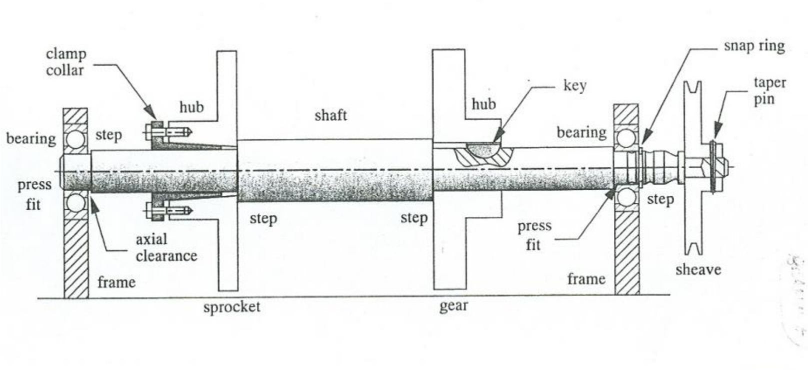
Mechanical Transmission Components – Selection and Application



Mechanical transmission components are most often COTS (commercial off the shelf), hence an engineering responsibility is a component selection which prevents failure. The engineer first determines loads transmitted through the component, required life and environmental factors, all of which can be the root cause of failure. For COTS parts the manufacturer's catalogs are the best source of design guidelines for component selection, which may often be based on an existing standard.

Contents

I. Fatigue (Progressive Growth of a Fracture)

See page of figures below. Fatigue failures have several defining characteristics:

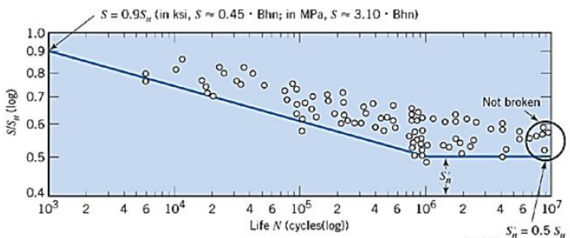


Figure 8.5 Generalized *S-N* curve for wrought steel with superimposed data points.

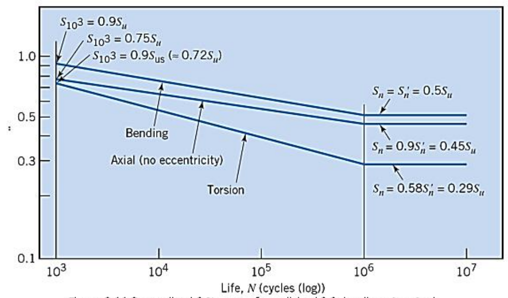
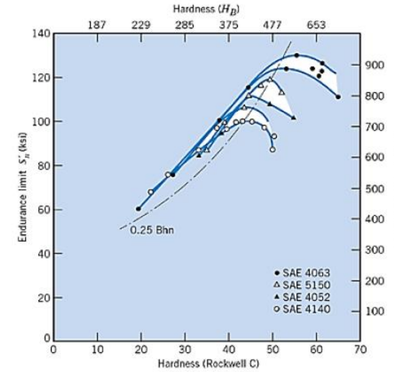


Figure 8.11 Generalized S-N curves for polished 0.3=in. diameter steel specimens (based on calculated elastic stresses ignoring possible yielding).



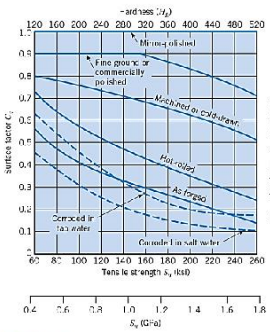


Figure 8.13 Reduction in endurance limit owing to surface finish–steel parts.

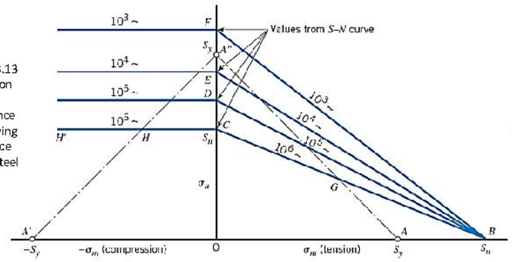


Figure 8.16 Constant-life fatigue diagram – ductile materials.

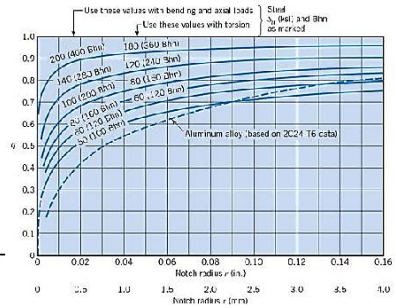
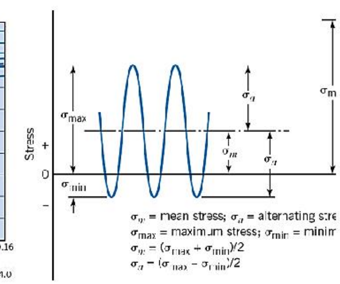


Figure 8.24 Notch sensitivity curves



σ_m = mean stress; σ_a = alternating stre
σ_{max} = maximum stress; σ_{min} = minim
$\sigma_m = (\sigma_{max} + \sigma_{min})/2$
$\sigma_a = (\sigma_{max} - \sigma_{min})/2$

TABLE 8.1 Generalized Fatigue Strength Factors for Ductile Materials (S–N curves)

a. 10^6-cycle strength (endurance limit)[a]

Bending loads: $S_n = S'_n C_L C_G C_S C_T C_R$
Axial loads: $S_n = S'_n C_L C_G C_S C_T C_R$
Torsional loads: $S_n = S'_n C_L C_G C_S C_T C_R$
where S'_n is the R.R. Moore, endurance limit,[b] and

	Bending	Axial	Torsion
C_L (load factor)	1.0	1.0	0.58
C_G (gradient factor): diameter < (0.4 in. or 10 mm)	1.0	0.7 to 0.9	1.0
(0.4 in. or 10 mm) < diameter < (2 in. or 50 mm)[c]	0.9	0.7 to 0.9	0.9
C_S (surface factor)	see Figure 8.13		
C_T (temperature factor)	Values are only for steel		
T ≤ 840 °F	1.0	1.0	1.0
840 °F < T ≤ 1020 °F	1 - (0.0032T − 2.688)		
C_R (reliability factor):[d]			
50% reliability	1.000	"	"
90% "	0.897	"	"
95% "	0.868	"	"
99% "	0.814	"	"
99.9% "	0.753	"	"

b. 10^3-cycle strength[e, f, g]

Bending loads: $S_f = 0.9S_u C_T$
Axial loads: $S_f = 0.75S_u C_T$
Torsional loads: $S_f = 0.9S_{us} C_T$
where S_u is the ultimate tensile strength and S_{us} is the ultimate shear strength.

[a] For materials not having the endurance limit, apply the factors to the 10^8 or 5×10^8-cycle strength.
[b] $S'_n = 0.5S_u$ for steel, lacking better data.
[c] For (2 in. or 50 mm) < diameter < (4 in. or 100 mm) reduce these factors by about 0.1. For (4 in. or 100 mm) < diameter < (6 in. or 150 mm), reduce these factors by about 0.2.
[d] The factor, C_R, corresponds to an 8 percent standard deviation of the endurance limit. For example, for 99% reliability we shift −2.326 standard deviations, and $C_R = 1 - 2.326(0.08) = 0.814$.
[e] No corrections for gradient or surface are normally made, but the experimental value of S_u or S_{us} should pertain to sizes reasonably close to those involved.
[f] No correction is usually made for reliability at 10^3 cycle strength.
[g] $S_{us} \approx 0.8S_u$ for steel; $S_{us} \approx 0.7S_u$ for other ductile metals.

