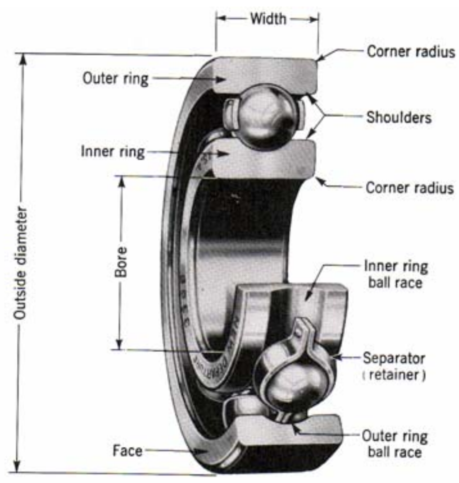
Rotary Rolling Contact Bearings

ME EN 7960 – Precision Machine Design
Topic 10

Ball Bearing Nomenclature



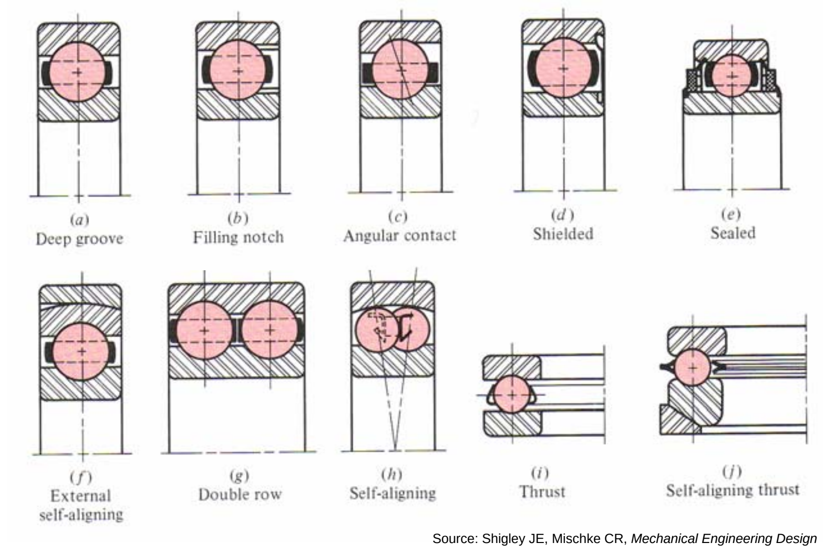
Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Rotary ball bearings are defined through geometry and performance.

Geometry:
- Outside diameter
- Inside diameter
- Width

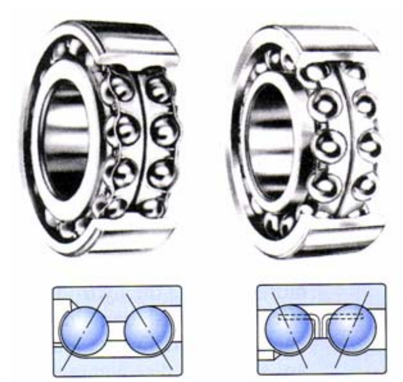
Performance:
- Load capacity (radial, axial, moment)
- Stiffness (radial, axial, moment)
- Runout
- Bearing life
- Allowable speed
- Lubrication
- etc.

Ball Bearing Types (Rotary)



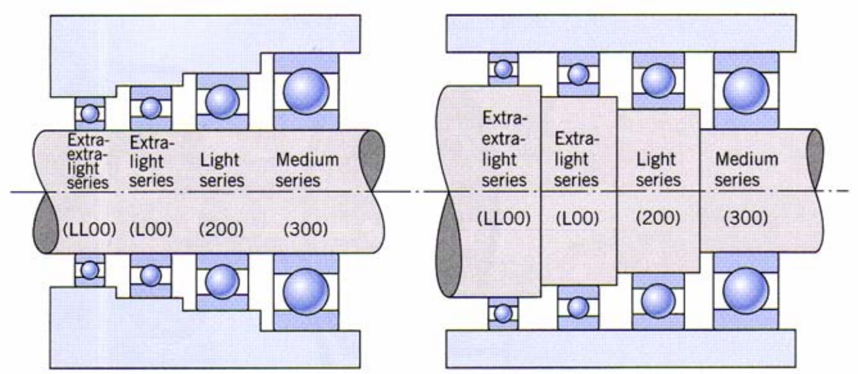
Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Ball Bearings (Rotary)



Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Bearing Sizes

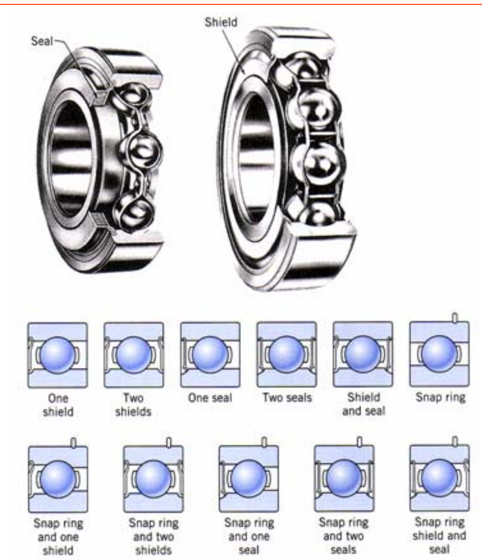


(a) Relative proportions of bearings with same bore dimension

(b) Relative proportions of bearings with same outside diameter

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Bearing Seals



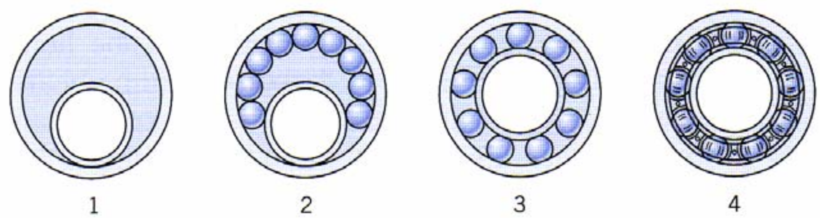
Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Ball Bearings – Details

