Needle Roller Bearings

Bearing Type Selection

Bearing type selection is made after the general design concept of the machine has been established and the magnitude of the loads and speeds estimated. Special conditions can directly affect bearing operation and must be considered. These include ambient or localized temperatures, shock or vibration, type of lubrication, dirt or abrasive contamination, difficulty in obtaining accurate alignment, space limitations, need for shaft rigidity, etc.

The fields of application for many types of bearings overlap, and the value of experience in bearing applications cannot be overemphasized.

Torrington's Sales and Engineering personnel have a wealth of experience in the design and application of rolling element bearings. Their knowledge can help solve your bearing application problems. **All new applications of Torrington Bearings should be submitted to Torrington for approval.**

These engineering services are rendered confidentially, and we urge you to use them since such procedure assures selection of the proper bearing in terms of maximum service and economy.

Each type of bearing has inherent features which determine its relative suitability for a specific application. Careful analysis of the features, and familiarization with the fundamental characteristics of each type of bearing, will help in selecting the proper bearing. The right is reserved to change design and specifications without notice.

The following information will provide guidance in your initial selection of the proper radial bearing. Thrust bearing type selection is discussed in the Thrust Bearing section.

FULL COMPLEMENT VERSUS CAGED NEEDLE ROLLER BEARINGS

Load rating – size for size, full complement bearings have more rollers. With more rollers contacting the raceway in the load zone, they generally have higher load ratings than caged bearings. This is particularly important under static, slow rotating, or oscillating conditions. Both types, however, have far greater capacity in less radial space than other types of rolling element bearings.

Slope Tolerance – Controlled Contour Rollers and the guidance of rollers supplied by the cage make the caged bearings preferable to the full complement bearing in applications where misalignment or shaft deflection is a factor.

Speed – for moderate speeds, both types are satisfactory. The caged bearing, however, permits higher speeds for a given shaft size.

Pregreased life – the expression refers to the maintenance-free life of a prelubricated bearing. It is significantly longer for a caged bearing because: (1) of its greater grease storage capacity, and (2) with less internal friction, it runs cooler.

Coefficient of friction - generally, the power loss in a rolling element bearing is so slight it can be ignored. Occasionally, the coefficient of friction must be known. The coefficient depends on many variables, such as speed, magnitude of load and lubricant viscosity and flow rate.

In a caged bearing, the frictional loss is less than in a full complement bearing because the cage improves roller guidance and reduces rubbing velocities by eliminating roller-to-roller contact.

No accurate method exists for predicting the coefficient of friction of a specific bearing under all conditions of operation. Approximate values for bearing friction under normal loads with oil lubrication have been established by test. From such tests, the coefficients of friction for radial bearings are:

$$\textbf{caged needle roller bearings} = 15 \cdot 10^{-4}$$
$$\textbf{full complement needle roller bearings} = 25 \cdot 10^{-4}$$

where the following formula defines the coefficient of friction:

$$\textbf{coefficient of friction} = \frac{\textbf{torque to turn bearing}}{\textbf{load on bearing} \cdot \textbf{bearing pitch radius}}$$

Cost – because of their lower cost, the various types of Drawn Cup bearings should be considered first for any application.

DRAWN CUP BEARINGS WITH DEEPLY HARDENED AND GROUND RACEWAYS

Load rating – the drawn cup bearing has an outer ring with a necessarily thin hardened case over a relatively ductile core. A bearing with a machined, deeply hardened and ground raceway therefore shows slightly higher load ratings than a drawn cup bearing of similar size and similar roller guidance method. However, a bearing with a deeply hardened ring, as compared with a drawn cup bearing, has more advantages when the applied loads are extremely heavy, beyond the strength of the case and core of the drawn cup bearing.

Cross section – a smaller, more compact cross section is provided by the drawn cup bearing, with its smaller roller diameter and thin steel outer ring, contrasted with the larger roller diameter and thicker outer ring of the heavy duty bearing.

Only caged needle roller assemblies and complements of loose needle rollers are more compact than drawn cup bearings, but they require the use of hardened outer and inner raceways.

Cost – because of their lower cost, the various types of Drawn Cup bearings should be considered first for any application which needs an integral hardened outer raceway.

