ME 308
MACHINE ELEMENTS II

CHAPTER 3

ROLLING CONTACT BEARINGS
PART_2

BİLYALI YATAKLAR

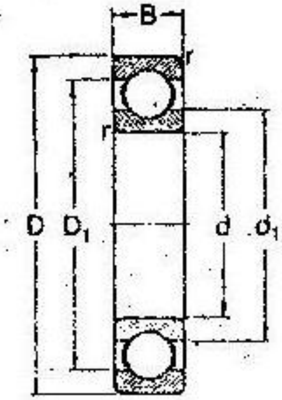
Yatak tipi	$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
Tek sıralı bilyalı yataklar 160, 60 62, 63, 64 serileri					
$\frac{F_a}{C_o}$ = 0,025				2	0,22
= 0,04				1,8	0,24
= 0,07	1	0	0,56	1,6	0,27
= 0,13				1,4	0,31
= 0,25				1,2	0,37
= 0,5				1	0,44
Bilyalı oynak yataklar 135, 126, 127, 108, 129	1	1,8	0,65	2,8	0,34
1200 — 1203		2		3,1	0,31
04 — 05		2,3		3,6	0,27
06 — 07		2,7		4,2	0,23
08 — 09	1	2,9	0,65	4,5	0,21
10 — 12		3,4		5,2	0,19
13 — 22		3,6		5,6	0,17
24 — 30		3,3		5	0,2
2200 — 2204		1,3		2	0,5
05 — 07		1,7		2,6	0,37
08 — 09		2		3,1	0,31
10 — 13	1	2,3	0,65	3,5	0,28
14 — 20		2,4		3,8	0,26
21 — 22		2,3		3,5	0,28
1300 — 1303		1,8		2,8	0,34
04 — 05		2,2		3,4	0,29
06 — 09	1	2,5	0,65	3,9	0,25
10 — 22		2,8		4,3	0,23
— 2301		1		1,6	0,63
2302 — 2304	1	1,2	0,65	1,9	0,52
05 — 10		1,5		2,3	0,43
11 — 18		1,6		2,5	0,39

MAKARALI YATAKLAR

Yatak tipi	$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
Makaralı oynak yataklar					
23944 — 239/670	1	3,7	0,67	5,5	0,18
239/710 — 239/950		4		6	0,17
23024 C — 23068 CA	1	2,9	0,67	4,4	0,23
23072 CA — 230/500 CA		3,3		4,9	0,21
24024C — 24080 CA	1	2,3	0,67	3,5	0,29
24084CA — 240/500CA		2,4		3,6	0,28
23120C — 23128C	1	2,4	0,67	3,6	0,28
23130C — 231/500 CA		2,3		3,5	0,29
24122 C — 24128C		1,9		2,9	0,35
24130 C — 24172 CA	1	1,8	0,67	2,7	0,37
24176 CA — 241/500 CA		1,9		2,9	0,35
22205 C — 22207 C		2,1		3,1	0,32
08 C — 09 C		2,5		3,7	0,27
10 C — 20 C	1	2,9	0,67	4,4	0,23
22 C — 44 C		2,6		3,9	0,26
48 — 64		2,4		3,6	0,28
23218 C — 23220 C	1	2,2	0,67	3,3	0,31
22 C — 64 CA		2		3	0,34
21304 — 21305		2,8		4,2	0,24
06 — 10		3,2		4,8	0,21
11 — 19	1	3,4	0,67	5	0,2
20 — 22		3,7		5,5	0,18
22308 C — 22310 C		1,8		2,7	0,37
11 C — 15 C		1,9		2,9	0,35
16 C — 40 C	1	2	0,67	3	0,34
44 — 56		1,9		2,9	0,35
Konik makaralı yataklar					
30203 — 30204				1,75	0,34
05 — 08				1,6	0,37
09 — 22	1	0	0,4	1,45	0,41

SKF

Deep groove ball bearings
d 17–40 mm



Equivalent bearing load
dynamic: $P = XF_r + YF_a$
static: $P_0 = 0{,}6\,F_r + 0{,}5\,F_a$
When $P_0 < F_r$ use $P_0 = F_r$

Boundary dimensions d	D	B	Basic load ratings dynamic C	static C_0	Limiting speeds Lubrication grease	oil	Mass	Designations
mm			N (1 N = 0,225 lbf)		r/min		kg	—
17	26	5	1 320	915	24 000	30 000	0,0082	61803
	35	8	4 650	2 600	19 000	24 000	0,032	16003
	35	10	4 650	2 600	19 000	24 000	0,039	6003
	40	12	7 350	4 500	17 000	20 000	0,065	6203
	47	14	10 400	6 550	16 000	19 000	0,12	6303
	62	17	17 600	11 800	12 000	15 000	0,27	6403
20	32	7	2 040	1 400	19 000	24 000	0,018	61804
	42	8	5 400	3 400	16 000	22 000	0,050	16004
	42	12	7 200	4 500	17 000	20 000	0,069	6004
	47	14	9 800	6 200	15 000	18 000	0,11	6204
	52	15	12 200	7 800	13 000	16 000	0,14	6304
	72	19	23 600	16 600	10 000	13 000	0,40	6404
25	37	7	2 280	1 700	17 000	20 000	0,022	61805
	47	8	5 850	4 000	14 000	17 000	0,060	16005
	47	12	8 650	5 600	15 000	18 000	0,080	6005
	52	15	10 800	6 950	12 000	15 000	0,13	6205
	62	17	17 300	11 400	11 000	14 000	0,23	6305
	80	21	27 500	19 600	9 000	11 000	0,53	6405

Calculation factors
dynamic

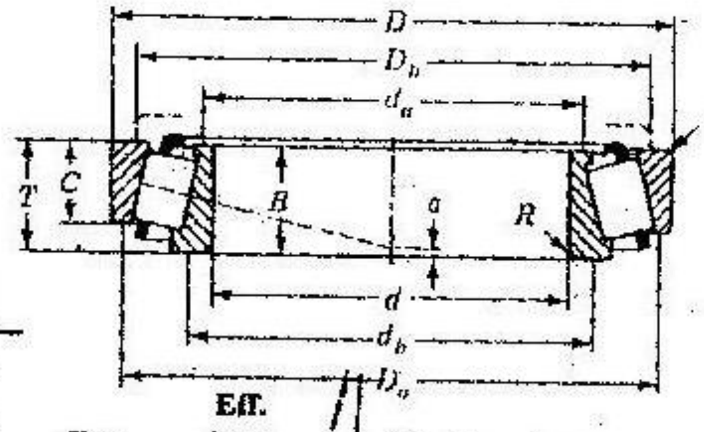
F_a/C_0	e	$F_a/F_r \leq e$ X	Y	$F_a/F_r > e$ X	Y
—					
0,025	0,22	1	0	0,56	2
0,04	0,24	1	0	0,56	1,8
0,07	0,27	1	0	0,56	1,6
0,13	0,31	1	0	0,56	1,4
0,25	0,37	1	0	0,56	1,2
0,5	0,44	1	0	0,56	1

a

Angular Contact Ball Bearings

Equivalent bearing load
dynamic: $P = XF_r + YF_a$
static: $P_0 = 0{,}5\,F_r + 0{,}26\,F_a$
When $P_0 < F_r$ use $P_0 = F_r$

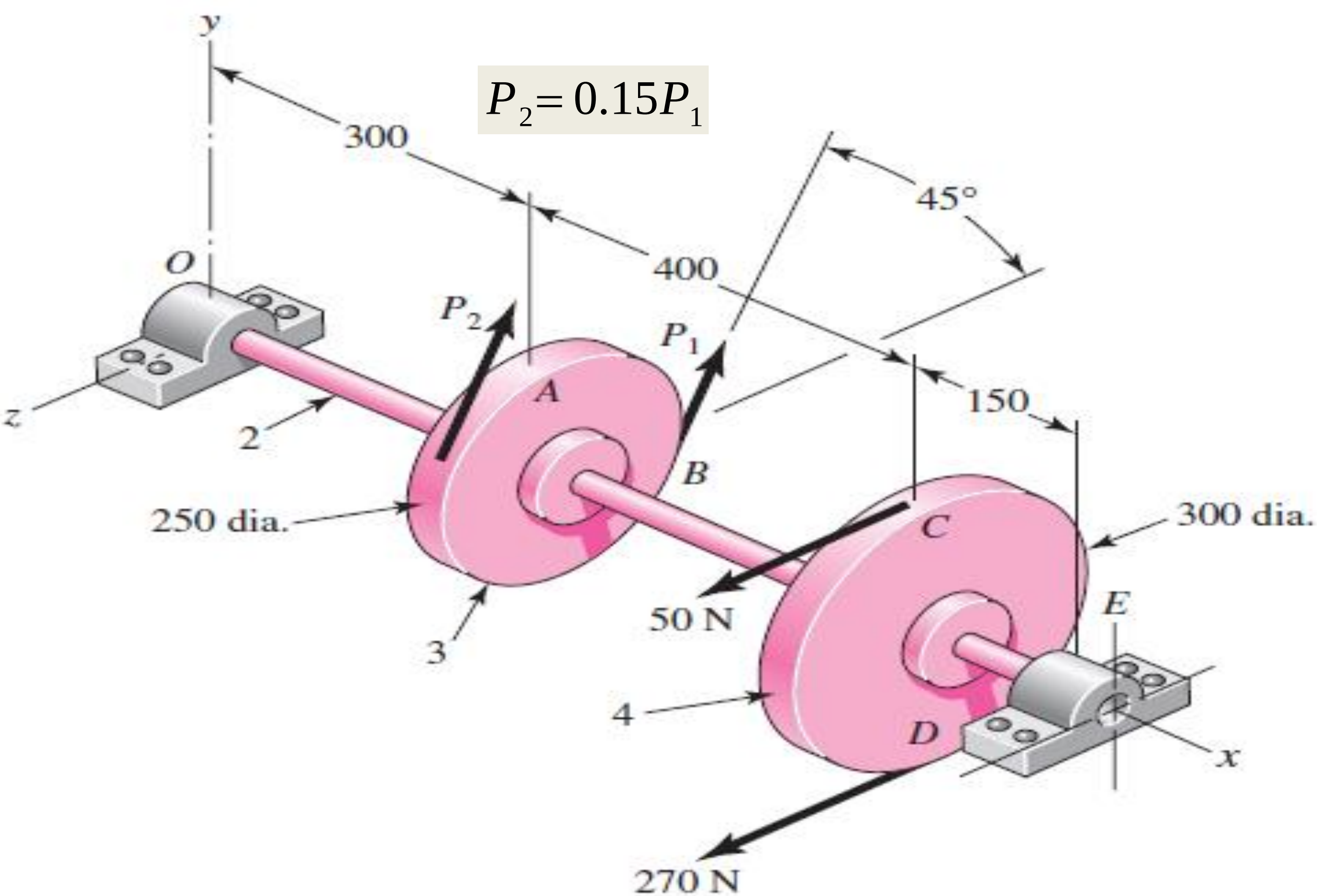
	Boundary dimensions			Basic load ratings		Limiting speeds Lubrication		Mass	Designation	Calculation factors dynamic				
				dynamic	static	grease	oil				$F_a/F_r \leq e$		$F_a/F_r > e$	
	d	D	B	C	C_0					e	X	Y	X	Y
	mm			N (1 N = 0,225 lbf)		r/min		kg	—	—				
10	10	30	9	3 800	2 120	19 000	28 000	0,031	7200 B	1,14	1	0	0,35	0,57
12	12	32	10	5 400	3 050	17 000	24 000	0,045	7201 B					
15	15	35	11	6 200	3 650	16 000	22 000	0,046	7202 B					
		42	13	9 000	5 300	14 000	19 000	0,090	7302 B					
17	17	40	12	7 650	4 650	14 000	19 000	0,070	7203 B					
		47	14	11 400	7 100	12 000	17 000	0,12	7303 B					
20	20	47	14	10 200	6 400	11 000	16 000	0,11	7204 B					
		52	15	13 400	8 150	10 000	15 000	0,15	7304 B					
	25	52	15	11 400	7 650	9 500	14 000	0,14	7205 B					
		62	17	19 000	12 200	8 500	12 000	0,24	7305 B					
	30	62	16	15 600	11 000	8 500	12 000	0,21	7206 B					
		72	19	24 000	16 600	7 500	10 000	0,36	7306 B					
	35	72	17	20 800	15 000	7 500	10 000	0,30	7207 B					
		80	21	26 000	20 000	7 000	9 500	0,48	7307 B					



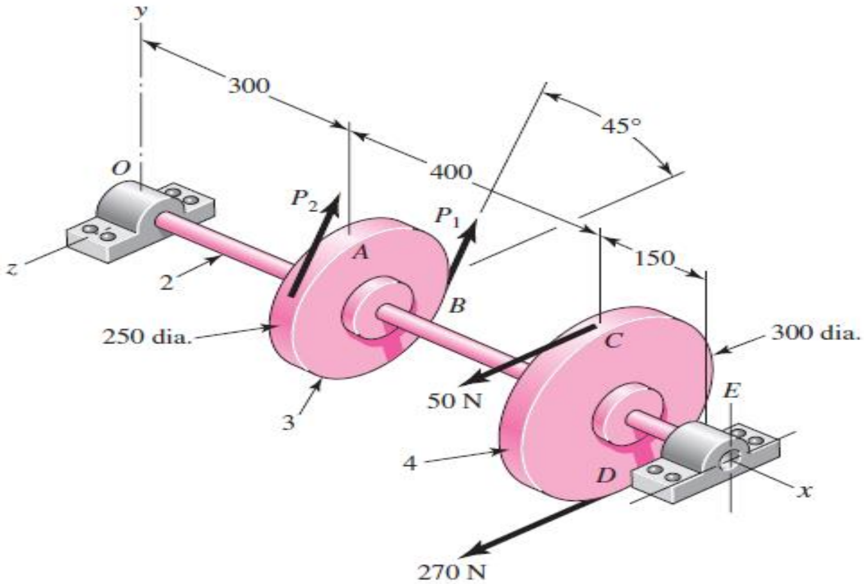
Bore d	Outside diameter D	Width T	Rating at 500 rpm for 3000 hours, L10: One row radial lb daN	Rating at 500 rpm for 3000 hours, L10: Thrust lb daN	Factor K	Eff. load center a†	Part numbers: Cone	Part numbers: Cup
0.3750 inch	1.2595	0.3940	435	301	1.44	−0.12	A2037	A2126
9.525 mm	31,991	10,008	192	134		−3,0		
‡0.4724	1.2595	0.3940	435	301	1.44	−0.12	A2047	A2126
‡12,000	31,991	10,008	192	134		−3,0		
0.4992	1.3775	0.4330	500	387	1.29	−0.10	A4049	A4138
12,680	34,988	10,998	222	172		−2,5		
0.5000	1.3775	0.4330	500	387	1.29	−0.10	A4050	A4138
12,700	34,988	10,998	222	172		−2,5		
0.5000	1.5000	0.5313	760	358	2.12	−0.20	00050	00150
12,700	38,100	13,495	338	159		−5,1		
‡0.5906	1.3775	0.4330	500	387	1.29	−0.10	A4059	A4138
‡15,000	34,988	10,998	222	172		−2,5		
0.6250	1.3775	0.4330	575	314	1.83	−0.13	L21549	L21511
15,875	34,988	10,998	256	140		−3,3		
0.6250	1.5745	0.4730	530	480	[illegible]	[illegible]	[illegible]	[illegible]

Example 3.2 (11.3)

The figure is a schematic drawing of a countershaft that supports two V-belt pulleys. The countershaft runs at shaft speed of 1100 rpm and the bearings are to have a life of 12 khrs, R= 99 % reliability using an application factor of unity. The belt tension on the loose side of pulley A is 15 % of the tension on the tight side. An analysis of this problem gave shaft forces at B and D of $\mathbf{F_B}$ = *-253**j**-253**kN*** and $\mathbf{F_D}$ = *-320**kN,*** with the belt pulls assumed to be parallel.



Based on bending deflection, a shaft diameter of 35 mm has been selected at both bearing points O and E. Select a suitable ball bearing to be used at both bearings are to be of the same size.