- The ball has a smaller radius than the groove
- The load capacity of a bearing is limited by the contact stresses between the rolling elements and the races

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

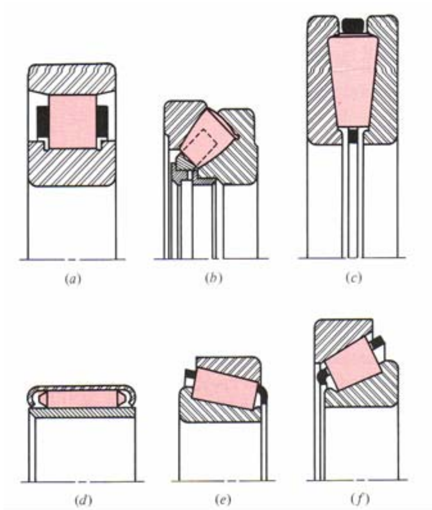
Ball Bearings - Assembly



Step 1: Move the inner race to one side of the out race
Step 2: Locate the largest gap between the two races and insert the balls in the groove
Step 3: Distribute the balls evenly, thereby centering the inner race with respect to the outer race
Step 4: Place the separator (retainer)

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Roller Bearings (Rotary)



a) Straight roller
b) Spherical roller
c) Tapered roller thrust
d) Needle
e) Tapered roller
f) Steep-angle tapered.

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Roller Bearings (Rotary)

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Ball vs. Roller Bearings

- Roller bearings are stiffer and have a higher load capacity than comparably sized ball bearings
  - This is due to the type of contact, line contact for rollers vs. point contact for balls
- Ball bearings have a lower friction
  - This also is a function of contact type.\
- Ball bearings can often be operated at higher speeds
- Most ball bearings can take modest axial loads for “free”
  - Only tapered rollers can take axial loads
- Ball bearings are less expensive than roller bearings

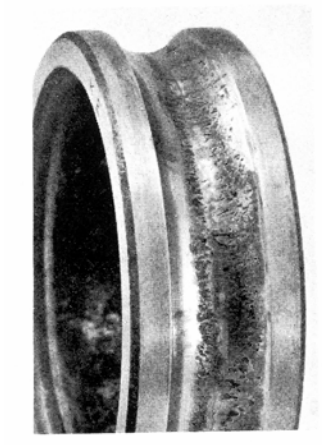
Bearing Characteristics

Bearing Type	Radial Capacity	Thrust Capacity	Limiting Speed	Radial Stiffness	Axial Stiffness
Deep-groove ball	Moderate	Moderate – both directions	High	Moderate	Low
Maximum-capacity ball	Moderate (plus)	Moderate – one direction	High	Moderate (plus)	Low (plus)
Angular contact ball	Moderate	Moderate (plus) – one direction	High (minus)	Moderate	Moderate
Cylindrical roller	High	None	Moderate (plus)	High	None
Spherical roller	High	Moderate – both directions	Moderate	High (minus)	Moderate
Needle roller	Moderate to high	None	Moderate to very high	Moderate to high	None
Single-row tapered	High (minus)	Moderate (plus) – one direction	Moderate	High (minus)	Moderate
Double-row tapered	High	Moderate – both direction	Moderate	High	Moderate
Four-row tapered	High (plus)	High – both direction	Moderate (minus)	High (plus)	High
Ball thrust	None	High – one direction	Moderate (minus)	None	High
Roller thrust	None	High (plus) – one direction	Low	None	High (plus)
Tapered roller thrust	Locational only	High (plus) – one direction	Low	None	High (plus)

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Failure Modes – Surface Fatigue

- Surface fatigue is the dominant failure mode
- The cyclic subsurface Hertzian shear stresses produced by the curved surfaces in rolling contact may initiate and propagate cracks that ultimately dislodge particles and generate surface pits
- Typically, the raceways pit first, resulting in noise, vibration, and heat



Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Failure Modes – Brinelling

- Static loads on stationary bearings may cause brinelling of the races
- The resulting local discontinuities cause vibration, noise, and heat

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Selection Criteria

- When selecting bearings, both failure modes need to be considered. As such, we need to examine:
  - Resistance to surface fatigue, expressed as *basic dynamic load rating* C_d
  - Resistance to brinelling, expressed as *basic static load rating* C_s

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Basic Load Ratings

- Basic load ratings are a standardized measure provided by the bearing industry to quantify the rolling element bearing's ability to resist surface fatigue and brinelling
- The basic static load rating C_s is a measure of the resistance to failure by brinelling
- The basic dynamic load rating C_d is a measure of resistance to failure due to surface fatigue

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

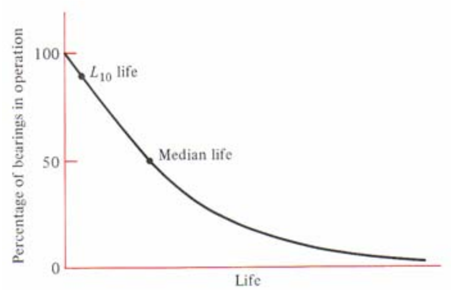
Bearing Life

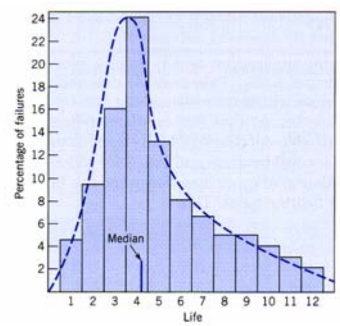
Two identical bearings tested under different loads P_1 and P_2 will have respective lives L_1 and L_2 according to:

$$\frac{L_1}{L_2} = \left(\frac{P_1}{P_2}\right)^a \quad (1)$$

Where:
a = 3 for ball bearings
a = 10/3 for roller bearings

Experiments with identical bearings under identical conditions reveal:





Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Example 1: Timken Bearing

A Timken ball bearing is rated as follows:
At a rated load of 2140 lb, the bearing has a life of 3000 hr at 500 rpm

Solution: Timken Bearing

A Timken ball bearing is rated as follows:
At a rated load of 2140 lb, the bearing has a life of 3000 hr at 500 rpm.