Split housings – the heavy duty bearing, with a relatively thick and rigid outer ring, can be used in a split housing. The drawn cup bearing, because of its thin, drawn outer ring construction, depends on the strength and roundness of the housing to achieve its final bearing roundness and operational clearances.

Therefore, while it is possible to use a split housing with a heavy duty bearing, it is recommended that the drawn cup bearing be first installed in a steel sleeve to round and size the bearing before assembly in the split housing.

Mounting – heavy duty bearings should be located axially in the housing by shoulders or other devices. Drawn cup bearings offer attractive manufacturing economies, because they are pressed into through-bored housings and do not require any other means for axial location.

Bearing Size Selection

From the foregoing discussion of the operating characteristics of the full complement and caged needle roller bearings listed in this catalog, the designer can determine the type of bearing most suitable for the specific conditions of an application. In many cases, more than one bearing type will meet the operating criteria. In these instances, the designer should select the most suitable size of each type and then make a final selection on the basis of mounting simplicity, available space and over-all economy. Needle roller bearings tabulated in this catalog meet tolerances specified for them in the respective ANSI/ABMA and ISO standards.

The basic parameters of bearing size selection are radial load, thrust load, speed, required life, and raceway hardness conditions. Other factors, such as misalignment, abnormal temperature, contamination and poor lubrication, can seriously reduce service life, but their effect is not easily determined. Instead of attempting to estimate their effect on bearing life, every attempt should be made to eliminate these conditions through proper design.

BEARING LIFE AND RELIABILITY

The life of a bearing is expressed as the number of revolutions (or the number of hours at a given speed) that a bearing will complete before failure. Life will vary from one bearing to another, but stabilizes into a predictable pattern when considering a large group of the same size and type bearing. The “L_{10}” or “rating life” of a group of such bearings is defined as the number of revolutions (or hours at a given constant speed) that 90% of the tested bearings will complete or exceed before the first evidence of failure develops. Thus it can be predicted with 90% reliability that a bearing will meet or exceed the calculated “L_{10}” life providing normal fatigue is the failure mode.

Some critical applications, however, require definition of life at reliabilities greater than 90%. To determine bearing life with reliabilities greater than 90%, the “L_{10}” life as calculated must be adjusted by a factor a_1 so that $L_n = a_1L_{10}$. The life adjustment factors shown in Table 1 are recommended.

Table 1 - Life Adjustment Factors

Reliability %	L_n Rating Life	a_1 Life Adjustment Reliability Factor
90	L_{10}	1
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

In some applications, when safety or maintenance economy is not critical and low initial bearing cost is the prime consideration, the reliability level can be reduced to 50%, or the “L_{50}” life. The “L_{50}” life may be as much as five times the “L_{10}” rating life.

DEFINITION OF LOAD RATINGS

Basic dynamic load rating – The “basic dynamic load rating” (C_r) for a radial roller bearing is that calculated, constant, radial load, which a group of apparently identical bearings with stationary outer ring can theoretically endure for a rating life of one million revolutions of the inner ring. For a thrust roller bearing (C_a) is that calculated, constant, centric thrust load, which a group of apparently identical bearings can theoretically endure for a rating life of one million revolutions of the bearing washers.The basic dynamic load rating is a reference value only, the base value of one million revolutions has been chosen for ease of calculation. Since applied loading as great as the basic dynamic load tends to cause local plastic deformation of the rolling surfaces, it is not anticipated that such heavy loading would normally be applied.

The dynamic load ratings tabulated in this catalog are based on extensive test and field experience. Torrington dynamic load ratings are not necessarily the same as those calculated using the recommendations of ISO and other standard institutes. Use of the Torrington load ratings results in the selection of bearings, which perform in actual service as predicted by this catalog as long as satisfactory conditions of lubrication, cleanliness, alignment, etc., are maintained. Torrington load ratings are denoted by the column heading Ⓣ . ISO 281 ratings are tabulated for comparison purposes only.