Use torque balance of the shaft to determine the value of P_1 and P_2 and reaction force on bearings

$$\sum M_{O_x} = 0;$$

$$0.85 \times P \times \frac{250}{2} = 220 \times \frac{300}{2}$$

$$P = 310.59\ N$$

Reaction force on Pulley A -B

$$R_B = 0.15P + P = 1.15P$$

$$R_B = 1.15 \times P = 1.15 \times 310.59 = 357.18\ N$$

$$F_{B_z} = \cos 45 \times 357.18 = -252.56\ N$$

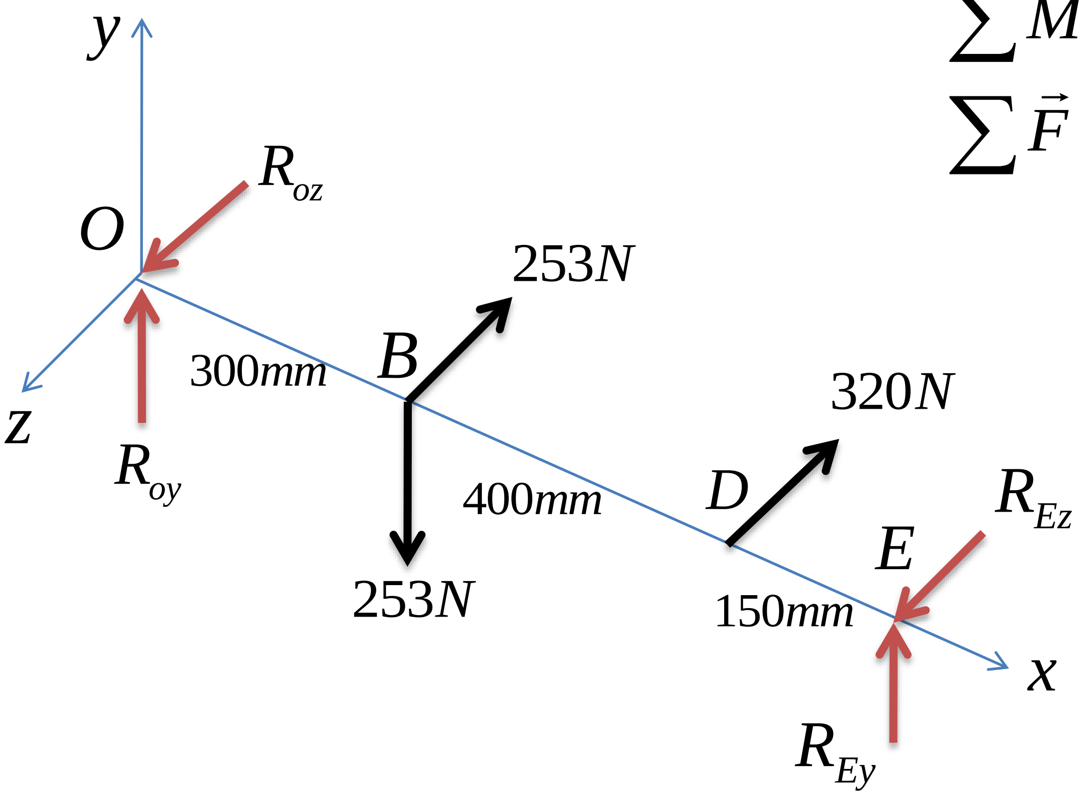
$$F_{B_y} = \sin 45 \times 357.18 = -252.56\ N$$

$$\vec{F}_B = -253\vec{j} - 253\vec{k}\ N$$

Reaction force on Pulley C -D

$$F_{D_z} = 270 + 50 = 320\ N$$

$$\vec{F}_D = -320\vec{k}\ N$$



$$\sum \vec{M} = 0 \,;$$

$$\sum \vec{F} = 0 \,;$$

<u>In x-y plane</u>

$$\sum M_{Oz} = 0;$$

$$253 \times 300 = R_{Ey} \times 850$$

$$R_{Ey} = 89.30 \; N$$

$$\sum \vec{F}_y = 0 \,;$$

$$R_{Oy} = 253 - R_{Ey}$$

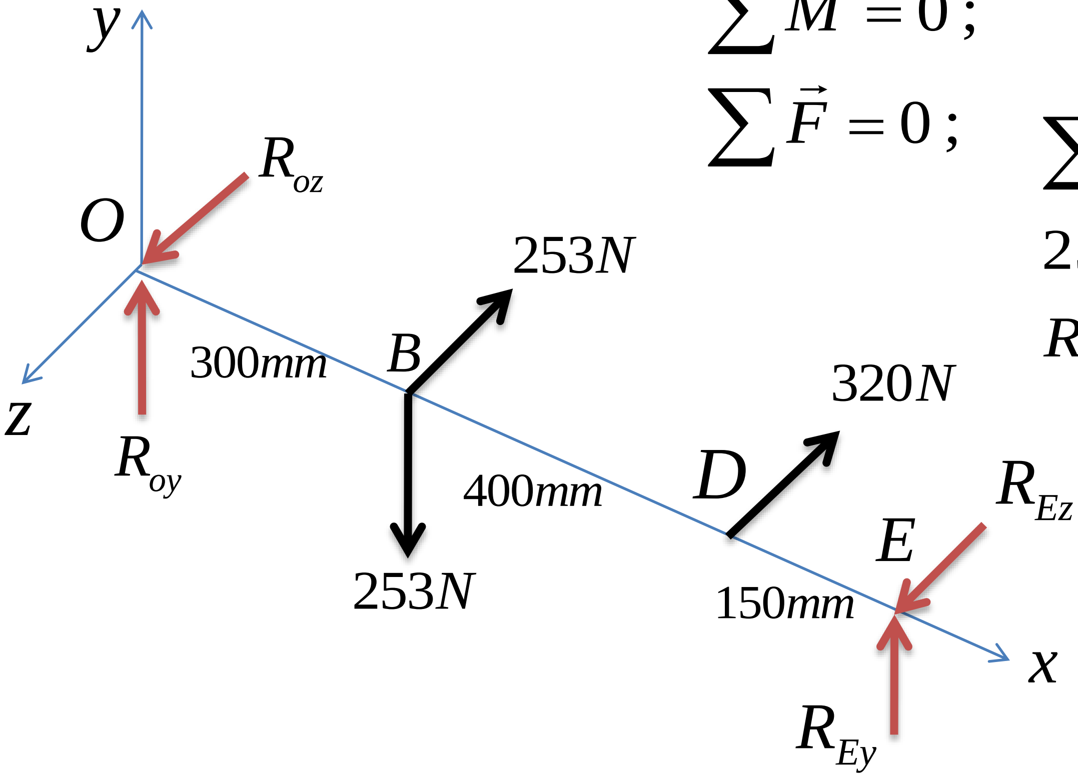
$$R_{Oy} = 253 - 89.30 = 163.70 \; N$$

$$R_{Ey} = 89.30 \; N$$

$$R_{Oy} = 163.70 \; N$$

$$\sum \vec{M} = 0 \, ;$$

$$\sum \vec{F} = 0 \, ;$$



In x-z plane

$$\sum M_{O_y} = 0;$$

$$253 \times 300 + 320 \times 700 = R_{Ez} \times 850$$

$$R_{Ez} = 352.82 \ N$$

$$\sum \vec{F}_z = 0 \, ;$$

$$R_{Oz} + R_{Ez} = 253 + 320$$

$$R_{Oz} = 573.0 - 352.82 = 220.18 \ N$$

$$R_{Ez} = 352.82 \ N$$

$$R_{Oz} = 220.18 \ N$$

$R_{Ey} = 89.30\ N$

$R_{Oy} = 163.70\ N$

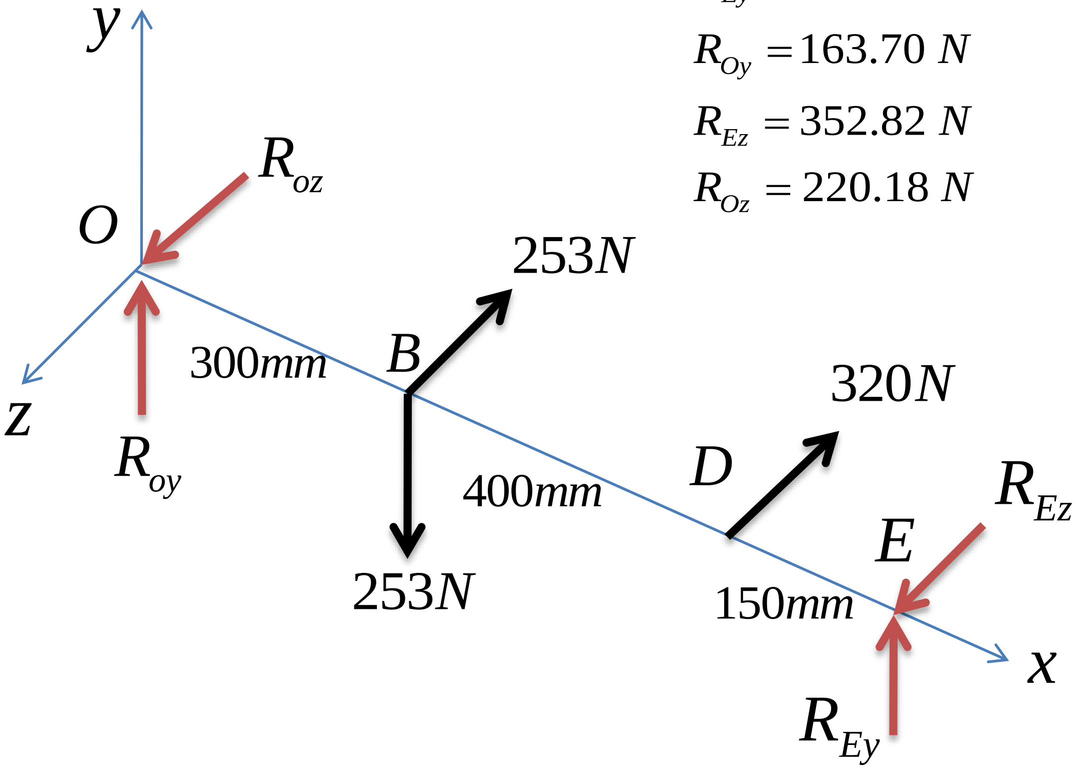
$R_{Ez} = 352.82\ N$

$R_{Oz} = 220.18\ N$

$$R_O = \sqrt{(R_{Oy})^2 + (R_{Oz})^2}$$

$$R_O = \sqrt{(163.70)^2 + (220.18)^2}$$

$$R_O = 274.36\ N$$



$$R_E = \sqrt{(R_{Ey})^2 + (R_{Ez})^2}$$

$$R_E = \sqrt{(89.30)^2 + (352.82)^2}$$

$$R_E = 363.94\ N$$

The shaft forces at B and D are given to be:

$\vec{F}_B = -253\vec{j} - 253\vec{k}\ N$

$\vec{F}_D = -320\vec{k}\ N$

$af = 1.0$

$n = 1100\ rpm$

$L_{10} = 12\ khrs\ (required)$

$R = 99\ \%$

Since there is no axial load but only radial load

$F_e = V \cdot F_r \quad where\ V = 1.0 \quad and \quad Also\ af = 1.0$

$F_{e_O} = R_O = 274.36\ N$

$F_{e_E} = R_E = 363.94\ N$

For ball bearings

$$R = \exp\left[-\left(\frac{L}{6.84 \times L_{10}} \right)^{1.17} \right]$$

$$\ln\left(\frac{1}{\mathrm{R}}\right) = +\left(\frac{L}{6.84 \times L_{10}} \right)^{1.17}$$

$$L_{10} = \frac{L_{req}}{6.84 \times \left(\ln \frac{1}{R} \right)^{\frac{1}{1.17}}} = \frac{12000}{6.84 \times \left(\ln \frac{1}{0.99} \right)^{\frac{1}{1.17}}}$$

$$L_{10} = 89467\ hrs \quad \text{required}$$

$$L_{hrs_{O/E}} = \left(\frac{C}{F_{eqv}} \right)^{a} \times \frac{16667}{n_{rpm}}$$

$$C_{req_{O/E}} = \left(\frac{L_{hrs} \times n_{rpm}}{16667} \right)^{\frac{1}{a}} \times F_{eqv}$$

$$C_{req_O} = \left(\frac{89467 \times 1100}{16667} \right)^{\frac{1}{3}} \times 274.36 = 4959\ N$$

$$C_{req_E} = \left(\frac{89467 \times 1100}{16667} \right)^{\frac{1}{3}} \times 363.94 = 6579\ N$$

Equivalent bearing load
dynamic $P = XF_r + YF_a$
static $P_0 = 0,6\,F_r + 0,5\,F_a$
When $P_0 < F_r$ use $P_0 = F_r$

Boundary dimensions			Basic load ratings		Limiting speeds Lubrication		Mass	Designations	Calculation factors dynamic					
			dynamic	static	grease	oil					$F_a/F_r \le e$		$F_a/F_r > e$	
d	D	B	C	C_0					F_a/C_0	e	X	Y	X	Y
mm			N (1 N = 0,225 lbf)		r/min		kg	—	—					
17	26	5	1 320	915	24 000	30 000	0,0082	61803	0,025	0,22	1	0	0,56	2
	35	8	4 650	2 800	19 000	24 000	0,032	16003	0,04	0,24	1	0	0,56	1,8
	35	10	4 650	2 600	19 000	24 000	0,039	6003	0,07	0,27	1	0	0,56	1,6
	40	12	7 350	4 500	17 000	20 000	0,065	6203	0,13	0,31	1	0	0,56	1,4
	47	14	10 400	6 550	16 000	19 000	0,12	6303	0,25	0,37	1	0	0,56	1,2
	62	17	17 600	11 800	12 000	15 000	0,27	6403	0,5	0,44	1	0	0,56	1
20	32	7	2 040	1 400	19 000	24 000	0,018	61804						
	42	8	5 400	3 400	16 000	22 000	0,050	16004						
	42	12	7 200	4 500	17 000	20 000	0,069	6004						
	47	14	9 800	6 200	15 000	18 000	0,11	6204						
	52	15	12 200	7 800	13 000	16 000	0,14	6304						
	72	19	23 600	16 600	10 000	13 000	0,40	6404						
25	37	7	2 280	1 700	17 000	20 000	0,022	61805						
	47	8	5 850	4 000	14 000	17 000	0,060	16005						
	47	12	8 650	5 600	15 000	18 000	0,080	6005						
	52	15	10 800	6 950	12 000	15 000	0,13	6205						
	62	17	17 300	11 400	11 000	14 000	0,23	6305						
	80	21	27 500	19 600	9 000	11 000	0,53	6405						
30	42	7	2 280	1 800	15 000	18 000	0,026	61806						
	55	9	6 650	5 850	12 000	15 000	0,085	16006						
	55	13	10 200	6 800	12 000	15 000	0,12	6006						
	62	16	15 000	10 000	10 000	13 000	0,20	6206						
	72	19	21 600	14 600	9 000	11 000	0,35	6306						
	90	23	33 500	24 000	8 500	10 000	0,74	6406						
35	47	7	2 360	2 000	13 000	16 000	0,030	61807						
	62	9	9 500	6 950	10 000	13 000	0,11	16007						
	62	14	12 200	8 500	10 000	13 000	0,16	6007						
	72	17	19 600	13 700	9 000	11 000	0,29	6207						
	80	21	25 500	18 000	8 500	10 000	0,46	6307						
	100	25	42 500	31 000	7 000	8 500	0,95	6407						
40	52	7	2 450	2 200	11 000	14 000	0,034	61808						

Deep Groove (single row) ball bearings
(cheap, most widely used in light applications)