TABLE 8.2 Construction of σ_a versus σ_m Constant Life Fatigue Diagram

Load Type	Directions
Bending	Construct diagram as shown; take points C, D, and so on from S–N curve for reversed bending.
Axial	Construct diagram as shown; take points C, and so on from S–N curve for reversed axial loads.
Torsional	Omit left half of diagram (*any* torsional mean stress is considered positive); take points C and so on from S–N curve for reversed torsion; use S_{sy} and S_{us} instead of S_y and S_u. (For steel, $S_{us} \approx 0.8S_u$, $S_{sy} \approx 0.58S_y$.)
General biaxial	Construct the diagram as for *bending* loads, and use it with *equivalent* load stresses, computed as follows. (Note that these equations apply to the generally encountered situation where σ_a and σ_m exist in one direction only. Corresponding equations for the more elaborate general case are also tentatively applicable.) 1. Equivalent alternating bending stress, σ_{ea}, is calculated from the *distortion energy theory* as being *equivalent* to the combination of existing *alternating* stresses: $\sigma_{ea} = \sqrt{\sigma_a^2 + 3\tau_a^2}$ (a) 2. Equivalent *mean* bending stress σ_{em}, is taken as the *maximum principal stress* resulting from the superposition of all existing static (mean) stresses. Use Mohr circle, or $\sigma_{em} = \frac{\sigma_m}{2} + \sqrt{\tau_m^2 + \left(\frac{\sigma_m}{2}\right)^2}$ (b) [For more complex loading, various other suggested equations for σ_{ea} and σ_{em} are found in the literature.]

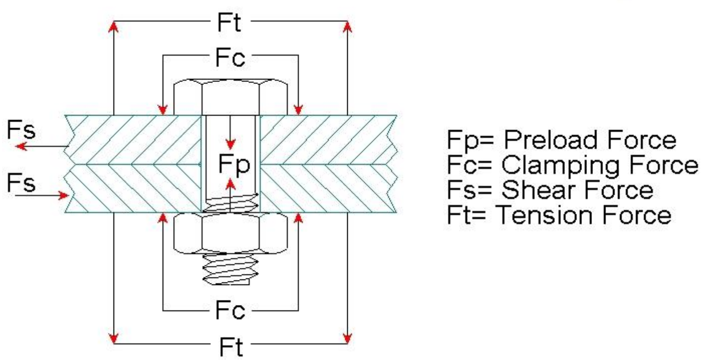
Table 8.2

II. Threaded Fasteners

TABLE 10.1 Basic Dimensions of Unified Screw Threads

Size	Major Diameter d (in.)	Coarse Threads—UNC Threads per Inch	Coarse Threads—UNC Minor Diameter of External Thread d_r (in.)	Coarse Threads—UNC Tensile Stress Area A_t (in.^{2})	Fine Threads—UNF Threads per Inch	Fine Threads—UNF Minor Diameter of External Thread d_r (in.)	Fine Threads—UNF Tensile Stress Area A_t (in.^{2})
0(.060)	0.0600	—	—	—	80	0.0447	0.00180
1(.073)	0.0730	64	0.0538	0.00263	72	0.0560	0.00278
2(.086)	0.0860	56	0.0641	0.00370	64	0.0668	0.00394
$\frac{1}{4}$	0.2500	20	0.1887	0.0318	28	0.2062	0.0364
$\frac{5}{16}$	0.3125	18	0.2443	0.0524	24	0.2614	0.0580
$\frac{3}{8}$	0.3750	16	0.2983	0.0775	24	0.3239	0.0878
$\frac{7}{16}$	0.4375	14	0.3499	0.1063	20	0.3762	0.1187
$\frac{1}{2}$	0.5000	13	0.4056	0.1419	20	0.4387	0.1599

Note: See ANSI standard B1.1-1974 for full details. Unified threads are specified as "$\frac{1}{2}$ in.–13UNC," "1 in.–12UNF."

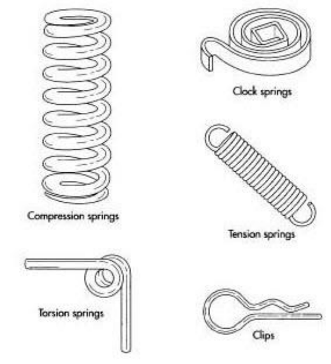


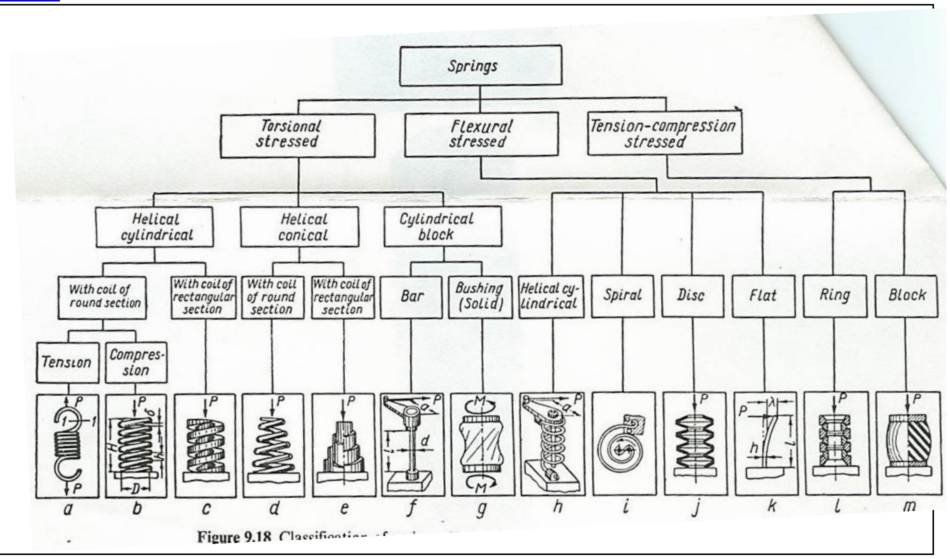
Some important equations:

III. Ball and Acme Screws

IV. Springs

Source: www.centuryspring.com





Formula for a helical spring:

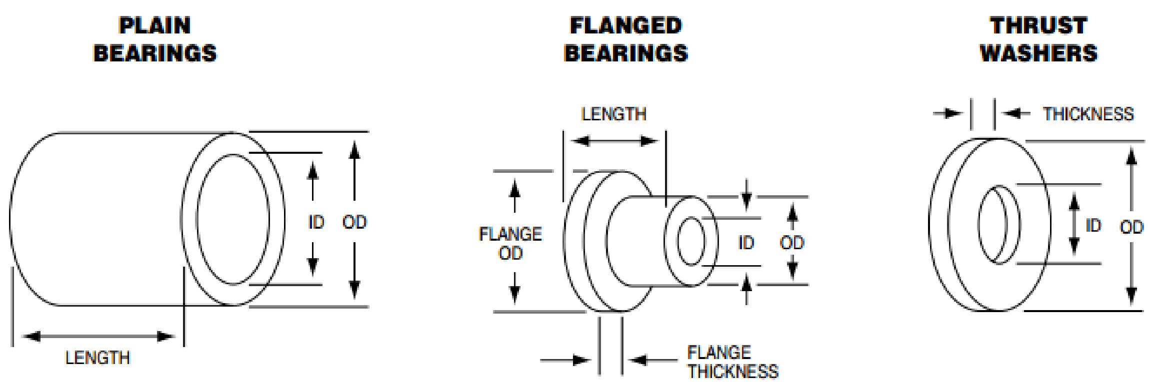
V. Bearings

A. Plain Bearings (aka Sleeve bearings, bushings)

Source: www.buntingbearings.com

Bearing Catalog Selection is based on load. PV is a means of measuring the performance capabilities of bearings. P is expressed as pressure or pounds per square inch on the projected area of the bearing. V is velocity in feet per minute of the wear surface (surface feet per minute).

