List of Equations & Tables

Tolerances & Fits:

Holes:

$$\boxed{D_{max} = D + \Delta D; \quad D_{min} = D}$$

Shafts:

Clearance:

$$\boxed{d_{max} = d + \delta; \quad d_{min} = d + \delta - \Delta d}$$

Interference:

$$\boxed{d_{max} = d + \delta + \Delta d; \quad d_{min} = d + \delta}$$

Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

Table A–11

A Selection of International Tolerance Grades—Metric Series (Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Millimeters)

Source: Preferred Metric Limits and Fits, ANSI B4.2-1978. See also BSI 4500.

Basic Sizes	Tolerance Grades IT6	IT7	IT8	IT9	IT10	IT11
0–3	0.006	0.010	0.014	0.025	0.040	0.060
3–6	0.008	0.012	0.018	0.030	0.048	0.075
6–10	0.009	0.015	0.022	0.036	0.058	0.090
10–18	0.011	0.018	0.027	0.043	0.070	0.110
18–30	0.013	0.021	0.033	0.052	0.084	0.130
30–50	0.016	0.025	0.039	0.062	0.100	0.160
50–80	0.019	0.030	0.046	0.074	0.120	0.190
80–120	0.022	0.035	0.054	0.087	0.140	0.220
120–180	0.025	0.040	0.063	0.100	0.160	0.250

Δ

Basic Sizes	Upper-Deviation Letter c	d	f	g	h	Lower-Deviation Letter k	n	p	s	u
0–3	−0.060	−0.020	−0.006	−0.002	0	0	+0.004	+0.006	+0.014	+0.018
3–6	−0.070	−0.030	−0.010	−0.004	0	+0.001	+0.008	+0.012	+0.019	+0.023
6–10	−0.080	−0.040	−0.013	−0.005	0	+0.001	+0.010	+0.015	+0.023	+0.028
10–14	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
14–18	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
18–24	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.041
24–30	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.048
30–40	−0.120	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.060
40–50	−0.130	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.070
50–65	−0.140	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.053	+0.087
65–80	−0.150	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.059	+0.102
80–100	−0.170	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.071	+0.124
100–120	−0.180	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.079	+0.144
120–140	−0.200	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.092	+0.170

δ

Mechanisms

Gruebler's Equation:

$$m = 3(n - g - 1) + \sum_{i=1}^{g} f_i$$

Where:

m = mobility, *n* = # of links

g = # of joints, *f* = DOF of joint *i*

Grashof's Criteria for type 1:

s + *l* < *p* + *q*

Where:

s = shortest link length, *l* = longest link length

p, q = lengths of other links

subtype:

1. Shortest link is the base: Double Crank
2. Shortest link is attached to base: Crank-Rocker
3. Shortest link is coupler: Double Rocker

Grashof Type 2: s + l > p + q

Linkages have no rotatable joints.

Grashof Neutral: s + l = p + q

Linkage can be 'flattened'.

Function Generation:

Chebyshev Spacing: For **n** precision points

$$x_i = \frac{x_f + x_0}{2} - \frac{x_f - x_0}{2}\cos\left(\frac{\pi}{2n}(2i-1)\right)$$

Where:

X_0 and x_f are starting and ending points, respectively

Precision points are given by: i = 1, 2,…, n

$$\theta_i = \frac{\theta_f - \theta_0}{x_f - x_0}(x_i - x_0) + \theta_0 \quad E$$

$$\varphi_i = \frac{\varphi_f - \varphi_0}{y_f - y_0}(y_i - y_0) + \varphi_0$$

Loop closure:

$$\boxed{\begin{bmatrix} 1 & \cos(\varphi_1) & -\cos(\theta_1) \\ 1 & \cos(\varphi_2) & -\cos(\theta_2) \\ 1 & \cos(\varphi_3) & -\cos(\theta_3) \end{bmatrix} \begin{Bmatrix} \lambda_1 \\ \lambda_2 \\ \lambda_3 \end{Bmatrix} = \begin{Bmatrix} \cos(\theta_1 - \varphi_1) \\ \cos(\theta_2 - \varphi_2) \\ \cos(\theta_3 - \varphi_3) \end{Bmatrix}}$$

$$\boxed{r_2 = \frac{1}{\lambda_2}}$$

$$\boxed{r_3 = \sqrt{1 + r_4^2 + r_2^2 - 2r_4 r_2 \lambda_1}}$$

$$\boxed{r_4 = \frac{1}{\lambda_3}}$$

– Note: these equations are scaled such that $|r_1| = 1$

Cam and Follower Motions:

θ is the angle from the start of each motion

β : is the total rise or return angle (Rad. or Degree)   L : is the total change height (m)

$\dot{y} = (dy/dt)$: follower velocity (m/s) $\ddot{y} = (d^2y/dt^2)$: follower acceleration (m/s^2)

ω: angular velocity (Rad./s)

$y' = (dy/d\theta)$ $y'' = (d^2y/d\theta^2)$

Uniform Motion:

$$\boxed{y = \frac{L}{\beta}\theta} \qquad \boxed{\dot{y} = \frac{L}{\beta}\omega} \qquad \boxed{\ddot{y} = 0}$$

Parabolic Motion:

Rise, First Half :

$$\boxed{y = \frac{2L}{\beta^2}\theta^2;} \boxed{\dot{y} = \frac{4L\omega}{\beta^2}\theta; \qquad \ddot{y} = \frac{4L\omega^2}{\beta^2}}$$

Rise, Second Half:

$$\boxed{y = L\left[1 - 2\left(1 - \frac{\theta}{\beta}\right)^2\right]; \quad \dot{y} = \frac{4L\omega}{\beta}\left(1 - \frac{\theta}{\beta}\right); \quad \ddot{y} = -\frac{4L\omega^2}{\beta^2}}$$

Return, First Half:

$$\boxed{y = L\left[1 - 2\left(\frac{\theta}{\beta}\right)^2\right] \quad \text{for} \quad 0 \le \theta \le \frac{\beta}{2}} \qquad \boxed{\dot{y} = \frac{4L\omega}{\beta^2}\theta; \qquad \ddot{y} = \frac{4L\omega^2}{\beta^2}}$$

Return, Second Half

$$\boxed{y = 2L\left[1 - \frac{\theta}{\beta}\right]^2 \quad \text{for} \quad \frac{\beta}{2} \le \theta \le \beta} \qquad \boxed{y = L\left[1 - 2\left(1 - \frac{\theta}{\beta}\right)^2\right]; \quad \dot{y} = -\frac{4L\omega}{\beta}\left(1 - \frac{\theta}{\beta}\right); \quad \ddot{y} = \frac{4L\omega^2}{\beta^2}}$$

Harmonic Motion:

Rise:

$$\boxed{y = \frac{L}{2}\left(1 - \cos\left(\frac{\pi\theta}{\beta}\right)\right)} \boxed{\dot{y} = \frac{\pi L\omega}{2\beta}\sin\left(\frac{\pi\theta}{\beta}\right)} \boxed{\ddot{y} = \frac{L}{2}\left(\frac{\pi\omega}{\beta}\right)^2 \cos\left(\frac{\pi\theta}{\beta}\right)}$$

Return:

$$\boxed{y = \frac{L}{2}\left(1 + \cos\left(\frac{\pi\theta}{\beta}\right)\right)} \quad \boxed{\dot{y} = -\frac{\pi L\omega}{2\beta}\sin\left(\frac{\pi\theta}{\beta}\right)} \boxed{\ddot{y} = -\frac{L}{2}\left(\frac{\pi\omega}{\beta}\right)^2 \cos\left(\frac{\pi\theta}{\beta}\right)}$$

Gears:

$$i = \frac{\omega_{in}}{\omega_{out}} = \frac{D_{out}}{D_{in}} = \frac{N_{out}}{N_{in}} = \frac{T_{out}}{T_{in}}$$

$$\boxed{T = rF}$$

$$P = \frac{N}{D}; \quad m = \frac{D}{N}$$

Compound Gear Train:

$$\boxed{i = i_{12} \times i_{34} = \frac{\omega_1}{\omega_2} \times \frac{\omega_3}{\omega_4} = \frac{N_2}{N_1} \times \frac{N_4}{N_3}}$$

Reverted:

$$\boxed{r_1 + r_2 = r_3 + r_4}$$

Power:

$$\boxed{P = \frac{\Delta W}{\Delta t}} \quad \boxed{P = T\omega} \quad \boxed{1\ [\text{RPM}] \times 2\pi\ [\text{rad/rev}] \times 1/60\ [\text{min/sec}] = 1\ [\text{rad/sec}]}$$