a) Calculate the bearing life *L*

$$L = tn = 3000\ hr \cdot 60 \frac{\min}{hr} \cdot 500 \frac{rev}{\min} = 90 \cdot 10^6\ rev$$

b) Determine the basic dynamic load rating

From (3):

$$\frac{L}{10^6} = \left(\frac{C_d}{P}\right)^3 \quad \rightarrow \quad C_d = \left(\frac{L}{10^6}\right)^{\frac{1}{3}} P = \left(\frac{90 \cdot 10^6}{10^6}\right)^{\frac{1}{3}} \cdot 2140\ lb = 8263\ lb$$

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Basic Dynamic Load Rating

The basic dynamic load rating C_d is defined to be the largest stationary radial load that 90 percent of a group of apparently identical bearings will survive for 1 million revolutions (inner race rotating, outer race fixed) with no evidence of failure by surface fatigue

The bearing life *L* for a given load *P* can be determined as follows:

$$L = \left(\frac{C_d}{P}\right)^a 10^6$$

L = bearing life (revolutions to failure)
C_d = basic dynamic load rating (90% reliability)
P = applied bearing load
a = exponent (*a* = 3 for ball bearings and a = 10/3 for roller bearings)

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Reliability Adjustments

Reliability adjustment factors, based on actual failure rate data, allows a designer to select bearings for reliabilities higher than 90%

$$L_R = K_R L \qquad (4)$$

L_R = reliability-adjusted bearing life
K_R = reliability life-adjustment factor
L = bearing life

Reliability Life-Adjustment Factor K_R for Bearing Reliabilities Different from R = 90%

Reliability *R* (%)	**Probability of Failure *P* (%)**	**K_R**
50	50	5.0
90	10	1.0
95	5	0.62
96	4	0.53
97	3	0.44
98	2	0.33
99	1	0.21

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Adjustments for Impact Loads

Rated load ratings are basic on static, continuous loading. Time-varying loads reduce the bearing life and need to be considered when selecting bearings

The applied equivalent dynamic load P_e is modified by the estimated *impact factor* I_F:

$$P = I_F \cdot P_e \qquad (5)$$

Estimated Impact Factors for Various Applications

Type of Application	**Impact Factor I_F**
Uniform load, no impact	1.0 – 1.2
Precision gearing	1.1 – 1.2
Commercial gearing	1.1 – 1.3
Toothed belts	1.1 – 1.3
Light impact	1.2 – 1.5
V-Belts	1.2 – 2.5
Moderate Impact	1.5 – 2.0
Flat belts	1.5 – 4.5
Heavy impact	2.0 – 5.0

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Selection Procedure

1. From free-body diagram, determine the *radial load* F_r and *axial thrust load* F_a
2. Determine the *design life requirement* L_d for the bearing
3. Determine the *reliability R* appropriate to the application and select the corresponding *life-adjustment factor* K_R
4. Assess the severity of any shock or impact associated with the application and select an appropriate *Impact Factor* I_F
5. Calculate the dynamic equivalent radial load P_e

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Selection Procedure (contd.)

$$P_{e1} = X_{d1}F_r + Y_{d1}F_a$$
$$P_{e2} = X_{d2}F_r + Y_{d2}F_a \quad (6)$$

Where: X_d = dynamic radial load factor
Y_d = dynamic axial (thrust) factor

If $P_{e1} > P_{e2}$ then $P_e = P_{e1}$, otherwise $P_e = P_{e2}$

Approximate Radial Load Factors for Selected Bearing Types								
Bearing Type	**Dynamic**				**Static**			
	X_{d1}	Y_{d1}	X_{d2}	Y_{d2}	X_{s1}	Y_{s1}	X_{s2}	Y_{s2}
Single-row radial ball bearing	1	0	0.55	1.45	1	0	0.6	0.5
Single-row angular contact bearing (shallow angle)	1	0	0.45	1.2	1	0	0.5	0.45
Single-row angular contact bearing (steep angle)	1	0	0.4	0.75	1	0	0.5	0.35
Double row radial ball bearing	1	0	0.55	1.45	1	0	0.6	0.5
Double-row angular contact bearing (shallow angle)	1	1.55	0.7	1.9	1	0	1	0.9
Double-row angular contact bearing (steep angle)	1	0.75	0.6	1.25	1	0	1	0.65
Straight roller bearing	1	0	-	-	1	0	1	0

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Selection Procedure (contd.)

6. Calculate the basic dynamic load rating requirement

$$C_d = \left(\frac{L}{K_R \cdot 10^6} \right)^{\frac{1}{a}} I_F P_e \qquad (7)$$

C_d = required dynamic load rating to give a bearing reliability of *R* percent
L = life (revolutions)
K_R = reliability adjustment factor
I_F = application impact factor
P_e = equivalent radial load
a = exponent equal to 3 for ball bearings or 10/3 for roller bearings

7. With C_d enter a basic load rating table and find the smallest bearing with a load rating of at least C_d

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Example 2: Rolling Element Bearing Selection

A support shaft for a new product has been designed with a diameter of 1.6 inches. The force analysis shows that:

Radial bearing load F_r = 370 lb
Axial bearing load F_a = 130 lb
Shaft speed n = 350 rpm
Design life specification is 10 years of operation, 50 days/year, 20hr/day
Design reliability specification is R = 95%
The shaft is V-belt driven

Solution: Rolling Element Bearing Selection

A support shaft for a new product has been designed with a diameter of 1.6 inches. The force analysis shows that:

1. Find design life in revolutions:

$$L = tn = 10\,yr \cdot 50\frac{days}{yr} \cdot 20\frac{hr}{day} \cdot 60\frac{\min}{hr} \cdot 350\frac{rev}{\min} = 2.1 \cdot 10^{8}\ rev$$

2. Find life-adjustment factor K_R for R = 95%

$$K_R = 0.62$$

3. Find impact factor I_F for application. Here, the shaft is V-belt driven.

$$I_F = 1.9$$

4. Calculate equivalent dynamic bearing load. Assume a single-row, deep-groove ball bearing.

Solution: Rolling Element Bearing Selection (contd.)