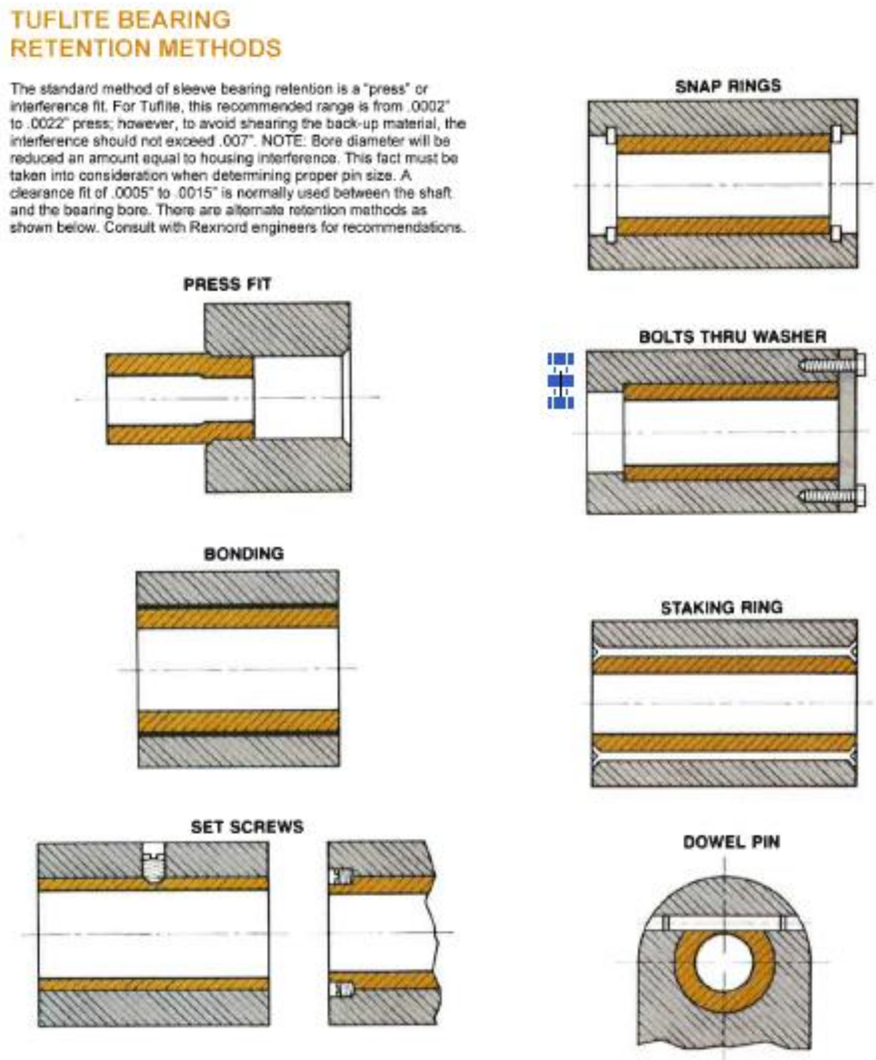
Examples: cast bronze (sleeve and flanged, can lubricate via grooves), powdered metal (bronze + oil, controlled porosity for self lubrication, press fit into housing, clearance fit .001”), nylon, PTFE (Teflon), composite braided bearings, etc.



Sintered metal (e.g. bronze/oil impregnated): advantages- inexpensive, handle moderate to high speeds, moderate temperatures, easy to use, machinable, readily available from many vendors. They do not require hardened shafting, although its a good idea to use it anyway. Disadvantages- they require a press fit, and must be reamed to size after installation. They don't handle shock/vibration well, and will wear out eventually (contaminants and eventual loss of oil usually cause this, although overloading doesn't help!), and field replacement is a problem due to the press fit and reaming required.

Plastics (e.g. Teflon): advantages: inexpensive, are easily installed with a light press fit (no need to ream them after installation!), run well with soft shafting, handle vibration and shock extremely well, do not lose their lubricating ability over time, and can be readily replaced in the field with a minimum of tools because of the light fit. They can handle corrosive atmospheres if properly selected, and are also electrical insulators. Disadvantages- they won't handle high speeds and/or pressures (defined as PV factor, pressure-velocity), because they're usually thermoplasics- and poor thermal conductors; a bad combination for some uses, although there are high temperature types for slow speeds.

Shafts for these bearings: The shafts must be round.   Cold rolled steel may need to be turned down to size,consider drill rod, or rods intended for used in linear bearings.

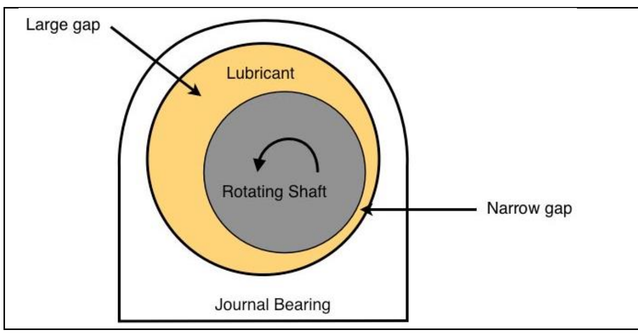


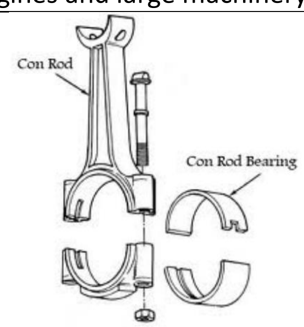
TUFLITE BEARING
RETENTION METHODS

The standard method of sleeve bearing retention is a "press" or interference fit. For Tuflite, this recommended range is from .0002" to .0022" press; however, to avoid shearing the back-up material, the interference should not exceed .007". NOTE: Bore diameter will be reduced an amount equal to housing interference. This fact must be taken into consideration when determining proper pin size. A clearance fit of .0005" to .0015" is normally used between the shaft and the bearing bore. There are alternate retention methods as shown below. Consult with Rexnord engineers for recommendations.

B. Journal Bearings

Journal bearings depend on a layer of lubricant to "float" the shaft, so must have a continuous source of lubricant (oil or grease), by being submerge in an oil bath or the oil is pumped into the bearing/shaft annular space. Often these are specially designed for the application, and not COTS. Usually bearing material is a softer metal than the shaft. Used in engines and large machinery.





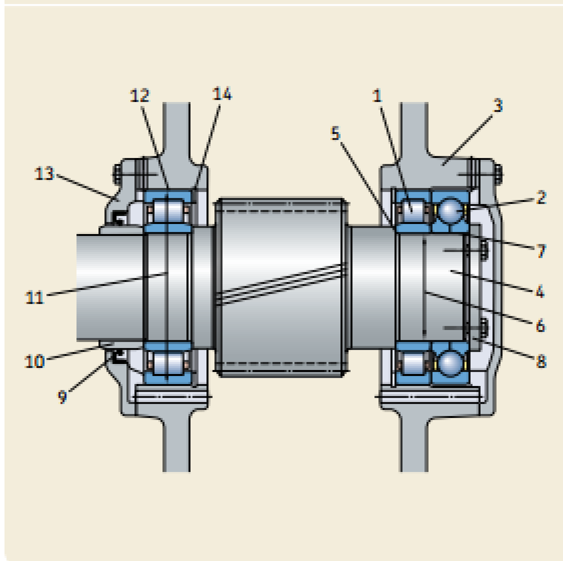
Journal bearing design is based on charts based on the solution of a partial differential equation from fluid mechanics (see Machine Design textbook for charts).