1 HP = 745.7 Watts (J/s) = 33,000 lb*ft / min

$$\boxed{T\ [\text{in} \cdot \text{lbs}] = \frac{D\ [\text{in}]}{2} W\ [\text{lbs}]} \quad \boxed{HP = T[\text{in} \cdot \text{lbs}] \cdot \frac{\text{RPM}}{63025}}$$

Gear Strength:

$$T[\text{in} \cdot \text{lbs}] = \frac{D}{2} W$$

$$hp = T[\text{in} \cdot \text{lbs}] \left(\frac{RPM}{63025}\right)$$

$$W[\text{lbs}] = \frac{S \cdot FW \cdot Y}{P} \left(\frac{600}{600 + V}\right)$$

$$V[\text{ft/min}] = 0.262 \cdot D \cdot RPM$$

Where:

- W= Tooth Load (Lbs.) (along the Pitch Line)
- S = Safe Material Stress (static) (psi)
- FW = Face Width (in.)
- Y = Tooth Form Factor
- P = Diametral Pitch
- V = Pitch Line Velocity (Ft. per Min.) = .262 x D x RPM
- D = Pitch Circle Diameter (in.)

Table 1: Tooth Form Factors for Lewis Formula

# Teeth	14.5° PA	20° PA	# Teeth	14.5° PA	20° PA	# Teeth	14.5° PA	20° PA
10	0.176	0.201	24	0.302	0.337	60	0.355	0.421
11	0.192	0.226	26	0.308	0.344	65	0.358	0.425
12	0.21	0.245	28	0.314	0.352	70	0.36	0.429
13	0.223	0.264	30	0.318	0.358	75	0.361	0.433
14	0.236	0.276	32	0.322	0.364	80	0.363	0.436
15	0.245	0.289	34	0.325	0.37	90	0.366	0.442
16	0.255	0.295	36	0.329	0.377	100	0.368	0.446
17	0.264	0.302	38	0.332	0.383	150	0.375	0.458
18	0.27	0.308	40	0.336	0.389	200	0.378	0.463
19	0.277	0.314	45	0.34	0.399	300	0.382	0.471
20	0.283	0.32	50	0.346	0.408	Rack	0.39	0.484
22	0.292	0.33	55	0.352	0.415			

Table 2: Safe Material Stress

Material	(S) psi	Material	(S) psi
Plastic	5000	Steel .20 Carbon (Case-hardened)	25000
Bronze	10000	Steel .40 Carbon (Untreated)	25000
Cast Iron	12000	Steel .40 Carbon (Heat-treated)	30000
Steel .20 Carbon (Untreated)	20000	Steel .40 C. Alloy (Heat-treated)	40000

Planetary Gears:

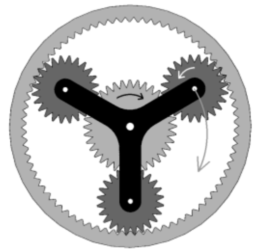
Na = Number teeth on annulus, Ns = number of teeth on sun,

Nc = Na + Ns (represents carrier teeth).

Gear ratio calculated as N_{out} / N_{in} as usual.

if none of them locked

$$\boxed{i = -\frac{\omega_s - \omega_c}{\omega_p - \omega_c} \times \frac{\omega_p - \omega_c}{\omega_a - \omega_c} = -\frac{\omega_s - \omega_c}{\omega_a - \omega_c}}$$



i: speed ratio
ω_s : angular velocity of sun gear
ω_a : angular velocity of annulus gear
ω_p : angular velocity of planet gear
ω_c : angular velocity of planet carrier

Band Analysis:

Torque of belt on drum:

$$\boxed{T = (F_1 - F_2)\frac{D}{2}}$$

Max contact pressure:

$$\boxed{p_{\max} = \frac{2F_1}{bD}}$$

Max belt tension:

$$\boxed{F_1 = F_2 \cdot \exp(\mu\theta)}$$

Power transmitted:

$$\boxed{H = T\omega = \omega(F_1 - F_2)\frac{D}{2}}$$

Where:
T = Torque, b = belt width, F1 = Max Belt Tension, F2 = Actuating Force Tension, μ = coefficient of friction, θ = wrap angle