From table for deep-groove bearing: X_{d1} = 1, Y_{d1} = 0, X_{d2} = 0.55, Y_{d2} = 1.45

$$P_{e1} = X_{d1}F_r + Y_{d1}F_a = 1.0 \cdot 370 + 0.0 \cdot 130 = 370\ lb$$
$$P_{e2} = X_{d2}F_r + Y_{d2}F_a = 0.55 \cdot 370 + 1.45 \cdot 130 = 392\ lb$$
$$\rightarrow P_e = 392\ lb$$

5. Calculate basic dynamic load requirement for R = 95%

$$C_d = \left(\frac{L}{K_R \cdot 10^6}\right)^{\frac{1}{3}} I_F P_e = \left(\frac{2.1 \cdot 10^8}{0.62 \cdot 10^6}\right)^{\frac{1}{3}} \cdot 1.9 \cdot 392 = 5192\ lb$$

Consulting manufacturer's tables, we need to find a deep-groove bearing with a basic dynamic load rating of at least 5192 lb, a bore diameter of at least 1.6 inches, and an operational speed of at least 350 rpm.

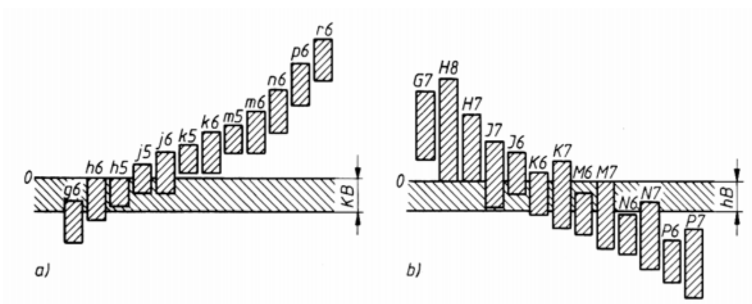
Bearing Quality

- Bearing quality (= tolerances) has been standardized by the Annular Bearing Engineers' Committee (ABEC)
- There are four primary grades of precision: ABEC 1, 5, 7, and 9
- ABEC 1 is the standard grade suited for most normal applications
  - For bearing bores between 35 and 50mm, tolerances are +0.0000 in. to -0.0005 in
- ABEC 9 is the highest grade for applications of extreme accuracy
  - For bearing bores between 35 and 50mm, tolerances are +0.00000 in. to -0.00010 in

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Bearing Fits

- Fits should be chosen according to manufacturer's recommendation. Alternatively, use the following rule:
  - The *rotating ring* should have an interference fit with its mating member (to ensure that the bearing ring will not slip or rotate)
  - The *non-rotating ring* should have a close push fit (to allow axial slip in case of thermal expansion)



K_b = tolerance for bearing inside diameter of inner race
h_b = tolerance for bearing outside diameter of outer race

Source: Matek W, Muhs D, Wittel H, *Roloff/Matek Maschinenelemente*

Typically Bearing Life Requirement

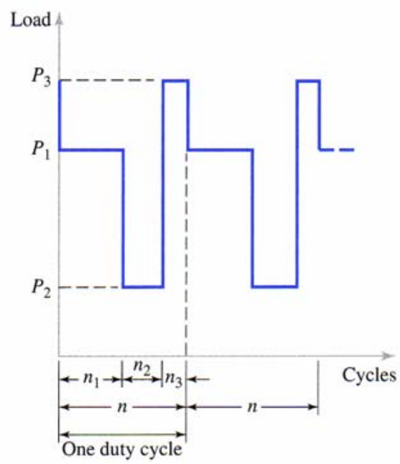
Typical Bearing Life Requirements	
Type of Application	**Life [hr]**
Instruments and apparatus for infrequent use	Up to 500
Aircraft engines	500 – 2,000
Machines for short or intermittent operation where service interruption is of minor importance	4,000 – 8,000
Machines for intermittent service where reliable operation is of great importance	8,000 – 14,000
Machines for 8 hr service which are not always fully utilized	14,000 – 20,000
Machines for continuous 24 hr service	50,000 – 60,000
Machines for continuous 24 hr service where reliability is of extreme importance	100,000 – 200,000

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Spectrum Loading

If a bearing is subjected to a spectrum of different applied loads during each duty cycle in the operation of a machine, two basic methods are available:

1. Assume that the largest load is applied to the bearing at every revolution, even though the actual load may be smaller during some segments of the operation
   – Bearing is oversized (not economical but safe)
2. Consider load spectrum
   – Requires thorough knowledge about spectrum



Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

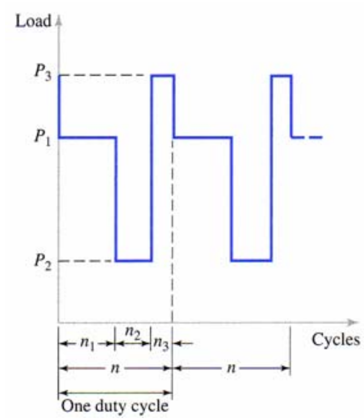
Spectrum Loading (contd.)

From bearing life equation:

$$[C_d]^a \cdot 10^6 = L_i P_i^a = L_1 P_1^a = L_2 P_2^a = \cdots$$

Where L_i = bearing life at a given load P_i

For a duty cycle as shown in the right figure, the total number of cycles is n, of which n_1 occur at a load P_1, and n_2 cycles occur at a load P_2, etc.



$$n_1 + n_2 + \cdots + n_i = n$$

$$\alpha_1 = \frac{n_1}{n} \equiv fraction\ \ of\ \ cycles\ at\ P_1$$

$$\alpha_2 = \frac{n_2}{n} \equiv fraction\ \ of\ \ cycles\ at\ P_2$$

$$\vdots$$

$$\alpha_i = \frac{n_i}{n} \equiv fraction\ \ of\ \ cycles\ at\ P_i$$

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Spectrum Loading (contd.)