C. Rolling-Element Bearings – Ball Type and Roller Type

Bearing systems (→ fig. 2)

1 Cylindrical roller bearing
2 Four-point contact ball bearing
3 Housing
4 Shaft
5 Shaft abutment shoulder
6 Shaft diameter
7 Shaft seat
8 End plate
9 Radial shaft seal
10 Distance ring
11 Housing bore diameter
12 Housing seat
13 Housing cover
14 Snap ring

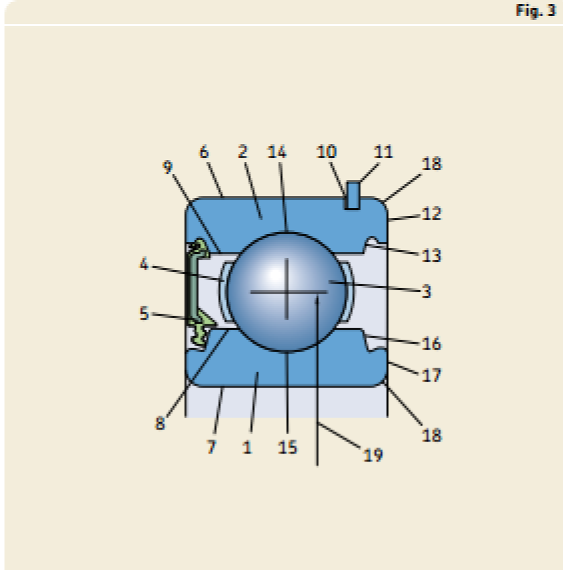
Fig. 2



Radial bearings (→ figs. 3 and 4)

1 Inner ring
2 Outer ring
3 Rolling element: ball, cylindrical roller, needle roller, tapered roller, spherical roller, toroidal roller
4 Cage
5 Capping device
Seal – made of elastomer
Shield – made of sheet steel
6 Outer ring outside surface
7 Inner ring bore
8 Inner ring shoulder surface
9 Outer ring shoulder surface
10 Snap ring groove
11 Snap ring
12 Outer ring side face
13 Recess for capping device

Fig. 3



Source Manufacturer: SKF catalog, http://www.skf.com/binary/168-121486/SKF-rolling-bearings-catalogue.pdf

1. Ball Type (see SKF table on last page)

Deep groove ball bearing- most common- moderate thrust and radial loads
Angular contact ball bearing- higher thrust loads, reduced radial, often intended to pre-load in thrust direction to remove axial clearance in pumps, ball screws, etc. Used singularly for one directional thrust, or in matched pairs. They are machined so that when their races are clamped flat, they pre- load for thrust in one direction. Require axial clamping.

Self aligning ball bearing – for shaft misalignment and shaft bending
Ball thrust bearing - handle thrust and nothing else.
Double row- two deep groove bearings in one housing- simplified arrangement for handling some torque perpendicular to the axis, or a little more radial load.

2. Roller Type

Cylindrical roller bearings- What you use when the heaviest duty ball bearing of the same physical dimensions won't meet your load/life requirements- these devices can handle several times the load of a comparably sized ball bearing- but they cost more. They come in pure cylindrical or spherical roller styles- the former for higher load with no angular compensation, the latter, which has a spherical housing, allows generous angular error.
Needle bearings- a roller bearing lacking one or both races. Low cost and minimal size, but you must provide whatever is lacking, a hardened and ground shaft (the inner race) and a machined housing.
Tapered roller bearings- handle large thrust and radial loads. Must be axially clamped.
Spherical roller bearings - large thrust and radial loads.
Roller thrust bearing- same as ball thrust, with higher loads. Straight rollers are cheap, but have some problems with the velocity differences (may gall at higher speeds), tapered rollers are better because velocities match.
Cam followers- essentially ball or roller bearings with thick inner and outer races designed to be used as wheels or to follow cam surfaces (a regular bearing might serve the same purpose if the forces are low, but higher forces would crack the outer race).

3. Lubrication

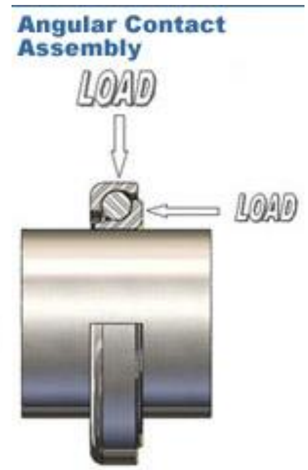
Many of the previous units are available with *metal shields* or *plastic seals*, and are pre- packed with grease, and if used properly, will need no additional lubrication. Open units should have some external sealing provided and either oil or grease lubrication provided.

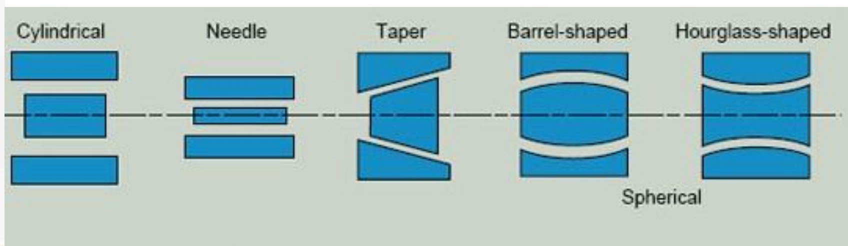
4. Shaft and Housing Fits

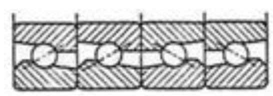
The previous examples require you to provide precision housing and shaft fits to evenly distribute the race loads and get good life- If the housing is too big (or not round), the outer race can crack or spin. The same for the shaft fit. If the housing is too small or the shaft too large, the balls are overloaded and deformed too far out of round, which causes them to rapidly fatigue. These housing/shaft fits are costly because they have to be more accurate than a lot of other things you'll be doing. The correct fit is also application specific and takes some practice.

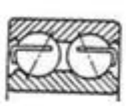
5. Housed Units (e.g. pillow block bearings, flanged pillow block bearings)

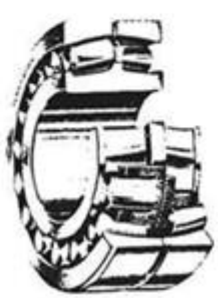
Many of the previously mentioned bearings are available in housings, and most have thicker inner races with some sort of shaft locking device. This means you can simply bolt these bearings to any mounting surface, and your shaft can be a little undersized (.001-.002" max, to use the full bearing load capability) which means easy installation. These housings are available in several materials/coatings for harsh environments, and many have spherical bores to allow the bearing inserts to accommodate angular misalignments. One additional bonus- most come with grease fittings to allow re-lubrication in demanding applications.

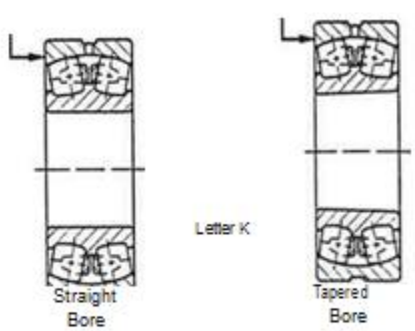

Angular Contact Assembly
LOAD
LOAD


Cylindrical
Needle
Taper
Barrel-shaped
Hourglass-shaped
Spherical








Straight Bore
Letter K
Tapered Bore

6. Important Equations

Basic rating life

The basic rating life of a bearing in accordance with ISO 281 is

$$L_{10} = \left(\frac{C}{P}\right)^p$$

If the speed is constant, it is often preferable to calculate the life expressed in operating hours using

$$L_{10h} = \frac{10^6}{60\,n}\,L_{10}$$

where

L_{10} = basic rating life (at 90% reliability) [million revolutions]

L_{10h} = basic rating life (at 90% reliability) [operating hours]

C = basic dynamic load rating [kN]

