^2}$ = **2.01 kN**

$\rightarrow$: R_{zc}= **1.67 kN**

Forces on Bearing B :

$\uparrow$: $\sqrt{R_{XC}{}^2 + R_{YC}{}^2}$ = **3.1 kN**

	Bearing C	Bearing D
Radial Force (Fr)	**2.01 kN**	**3.1 kN**
Axial Force (Fa)	**1.67 kN**	

Static Analysis

- **.Minimum load Fr ≥Fr_{min}**

  to check if radial load Fr is greater than minimum load requirements Fr_{min} for the bearing,

For both bearings $\mathbf{Fr_{min} = K_r \cdot \left(\frac{v_n}{1000}\right)^{\frac{2}{3}} + \left(\frac{d_n}{100}\right)^2}$

where, K_r from SKF data is 0.03 for 6308

$$\mathbf{K_r = 0.03}$$

From SKF data inner diameter **d = 40mm**, and outer diameter **D = 90mm**

$$\mathbf{d_m = \frac{d+D}{2} = 65mm}$$

d_m(Bearing mean diameter) and shaft rotation speed is **n=295.5rpm**

For oil lubricant, actual operating viscocity is 65.4mm^2/s

$$\mathbf{\nu = 65.4 mm^2/s}$$

Fr_{min}= **0.09KN** for both bearings

Thus, as Fr_{min} is smaller than Fr of either Bearings(2.01 kN on bearing C and 3.1 kN on bearing D), implies that both Bearing are operating over Fr_{min} threshold, which is desired by the operation

---

Axial Load capacity

Bearing D operated under no axial load, thus, no need to verify its axial loading capacity

As for bearing C, $F_a = 1.67KN$

Condition is to have $\mathbf{F_a < 0.5C_o}$

From catalogue, $\mathbf{C_o = 24KN}$ for Bearing 6308

$\mathbf{0.5C_o = 12KN}$

$F_a < 0.5C_0$ Condition for Bearing C is met design is verified

Both the bearing meets the design requirement for axial load capacity for given working conditions

---

Equivalent static bearing load

For deep groove ball bearing , equivalent static bearing load P_0 is obtained as,

$$P_o = \begin{cases} 0.6Fr + 0.5F_a \\ Fr & \text{if } P_o < Fr \end{cases}$$

for bearing C, $\mathbf{P_o = 2.04KN}$

For bearing D, $\mathbf{P_o = 3.10KN}$

Static safety for bearing with formula . $S_0 = \frac{C_o}{P_o}$

S_0 for bearing C : 11.7

S_0 for bearing D : 7.7

Bearing life cycle analysis(millions of cycle)

$$L_{10M} = a.a_{skf}.\left(\frac{C}{P}\right)^{p}$$

where, C is basic dynamic load rating given in SKF catalogue for 6308

C = 42.3 kN

P for ball bearing is 3, **p = 3**

For evaluation of eq. dynamic load acting on bearing

P_o = {. Fr if $F_a/Fr \leq e$

. $xFr + yF_a$ if $F_a/Fr > e$

no F_a operating on bearing D, thus dynamic load on bearing D is **P= Fr= 3.1KN**

As for bearing C,

Calculation factors for deep groove ball bearings

	Single row and double row bearings Normal clearance		
f_0F_a/C_0	e	X	Y
0,172	0,19	0,56	2,3
0,345	0,22	0,56	1,99
0,689	0,26	0,56	1,71
1,03	0,28	0,56	1,55
1,38	0,3	0,56	1,45
2,07	0,34	0,56	1,31
3,45	0,38	0,56	1,15
5,17	0,42	0,56	1,04
6,89	0,44	0,56	1

f_o from SKF catalogue is **13**

$f_o.\frac{F_a}{C_o}$ **= 0.91** which gives,

e≈0.27

X=0.56

Y=1.63

for bearing C, **P_a/Fr = 0.83 >0.27**

so, **F_a/F_b > e**

thus for bearing C, **P = XFr + YFa = 3.85KN**

With given 90% reliability, life adjustment factor $\mathbf{a_1=1}$

from the given table

Reliability	Failure probability n	SKF rating life L_{nm}	Life adjust-ment factor a_1
%	%	-	-
90	10	L_{10m}	1
95	5	L_{5m}	0,62
96	4	L_{4m}	0,53
97	3	L_{3m}	0,44
98	2	L_{2m}	0,33
99	1	L_{1m}	0,21