Where: L_1 = cycles to failure if all cycles were loaded at P_1
L_2 = cycles to failure if all cycles were loaded at P_2
L_i = cycles to failure if all cycles were loaded at P_i

$$\alpha_i L_d = total\ cycles\ at\ P_i$$

Palmgren-Miner linear damage rule, failure can be expected if:

$$\sum \frac{\alpha_i L}{L_i} = 1 \qquad (8)$$

With:

$$L_i = \frac{[C_d]^a \cdot 10^6}{P_i^a} \qquad (9)$$

(9) in (8):

$$\sum \frac{(\alpha_i L_d) P_i^a}{[C_d]^a \cdot 10^6} = 1 \qquad (10)$$

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Spectrum Loading (contd.)

Or: $$\alpha_1 P_1^a + \alpha_2 P_2^a + \cdots + \alpha_i P_i^a = \frac{[C_d]^a \cdot 10^6}{L_d} \qquad (11)$$

Or: $$C_d = \left[\frac{L}{K_R \cdot 10^6}\right]^{\frac{1}{a}} \left[\sum \alpha_i \left((I_F P_e)_i\right)^a\right]^{\frac{1}{a}} \qquad (12)$$

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Example: Spectrum Loading

The load analysis of a newly proposed rotating shaft has been found as shown in the table below.
The total design life is to be 10^7 revolutions with a reliability of 97%.
A single-row deep groove bearing is preferred.

a) Find the basic dynamic load rating required for this application based on the load spectrum

b) Find the basic dynamic load rating required for this application based on the maximum load

Duty Cycles For Application			
Variable	**Segment 1**	**Segment 2**	**Segment 3**
Radial load F_r [lb]	2000	1000	5000
Axial load F_a [lb]	500	900	0
Shock loading I_F	Light shock	Moderate shock	Moderate shock
Cycles n_i/duty cycle [number of cycles]	1000	2000	100
Operational speed N_{op} [rpm]	3600	7200	900

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Solution: Spectrum Loading

a) Find the basic dynamic load rating required for this application based on the load spectrum

For a reliability of 97%, the *life-adjustment factor* K_R = 0.44

The *impact factor* I_F for light shock is chosen as I_F = 1.4, and for medium shock I_F = 1.8

Total number cycles: $n = n_1 + n_2 + n_3 = 1000 + 2000 + 100 = 3100$

Cycle fractions: $\alpha_i = \frac{n_i}{n} \rightarrow \alpha_1 = \frac{n_1}{n} = \frac{1000}{3100} = 0.32$

Similarly for α_2 and α_3: $\alpha_2 = \frac{n_2}{n} = \frac{2000}{3100} = 0.65 \quad \textit{and} \quad \alpha_3 = \frac{n_3}{n} = \frac{100}{3100} = 0.03$

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Solution: Spectrum Loading (contd.)

For a deep groove bearing:

$$X_{d1} = 1 \quad Y_{d1} = 0 \quad X_{d2} = 0.55 \quad Y_{d2} = 1.45$$

Determine equivalent dynamic bearing loads P_{e1} and P_{e2}:

$$P_{e1} = X_{d1} F_r + Y_{d1} F_a$$
$$P_{e2} = X_{d2} F_r + Y_{d2} F_a$$

	Segment 1	**Segment 2**	**Segment 3**
P_{e1} [lb]	2000	1000	5000
P_{e2} [lb]	1825	3450	2750
P_e [lb]	2000	3450	5000

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Solution: Spectrum Loading (contd.)

Using (12):

$$C_d = \left[\frac{L}{K_R \cdot 10^6}\right]^{\frac{1}{a}} \left[\sum \alpha_i \left(\left(I_F P_e\right)_i\right)^a\right]^{\frac{1}{a}}$$

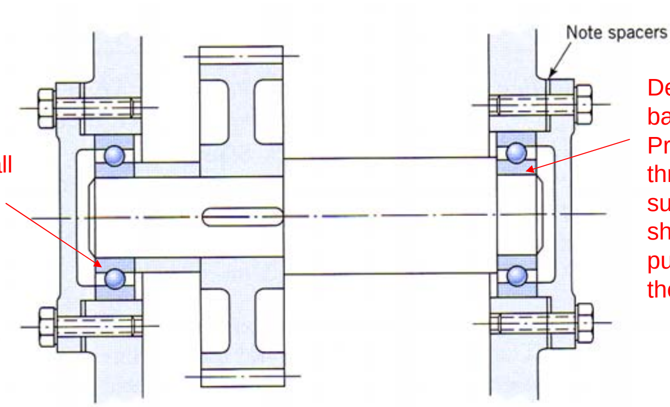
$$C_d = \left[\frac{10^7}{0.44 \cdot 10^6}\right]^{\frac{1}{3}} \sqrt[3]{\left(0.32[1.4 \cdot 2000]^3 + 0.65[1.8 \cdot 3450]^3 + 0.03[1.8 \cdot 5000]^3\right)} = 15{,}460\ lb$$

b) Find the basic dynamic load rating required for this application based on the maximum load.

$$C_d = \left[\frac{10^7}{0.44 \cdot 10^6}\right]^{\frac{1}{3}} \cdot 1.8 \cdot 5000 = 25{,}470\ lb$$

Source: Collins JA, *Mechanical Design of Machine Elements and Machines*

Bearing Mount for Thrust



Deep-groove ball bearing. Provides thrust support if shaft is pushed to the left

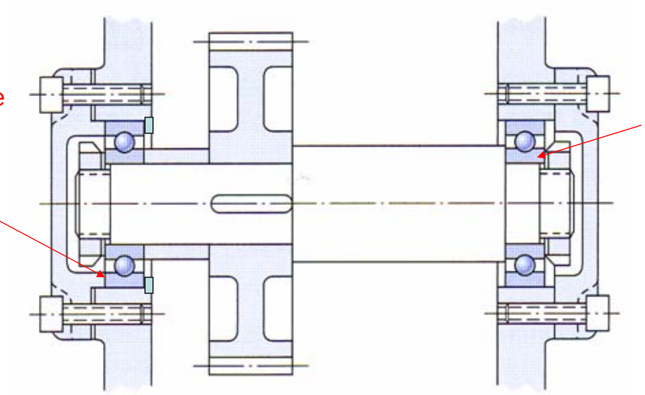
Deep-groove ball bearing. Provides thrust support if shaft is pushed to the right

WARNING!!!!!
Thermal expansion of the shaft creates thrust force, possibly overloading the bearings.