P = equivalent dynamic bearing load [kN] (→ **page 85**)

n = rotational speed [r/min]

p = exponent of the life equation

- for ball bearings, $p = 3$
- for roller bearings, $p = 10/3$

Example 1: Basic rating life and SKF rating life

An SKF Explorer 6309 deep groove ball bearing is to operate at 3 000 r/min under a constant radial load F_r = 10 kN. Oil lubrication is to be used, the oil has an actual kinematic viscosity $\nu = 20$ mm²/s at normal operating temperature. The desired reliability is 90% and it is assumed that the operating conditions are very clean. What are the basic and SKF rating lives?

a) The basic rating life for 90% reliability is

$$L_{10} = \left(\frac{C}{P}\right)^3$$

From the product table for bearing 6309, C = 55,3 kN. Since the load is purely radial, $P = F_r$ = 10 kN (→ *Equivalent dynamic bearing load*, **page 85**).

$$L_{10} = \left(\frac{55,3}{10}\right)^3$$

$$= 169 \text{ million revolutions}$$

or in operating hours, using

$$L_{10h} = \frac{10^6}{60\,n}\,L_{10}$$

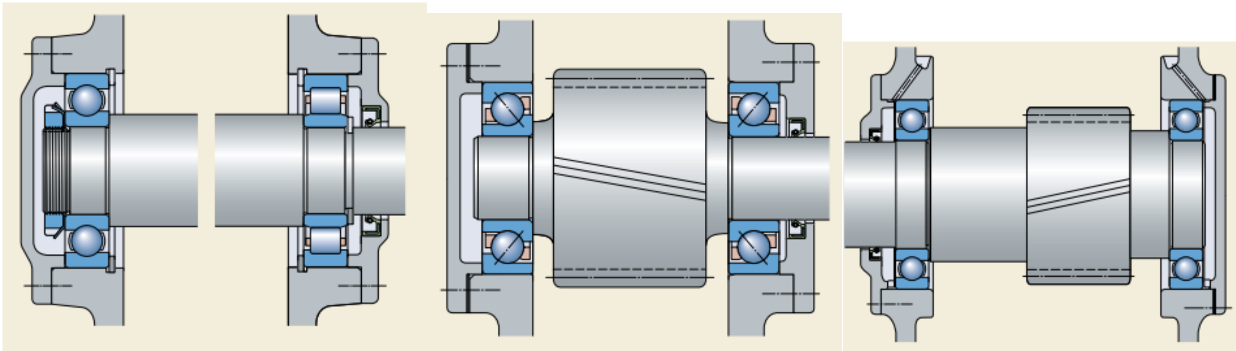
$$L_{10h} = \frac{1\,000\,000}{60 \times 3\,000} \times 169$$

$$= 940 \text{ operating hours}$$

7. Bearing Selection Guidelines, Mounting Methods, Examples

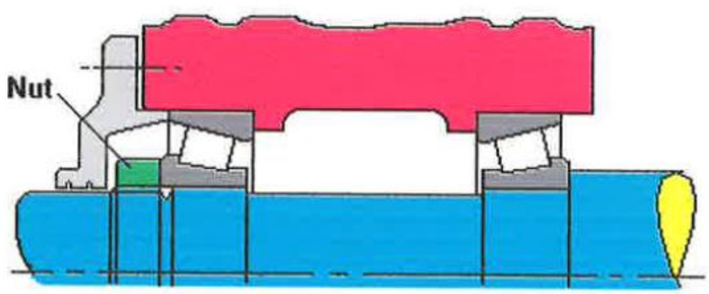
Bearings are chosen based on load, life and environment. A bearing may need to support a radial load (perpendicular to shaft), thrust load (along shaft), or combination. Model bearings as simply supported end condition (e.g. a pinned end condition) in most situations, two bearings side-by-side are required to handle a moment load. Bearings can be selected that can handle shaft misalignment from bending, Generally ball bearings can handle small radial load, with the wider bearing for larger load (why?).

From http://www.agstech.net/web-storage/webstorage2/Engineering_Guide_for_Bearing_Selection-AGS-TECH%20Inc.pdf :

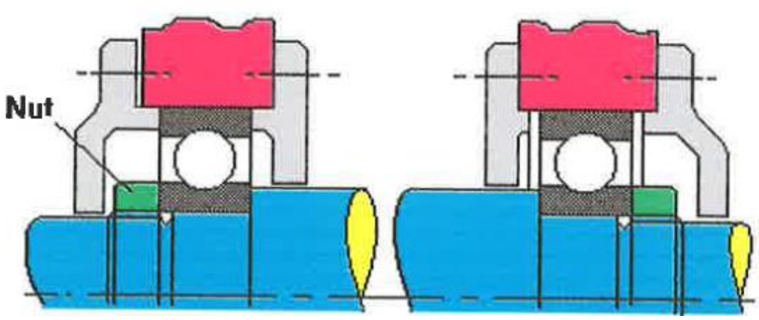


Left: Locating/Nonlocating System – for thermal expansion and contraction of the shaft. Left bearing "locates", right bearing is "nonlocating".
Middle: Cross-Locating System - the shaft is located axially in one direction by one bearing arrangement and in the opposite direction by the other. This system is referred to as "crosslocated".
Right: Floating bearing system - for applications where axial stability of the shaft is less demanding or where other components on the shaft locate it axially. Although hard to see, there is a small gap between the outer race and the endcaps.

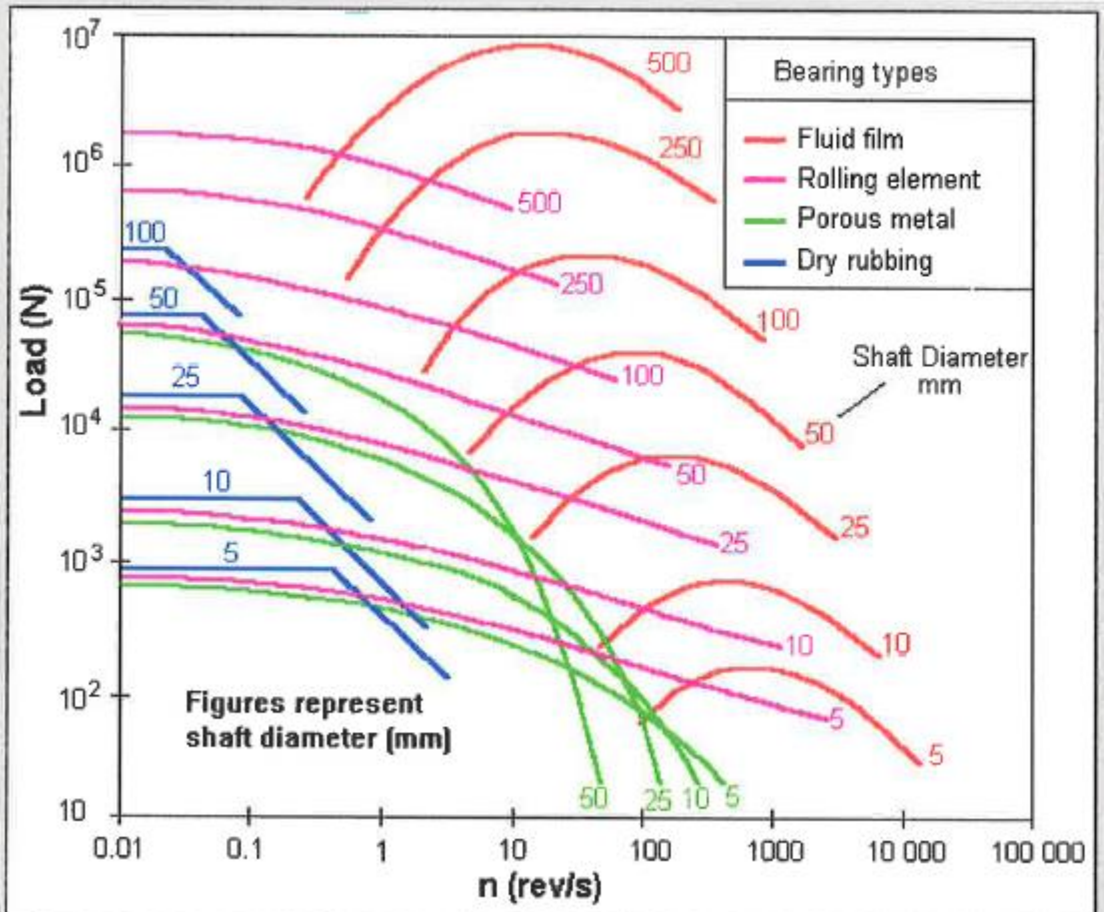
Car Wheel Bearing Arrangement – Wheel hub (pink) is accurately located between the two bearings with small clearance in the bearings. When the nut is tightened, the bearing is positioned through the inner race. Notice the "O" arrangement. In this situation, the wheel hub is free to spin, blue spindle is stationary, the outer rings are press fit into the hub, inner rings sliding fit onto the blue shaft (spindle), nut is hand tightened and with a jam nut behind to lock into place. NOTE: Human Exploration Rover is similar, except the shaft is rotating inside the knuckle (pink).