Basic static load rating – for a radial roller bearing suitably manufactured from a good quality hardened alloy steel, the static radial load C_or is that uniformly distributed static radial bearing load, which produces a maximum contact stress of 4000 megapascals (580,000 PSI) acting at the center of contact of the most heavily loaded rolling element.The static axial load rating (C_{oa}) is that uniformly distributed static centric axial bearing load, which produces a maximum contact stress of 4000 megapascals (580,000 PSI) acting at the center of contact of the each rolling element.
Note: For a contact stress of 4000 MPa, a total permanent deformation of roller and raceway occurs, which is approximately 0.0001 of the roller diameter.

Dynamic working load – For most industrial applications, the maximum dynamic working load should not exceed the basic dynamic load rating Ⓣ or the tabulated working load whichever is smaller.

Load ratings for bearings used in airframes – Airframe designers commonly use the terms “limit load” rating and “ultimate” or “static fracture load” rating.

Limit load rating – (the working load for airframe bearings) can be defined as the maximum radial load which can be applied to a bearing without impairing the subsequent functioning of the bearing in airframe applications.

Ultimate or static fracture load rating is not less than 1.5 times the limit load rating.

Aircraft static capacity, where listed, is based on the rolling elements of the bearing only. For properly housed bearings, the aircraft static capacity corresponds to the ultimate or static fracture load rating.

For capacities of airframe bearings operating as track rollers and for capacities of tracks on which the bearings roll please refer to the airframe bearings section of this catalog.

Bearing Size Selection *(continued)*

LIFE AND LOAD RELATIONSHIP

Empirical calculations and experimental data point to a predictable relationship between bearing load and life. This relationship may be expressed by a formula. In this empirical formula, the bearing life is found to vary inversely as the applied load to an exponential power. The assigned value of the exponent depends on the basic type of rolling element. For all types of Torrington needle roller bearings the formula is:

$$\underset{\text{(millions of revolutions)}}{\text{life}} = \left(\frac{\text{basic dynamic load rating}}{\text{applied load}}\right)^{10/3}$$

Consequently, if the load applied to a given bearing is increased by a factor of 2 the life is decreased by a factor of 10.

Since bearing life in revolutions equals rpm • 60 • life in hours, an increase in speed results in a decrease in hours of life as long as other factors are the same. For example, if the speed is doubled, the hours of life are halved. Conversely, if the speed is halved the hours of life are doubled.

When the applied load is greater than one-half of the basic dynamic load rating, the above load-life relationship is no longer valid. Consult the Torrington engineering department for recommendations.

Table 2- Working Load Factors (Static)

Raceway Hardness HRC	Working Load Factor (WLF)
58	1.00
57	1.06
56	1.13
55	1.21
54	1.29
53	1.37
52	1.46
51	1.55
50	1.65
49	1.76
48	1.88
47	2.00
46	2.13
45	2.27
44	2.41
43	2.57
42	2.74
41	2.92
40	3.10

EFFECT OF RACEWAY HARDNESS

Both the dynamic and static load ratings of bearings in this catalog are based on a minimum raceway hardness equivalent to 58 HRC (ref.ASTM, E-18). If the raceway hardness must be decreased, then these load ratings must be reduced.

When the loading is static, it is usually recommended that the static load rating (or limit load rating for airframe bearings) divided bythe appropriate factor as shown in Table 2.

The raceway hardness also affects the life of the bearing application even though the applied load is less than the maximum recommended working load discussed above. Table 3 shows the Hardness Factors (HF) by which the basic dynamic load rating should be divided where the raceway hardness must be less than 58 HRC or equivalent.

Table 3 - Hardness Factors to Modify Basic Dynamic Load Rating

Raceway Hardness HRC	Hardness Factor HF
58	1.00
57	1.02
56	1.04
55	1.07
54	1.12
53	1.19
52	1.28
51	1.41
50	1.59
49	1.78
48	2.00
47	2.24
46	2.50
45	2.76
44	3.06
43	3.39
42	3.77
41	4.16
40	4.55

Bearing Life Calculations

To calculate L_{10} life for a selected bearing under given conditions of speed, load and raceway hardness, use the general formula:

$$LF = \frac{C_r \text{ (or } C_a)}{SF \cdot P \cdot HF}$$

to obtain the Life Factor (LF). Then simply read off the hours of L_{10} life from the Life Chart shown at the right (and on tabular pages). In this formula:

C_r = Basic dynamic load rating as found on the tabular pages (C_a for thrust bearings).
SF = Speed Factor, obtained from speed chart shown at right (and on tabular pages).
P = Applied Load
HF = Hardness Factor (See Table 3)

In those instances where calculating the required basic dynamic load rating facilitates the choice of a bearing, use the same formula restated as follows:

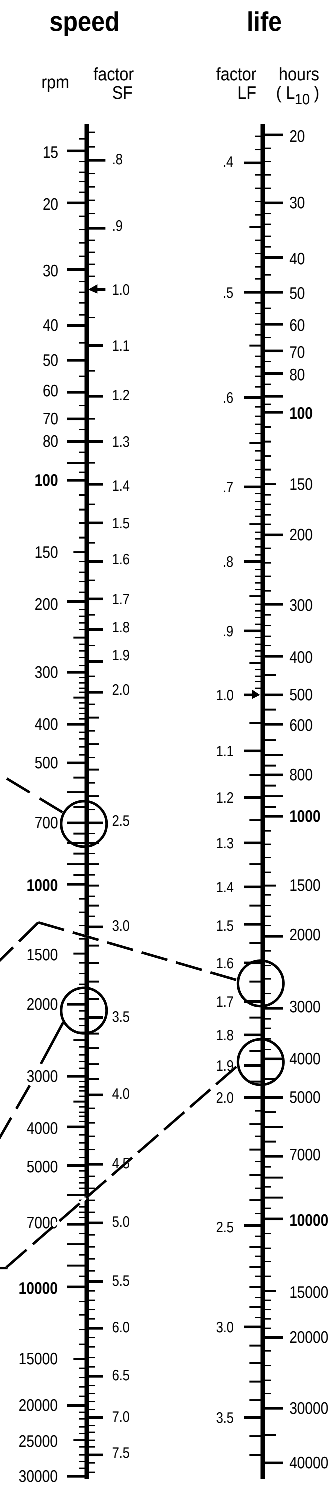
$$\text{Required } C_r \text{ (or } C_a) = P \cdot SF \cdot LF \cdot HF$$

Similarly, the load a bearing will support under various conditions of speed and life can be calculated from this restatement of the basic formula:

$$\text{allowable load at given conditions} = \frac{C_r \text{ (or } C_a)}{SF \cdot LF \cdot HF}$$

The equations above are based on ideal conditions, and if excessive vibration, lubrication problems, shock loading, extreme temperatures, unusual misalignment, or any other abnormal conditions are present, consult Torrington for recommendations. In oscillating applications, especially those of very small amplitudes, fretting corrosion can be a problem requiring consultation.

Consult our Bearing Engineering Department before final bearing selection is made.



Example: A review of operating conditions and other factors (see Bearing type Selection) leads to the selection of a full complement drawn cup needle roller bearing. The available space accommodates the B-1212 bearing. Find the L_{10} life under the following conditions:

Speed (rpm) = 700 (SF = 2.5)
Applied radial load (P) = 490 lbf* *See note below.
Shaft raceway hardness = 50 HRC
(HF = 1.59)
Working Load (max.) = 4710 lbf
Basic dynamic load rating (C_r) = 3160 lbf
* Applied load of 490 lbf is satisfactory since it is less than the basic dynamic load rating C_r.

$$LF = \frac{3160}{2.50 \cdot 490 \cdot 1.59} = 1.63$$

From nomograph L_{10} = 2530 hours

Example: Find the required basic dynamic load rating (C_r) for a bearing which will carry an applied radial load (P) of 260 lbf for 4000 hours minimum life (L_{10}) at 2000 rpm and raceway hardness of 58 HRC:

Applied radial load (P) = 260 lbf
Speed = 2000 rpm (SF = 3.41)
Minimum life (L_{10}) = 4000 hrs. (LF = 1.87)
Raceway hardness = 58 HRC (HF = 1.00)
$C_r = 260 \cdot 3.41 \cdot 1.87 \cdot 1.00 = 1658$ lbf
* Note: When using the required C_r to select a type DC bearing, be sure to match against values in the Ⓣ column of the tables of dimensions and capacities.
Thus, the J-1212 with C_r = 2320 lbf might be chosen.
The applied load is well below the allowable working load of the J-1212 which, on the shaft raceway of 58 HRC hardness, is the C_r value of 2320 lbf.