Choose SKF 6007 (d= 35 mm)

$$L_{hrs} = \left(\frac{C}{F_{eqv}} \right)^a \times \frac{16667}{n_{rpm}}$$

$$C = 12200\ N$$

$$C_o = 8500\ N \rightarrow L_{10\,\underset{O}{hrs}} = \left(\frac{12200}{274.36} \right)^3 \frac{16667}{1100} = 1{,}332{,}238\ hrs >>> 89467\ hrs$$

too much life

Choose SKF 61807

$$C = 2360 N$$

$$C_o = 2000 N \rightarrow L_{\underset{O}{hrs}} = \left(\frac{2360}{274.36} \right)^3 \frac{16667}{1100} = 9643 hrs < 89467$$

not satisfactory

<u>Choose SKF 16007</u>

$$L_{hrs} = \left(\frac{C}{F_{eqv}}\right)^a \times \frac{16667}{n_{rpm}}$$

$C = 9500\ N$

$C_o = 6950\ N \rightarrow \underset{O}{L_{hrs}} = \left(\frac{9500}{274.36}\right)^3 \frac{16667}{1100} = 629032\ hrs > 89467\ hrs$

satisfactory

Same bearing (<u>SKF 16007)</u> for point E provides a life of

$$\underset{E}{L_{hrs}} = \left(\frac{9500}{363.94}\right)^3 \frac{16667}{1100} = 269492\ hrs > 89467\ hrs$$

satisfactory

If we intend to use <u>angular contact ball bearing</u> (d=35mm) the option with smallest C value is SKF 7207 B

$$C = 20800N$$

$$C_o = 15000N \rightarrow L_{\substack{hrs \\ E}} = \left(\frac{20800}{364}\right)^3 \frac{16667}{1100} = 2985420hrs >>> 89467$$

too much life

If we intend to use straight roller bearings (F_a=0)
The option with smallest C value is SKF NU1007

$$C = 19000N$$

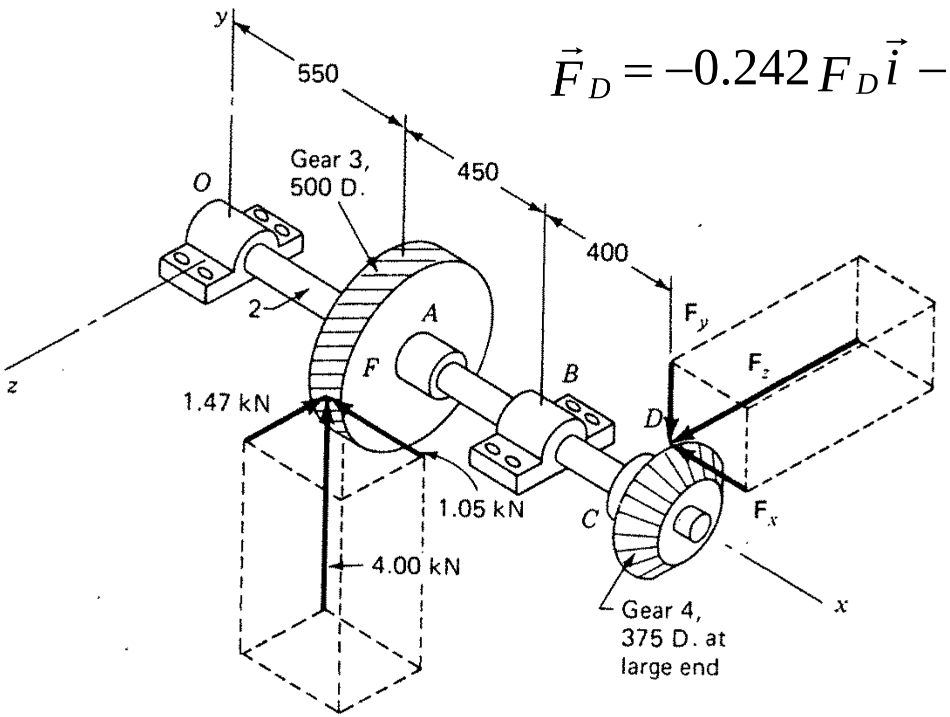
$$C_o = 11600N \rightarrow L_{hrs} = \left(\frac{19000}{364}\right)^3 \frac{16667}{1100} \cong 2275500hrs >>> 89467hrs$$

too much life

<u>Thus better to use SKF 16007 single row deep groove ball bearing which satisfies a life of</u>
<u>269492 hrs for bearing E &</u>
<u>629032 hrs for bearing O (both in excess of 89467 hrs)</u>

Example 3.3 (11.7 <u>Ex for angular contact ball bearing and straight roller bearing</u>

$$\vec{F}_D = -0.242\,F_D\vec{i} - 0.242\,F_D\vec{j} + 0.940\,F_D\vec{k}$$

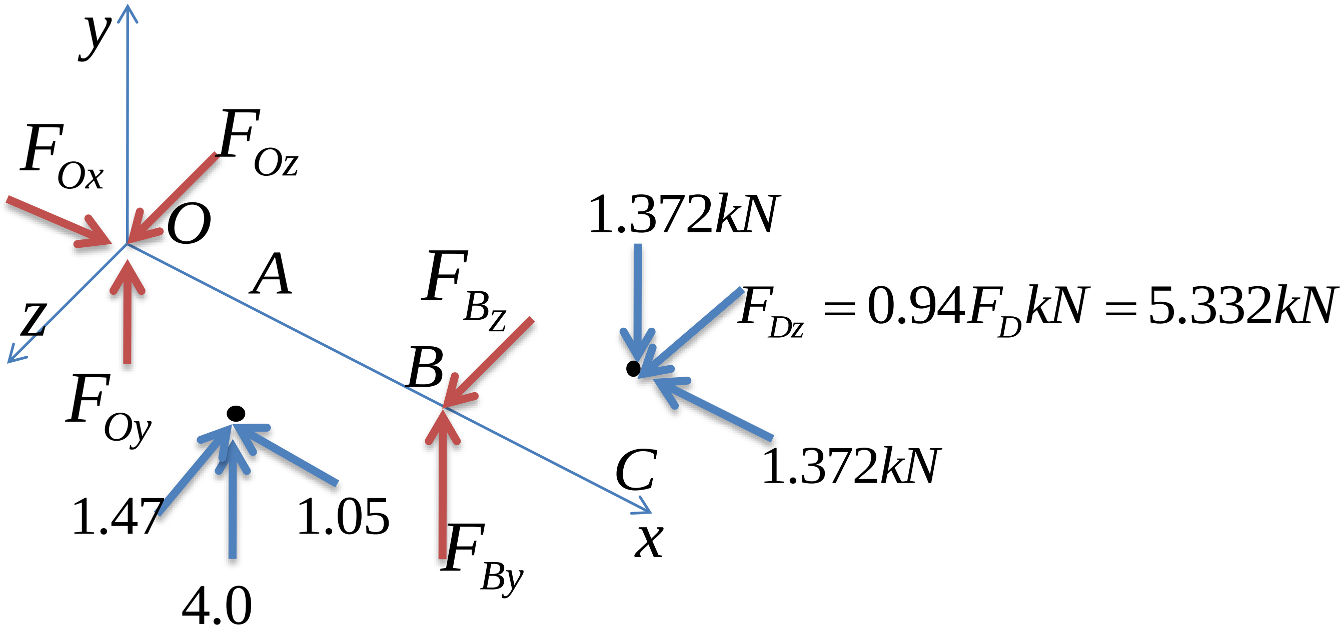


An angular-contact ball bearing with shallow angle is to be housed at O to take both radial and thrust loads.

The bearing at B is to be a straight roller bearing.

a) Determine the required ratings of each bearing based on an L_{10} life of 36 khrs at a shaft speed of 900 rpm.

b) Select suitable bearings for both points. If the bearing bores are 90 mm (or 75 mm) at O and 60 mm at B.



$$\vec{F}_D = -0.242\,F_D\vec{i} - 0.242\,F_D\vec{j} + 0.940\,F_D\vec{k}$$

Use torque balance of the shaft to determine force F_D

$$\sum M_{O_x} = 0;$$

$$4 \times \frac{500}{2} = 0.94F_D \times \frac{375}{2}$$

$$F_D = 5.673\,kN$$

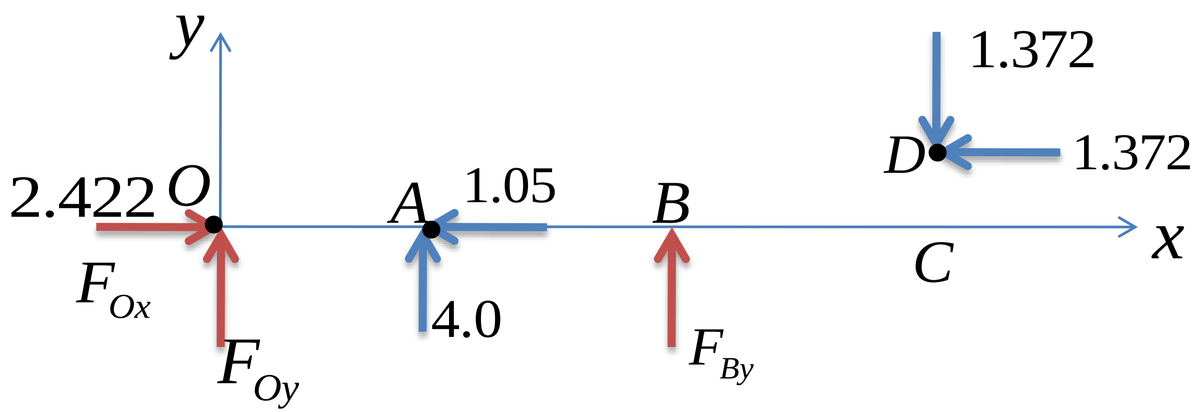
$$F_{Dx} = -0.242F_D = 1.372kN$$

$$F_{Dy} = 1.372kN$$

$$F_{Dz} = 5.332kN$$

$$\sum F_x = 0 \rightarrow F_{Ox} = 1.05 + 1.372 = 2.422kN$$

In x-y plane



$$\sum M_{O_z} = 0;$$

$$4 \times 550 + F_{By} \times 1000 + \frac{375}{2} \times 1.372 = 1400 \times 1.372$$

$$F_{By} = -0.534\ kN$$

$$\sum F_y = 0;$$

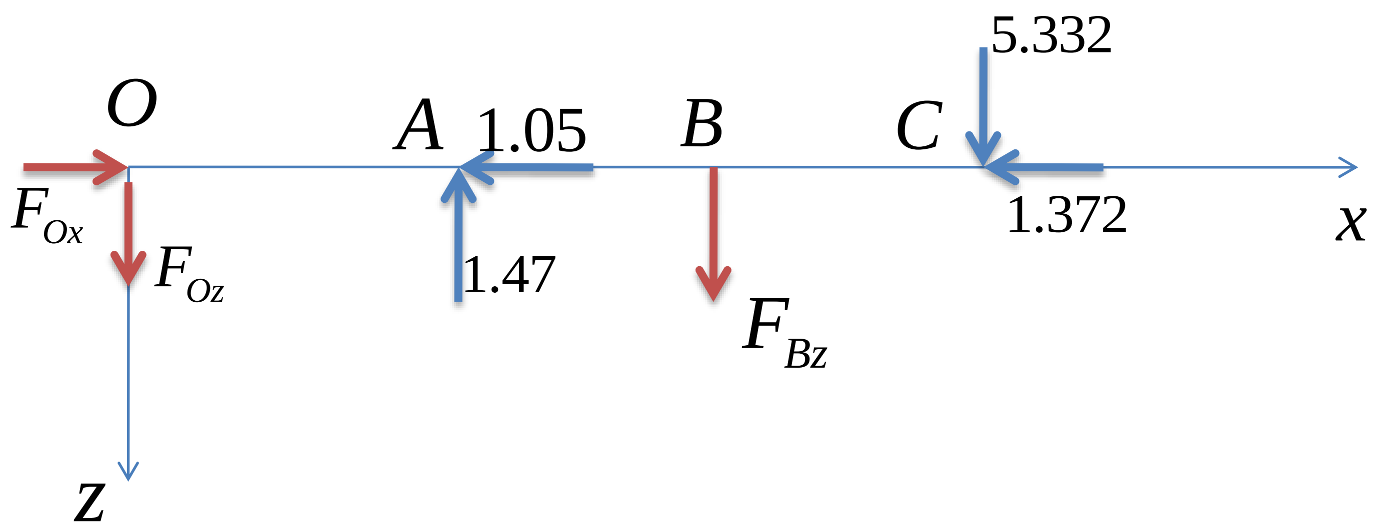
$$F_{Oy} + 4.0 + F_{By} - 1.372 = 0$$

$$F_{Oy} = 1.372 - 4.0 - F_{By}$$

$$F_{Oy} = 1.372 - 4.0 - (-0.534)$$

$$F_{Oy} = -2.093 kN$$

In x-z plane



$$\sum M_{O_y} = 0;$$

$$1.47 \times 550 + F_{Bz} \times 1000 - 1400 \times 5.332 = 0$$

$$F_{Bz} = -6.656\ kN$$

$$\sum F_z = 0;$$

$$F_{Oz} - 1.47 + F_{Bz} + 5.332 = 0$$

$$F_{Oz} = 1.47 - F_{Bz} - 5.332$$

$$F_{Oz} = 2.794 kN$$

Thus axial and radial loads at bearings O and B are:

$$F_{a_O} = F_{Ox} = 2.422kN$$

$$F_{r_O} = \sqrt{F_{Oy}{}^2 + F_{Oz}{}^2} = 3.491kN$$

For angular contact ball bearing at point O with both radial and thrust (axial) load

$$F_{a_B} = 0$$

$$F_{r_B} = \sqrt{F_{By}{}^2 + F_{Bz}{}^2} = 6.677kN$$

For roller contact bearing at point B with only radial load and no thrust (axial) load

$$\rightarrow a) C_{req} = ?, C_{req} = \left(\frac{L_{hr} x n_{rpm}}{16667} \right)^{\frac{1}{a}} x F_{eqv}$$

$$F_{eqv} = V \times F_r$$

$$F_{eqv} = V \times XF_r + YF_a$$

$$F_{eqv} = af \times F_e$$

For commercial gearing
af= 1.1-1.3
Thus we use af=1.2

a) For angular contact ball bearing at O

$$F_a = 2.422\ kN$$

$$F_r = 3.491\ kN, \quad \frac{F_a}{F_r} = 0.693 \text{ and } e = 1.14$$

Thus $\frac{F_a}{F_r} < e;$ $X = 1.0$, $Y = 0.0$ from sheet for 72B and 73B series

$$F_e = 1.0 \times 1.0 \times 3.491 + 0.0 \times 2.422$$

$$F_e = 3.491\ kN$$

$$F_{eqv} = af \times F_e = 1.2 \times 3.491 = 4.1892\ kN$$

$$C_{req} = \left(\frac{L_{hrs} \times n_{rpm}}{16667} \right)^{\frac{1}{a}} \times F_{eqv}$$

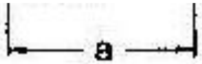
$$C_{req} = \left(\frac{36000 \times 900}{16667} \right)^{\frac{1}{3}} \times 4.1892 = 52.282\ kN = 52282\ N$$

For straight roller bearings at B

$$F_a = 0 \rightarrow F_e = V \times F_r = 1 \times 6.677\ kN = 6.677\ kN$$

$$F_{eqv} = 1.2 \times F_e = 1.2 \times 6.677\ kN = 8.012\ kN$$

$$C_{req} = \left(\frac{36000 \times 900}{16667} \right)^{\frac{3}{10}} \times 8.012 = 77.686\ kN = 77686\ N$$



Angular Contact Ball Bearings

Equivalent bearing load
dynamic: $P = XF_r + YF_a$
static: $P_0 = 0{,}5\ F_r + 0{,}26\ F_a$
When $P_0 < F_r$ use $P_0 = F_r$

Boundary dimensions			Basic load ratings		Limiting speeds Lubrication		Mass	Designation	Calculation factors dynamic					
			dynamic	static	grease	oil				$F_a/F_r \leq e$		$F_a/F_r > e$		
d	D	B	C	C_0					e	X	Y	X	Y	
mm			N (1 N = 0,225 lbf)		r/min		kg	—	—					
10	30	9	3 800	2 120	19 000	28 000	0,031	7200 B	1,14	1	0	0,35	0,57	[illegible]
12	32	10	5 400	3 050	17 000	24 000	0,045	7201 B						
15	35	11	6 200	3 650	16 000	22 000	0,048	7202 B						
	42	13	9 000	5 300	14 000	19 000	0,090	7302 B						
17	40	12	7 650	4 650	14 000	19 000	0,070	7203 B						
	47	14	11 400	7 100	12 000	17 000	0,12	7303 B						
20	47	14	10 200	6 400	11 000	16 000	0,11	7204 B						
	52	15	13 400	8 150	10 000	15 000	0,15	7304 B						
25	52	15	11 400	7 650	9 500	14 000	0,14	7205 B						
	62	17	19 000	12 200	8 500	12 000	0,24	7305 B						
30	62	16	15 600	11 000	8 500	12 000	0,21	7206 B						
	72	19	24 000	16 600	7 500	10 000	0,36	7306 B						
35	72	17	20 800	15 000	7 500	10 000	0,30	7207 B						
	80	21	[illegible]	[illegible]	7 000	9 500	0,48	7307 B						

b) At point O (angular contact ball bearing)