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Bearing Mount for Thrust (contd.)

Deep-groove ball bearing. Provides thrust support if shaft is pushed to the left and right.



Deep-groove ball bearing. Provides no thrust support because it can slide inside the bore.

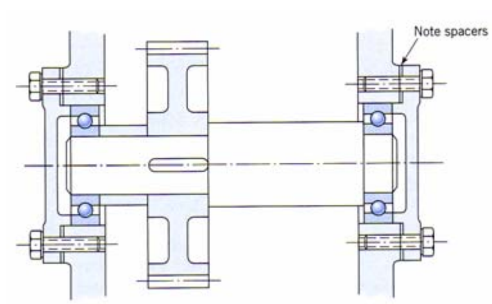
RECOMMENDED!!!!
The thrust force is provided by a single bearing. The second bearing only takes radial loads because it is allowed to slide axially.

Source: Juvinall RC, Marshek KM, *Fundamentals Of Machine Component Design*

Example: Thrust from Thermal Expansion

A shaft with a diameter of 40 mm and a length of 200 mm is rigidly supported at both ends with a single-row deep-groove bearing (#6008). The bearings have a rated basic dynamic load rating of $C_{d(90)}$ = 16.8 kN. The gear is creating a radial load of 5000 N while the thrust force is negligible. The shaft spins at 350 rpm.

a) Assuming that radial load on the left bearing is 3000 N, what is the bearing life for 98% reliability?
b) Assuming that the shaft heats up to 5°C above room temperature, what is the expected bearing life? (Assume the housing to be rigid and at room temperature).



Solution: Thrust from Thermal Expansion

a) Assuming that radial load on the left bearing is 3000 N, what is the bearing life for 98% reliability?

For a deep groove bearing:

$$X_{d1} = 1 \quad Y_{d1} = 0 \quad X_{d2} = 0.55 \quad Y_{d2} = 1.45$$

Determine equivalent dynamic bearing loads P_{e1} and P_{e2}:

$$P_{e1} = X_{d1}F_r + Y_{d1}F_a$$
$$P_{e2} = X_{d2}F_r + Y_{d2}F_a$$

Here: $P_e = 3000N$

Solution: Thrust from Thermal Expansion (contd.)

Bearing Equation: $$C_d = \left(\frac{L}{K_R \cdot 10^6}\right)^{\frac{1}{a}} I_F P_e$$

Solve for bearing life L:

$$L = \left(\frac{C_d}{I_F \cdot P_e}\right)^3 \cdot K_R \cdot 10^6 = \left(\frac{16800}{1.2 \cdot 3000}\right)^3 \cdot 0.33 \cdot 10^6 = 3.35 \cdot 10^7 \, rev$$

Or in terms of operating time:

$$t = \frac{L}{n} = \frac{3.35 \cdot 10^7 \, rev}{350 \frac{rev}{\min} \cdot 60 \frac{\min}{hr}} = 1{,}600 hr$$

Solution: Thrust from Thermal Expansion (contd.)

b) Assuming that the shaft heats up to 5°C above room temperature, what is the expected bearing life? (Assume the housing to be rigid and at room temperature.

Thermal expansion coefficient for steel: $\alpha = 12 \cdot 10^{-6} \frac{1}{K}$

Linear expansion: $\Delta l = \alpha \cdot \Delta T \cdot L = 12 \cdot 10^{-6} \cdot 5 \cdot 0.2 = 12 \cdot 10^{-6} m = 12 \mu m$

How much force is created if the shaft is prevented to expand 12μm?

For a bar, the stiffness is given as: $k = \frac{AE}{L}$

In general: $k = \frac{F}{\delta}$

Solution: Thrust from Thermal Expansion (contd.)

Combine:

$$F = \frac{AE \cdot \Delta L}{L} = \frac{\pi d^2 E \cdot \Delta L}{4L} = \frac{\pi (0.04m)^2 \cdot 207 \cdot 10^9 \frac{N}{m^2} \cdot 12 \cdot 10^{-6} m}{4 \cdot 0.2m} = 15607N$$

For a deep groove bearing:

$$X_{d1} = 1 \quad Y_{d1} = 0 \quad X_{d2} = 0.55 \quad Y_{d2} = 1.45$$

Determine equivalent dynamic bearing loads P_{e1} and P_{e2}:

$$P_{e1} = X_{d1} F_r + Y_{d1} F_a = 3000N$$

$$P_{e2} = X_{d2} F_r + Y_{d2} F_a = 24280N$$

Here:

$$\rightarrow \quad P_e = 24280N$$

Example: Thrust from Thermal Expansion (contd.)

Solve for bearing life L:

$$L = \left(\frac{C_d}{I_F \cdot P_e}\right)^3 \cdot K_R \cdot 10^6 = \left(\frac{16800}{1.2 \cdot 24280}\right)^3 \cdot 0.33 \cdot 10^6 = 6.32 \cdot 10^4 rev$$

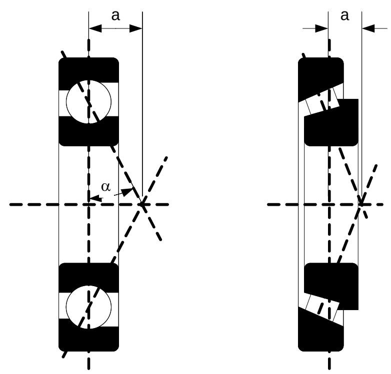
Or in terms of operating time:

$$t = \frac{L}{n} = \frac{6.23 \cdot 10^4 rev}{350 \frac{rev}{\min} \cdot 60 \frac{\min}{hr}} = 3.0 hr$$

The thermal expansion reduces the bearing life by a factor of 532!!!!