Electric Motor Bearing Arrangement – Required to be simple and quiet, and operate at high speed. The left hand bearing is a deep groove ball bearing and axially locates the shaft. The right hand bearing is the output torque end and notice that the bearing floats in the housing.



8. Three Summarizing Charts for the Bearing Designer



Reproduced from the Tribology Handbook by M. J. Neale https://www.qbcbearings.com/TechInfo/PDF/TechT5_6.pdf

Table 1-1 Bearing Selection Factors*

Relative rating

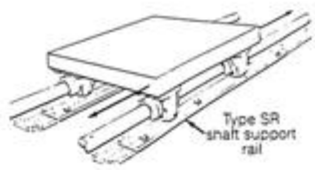
- ■ good
- ◪ intermediate
- • poor
- ▬ none
- ☒ does not apply
- □ not available or known

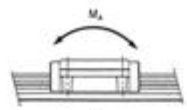
NOTE: This table is only a general guide – ratings may change when considering special types or treatments.

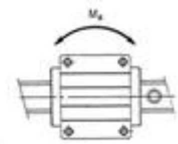
Selection Factors	Ball						Roller						Journal						Thrust				Ext. press.⑤		Gas ⑤	
	2a	2b	2c	2d	2e	2f	3a	3b	3c	3d	3e	3f	8a	8b	8c	8d	8e	8f	12a	12b	12c	12d				
	deep groove	deep groove – fill	slot	angular contact	2-row angular	contact	cylindrical	cylindrical–locating	barrel	spherical thrust	tapered	needle	full	partial	axial slotted	elliptical	three-lobed	pivoted shoe	parallel	step	tapered land	tilt shoe	capillary ⑥	variable flow	pneumostatic	pneumodynamic
Low starting friction	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	•	•	•	•	•	•	•	•	•	•	■	■	■	•
Low running friction	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	•	◪	◪	◪	■	◪	■	■
Low noise	•	•	•	•	•	•	•	•	•	•	•	•	■	■	■	■	■	■	■	■	■	■	■	■	■	■
Small diameter ①	•	•	•	•	•	•	◪	◪	◪	◪	◪	■	■	■	■	■	■	◪	•	◪	◪	◪	◪	◪	•	•
Short length ①	■	■	■	◪	■	■	■	■	■	■	■	◪	◪	◪	◪	◪	◪	◪	■	■	■	◪	◪	◪	•	•
High accuracy	◪	•	■	■	◪	◪	◪	◪	◪	◪	◪	◪	•	•	•	•	•	◪	•	•	•	•	■	■	■	•
Most available	■	■	■	■	■	■	■	■	■	◪	■	■	■	■	•	◪	•	•	■	•	◪	•	•	•	•	•
High radial load ②	◪	◪	◪	◪	◪	▬	■	■	■	▬	■	▬	■	■	◪	◪	◪	◪	▬	▬	▬	▬	◪	◪	•	•
High thrust load ②	◪	•	■	■	•	■	▬	•	◪	■	■	▬	▬	▬	▬	▬	▬	▬	•	■	■	■	◪	◪	•	•
High dynamic load ②	◪	◪	◪	◪	◪	◪	■	■	■	■	■	■	■	◪	◪	◪	•	◪	■	◪	◪	◪	◪	◪	•	•
Tolerate misalignment	◪	◪	◪	•	■	◪	•	•	■	■	•	•	◪	◪	◪	◪	◪	■	◪	◪	◪	■	◪	◪	•	•
Tolerate dirt	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	■	■	■	■	■	■	■	■	■	■	•	◪	•	•
Low initial cost	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	■	■	■	◪	◪	•	■	■	■	•	•	•	•	◪
High speed	■	•	■	◪	◪	•	■	■	•	•	•	•	◪	◪	◪	■	■	■	•	■	■	■	■	◪	■	■
High temperature ③	◪						◪						•												■	◪
Simple lube system	■	■	■	■	■	■	■	■	■	■	■	■	◪	◪	◪	◪	◪	◪	◪	◪	◪	◪	•	•	•	■
High stability ④	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	☒	•	•	◪	■	■	■	■	■	■	■	■	◪	•	•
Easy for designer	■	■	■	■	■	■	■	■	■	■	■	■	■	■	◪	◪	◪	•	■	◪	◪	◪	•	•	•	•

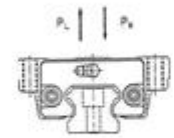
① with a given load (radial or thrust) ② with a given size ③ above 450°F ④ applies to high speed fluid-film bearings
⑤ journal or thrust type ⑥ restrictor controlled (liquid)

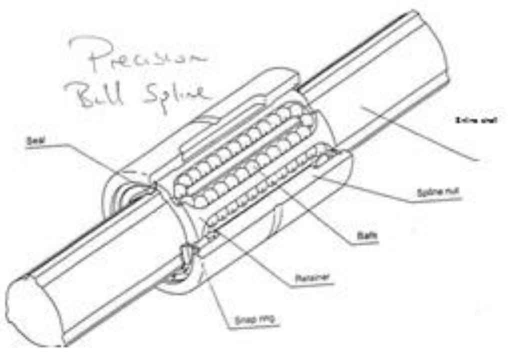
9. Linear Bearing Systems











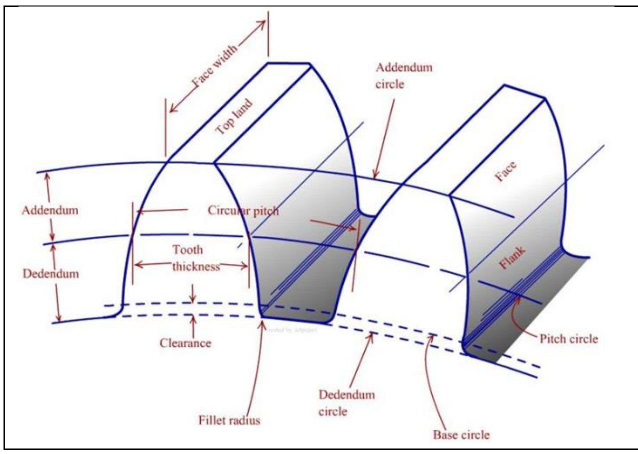
VI. Gears

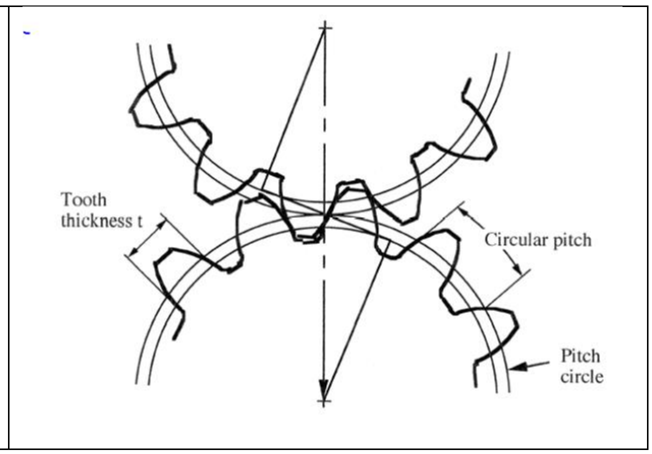
Gears can be imaged as two rolling without slipping cylinders or cones; the imagined circle is call the **pitch circle**, with it associated **pitch diameter**/radius for kinematic calculations. Tooth Size Measures - The **module 'm'** is the ratio of the pitch diameter to the number of teeth on the gear. The unit of module in SI system is mm, and is commonly used for metric gears. For English units, the **diametral pitch 'P'** is the ratio of the number of teeth on the gear to its pitch circle diameter. It is the reciprocal of module. The diametral pitch is usually expressed as 'teeth per inch'. Both meshing gears must have same diametrical pitch or module.