SKF 7218 B with d= 90 mm and C= 81 500 N (>52 282 N) or
SKF 7215 B with d= 75 mm and C= 55 000 N (>52 282 N)
will suit the application.
The life of the bearing will then be

$$L_{hrs} = \left(\frac{C}{F_{eqv}}\right)^a \times \frac{16667}{n_{rpm}} = \left(\begin{matrix} 81500 \\ \dfrac{55000}{4189} \\ \end{matrix}\right)^3 \frac{16667}{900}$$

$$L_{hrs} = 136380 hrs >> 36000 hrs$$ with SKF 7218 B

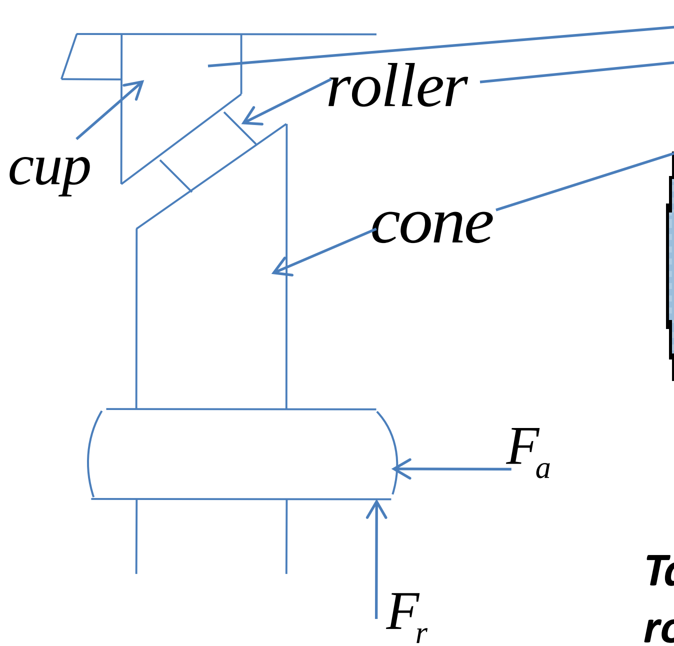
$$L_{hrs} = 41915 hrs >> 36000 hrs$$ with SKF 7215 B

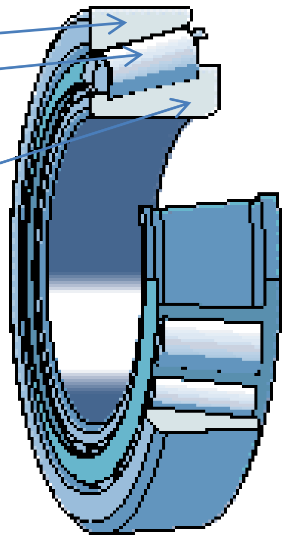
At point B, straight roller bearing

SKF NU2212 (or NJ2212) with d=60 mm and **C=88 000 N** (>77 686 N) will suit the application. The life of the bearing will be

$$L_{hrs} = \left(\frac{88000}{8012}\right)^{\frac{10}{3}} \frac{16667}{900} = 54544 hrs > 36000 hrs$$

Selection of Tapered Roller Bearings





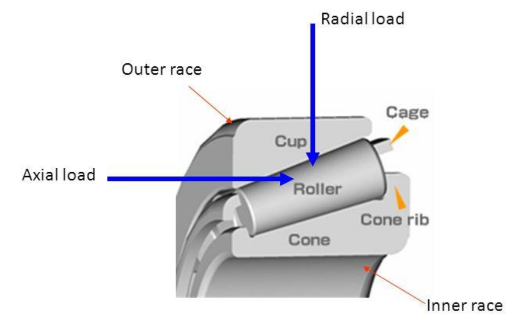
Tapered roller bearing

Tapered roller bearings can carry both

- *radial loads and
- *axial (thrust) loads or
- *combination of radial & axial loads

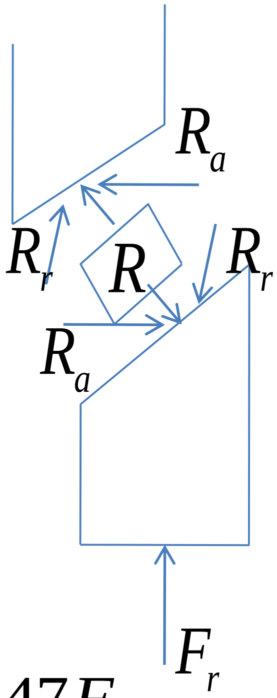
in relatively high capacities compared with other bearings

Tapered roller bearings (takes on radial as well as thrust/axial loads)



Ref: http://www.jtekt.co.jp/e/company/news/images/20060404_3e.jpg

Even, in cases:
when an <u>external axial</u> (thrust) load <u>is not present,</u>
the radial load itself will induce (create) a thrust reaction/force within the bearing because of taper geometry.



$$R_a = \frac{0.47 F_r}{K}$$

R_a is created by F_r due to the taper geometry and this R_a axial force can be thought as a force trying to separate the races (rings) from the rollers.

To avoid separation of the races and rollers this thrust force R_a must be resisted/reacted by an equal and opposite force.

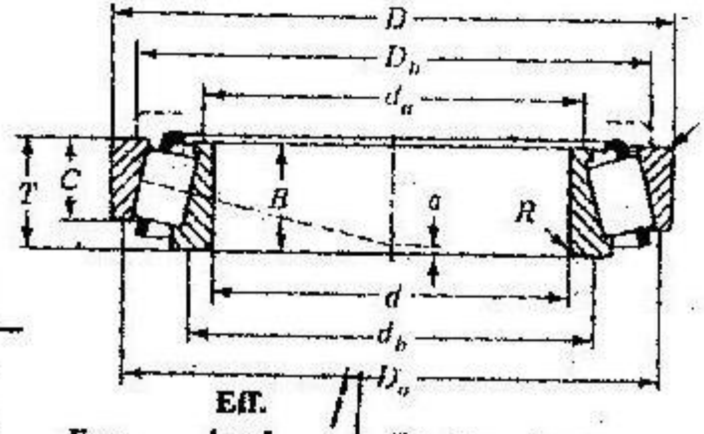
One way of generating this opposing force is to always use at least two tapered roller bearings on a shaft with opposite mounting.

Above equation is given by TIMKEN, one of the largest tapered roller bearing manufacturer.

$$K \cong 1.5 \text{ for radial bearings.}$$

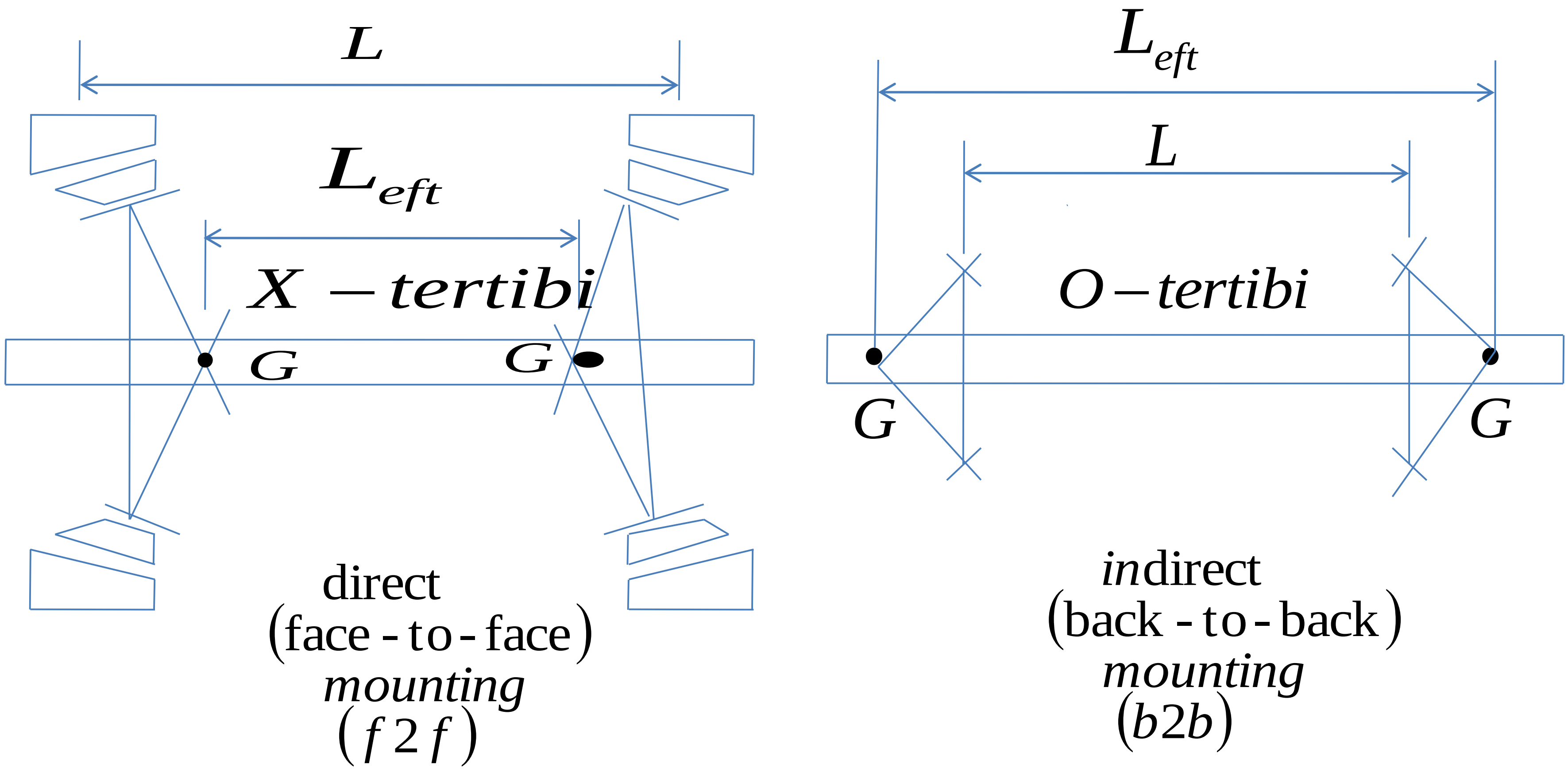
$$K \cong 0.75 \text{ for steep angle bearings.}$$

Exact values for K's are given in catalogues.



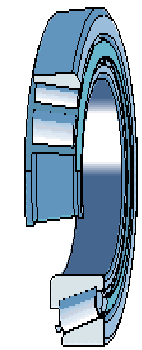
Bore d	Outside diameter D	Width T	Rating at 500 rpm for 3000 hours, L10: One row radial lb daN	Thrust lb daN	Factor K	Eff. load center a†	Part numbers: Cone	Cup
0.3750 inch	1.2595	0.3940	435	301	1.44	−0.12	A2037	A2126
9.525 mm	31,991	10,008	192	134		−3,0		
‡0.4724	1.2595	0.3940	435	301	1.44	−0.12	A2047	A2126
‡12,000	31,991	10,008	192	134		−3,0		
0.4992	1.3775	0.4330	500	387	1.29	−0.10	A4049	A4138
12,680	34,988	10,998	222	172		−2,5		
0.5000	1.3775	0.4330	500	387	1.29	−0.10	A4050	A4138
12,700	34,988	10,998	222	172		−2,5		
0.5000	1.5000	0.5313	760	358	2.12	−0.20	00050	00150
12,700	38,100	13,495	338	159		−5,1		
‡0.5906	1.3775	0.4330	500	387	1.29	−0.10	A4059	A4138
‡15,000	34,988	10,998	222	172		−2,5		
0.6250	1.3775	0.4330	575	314	1.83	−0.13	L21549	L21511
15,875	34,988	10,998	256	140		−3,3		
0.6250	1.5745	0.4730	530	480	[illegible]	[illegible]	[illegible]	[illegible]

Bore d	Outside diameter D	Width T	row radial lb daN	Thrust lb daN	Factor K	Eff. load center a†	Part numbers Cone	Part numbers Cup
0.3750 inch	1.2595	0.3940	435	301	1.44	−0.12	A2037	A2126
9.525 mm	31,991	10,008	192	134		−3,0		
‡0.4724	1.2595	0.3940	435	301	1.44	−0.12	A2047	A2126
‡12,000	31,991	10,008	192	134		−3,0		
0.4992	1.3775	0.4330	500	387	1.29	−0.10	A4049	A4138
12,680	34,988	10,998	222	172		−2,5		
0.5000	1.3775	0.4330	500	387	1.29	−0.10	A4050	A4138
12,700	34,988	10,998	222	172		−2,5		
0.5000	1.5000	0.5313	760	358	2.12	−0.20	00050	00150
12,700	38,100	13,495	338	159		−5,1		
‡0.5906	1.3775	0.4330	500	387	1.29	−0.10	A4059	A4138
‡15,000	34,988	10,998	222	172		−2,5		
0.6250	1.3775	0.4330	575	314	1.83	−0.13	L21549	L21511
15,875	34,988	10,998	256	140		−3,3		
0.6250	1.5745	0.4730	530	480	1.11	−0.06	A6062	A6157
15,875	39,992	12,014	236	212		−1,5		
0.6250	1.6250	0.5625	890	475	1.88	−0.20	03062	03162
15,875	41,275	14,288	396	210		−5,2		
0.6250	1.6875	0.5625	725	875	0.83	−0.05	11590	11520
15,875	42,862	14,288	324	390		−1,1		
0.6250	1.6875	0.6563	1150	655	1.76	−0.23	17580	17520
15,875	42,862	16,670	515	292		−6,0		
0.6250	1.7500	0.6100	1020	620	1.64	−0.15	05062	05175
15,875	44,450	15,494	450	276		−3,9		
0.6250	1.9380	0.7813	1600	730	2.20	−0.36	09062	09195
15,875	49,225	19,845	715	324		−9,1		
0.6250	2.1250	0.8750	1710	1730	0.99	−0.23	21063	21212
15,875	53,975	22,225	760	770		−5,8		
‡0.6299	†1.8504	0.8268	1560	1460	1.07	−0.24	HM81649	HM81610
‡16,000	†47,000	21,000	695	650		−6,0		
0.6690	1.6250	0.4687	530	480	1.11	−0.06	A6067	A6162
16,993	41,275	11,905	236	212		−1,5		
0.6872	1.4380	0.4375	500	420	1.20	−0.08	A5069	A5144



B2b mounting provides stiffer structure than f2f mounting and it is recommended if there is tilting moment on the bearings.

For a typical tapered roller bearing application under the effect of external radial load F_r and external thrust load T_e ;
the roller radial reactions at G_A and G_B are calculated by using the moment equations at effective load center G_A and G_B by the help of distances r_A and r_B .