Evaluation of the coefficient a_{skf}

To evaluate the value of coefficient a_{skf} ,

a_{skf} is a function of viscosity ratio $\mathbf{K = \nu/\nu_1}$, where ν is lubricant viscosity

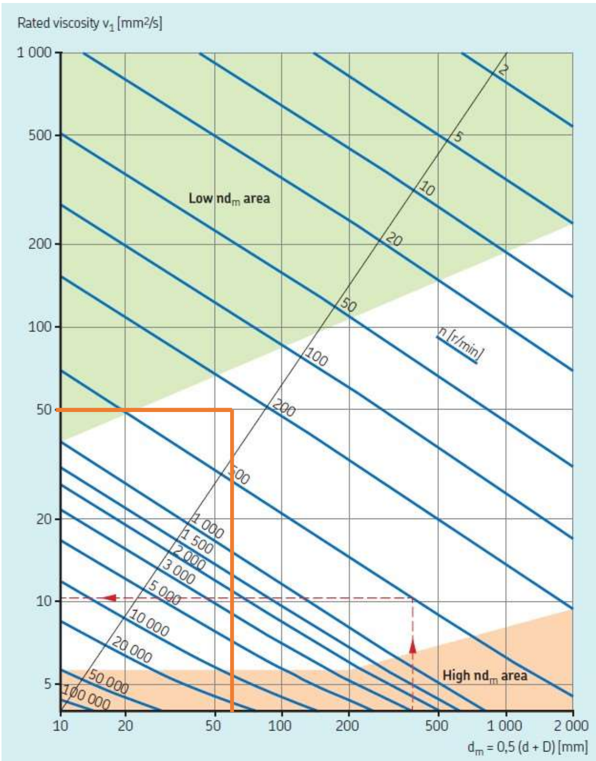
and ν_1 is minimum required viscosity

For the given operating conditions,

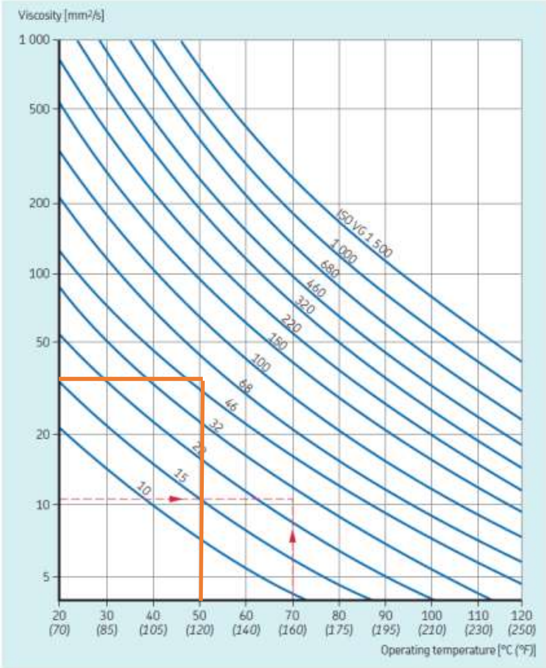
ν_1 is found with dm and n (shaft rotational speed) using given reference diagram

$\mathbf{d_m = \frac{d+D}{2} = 65mm}$

$\mathbf{\nu_1 = 50mm^2/s}$



The value of v is evaluated as function of v_1 and operating temperature which is 50°c



***v* = 35mm²/s**

thus, **K =v/v_1 = 35/50 =0.7**

- **Evaluation of the contamination**

Using the following table

Guideline values for factor η_c for different level of contamination

Conditions	Factor η_c [1)] for bearings with diameter d_m < 100	$d_m \geq 100$ mm
Extreme cleanliness • Particle size of the order of the lubricant film thickness • Laboratory conditions	1	1
High cleanliness • Oil filtered through an extremely fine filter • Typical conditions: sealed bearings that are greased for life	0,8 ... 0,6	0,9 ... 0,8
Normal cleanliness • Oil filtered through a fine filter • Typical conditions: shielded bearings that are greased for life	0,6 ... 0,5	0,8 ... 0,6
Slight contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles and slight ingress of contaminants	0,5 ... 0,3	0,6 ... 0,4
Typical contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles, and ingress from surroundings	0,3 ... 0,1	0,4 ... 0,2
Severe contamination • Typical conditions: high levels of contamination due to excessive wear and/or ineffective seals • Bearing arrangement with ineffective or damaged seals	0,1 ... 0	0,1 ... 0
Very severe contamination • Typical conditions: contamination levels so severe that values of ηc are outside the scale, which significantly reduces the bearing life	0	0