Angular Contact and Tapered Roller Bearings

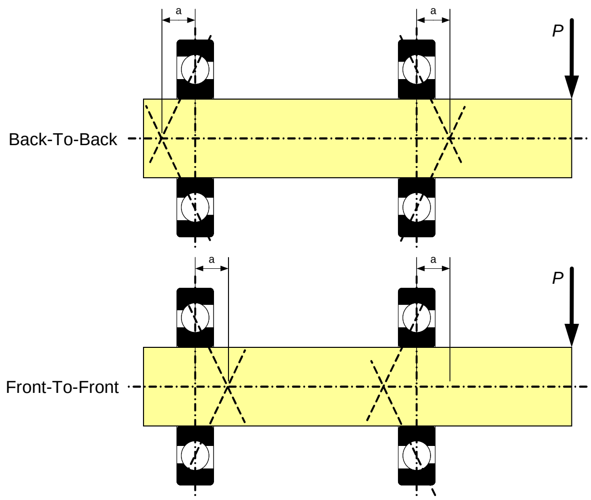
- All tapered bearings including angular contact ball bearings transmit loads from the inner to the outer ring at an angle with respect to the center line
- This angle creates an offset *a* from the center of the bearing



Angular Contact Bearing

Tapered Roller Bearing

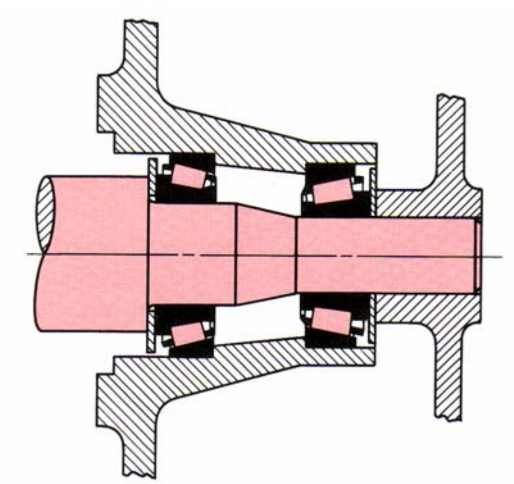
Two possible mounts: Back-To-Back (DB) and Front-To-Front (DF)



Back-To-Back mounting increases the effective spacing between the bearings, thereby reducing the bearing loads required to counteract the applied moment

Front-to-Front has reduced moment stiffness compared to Back-to-Back. For the case where the load lines intersect at the same point, moment stiffness is zero and the bearing is "self-aligning".

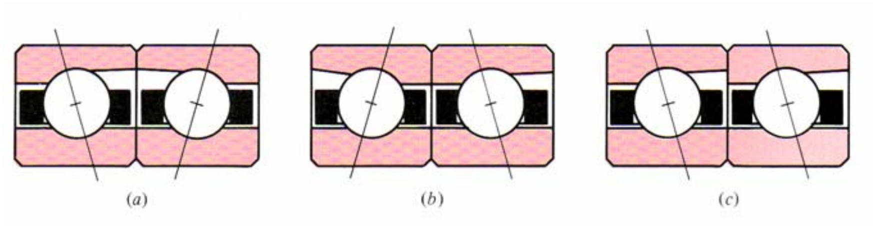
Back-To-Back Examples



Spindle of Front Loader Washing Machine

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

Angular Contact Bearing Arrangements



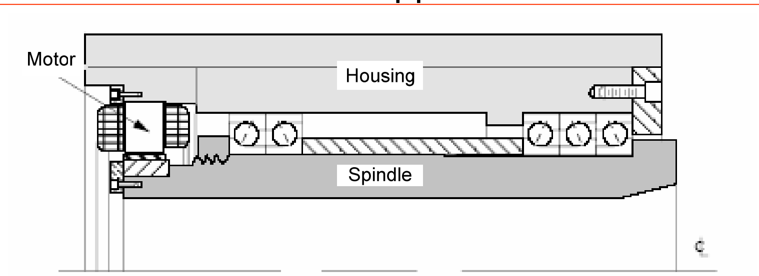
a) Front-To-Front (DF) mount. Reduced moment stiffness results in "self-alignment"
   - The system is thermally unstable for a rotating inner race
   - Differential radial and axial growth both tend to increase the preload

b) Back-To-Back (DB) mount. Increased moment stiffness results in rigid bearing support
   - For a fixed outer race, the shaft expands axially and radially more than the housing
   - The preload remains relatively constant

c) Parallel (DT) mount

Source: Shigley JE, Mischke CR, *Mechanical Engineering Design*

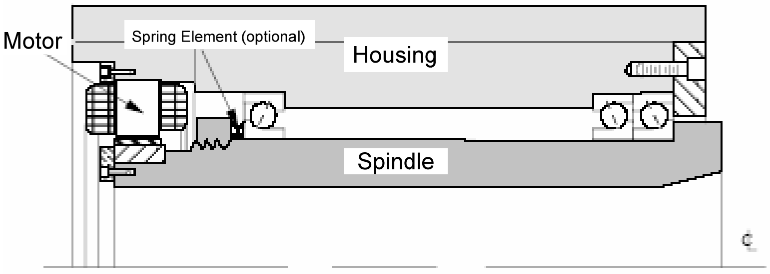
Mounting Strategies for Spindles Fixed-Supported



Source: Alexander Slocum, *Precision Machine Design*

- This is the most common, least expensive, and least desirable method
- It can lead to decreased radial stiffness and accuracy if the fit is too loose
- Early bearing failure if the fit is too tight
- Consider using a surface treatment in the bore to decrease friction

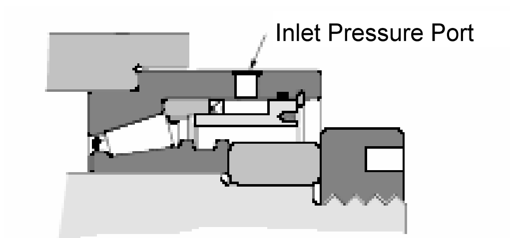
Mounting Strategies for Spindles Fixed-Thermocentric



Source: Alexander Slocum, *Precision Machine Design*

- The radial and axial spacing are set so thermal growths cancel each other
- Optionally, a spring element is used to preload the system to allow for deviations from ideal growth
- This is a very difficult configuration to design and thus is not often used for a multi-purpose spindle