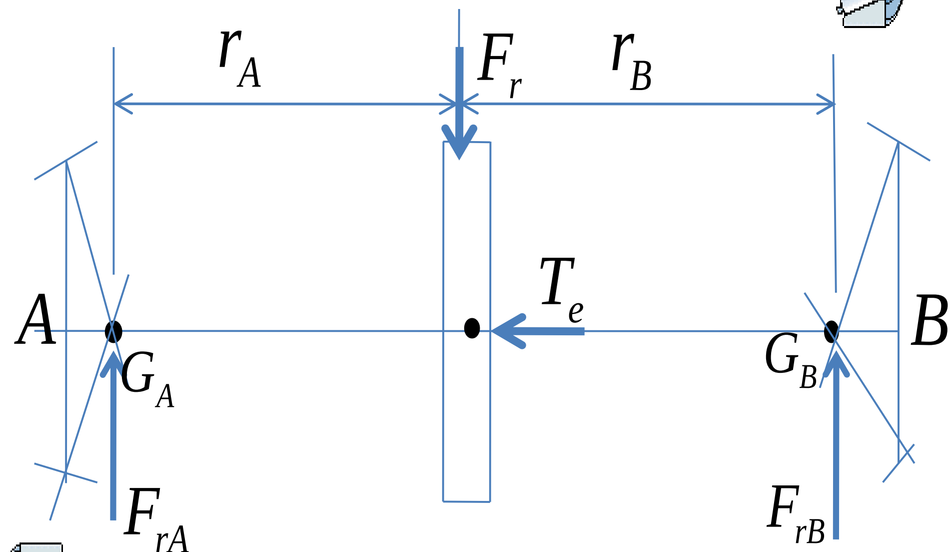
Thus;

$$\sum M_B = 0, \quad F_{rA}(r_A + r_B) = F_r \times r_B$$

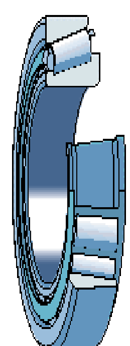
$$F_{rA} = F_r \frac{r_B}{r_A + r_B}$$

Also from $\sum M_A = 0, \quad or \sum F_y = 0$

$$F_{rB} = F_r \frac{r_A}{r_A + r_B}$$



The external thrust load T_e , on the other hand, is carried by the bearing at A only due to the face-to-face arrangement.



Tapered roller bearing

From $\sum F_x = 0$

$$F_{a_A} = T_e$$

In ball bearings the equivalent radial load was calculated by the relation:

$$F_e = V \cdot X \cdot F_r + Y \cdot F_a$$

The same relation is used for tapered roller bearings with a few modifications:

V=1.0 (rotation factor) and X becomes 0.4 and Y is the K value of bearing:

Total thrust load

$$F_{eA} = 1.0 \times 0.4 \times F_{rA} + K_A \left(\frac{0.47 \times F_{rB}}{K_B} \pm T_e \right)$$

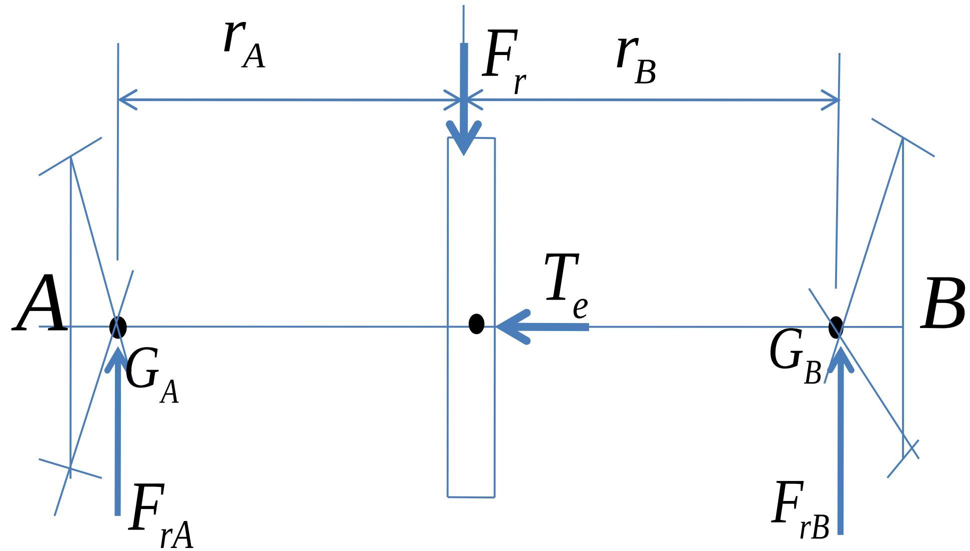
External thrust load

Thrust load
created by the other bearing

For a f2f configuration as seen:

$$F_{eA} = 0.4F_{rA} + K_A\left(\frac{0.47F_{rB}}{K_B} + T_e\right)$$

$$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} - T_e\right)$$



However, we have to check if new equivallent radial load is more or less than the actual radial load on the bearing:

If, $F_{eA} < F_{rA}$ then $F_{eA} = F_{rA}$

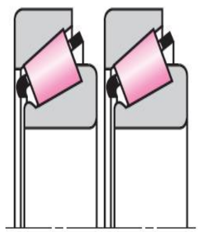
If, $F_{eB} < F_{rB}$ then $F_{eB} = F_{rB}$

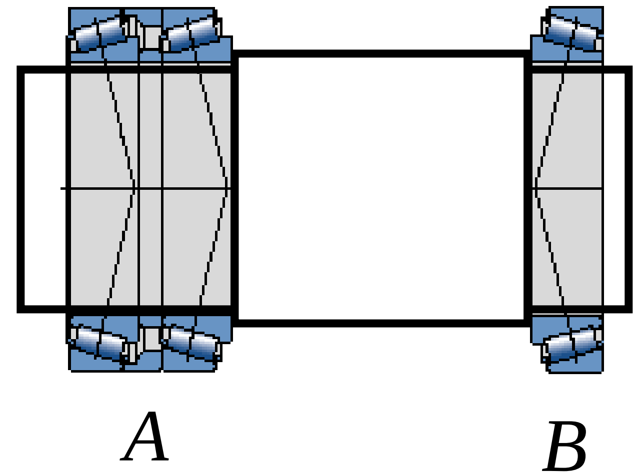
Also try not to forget using application factor as a load safety factor

$$F_{eq_A} = af \times F_{e_A}$$

$$F_{eq_B} = af \times F_{e_B}$$

If a <u>tandem</u> arrangement (means two bearings together) is used at A, and a single bearing at B:





$$F_{eA} = 0.4F_{rA_1} + K_A\left(\frac{0.47F_{rB}}{2xK_B} + T_{e_1}\right)$$

$$F_{rA_1} = \frac{F_{rA}}{2}$$

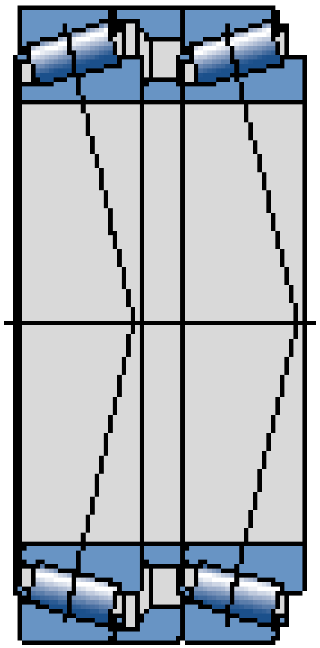
Thrust load
Created by bearing at B and carried by one bearing at A

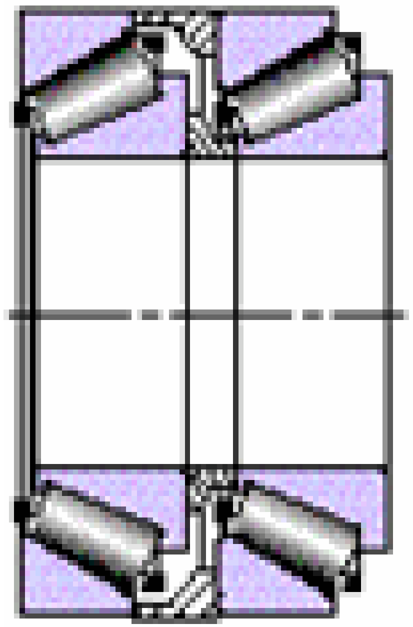
Thrust load carried by one bearing at A

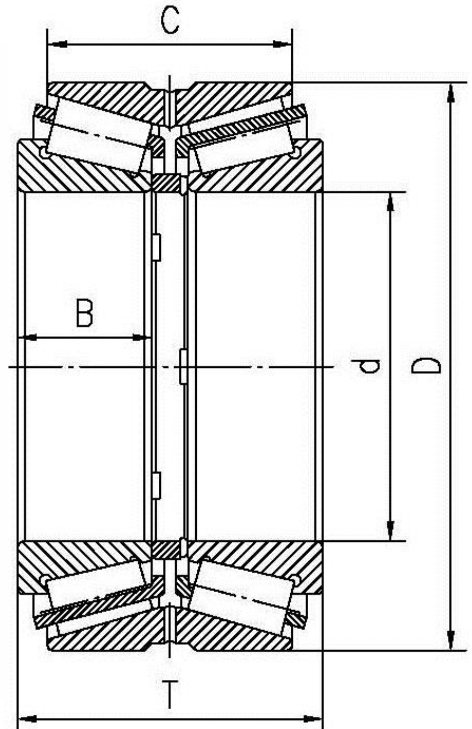
$$T_{e_1} = \frac{T_e}{2}$$

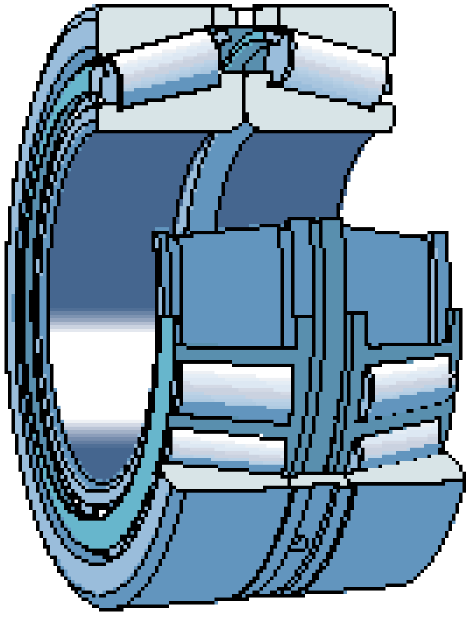
For bearing B (if a single bearing is used):

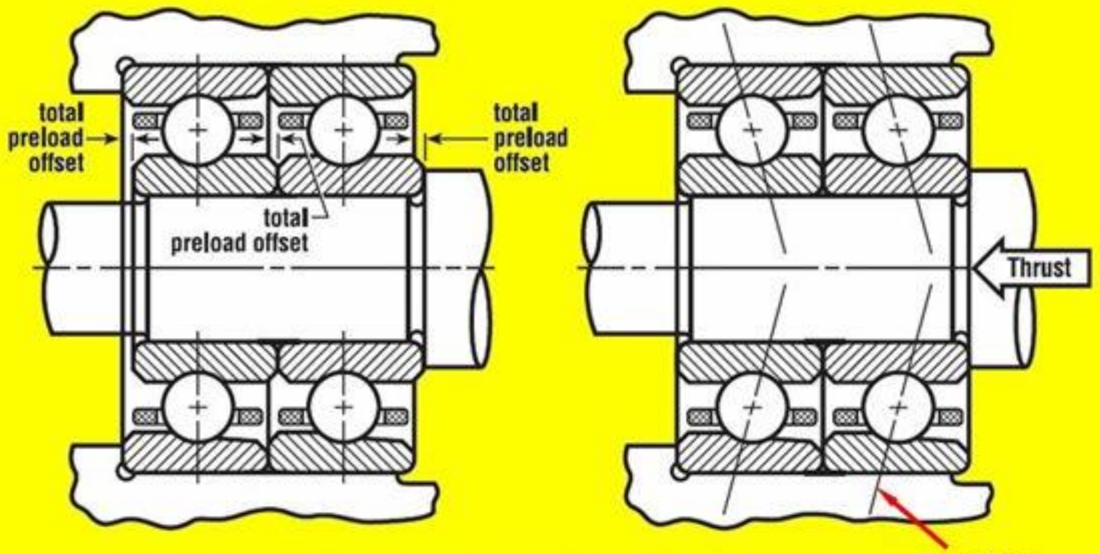
$$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} - T_e\right)$$



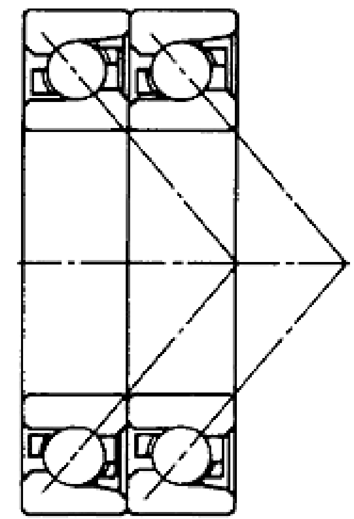








Tandem Arrangement of Duplex Bearings



Tandem arrangement (DT)

Along with the equations given;

once external force and bearing reaction values are known, <u>a trial & error method</u> can be used to select a suitable tapered roller bearing for the information generally given as shaft speed and shaft diameter.

Then the life of bearing selected can be compared with the life required for the application.

$$L_D = \frac{L_R . n_R}{n_D} \left(\frac{C_R}{F} \right)^a$$

If bearing selected provides a life reasonably larger than the life required then selection is correct,

$$L_D \quad >? \quad L_{req}$$

If NOT, then try a stronger bearing and check life again

Tapered roller bearings produced by TIMKEN do obey the relation of

$$C_R = F_{eq}\left[\frac{L_D n_D}{L_r n_r}\right]^{\frac{1}{a}}$$

$$C_{R_{TIMKEN}} = F_{eq_{A,B}}\left[\frac{L_D n_D}{L_R n_R}\right]^{\frac{1}{a}}$$

$$C_{R_{general}} = F_{eqv}\left[\frac{L_D}{10^6}\right]^{\frac{1}{a}}, a = \frac{10}{3}$$

where

L_D design (required) life of bearing in hrs

n_D design speed (in rpm) of the shaft

L_r (rating life) 3000 hrs

n_r (rating speed) 500 rpm

F_{eq} radial equivalent load to be carried

C_R (rating load) for bearings

These equations help calculate required rating load to be satisfied in choosing roller bearings from catalogue.

Selection procedure for TIMKEN Tapered Roller Bearings.

1) Calculate radial loads at related bearing points from statics
2) Calculate thrust load (T_e) and set the direction
3) Assume a K value of nearly 1.5 for both bearings $K_A = 1.5$ *and* $K_B = 1.5$
4) Calculate F_{eA} and F_{eB} values based on above values by using relations.

$$F_{eA} = 0.4\frac{F_{rA}}{n} + K_A\left(\frac{0.47F_{rB}}{nxK_B} \pm \frac{T_e}{n}\right)$$

$$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} \pm T_e\right)$$

Use correct directions for thrust load and also take into consideration tandem conditions if it exists *n* is the number of bearings at *A* (if tandem exists at *A*)

If $F_{e_{A,B}} < F_{r_{A,B}}$ then $F_{e_{A,B}} = F_{r_{A,B}} \Rightarrow F_{eqv} = af \times F_e$

5) Use F_{eqv} values in load-life relation along with L_D & n_D values to calculate required rating load of the bearings to be used is selection from catalogue.

Here are the relations used to select TIMKEN tapered roller bearings

$$C_R = F\left[\frac{L_D x n_D}{L_R x n_R}\right]^{\frac{1}{a}} \qquad C_R = F\left[\frac{L_D x n_D}{L_R x n_R}\left(\frac{1}{6.84}\right)\right]^{\frac{1}{a}} x \frac{1}{\left(\ln\frac{1}{R}\right)^{\frac{1}{1.17a}}}$$

6) Choose a bearing (both cup-cone pair) from the catalogue which satisfies both diameter and rating load (C_R) requirements;

$$C_{Rcat} > C_R$$

7) Use the new and correct K values of selected bearings (K_A, K_B) in step 4 to recalculate the F_{eA} and F_{eB} values and then use *af* if required.

8) Use the new $F_{eqv\,A,B_{new}}$ values in life equation to calculate the new life which the selected bearing can run for (with 90 % reliability). This life has to be more than what is required.

$$L_D = L_R \frac{n_R}{n_D}\left(\frac{C_R}{F_{eq_{new}}}\right)^a_{A,B}$$

9) Compare the newly calculated life with the life required;

If it is more than what is required, then selection is correct,

If NOT, then selection is wrong, TRY a stronger bearing until the life requirement is satisfied.