Bearing Life Calculations *(continued)*

BEARING LIFE UNDER VARIABLE LOADS AND SPEEDS, EQUIVALENT LOAD

When a bearing is subjected to varying loads and speeds the equivalent (constant) load (P_e) on the bearing can be determined using the following formula:

$$P_e = \sqrt[10/3]{\frac{(F_1)^{10/3}\ pm_1 \bullet t_1 + (F_2)^{10/3} \bullet rpm_2 \bullet t_2 + \ldots (F_n)^{10/3} rpm_n \bullet t_n}{N}}$$

where F_1, F_2,…F_n represent loads acting at speeds rpm1, rpm_2,… rpm_n and t_1,t_2…t_n represent the decimal portion of the total time that F_1 is acting at rpm_1, F_2 is acting at rpm_2…F_n is acting at rpm_n. The weighted rpm, N, is the numerical summation of t_1 rpm_1 plus t_2 rpm_2 plus…t_n rpm_n.

BEARING LIFE FOR GIVEN LOAD AND SPEED

When a bearing must be selected to meet given load and speed conditions, the following formula is used to compute the required basic dynamic load rating:

C_r (or C_a) = P_e • LF • SF

where C_r (or C_a) = basic dynamic load rating

LF = life factor for the total hours the bearing must operate

SF = speed factor for weighted rpm (N)

P_e = equivalent load

If, however, the bearing to be used has already been selected, the following formula is used to determine whether the total desired hours can be achieved:

$$LF = \frac{C_r \text{ (or } C_a)}{P_e \bullet Sf}$$

Additional Factors Affecting Bearing Operation

Lubrication

The basic dynamic load ratings listed in this catalog assume the use of mineral or synthetic oils with sufficient lubricity and a minimum viscosity of 100 Saybolt Universal Seconds (S.U.S.) or 20 centistokes (cSt) at the bearing operating temperature. If the viscosity is less than this, or the oils lack lubricity, the life of the bearing may be reduced.

Since the lubricant affects bearing life and operation, selecting the proper lubricant is an important design function. The purpose of lubrication in bearing application is :

(1) to minimize friction at points of contact within the bearings

(2) to protect the highly finished bearing surfaces from corrosion

(3) to dissipate heat generated within the bearings

(4) to remove or prevent entry of foreign matter.

Either oil or grease may be used with rolling element bearings. Each has its advantages and limitations.

Since oil is a liquid, it lubricates all surfaces and dissipates heat from these surfaces more readily. It is generally used for high speed applications. Oil lubricants can be circulated, cleaned and cooled for more effective lubrication.

Grease, which is easier to retain in the bearing housing, aids as a sealant against foreign matter.

However, the grease must also be compounded with mineral or synthetic oils whose viscosity is greater than 100 S.U.S. or 20 centistokes at the operating temperature and have good lubricity.

Frequent replenishing of the grease may be necessary for optimum performance.

Some bearing types in this catalog are prepacked with a grease suitable for their normal application. For instance, airframe bearings are supplied prepacked with a special wide temperature range aircraft grease. The stud and yoke type track rollers are prepacked with a medium temperature grease.

The remainder of the bearings are normally shipped protected with corrosion-preventative compound which is not a lubricant. Such bearings may be used in oil lubricated applications without removal of the corrosion preventative compound. The packing label indicates the type of lubricant or corrosion preventative compound within the bearing.

When specified by the customer, other Torrington bearings may be ordered prelubricated with suitable greases and oils. Care must be exercised in lubricant selection since different lubricants are often incompatible.

LIMITING SPEEDS

Since load, lubrication method, temperature and other factors affect the maximum speeds at which bearings will operate, it is impossible to determine precise limiting speeds. Th elimiting speeds listed in this catalog can readily be reached by moderately loaded bearings using normal splash or drip feed mineral oils. Heavy loads or oils of less lubricity may require increased lubricant flow rate to dissipate heat at high bearing speed.

With carefully controlled geometry and improved lubrication provisions, the limiting speeds may be exceeded.