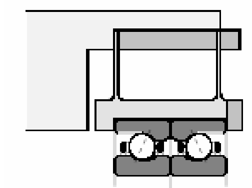
Mounting Strategies for Spindles - Fixed Preload



Source: Alexander Slocum, *Precision Machine Design*

- Use a hydraulic device to maintain a fixed preload on the bearings (Hydra-Rib™)
- This is an effective, but more expensive, method

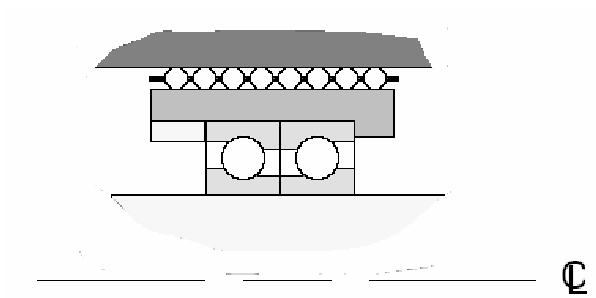
Mounting Strategies for Spindles
Fixed-Diaphragm Flexure



Source: Alexander Slocum, *Precision Machine Design*

- Use a diaphragm flexure to support the bearing set which must have freedom to move axially
  - This is a moderate cost method that does require extra room and thus is not often used
- Method for bearing mounting that provides:
  - Good radial stiffness
  - Low axial stiffness
  - Accommodation of axial thermal growth of the supported shaft

Mounting Strategies for Spindles
Fixed-Rolling Element

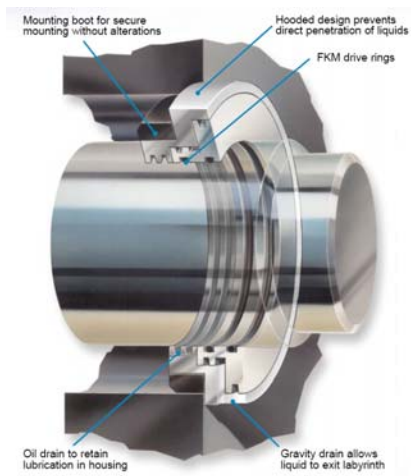


Source: Alexander Slocum, *Precision Machine Design*

- Use a die-set bushing as a micro-motion flexural rolling element bearing

Seals

- Rotary motion rolling element bearings are relatively easy to seal using labyrinth, wiper, or screw-thread type seals
- At high speeds, non-contact seals (Hydra Seals) are required to avoid heat generation
  - Even better sealing can be achieved with positive air pressure
- High pressure coolant bouncing off of a part can force its way past non-contact

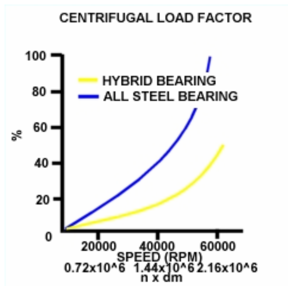


Accuracy

- Total axial and radial error motions can be as low as ¼ µm but typically are on the order of ½ - 1 µm
- Commercially available spindles, are available with total error motions (TIR/2) of 1/4 - 1/2 µm
- Angular contact bearings typically have three times the speed capability of radial contact bearings
- Angular contact bearings are generally the most often used type of ball bearing in precision machinery
- For high-speed spindles, ceramic balls are well suited

Ceramic Ball Bearings

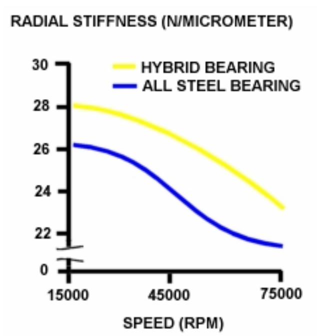
- 40% as dense as steel, the resulting reduction in weight reduces centrifugal forces imparted on the rings, reducing skidding, allowing 30 to 50% higher running speeds with less lubrication
- Ceramic balls have a smoother finish than steel, vibration and spindle deflection is reduced allowing higher speeds

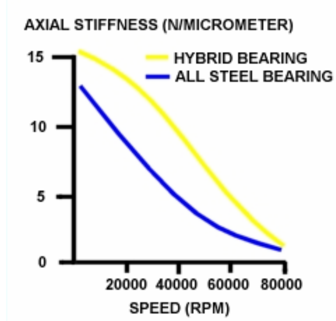


Source: Bearing Works

Ceramic Ball Bearings (contd.)

- Silicon nitride balls have a 50 % higher modulus of elasticity than steel, which means a 15 to 20% increase in rigidity, improving accuracy

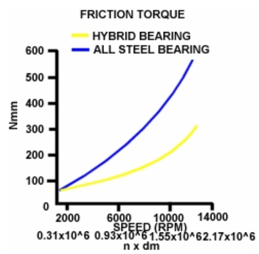


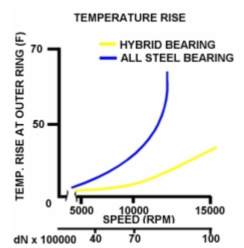


Source: Bearing Works

Ceramic Ball Bearings (contd.)

- A low thermal expansion coefficient allows HYBRIDS with ceramic balls to undergo smaller changes in contact angle reducing preload variations improving life and maintaining tolerances
- Ceramic Hybrid Ball Bearings truly are “anti-friction“
- Lower friction leads to less wear, less lubrication, less energy consumption, reduced sound level and extends life

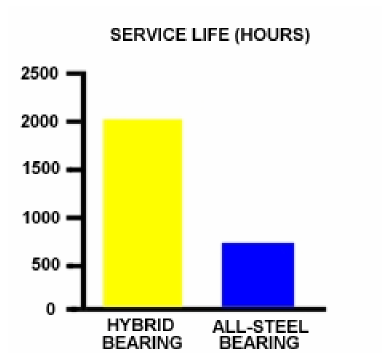




Source: Bearing Works

Ceramic Ball Bearings (contd.)

- With their numerous advantages ceramic hybrid ball bearings typically yield 5 to 10 times longer life than conventional steel-steel ball bearings in most applications



Source: Bearing Works