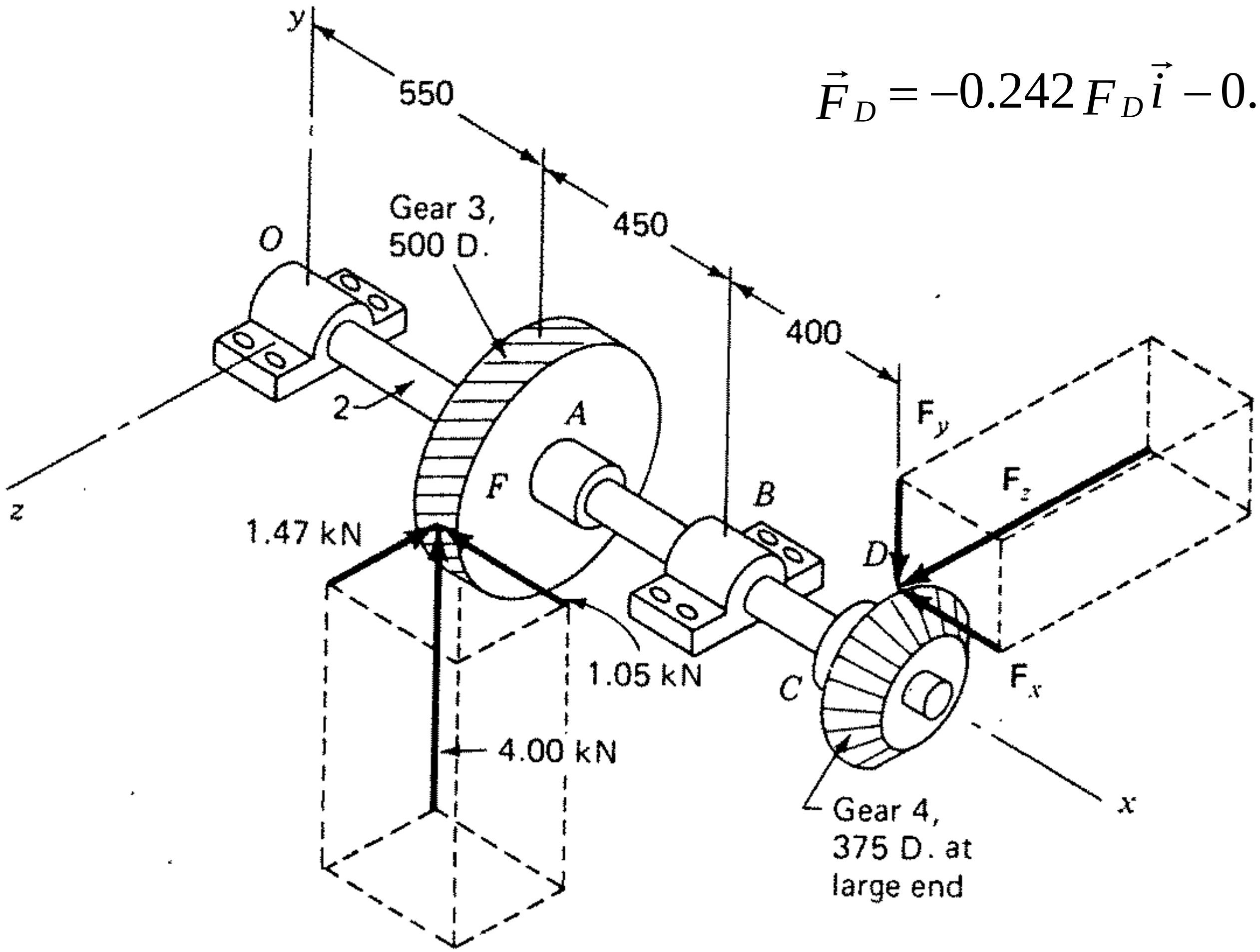
If $L_{D_{new}} > L_D$ selection is OK

If $L_{D_{new}} < L_D$ re-try a stronger bearing

Example 3.4 (11.7) same as Fig. in Ex. 3.3

Example for tapered roller bearings 11.9 (same figure as in 11.10)

$$\vec{F}_D = -0.242\, F_D \vec{i} - 0.242\, F_D \vec{j} + 0.940\, F_D \vec{k}$$



Tapered roller bearings are to be mounted in housings at points O and B with the bearing at O intended to take out the major thrust component.

a) The bearings are to have an L_{10} life of 36 000 hrs corresponding to a shaft speed of 900 rpm. Use 1.5 for *K* factors, unity for the application factor and find the required radial rating of each bearing,

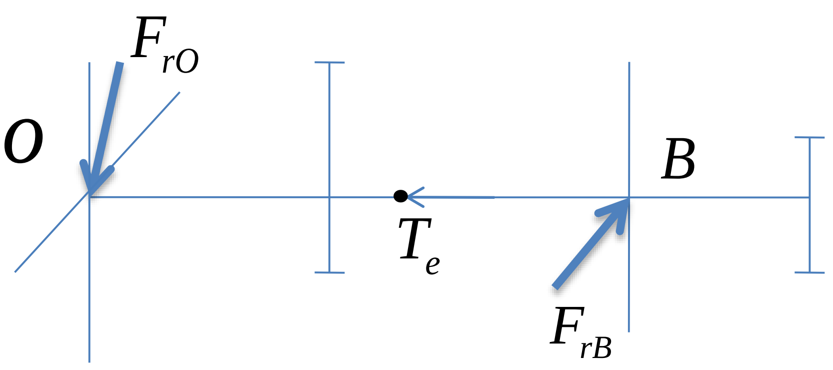
b) Choose suitable bearings for both housings with diameter about 34 mm at O and 34 mm at B, for a life of L_{10} = 10 khrs at a shaft speed of 500 rpm from TIMKEN catalogue.

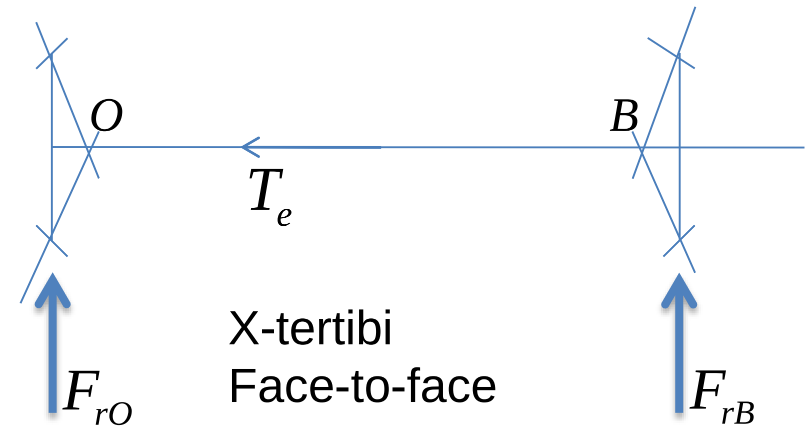
The reaction forces were found as shown in the figure.

$$T_e = 2.422\ kN$$

$$F_{r_O} = 3.491\ kN$$

$$F_{r_B} = 6.677\ kN$$





So,

$$F_{eO} = 0.4F_{rO} + K_o(\frac{0.47F_{rB}}{K_B} + T_e) = 0.4x3.491 + 1.5(\frac{0.47x6.677}{1.5} + 2.422) = 8.167kN$$

$$F_{eB} = 0.4F_{rB} + K_B(\frac{0.47F_{rO}}{K_O} - T_e) = 0.4x6.677 + 1.5(\frac{0.47x3.491}{1.5} - 2.422) = 0.678kN$$

Since $F_{e_B} < F_{r_B};\quad F_{e_B} = F_{r_B} = 6.677\ kN$

The required ratings for the bearings are:

$$C_R = F_{eq}\left[\frac{L_D n_D}{L_r n_r}\right]^{\frac{1}{a}} \rightarrow$$

$$C_{R_O} = 8167\left[\frac{36000 \times 900}{3000 \times 500}\right]^{\frac{3}{10}} = 20530\ N$$

$$C_{R_B} = 6677\left[\frac{36000 \times 900}{3000 \times 500}\right]^{\frac{3}{10}} = 16784\ N$$

b) For $L_D = 10000 hrs \,\&\, n_D = 500 rpm$

$$C_{R_O} = 8.167\left[\frac{10000x500}{3000x500}\right]^{\frac{3}{10}} = 11.72kN(= 2631lb)$$

$$C_{R_B} = 6.677\left[\frac{10000x500}{3000x500}\right]^{\frac{3}{10}} = 9.58kN(= 2151lb)$$

From TIMKEN catalogue on page 502 with a diameter of 1.375“ =34.92 mm> 34 mm different C values are possible the required values are 2150 lb & 2630 lb.

1)Try bearings with cone M38549 & cup 38510 with K=1.66 & C=2520lb= 11223 N for both points O & B.

Now re-calculate

$$F_{eO} \,\&\, F_{eB}$$

Since K_A and K_B values changed from 1.5 to 1.66

$$F_{eO} = 0.4x3491 + 1.66(\frac{0.47x6677}{1.66} + 2422) = 8555N$$

$$F_{eB} = 0.4x6677 + 1.66(\frac{0.47x3491}{1.66} - 2422) = 291N \rightarrow F_{eB} = 6677N$$

$$L_{D_{new_O}} = L_R \frac{n_R}{n_D}\left(\frac{C_R}{F_{eO}}\right)^{\frac{10}{3}} = 3000x\frac{500}{500}\left(\frac{11223}{8555}\right)^{\frac{10}{3}} = 7414hrs < 10000hrs$$

not satisfactory

2) Try bearings with ; cone 02877 cup 02820 K=1.29
C=2620 lb=11668N for both O & B
Re-calculate

$$F_{eO} \,\&\, F_{eB}$$

$$F_{eB} = 0.4F_{rB} + K_B\left(\frac{0.47F_{rA}}{K_A} - T_e\right)$$

$$F_{eA} = 0.4F_{rA} + K_A\left(\frac{0.47F_{rB}}{K_B} \pm T_e\right)$$

$$F_{eO} = 0.4 \times 3491 + 1.29\left(\frac{0.47 \times 6677}{1.29} + 2422\right) = 7659\ N$$

$$F_{eB} = 0.4 \times 6677 + 1.29\left(\frac{0.47 \times 6677}{1.29} - 2422\right) = 2684.6\ N < F_{rB} \rightarrow F_{rB} = 6677\ N$$

$$L_{Do} = 3000x\frac{500}{500}\left(\frac{11668}{7659}\right)^{\frac{10}{3}} = 12200 hrs > 10000 hrs;$$

OK for point O

$$L_{D_B} = 3000x\frac{500}{500}\left(\frac{11668}{6677}\right)^{\frac{10}{3}} = 19280 hrs > 10000 hrs;$$

OK for point B

So TIMKEN tapered roller bearing with d=1.375”=34.92 mm.
Cone 02877 K=1.29
Cup 02820 C=2620 lb= 11668 N will satisfy the life & load requirements both at points O & B.

If we use SKF tapered instead of TIMKEN, we use equation

$$F_{eqv} = X \times F_r + Y \times F_a \rightarrow L_{hrs} = \left(\frac{C}{F_{eqv}} \right)^a \frac{16667}{n_{rpm}}; \quad a = 10/3$$ for tapered rollers.

X , *Y* factors are taken from SKF catalogue.
Let d ≥ 34 mm; In SKF we have bearings.

d *(mm)*	**Designation**	***C*** *(N)*	***C_o*** *(N)*	***e***		$\frac{F_a}{F_r} \le e$ ***X***	***Y***		$\frac{F_a}{F_r} > e$ ***X***	***Y***
35	32007X	36500	30500	0.46		1	0		0.4	1.3
	30207	44000	32500	0.37		1	0		0.4	1.6
	32207	56000	45000	0.37		1	0		0.4	1.6
	33207	72000	62000	0.35		1	0		0.4	1.7
	30307									
	31307									
	32307									

Single row taper roller bearings

Equivalent dynamic bearing load

$P = F_r$ when $F_a/F_r \leq e$
$P = 0{,}4\ F_r + Y\ F_a$ when $F_a/F_r > e$

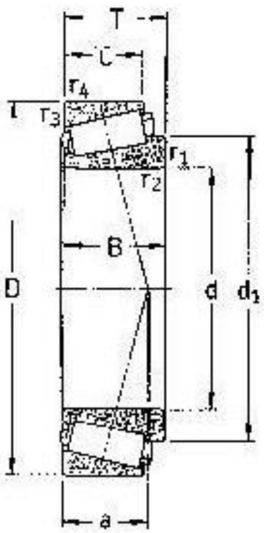
The values of the calculation factors e and Y can be found in the product tables.

Equivalent static bearing load

$P_0 = 0{,}5\ F_r + Y_0\ F_a$

When $P_0 < F_r$, $P_0 = F_r$ should be used. The value of the calculation factor Y_0 can be found in the product tables.

Metric single row taper roller bearings
d 35 – 40 mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Calculation factors		
d	D	T	C	C_0	P_u					e	Y	Y_0
mm			kN		kN	r/min		kg	–	–		
35	62	18	49	54	5,85	8 500	11 000	0,22	* 32007 X/Q	0,46	1,3	0,7
	62	18	42,9	49	5,2	8 000	11 000	0,22	32007 J2/Q	0,44	1,35	0,8
	72	18,25	51,2	56	6,1	7 000	9 500	0,32	30207 J2/Q	0,37	1,6	0,9
	72	24,25	66	78	8,5	7 000	9 500	0,43	32207 J2/Q	0,37	1,6	0,9
	72	28	84,2	106	11,8	6 300	9 500	0,56	33207/Q	0,35	1,7	0,9
	80	22,75	72,1	73,5	8,3	6 700	9 000	0,52	30307 J2/Q	0,31	1,9	1,1
	80	22,75	61,6	67	7,8	6 000	8 500	0,52	31307 J2/Q	0,83	0,72	0,4
	80	32,75	95,2	106	12,2	6 300	9 000	0,73	32307 J2/Q	0,31	1,9	1,1
	80	32,75	93,5	114	13,2	6 000	8 500	0,80	32307 BJ2/Q	0,54	1,1	0,6
37	80	32,75	93,5	114	13,2	6 000	8 500	0,85	32307/37 BJ2/Q	0,54	1,1	0,6

Let’s try SKF 30207 bearings:

For point O & B

$$F_{rO} = 3491\ N$$

$$F_{rB} = 6677\ N$$

$$\left(\frac{F_a}{F_r}\right)_O = \frac{2422 + \dfrac{0.47 \times F_{r_B}}{K_B}}{3491} = 1.2556 > e = 0.37 \rightarrow X = 0.4,\ Y = 1.6$$ O

$$\left(\frac{F_a}{F_r}\right)_B = \frac{\dfrac{0.47 \times F_{r_O}}{K_O}}{6677} = \frac{1025}{6677} = 0.15 < e = 0.37 \rightarrow X = 1.0,\ Y = 0.0$$ B

$$F_{eO} = 0.4 \times F_{r_O} + 1.6 \times \left(\frac{0.47 \times F_{r_B}}{K_B} + F_a\right) = 0.4 \times 3491 + 1.6 \times \left(\frac{0.47 \times F_{r_B}}{1.6} + 2422\right)$$

$$F_{eO} = 0.4 \times 3491 + 1 \times (0.47 \times 6677 + 1.6 \times 2422) = 8410\ N$$

$$af = 1.0 \rightarrow F_{eqvO} = F_{eO} = 8410\ N$$

$$L_{hrs} = \left(\frac{C}{F_{eqv}}\right)^a \times \frac{16667}{n_{rpm}}$$

$$L_{hrs_O} = \left(\frac{44000}{8410}\right)^{\frac{10}{3}} \frac{16667}{500} = 8\,286\ hrs < 10\,000\ hrs \rightarrow \text{ FAILS.}$$

$$Try \quad SKF\ 32207 \quad (C = 56\,000\ N)$$

$$F_{eO} = 0.4 \times 3491 + 1.0 \times (0.47 \times 6677 + 1.6 \times 2422) = 8410\ N$$

$$L_{hrs_O} = \left(\frac{56000}{8410}\right)^{\frac{10}{3}} \frac{16667}{500} = 18\,515\ hrs > 10\,000\ hrs \rightarrow \text{OK} \quad \text{at O} \quad \text{SKF 32207.}$$

$$F_{eB} = 1.0 \times F_{r_B} + 0 \times \left(\frac{0.47 \times F_{r_O}}{K_O} + F_a\right) = F_{r_B} = 6677\ N$$

$$SKF\ 30207 \quad (C = 44\,000\ N)$$

$$L_{hrs_B} = \left(\frac{44000}{6677}\right)^{3.33} \frac{16667}{500} = 17\,882\ hrs > 10\,000\ hrs \rightarrow \text{OK} \quad \text{at B} \quad \text{SKF 30207.}$$