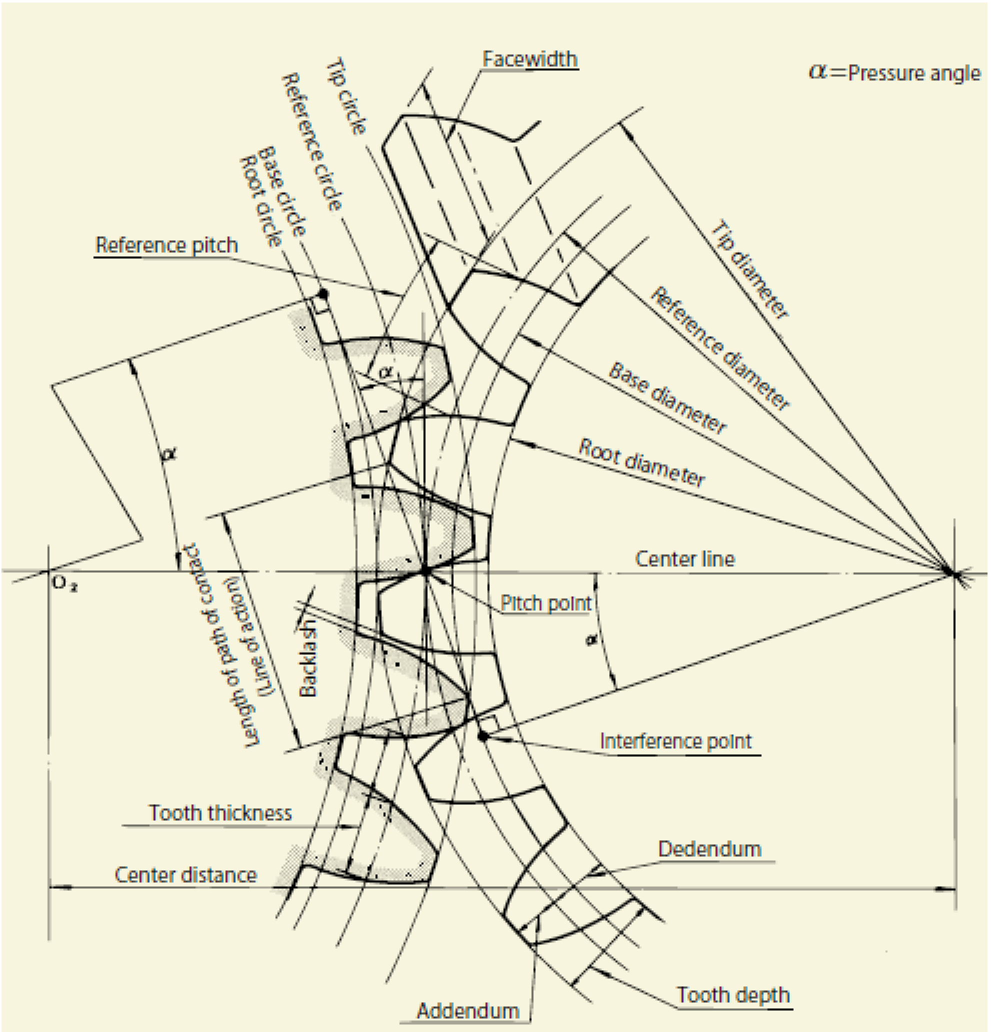
The **addendum circle** is the largest circle on the gear. The **addendum 'a'** is the radial distance between the pitch circle and the addendum circle. The **dedendum circle** is usually the smallest circle on the gear. The **dedendum 'b'** is the radial distance between the pitch circle and the dedendum circle. The dedendum is larger than the addendum; i.e., b > a. The **depth of tooth 'h'** is the sum of the addendum and dedendum; i.e., h = (a + b). The **base circle** or clearance circle of a gear is tangent to the base circle of its meshing gear. The angle of this line with respect a perpendicular to the centerline between the two gears is called the **pressure angle.**

Gears and Gear Reducers, P = Diametral pitch, z = Number of teeth, d = Pitch circle diameter, m = Module (mm), p = Circular pitch (mm), a = Addendum, b = Dedendum, c = Clearance = b – a, do = Diameter of addendum circle or Outside diameter =(d + a)

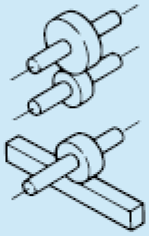
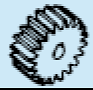
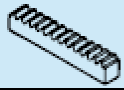
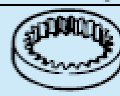
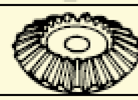
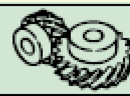
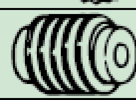
Failure mode that are the basis of a gear design: surface fatigue and root bending fatigue failure. Smaller gear in a mesh is the "pinion", the larger gear is the "gear".







http://www.martinsprocket.com/docs/default-source/brochures---gears/martin-gear-manual.pdf

Categories of Gears	Types of Gears	Efficiency (%)	Isometrics
Parallel Axis Gears 	Spur Gear	98.0–99.5	
	Helical Gear		
	Rack · Helical Rack		
	Internal Gear		
Intersecting Axis Gears 	Miter Gear	98.0–99.0	
	Straight Bevel Gear		
	Spiral Bevel Gear		
Nonparallel and Nonintersecting Axis Gears 	Screw Gear (Crossed Helical Gear)	70.0–95.0	
	Worm	30.0–90.0	
	Worm Wheel		

http://www.khkgears.co.jp/en/gear_technology/guide_info.html
http://www.khkgears.co.jp/en/gear_technology/pdf/455-461.pdf