Belt analysis

pully of Diameter D is rotating with ω (rad/s)

the non slip belt moving speed V = ω * D/2

F_c (The centrifugal force) = m * V^2

Where m is The belt mass per unit length

Belt equation $\frac{F_1 - F_c}{F_2 - F_c}$ = exp (μ θ) Where θ is the wrap angle

T (Torque) = $(F_1 - F_2)$ * D/2

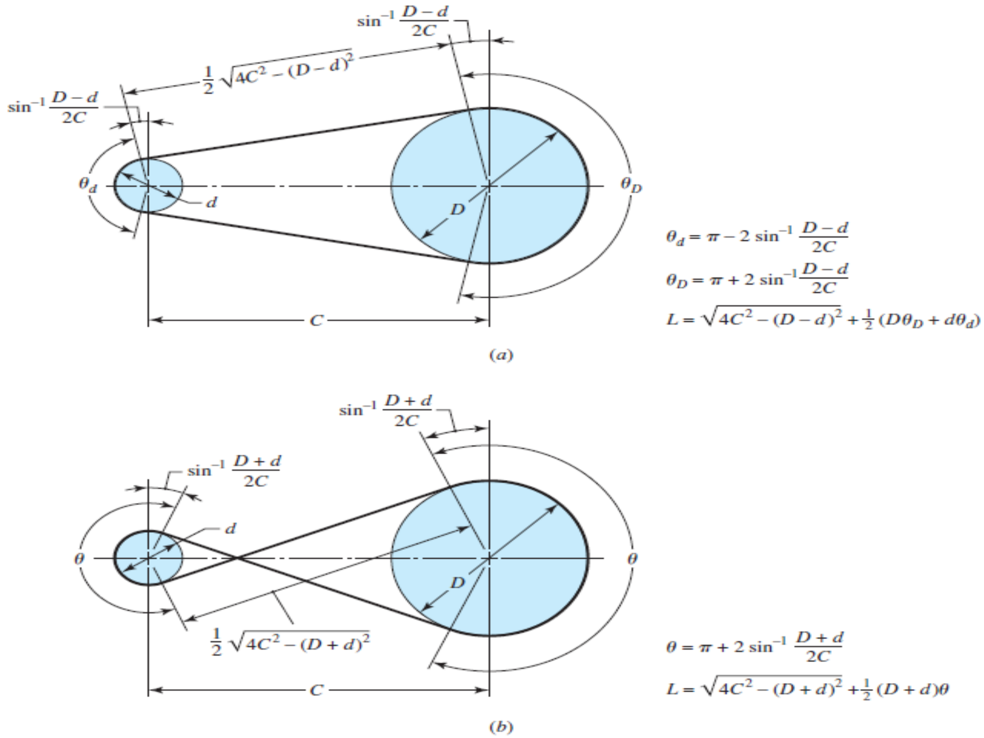
Power = T * ω = $(F_1 - F_2)$ * ω

F_i = initial tension on the belt to avoid slip on the pully =

$$F_i = \frac{F_1 + F_2}{2} - F_c$$

Belt length:

Flat-belt geometry. (*a*) Open belt. (*b*) Crossed belt.



Fluid Power:

$$\boxed{p = \frac{F_{\text{in}}}{A_{\text{in}}} = \frac{F_{\text{out}}}{A_{\text{out}}}} \quad \boxed{d_{\text{out}} = d_{\text{in}}\frac{A_{\text{in}}}{A_{\text{out}}}} \quad \boxed{W = Fd} \quad \boxed{P = Fv\,[Nm/s = W]}$$

$$\boxed{MA = \frac{F_{\text{out}}}{F_{\text{in}}} = \frac{A_{out}}{A_{in}}}$$

Where:

p = pressure, F = force, A = area, d = piston displacement

W= The work done, P = power, υ = velocity, MA = mechanical advantage

Chains in bikes

$$\boxed{\frac{\omega_{rear}}{\omega_{front}} = \frac{N_{front}}{N_{rear}} = \frac{T_{front}}{T_{rear}}}$$

Power = T * ω $V_{bike} = \omega_o$ * Rear wheel radius

Bolt Loading:

$F_p = (S_p * A_t)$ / factor of safety

Where:
F_p = proof load per bolt
S_p = proof stress (grade table)
A_t = tensile stress area (from standard table 8.1 or 8.2)

Clamping preload force (if no gasket)
F_i = bolt preload
$F_i = 0.75F_p$ for non-permanent joints
$F_i = 0.9F_p$ for permanent joints