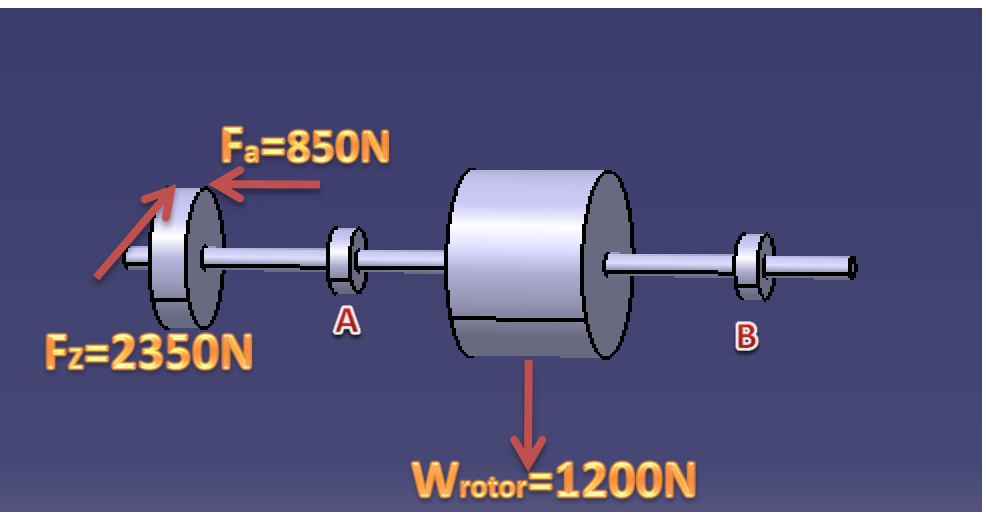
Example 3.5

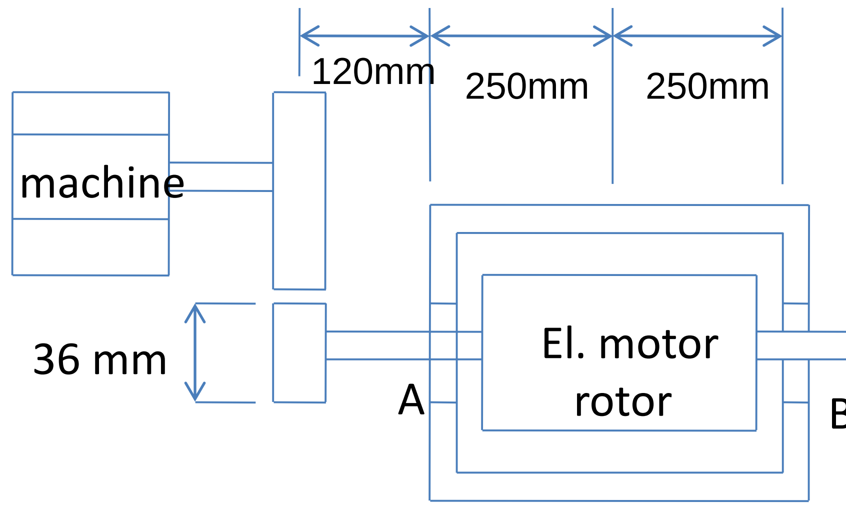
Example for ball bearings

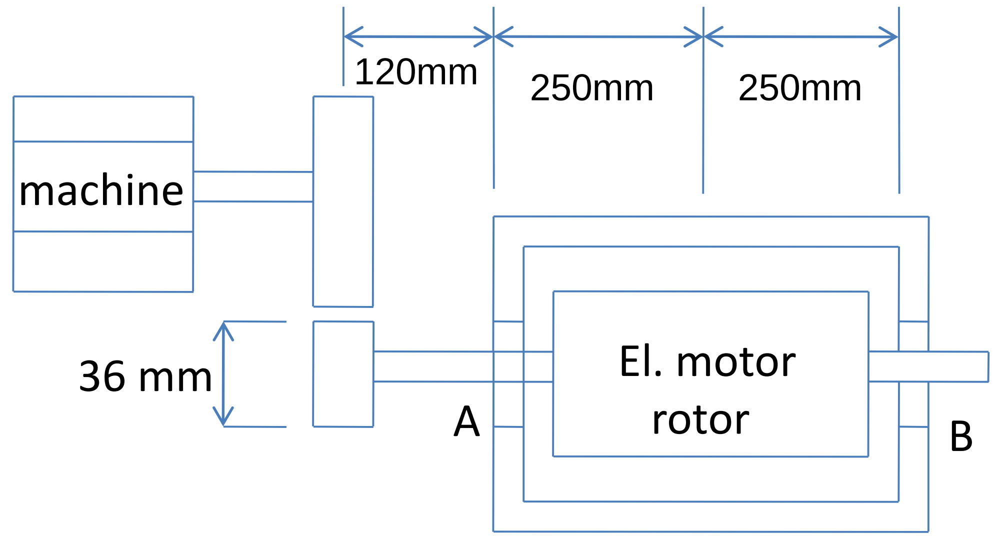
The electric motor rotor rests on two bearings at A and B. It drives a machine via a pair of helical gear. The rotor itself weighs 1200 N and is carried by the two bearings symmetrically.

At the mesh point of the two gears a tangential load of 2350 N and an axial load of 850 N exist as shown in the figure. The diameters of the rotor at bearings points are 40 mm and in between where rotor is fixed, it is 45 mm. The motor power is 24.5 HP and rotor rotates at 1500 rpm.

Based on these information select bearings at A and B.







The required life for the bearing is not stated in the question.
In such cases we can make use of suggestions given in Tables.
Determine the required life for the electric motor from Table 11.6

- *Machines for intermittent service, reliable 8-14 khr (L_{10})*
- *Machine for 8-h service not always fully utilized 14-20 khr*

A bearing with rating load C will be chosen from catalogue and life will be checked.

$$F_e = ?,\ \ F_e = V \times XF_r + YF_a$$

$$F_{eqv} = af \times F_e \quad af = ? \quad 1.2$$

$$L_{hrs} = \left(\frac{C}{F_{eqv}} \right)^a \frac{16667}{n_{rpm}}$$

$$F_r = ?$$ At A & B

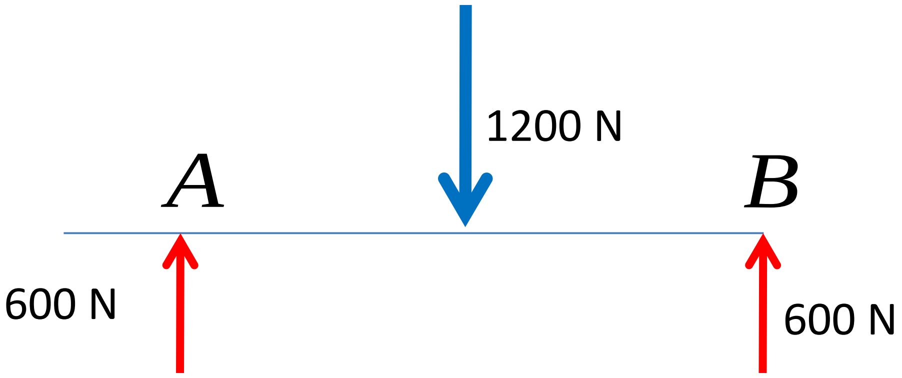
$$F_a = ?$$

Given 1500rpm

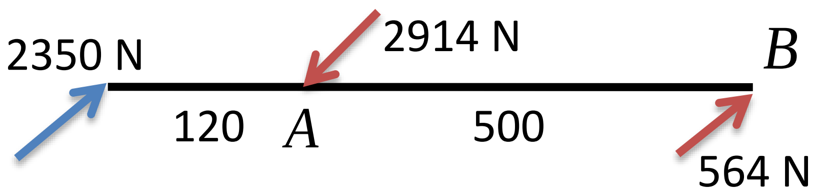
Table 11-6 BEARING-LIFE RECOMMENDATIONS FOR VARIOUS CLASSES OF MACHINERY

Type of application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service which are not always fully utilized	14–20
Machines for 8-h service which are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

Due to weight of the rotor (1200 N)



Due to tangential load at gear (2350N)



$$\sum M_A = 0;$$

$$2350 \times 120 = F_{Bz} \times 500$$

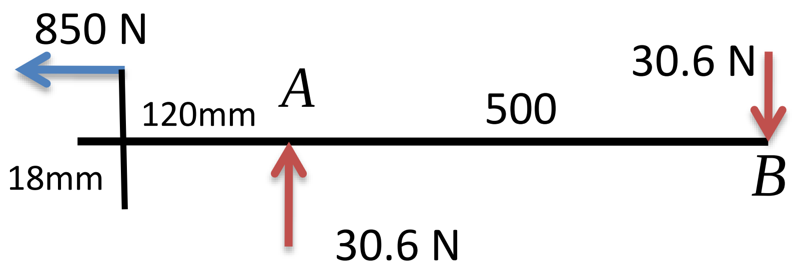
$$F_{Bz} = 564\ N$$

$$\sum F_Z = 0;$$

$$F_{Az} = 2350 + F_{Bz}$$

$$F_{Az} = 2350 + 564 = 2914\ N$$

Due to axial load (850N)



$$\sum M_A = 0;$$

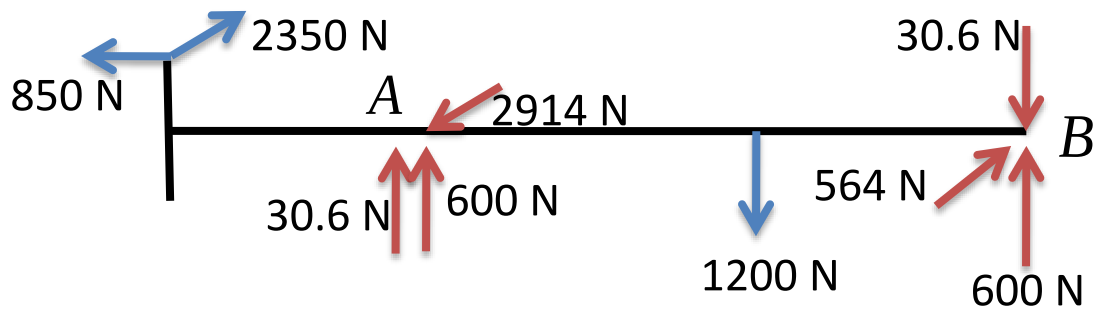
$$850 \times 18 = F_{By} \times 500$$

$$F_{By} = 30.6\ N$$

$$\sum F_Y = 0;$$

$$F_{Ay} = F_{By} = 30.6\ N$$

Resultant Forces at points A and B



$$R_A = \sqrt{(R_{Ay})^2 + (R_{Az})^2}$$

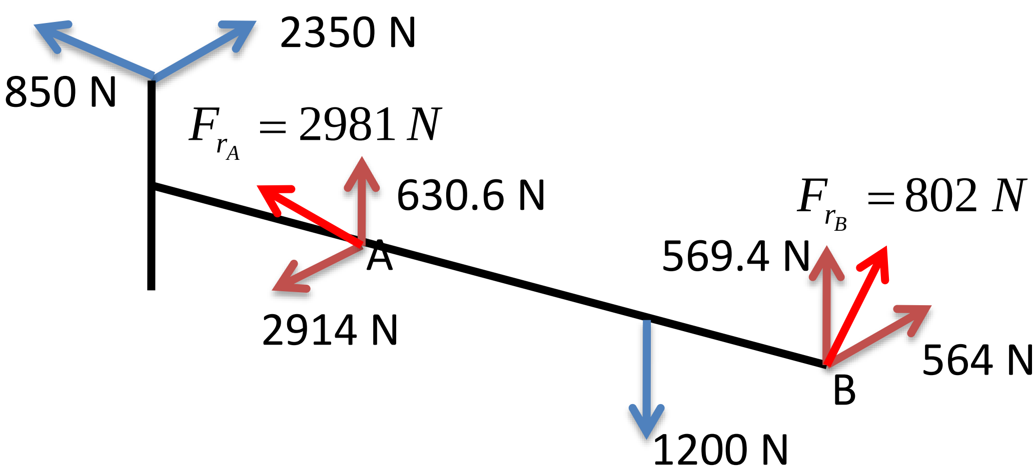
$$R_A = \sqrt{(630.6)^2 + (2914)^2}$$

$$R_A = 2981\ N$$

$$R_B = \sqrt{(R_{By})^2 + (R_{Bz})^2}$$

$$R_B = \sqrt{(569.4)^2 + (564)^2}$$

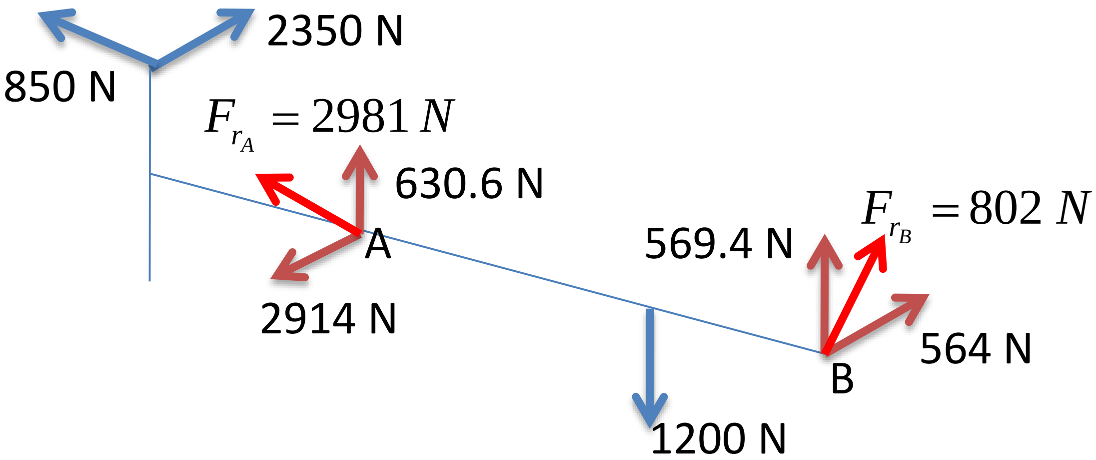
$$R_B = 802\ N$$



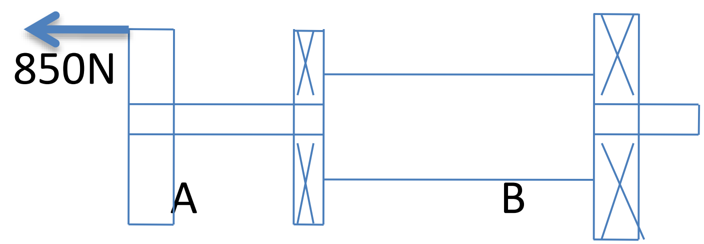
$$F_{a_A} = 850\ N$$

$$F_{r_A} = 2981\ N$$

$$F_{r_B} = 802\ N$$



Due to geometry of the EM rotor and bearing configuration bearing A should carry the axial load 850 N.



Thus

Bearing A $F_a = 850N$

$F_r = 2981N$

More critical due to both axial and radial loads

Bearing B $F_a = 0N$

$F_r = 802N$

For Bearing A from catalogue

With d= 40 mm available bearings of deep groove (cheap) ball type are:

6008	C=12900 N	C_o= 9300 N
6208	C=23600 N	C_o=16600 N
6308	C=31500 N	C_o=22400 N
6408	C=49000 N	C_o=36500 N

Let's use **6208**; C=23600 N   d= 40 mm.