Table 8.1 Forces acting upon a gear

Types of gears		F_t : Tangential force	F_x : Axial force	F_r : Radial force
Spur gear		$F_t = \frac{2000T}{d}$	———	$F_t \tan\alpha$
Helical gear			$F_t \tan\beta$	$F_t \frac{\tan\alpha_n}{\cos\beta}$
Straight bevel gear		$F_t = \frac{2000T}{d_m}$	$F_t \tan\alpha \sin\delta$	$F_t \tan\alpha \cos\delta$
Spiral bevel gear		d_m is the central reference diameter $d_m = d - b\sin\delta$	When convex surface is working:	
			$\frac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta - \sin\beta_m \cos\delta)$	$\frac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta + \sin\beta_m \sin\delta)$
			When concave surface is working:	
			$\frac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta + \sin\beta_m \cos\delta)$	$\frac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta - \sin\beta_m \sin\delta)$
Worm gear pair	Worm (Driver)	$F_t = \frac{2000T_1}{d_1}$	$F_t \frac{\cos\alpha_n \cos\gamma - \mu \sin\gamma}{\cos\alpha_n \sin\gamma + \mu \cos\gamma}$	$F_t \frac{\sin\alpha_n}{\cos\alpha_n \sin\gamma + \mu \cos\gamma}$
	Worm Wheel (Driven)	$F_t \frac{\cos\alpha_n \cos\gamma - \mu \sin\gamma}{\cos\alpha_n \sin\gamma + \mu \cos\gamma}$	F_t	
Screw gear $\left(\begin{array}{c}\Sigma = 90^\circ \\ \beta = 45^\circ\end{array}\right)$	Driver gear	$F_t = \frac{2000T_1}{d_1}$	$F_t \frac{\cos\alpha_n \sin\beta - \mu \cos\beta}{\cos\alpha_n \cos\beta + \mu \sin\beta}$	$F_t \frac{\sin\alpha_n}{\cos\alpha_n \cos\beta + \mu \sin\beta}$
	Driven gear	$F_t \frac{\cos\alpha_n \sin\beta - \mu \cos\beta}{\cos\alpha_n \cos\beta + \mu \sin\beta}$	F_t	

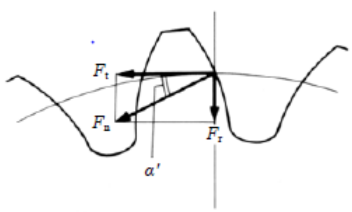


Fig.8.1 Forces acting on a spur gear mesh

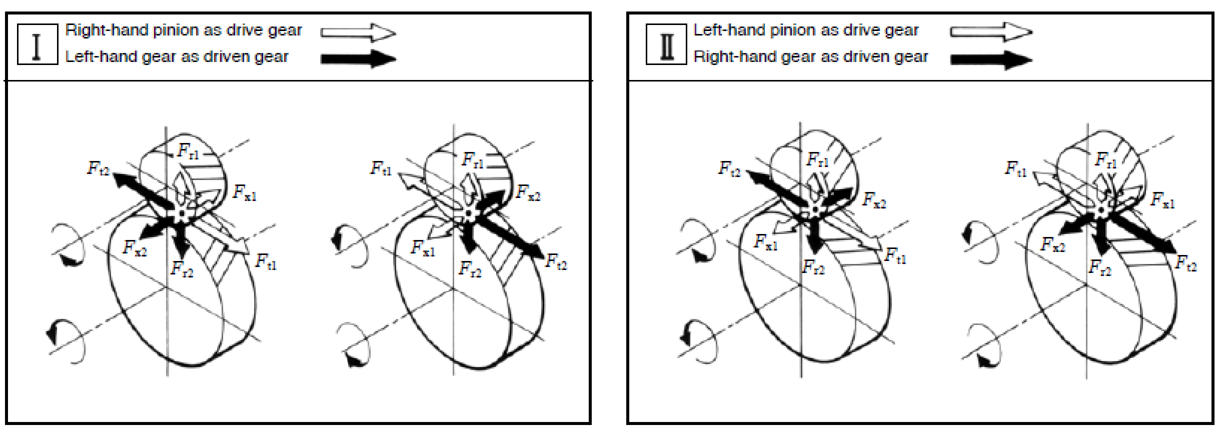
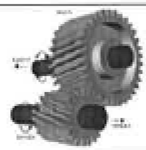


Fig.8.4 Directions of forces acting on a helical gear mesh

Gear Type	Figure	Geometric Description	Power Transmission Between:	Key Points in Engineering Design	Force Modelling
Spur Gears		Teeth parallel to cylinder axis	Parallel shafts	Most common type, moderate to low speeds	Radial, tangential, no axial force
Helical Gears		Teeth that follow on helical paths on the cylinder	Parallel and nonparallel nonintersecting shafts	Quieter operation because teeth load gradually and more teeth in contact, higher speeds, higher loads	Radial, axial and tangential
Herringbone Gears					
Internal Gears					
Bevel Gears					
Spiral Bevel Gears					
Miter Gears					
Hypoid Gears					
Worm + Worm Gear		A worm is a screw meshing with a spur-like gear called the worm gear			

http://www.martinsprocket.com/docs/default-source/brochures---gears/martin-gear-manual.pdf

VII. Shafts Parts

VIII. Brakes and Clutches

IX. Belts and Chains

X. Shock Absorbers and Shock Mounts

XI. Gearboxes and Gear Reducers
XII. Motors

XIII. Homework Problems

aring types - design and characteristics

The matrix can only provide a rough guide so that in each individual case it is necessary to make a more qualified selection referring to the information given in the catalogue

Symbols

+++ excellent - poor
++ good -- unsuitable
+ fair ← single direction
←→ double direction

Design

1 Tapered bore
2 Shields or seals
3 Self-aligning
4 Non-separable
5 Separable

Characteristics

Suitability of bearings for

6 Purely radial load
7 Purely axial load
8 Combined load
9 Moment load
10 High speed
11 High running accuracy
12 High stiffness
13 Quiet running
14 Low friction
15 Compensation for misalignment in operation
16 Compensation for errors of alignment (initial)
17 Locating bearing arrangements
18 Non-locating bearing arrangement
19 Axial displacement possible in bearing

		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Deep groove ball bearings	a, b		a				+	+ ←→	+ ←→	- b+	+++ b+	+++ b+	+	+++	+++	-	-	++ ←→	+	--
Angular contact ball bearings							+	+ ←	++ ←	-	++	+++	+	++	++	-	-	++ ←	--	--
	a, b, c		b		a, b	c	++	+ ←→	++ ←→	+	+	++	++	+	+	--	--	++ ←→	+	--
							-	++ ←→	+ ←→	+	++	+	+	+	+	--	--	++ ←→	-	--
Self-aligning ball bearings							+	-	-	--	+++	++	-	++	+++	+++	+++	+ ←→	+	--
Cylindrical roller bearings							++	--	--	--	++	++	++	++	++	-	-	--	+++	+++
	a, a, b, b						++	+ a← b←→	+ a← b←→	--	++	++	++	+	++	-	-	++ a← b←→	+ a←	+ a←
full complement	a, b				a	b	+++	-	+ ←	--	-	+	+++	-	-	-	-	+ ←	+	+
	a, a, b, c		a				+++	-	+ a←→ b←	+	-	+	+++	-	-	--	--	+ a←→ b←	+ c	+ b← c←→
Needle roller bearings	a, b, c		a	c			++	--	--	--	+	+ a++	a++ b++	+	-	--	-- c++	--	+++	+++
	a, b, c		b,c				++	--	--	--	+	+	++	+	-	--	--	--	+++	+++
	a, b, c		b,c				+	+ c++ ←	+ ←	-	+	+	++	+	-	--	--	+ ←	--	--
Tapered roller bearings							++	++ ←	+++ ←	-	+	+	++	+	+	-	-	+++ ←	--	--
	a, b						+++	++ ←→	+++ ←→	+	+	+	+++	+	+	-	--	+++ ←→	-	--
Spherical roller bearings							+++	+ ←→	+++ ←→	--	+	+	++	+	+	+++	+++	++ ←→	+	--
CARB bearings							+++	--	--	--	+	+	++	+	+	+++	+++	--	+++	+++
full complement							+++	--	--	--	-	+	+++	+	+	+++	+++	--	+++	+++
Thrust ball bearings	a, b						--	+ a← b←→	--	--	-	++ a	+	-	+	-	--	++ a← b←→	--	--
	a, b						--	+ a← b←→	--	--	-	+	+	-	+	-	++	++ a← b←→	--	--
Needle Cylindrical	a, b roller thrust bearings						--	++ ←	--	--	-	a+ b++	++	-	-	--	--	++ ←	--	--
Spherical roller thrust bearings							--	+++ ←	+ ←	--	-	+	++	-	+	+++	+++	+++ ←	--	--