and **dm = 65mm <100mm**

n_c =0.5 value taken as **normal contamination**

Fatigue load limit P_u from SKF catalogue is obtained

P_u=1.02KN

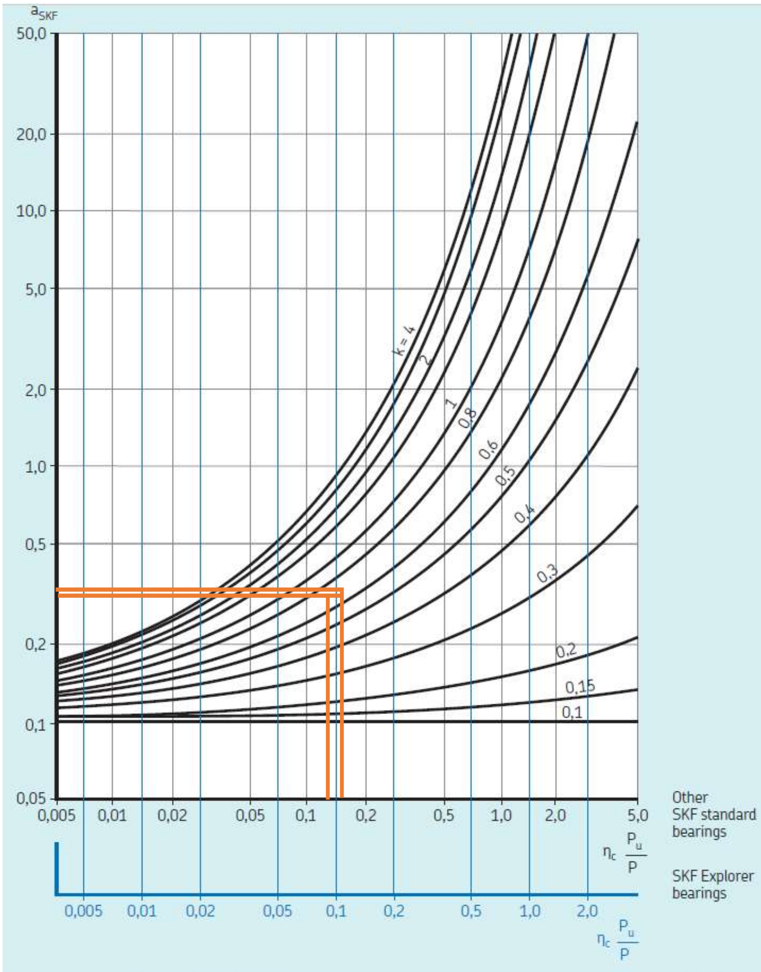
. for bearing C : **$n_c \cdot P_u/P$ =0.13**

. for bearing D : **$n_c \cdot P_u/P$ =0.16**

With diagram ,

For standard bearings,

And value of K and $n_c \cdot P_u/P$ obtained above,



a_{skf} of bearing c = 0.35

a_{skf} of bearing D = 0.35

Bearings life cycle (in millions of cycles)

with all the values obtained above and the formula for bearing life cycles in millions ,

$$L_{10,M} = a.\ a_{skf}\ (\frac{C}{P})^{p}$$

We get both bearings' lives as,

	$L_{10,M}$
Bearing C	**462.76 million cycles**
Bearing D	**888.52 million cycles**

Bearing will need to be replaced often with respect to their respective life cycles in million

---

Bearing life in operating hours

with formula , $$L_{10Mh} = \frac{10^6}{60.n} L_{10,M}$$

where n is rotating speed of bearing as **n= 295.45 rpm**

We get,

	L_{10Mh}
Bearing C	**26104.5 hours**
Bearing D	**50121.4 hours**

Under the given conditions each bearing will sustain only until their respective working hours, for further working of gearbox, the bearing will need to be changed

---

Concluding Remarks

With calculations of multiple verifications done on shaft A2, gear 3 and bearings C and D on shaft A2, findings show that the design shows optimal results for the given working conditions. Thus, we can confidently pass the given shaft design for gear box workings.