6008	C=12900 N	C_o= 9300 N
6208	C=23600 N	C_o=16600 N
6308	C=31500 N	C_o=22400 N
6408	C=49000 N	C_o=36500 N

$$\frac{F_a}{C_o} = \frac{850}{16600} = 0.0512 \rightarrow e \cong 0.25$$

$$\frac{F_a}{F_r} = \frac{850}{2981} = 0.285 > e = 0.25$$

$$So; \rightarrow X = 0.56, \quad Y = 1.7$$

$$F_e = V \times XF_r + YF_a = 1.0 \times 0.56 \times 2981 + 1.7 \times 850$$

$$F_e = 3114\ N\ (> F_r)$$

$$F_{eqv} = af \times F_e = 1.2 \times 3114 = 3737\ N$$

$$L_{hrs} = \left(\frac{23600}{3737}\right)^3 \times \frac{16667}{1500} = 2798\ hrs <<< 8-14\ khrs$$ Not satisfactory

Re-choose 6308

$$C = 31500N, \quad C_0 = 22400N$$

$$F_a = 850N$$

$$F_r = 2981N$$

$$\frac{F_a}{C_0} = 0.038 \rightarrow e \cong 0.24; \quad \frac{F_a}{F_r} = 0.285 > e; \quad X = 0.56, \quad Y = 1.8$$

$$F_e = 0.56 \times 2981 + 1.8 \times 850 = 3200\ N(> F_r)$$

$$F_{eqv} = 1.2 \times 3200 = 3840\ N$$

$$L_{hrs} = \left(\frac{31500}{3840}\right)^3 \frac{16667}{1500} = 6133\ hrs < 8\ khrs$$

Not satisfactory

Re-choose 6408

$$C = 49000N, \quad C_0 = 36500N$$

$$\frac{F_a}{C_0} = 0.023 \rightarrow e \cong 0.22; \quad \frac{F_a}{F_r} = 0.285 > e; \quad X = 0.56, \quad Y = 2.0$$

$$F_e = 0.56 \times 2981 + 2.0 \times 850 = 3370N(> F_r)$$

$$F_{eqv} = 1.2 \times 3370 = 4044\ N$$

$$L_{hrs} = \left(\frac{49000}{4044}\right)^3 \frac{16667}{1500} = 19766\ hrs > 14\ khrs$$

satisfactory

(use 6408 deep groove for both A & B)

Time Varying Loads

If the load on the bearing is not constant over the life-time of the bearing but varies with time or revolution of bearing then we have to find an equivalent or mean load which is assumed to be constant over the life-time of the bearing and use it in following calculations.

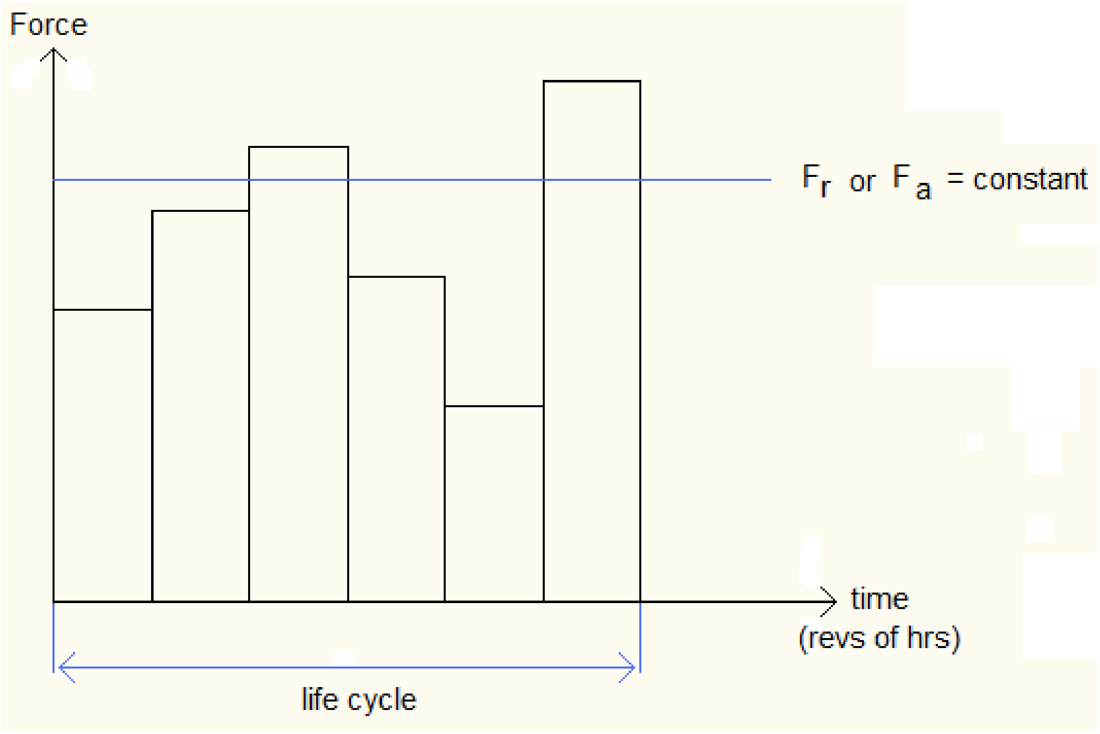


Fig. 3.2 Time varying loads on bearings

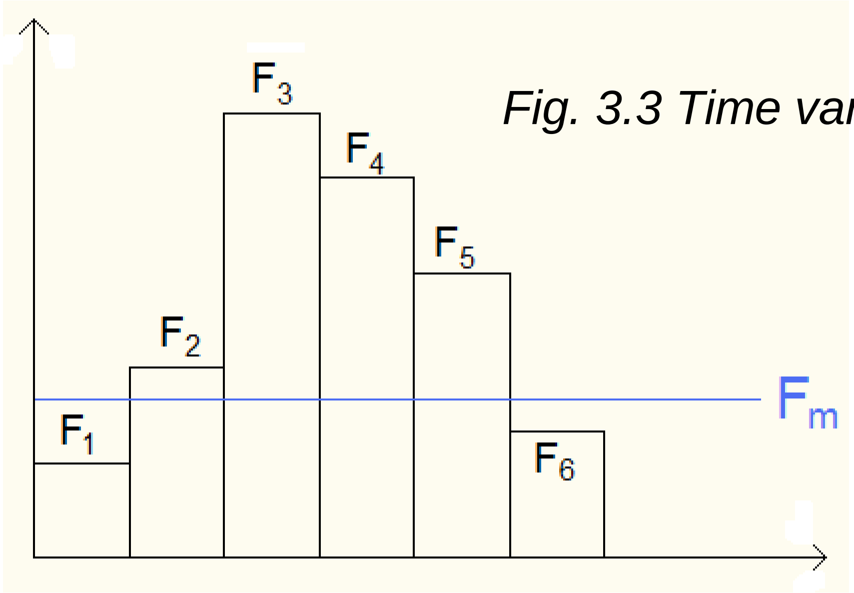


Fig. 3.3 Time varying loads on bearings

By using the load life relation of the bearing:

Load	Revs, hrs
F_1	N_1
F_2	N_2
F_3	N_3
F_4	N_4
.	.
.	.

$$L_1^a \cdot N_1 = L_2^a \cdot N_2$$

$$F_1^{\,a} \cdot N_1 = F_2^{\,a} \cdot N_2 = \sum_i^n F_i^{\,a} \cdot N_i$$

$$F_m^{\,a} \cdot N = F_1^{\,a} \cdot N_1 + F_2^{\,a} \cdot N_2 + F_3^{\,a} \cdot N_3 + \cdots$$

$$F_m^{\,a} \cdot N = \sum F_i^{\,a} \cdot N_i$$

or

$$F_m = \left[\frac{\Sigma F_i^{\,a} . N_i}{N} \right]^{\frac{1}{a}}$$

where

$$N = N_1 + N_2 + N_3 + \ldots\ldots$$

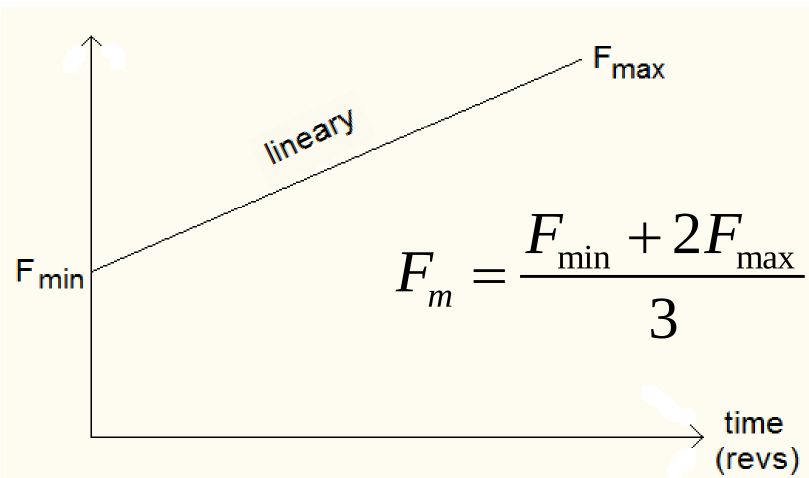
a=3 for balls
a=10/3 for rollers

Mean load can be found for varying axial load as well as for varying radial load

Then use F_m in other equations such as

$$F_{eqv} = V.XF_{mr} + Y.F_{ma}$$

$$L_{revs} = \left(\frac{C}{F_{eqv}} \right)^a x10^6$$



$$F_m = \frac{F_{\min} + 2F_{\max}}{3}$$

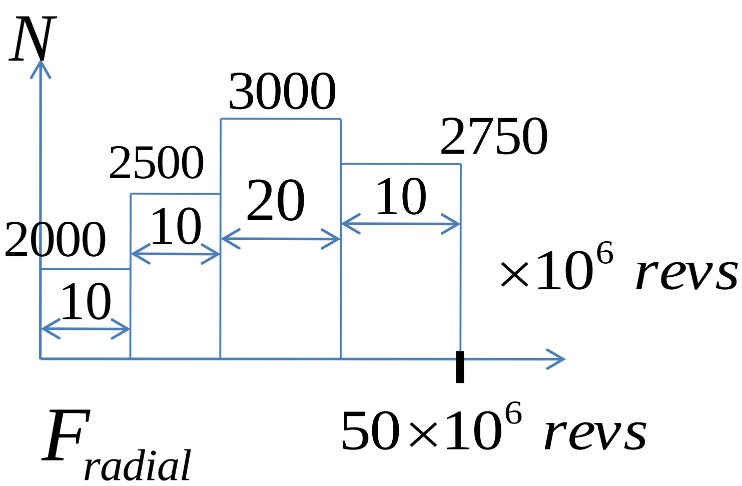
Fig. 3.4 Time varying loads on bearings

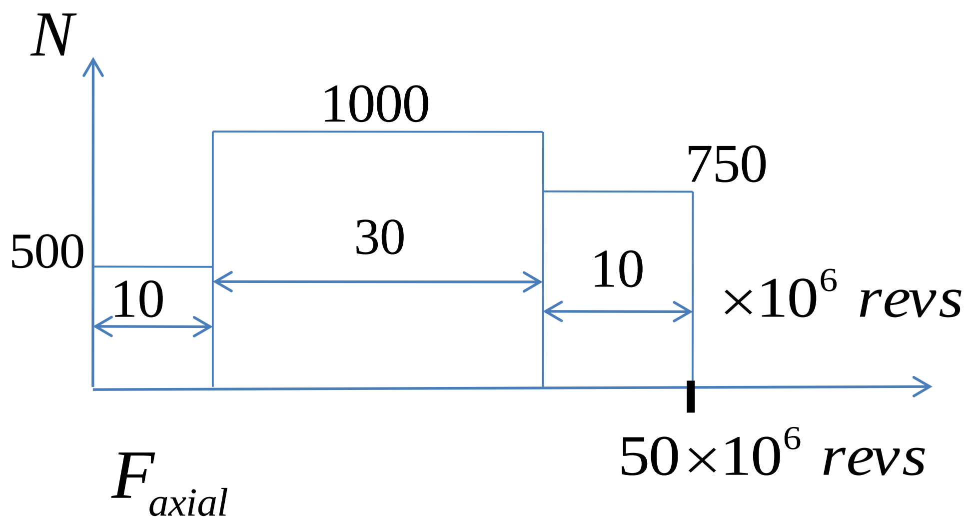
Example 3.6 Example for varying load application

A bearing during operation carries loads as shown in figures for its full life. Both the radial & the axial loads are of varying nature.
Choose suitable bearings for the application if the shaft speed is 1500 rpm and the shaft diameter is:

a) 40 mm b) 35 mm c) 25 mm

i) Deep groove balls
ii) Angular contact balls
iii) Cylindrical roller (straight)





Since the loads are varying type we have to calculate mean values for both radial and axial loads F_{m_r} and F_{m_a}

$$F_m = \left[\frac{\sum F_i^{\,a} \cdot N_i}{N} \right]^{\frac{1}{a}}$$

let's assume that we will use ball bearings: so a= 3.

$$F_{m_r} = \left[\frac{(2000^3 \times 10 \times 10^6) + (2500^3 \times 10 \times 10^6) + (3000^3 \times 20 \times 10^6) + (2750^3 \times 10 \times 10^6)}{(10+10+20+10) \times 10^6} \right]^{\frac{1}{3}} = 2700\ N$$

$$F_{m_a} = \left[\frac{(500^3 \times 10 \times 10^6) + (1000^3 \times 30 \times 10^6) + (750^3 \times 10 \times 10^6)}{(10+30+10) \times 10^6} \right]^{\frac{1}{3}} = 892\ N$$

Now, we use F_{m_r} and F_{m_a} to calculate equivalent load F_e and the life to run for different bearings.

a) Using deep groove ball bearings (cheap and most widely used) with d= 40 mm we have alternatives of

d (mm)	*Designation*	*C (N)*		*C_o (N)*
40	61808	2400		2200
	16008	10200		7800
	6008	12900		9300
	6208	23600		16600
	6308	31500		22400
	6408	49000		36500

Try 6008 (C=12900 N)

$$\frac{F_a}{C_o} = \frac{892}{9300} = 0.096 \rightarrow e \cong 0.28$$

$$\frac{F_a}{F_r} = \frac{892}{2700} = 0.33 > e \rightarrow X = 0.56, Y \cong 1.55 \text{ and let } af = 1.2$$

$$F_e = 0.56 \times 2700 + 1.55 \times 892 = 2895\, N \; (> F_r = 2700\; N)$$

$$F_{eqv} = af \times F_e = 1.2 \times 2895 = 3475\, N$$

$$L_{hrs} = \left(\frac{C}{F_{eqv}} \right)^a \times \frac{16667}{n_{rpm}} = \left(\frac{12900}{3475} \right)^3 \frac{16667}{1500} = 568\; hrs$$

$$= 568\; hrs \times \frac{60\; min}{1\, hrs} \times \frac{1500\; revs}{min} = 51157980\; revs = 51.15 \times 10^6\; revs > \sum N_i$$

Since $L_{revs} = 51.15 \times 10^6 \geq 50 \times 10^6 \; total\; life \quad \underline{\underline{OK}}$

OR

$$L_{revs} = \left(\frac{C}{F_{eqv}} \right)^a \times 10^6 \ revs = \left(\frac{12900}{3475} \right)^3 \times 10^6$$

$$= 51.15 \times 10^6 \ revs > \sum N_i \quad \underline{\underline{OK}}$$

Do other diameters yourself

Try also

ii) angular contact ball bearings?

iii) cylindrical roller (straight) bearings (a=10/3) ? ! !

ii) Choose angular contact ball bearings with d= 40 mm

SKF

d, mm	designation	C , N	C_0 , N
40	7208B	24500	18600
	7308B	34500	25000

e= 1.14

$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$	
X	Y	X	Y
1	0	0.35	0.57

$$\frac{F_a}{F_r} = \frac{892}{2700} = 0.33 < e = 1.14 \rightarrow X = 1.0, Y = 0.0, Let\ af = 1.2$$

$$F_e = X \times F_r = 2700\ N \rightarrow F_{eqv} = 1.2 \times 2700 = 3240\ N$$

$$C_{req} = \left(\frac{L_{hrs} \times n_{rpm}}{16667} \right)^{\frac{1}{a}} \times F_{eqv} \quad or \quad C_{req} = \left(\frac{L_{revs}}{10^6} \right)^{\frac{1}{a}} \times F_{eqv}$$

$$C_{req} = \left(\frac{50 \times 10^6}{10^6} \right)^{\frac{1}{3}} \times 3240 = 11935 \rightarrow 7208B\ with\ C = 24500\ N$$

will satisfy

$$L_{revs} = \left(\frac{C_{req}}{F_{eqv}} \right)^{a} \times 10^6 = \left(\frac{24500}{3240} \right)^{3} \times 10^6 = 432 \times 10^6\ revs > 50 \times 10^6\ revs$$

iii) Cylindrical straight roller bearing!!