Clamping preload force (if there is a gasket)
F_i = bolt preload = maximum allowable pressure on the gasket * Cross section area of the gasket

Clamping Torque
T = 0.2 blot diameter * F_i*

Bolted Joints in Tension:

$$F_b = F_i + P\frac{k_b}{k_j+k_b}$$

$$F_j = F_i - P\frac{k_j}{k_j+k_b}$$

Where:
F_b = force in all bolts, F_j = force in the joint
k_b = bolt stiffness, k_j = joint stiffness
P = total external load, F_i = preload on all bolts

Grade	Proof	Yield	Ultimate
SAE Grade 2	55 ksi	57 ksi	74 ksi
SAE Grade 5	85 ksi	92 ksi	120 ksi
SAE Grade 8	120 ksi	130 ksi	150 ksi
ISO Grade 8.8	600 MPa	640 MPa	800 MPa
ISO Grade 10.9	830 MPa	940 MPa	1040 MPa

Table 8–1

Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series: Pitch p mm	Coarse-Pitch Series: Tensile-Stress Area A_t mm²	Coarse-Pitch Series: Minor-Diameter Area A_r mm²	Fine-Pitch Series: Pitch p mm	Fine-Pitch Series: Tensile-Stress Area A_t mm²	Fine-Pitch Series: Minor-Diameter Area A_r mm²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470

Table 8–2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC Threads per Inch N	Coarse Series—UNC Tensile-Stress Area A_t in^2	Coarse Series—UNC Minor-Diameter Area A_r in^2	Fine Series—UNF Threads per Inch N	Fine Series—UNF Tensile-Stress Area A_t in^2	Fine Series—UNF Minor-Diameter Area A_r in^2
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

*This table was compiled from ANSI B1.1-1974. The minor diameter was found from the equation $d_r = d - 1.299\,038p$, and the pitch diameter from $d_p = d - 0.649\,519p$. The mean of the pitch diameter and the minor diameter was used to compute the tensile-stress area.

Adhesives – Types of Loading

Shear Loading (strongest): when force is parallel to adhesive area

$$\boxed{\tau = \frac{F}{parallel\ adhesive\ area}}$$

Tensile Loading: when force is normal to adhesive area

$$\boxed{\sigma = \frac{F}{Normal\ adhesive\ area}}$$

Peel Loading (weakest): when force is causing momentum on adhesive area

$$\boxed{W = \frac{F}{b}}$$

Where: b = width of adhesive area

Rolling and ball Bearing Life:

Dynamic loading

$$\boxed{C_{10} = LAF \cdot F_D \left(\frac{L_D n_D 60}{L_{10}}\right)^{1/a} = LAF \cdot F_D \left(\frac{L_D n_D 60}{10^6}\right)^{1/a}}$$

Where:

LAF = Load Application Factor

F_D = Desired radial load (N or lb) or F_e if there is a thrust load too

L_D = Desired Life (hours)

n_D = Desired Speed (rpm)

a = 3 for ball bearing, 10/3 for roller bearing

V = 1.0 for inner ring rotates, 1.2 if outer rotates

Equivalent Loading: (Radial and axial loading)

$$F_e = X_i V F_r + Y_i F_a$$

Where V = 1 if inner ring rotates, 1.2 if outer

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Bearing No.	Principal Dimensions (mm)			Basic Load Ratings (kN)	
	Bore Diameter	Outside Diameter	Width	C_{10}	C_0
623	3	10	4	0.63	0.218
624	4	13	5	1.3	0.485
625	5	16	5	1.88	0.68
626	6	19	6	2.34	0.885
627	7	19	6	3.35	1.4
628	8	24	8	4	1.59
629	9	26	8	4.55	1.96
6200	10	30	9	5.1	2.39
6201	12	32	10	6.1	2.75
6202	15	35	11	7.75	3.6
6203	17	40	12	9.6	4.6
6204	20	47	14	12.8	6.65
6205	25	52	15	14	7.85
6206	30	62	16	19.5	11.3
6207	35	72	17	25.7	15.3
6208	40	80	18	29.1	17.8
6209	45	52	19	32.5	20.4
6210	50	90	20	35	23.2

Plain Bushing Bearings:

P = W/A (load / projected area) A = shaft diameter * bearing length

V = ω * r (angular velocity * shaft radius)