Grease lubricated bearings can reach the listed limiting speeds as long as the loads are light and the frictional heat can be readily dissipated through the shaft and housing.

Only testing will determine the ultimate speed which can be reached in a specific application.

MISALIGNMENT AND OPERATING DEFLECTION

The effect of misalignment on operating deflection may result in the shaft raceway or inner ring having a slope relative to the centerline of the path of rollers in the bearing. Such slope causes an unequal distribution of contact stress along the length of the rollers and a subsequent reduction in bearing life.

The load ratings in this catalog are sufficiently conservative so that if the slope values in Table 4 are not exceeded, the calculated bearing life should be achieved.

With reasonable attention to design, deflection of the housing and shaft can be balanced so that in most applications the slopes tabulated need not be exceeded. If such design cannot be achieved, bearing function will be adversely affected.

Table 4 – Slope Values

Bearing Width	Maximum Slope	
	Caged	Full Complement
>2 in. or 50 mm	$5 \cdot 10^{-4}$	$5 \cdot 10^{-4}$
1-2 in. or 25 - 50 mm	$10 \cdot 10^{-4}$	$5 \cdot 10^{-4}$
<1 in. or 25 mm	$15 \cdot 10^{-4}$	$10 \cdot 10^{-4}$

Shaft Design

When the shaft is used as the inner raceway for needle roller bearings the following specifications must be met:

1. metallurgy – either case hardening or through hardening grades of good bearing quality steel are satisfactory for raceways. Steels which are modified for free machining, such as those high in sulfur content and particularly those containing lead, are seldom satisfactory for raceways.

To realize full bearing capacity, the raceway area must be at least surface hard with a reasonable core strength. The preferred surface hardness is equivalent to 58 HRC (ref. ASTM, E18). If the raceway is of lesser hardness, see the modification factors shown in Tables 2 and 3.

Shaft raceways for all needle roller bearings, in diameters up to 3.5 inches or 90mm should have an effective case depth of 0.030 inch or 0.8mm. (Effective case depth is defined as the distance from the surface, after final grinding, to the 50 HRC hardness level.) For raceways larger than 3.5 inches or 90 mm in diameter the effective case depth should be 0.050 inch or 1.3 mm.

2. strength – the shaft must be of sufficient size to keep the operating deflections within the limits outlined in Table 4.

3. tolerance – the recommended shaft diameter tolerances for each type of needle roller bearing are indicated on the tabular pages.

4. taper – the taper within the length of the bearing raceway should not exceed 0.0003 inch (0.008 mm), or one-half the diameter tolerance, whichever is smaller.

5. out-of-roundness – the radial deviation from true circular form of the raceway should not exceed .0001 inch (0.0025mm) for diameters up to and including 1.0 inch (25mm). For raceways greater than 1.0 inch or 25mm the allowable radial deviation may be greater than .0001 inch (0.0025mm) by a factor of raceway diameter (in inches) divided by 1.0 or a factor of raceway diameter (in mm) divided by 25.

6. surface finish – the raceway finish should not exceed 16 microinches aa (arithmetic average) or 0,4 µm (on the Ra scale). In addition, the raceway area must be free of nicks, scratches and dents. Oil holes are permissible in the raceway area but care must be taken to blend the edges gently into the raceway.

Care must be taken to prevent grind reliefs, fillets, etc., from extending into the raceway area. If the rollers overhang a grind relief or step on the shaft, there will be high stress concentration with resultant early failure.

7. end chamfer – for most effective assembly of the shaft into a bearing, the end of the shaft should have a large chamfer or rounding. This should help in preventing damage to the roller complement, scratching of the raceway surface and nicking of the shaft end.

8. sealing surface – in some instances bearings have integral or immediately adjacent seals that operate on the surface ground for the bearing raceway. Here, particular attention should be paid to the pattern of the shaft finish. In no instance should there be a “lead”, or spiral effect, as often occurs with through feed centerless grinding. Such a “lead” may pump lubricant past the seal.

When it is undesirable or impractical to prepare the shaft to be used as a raceway, inner rings are available as listed in the tabular pages. If the shaft is not used directly as a raceway, the following design specifications must be met:

1. **strength** – the shaft must be of sufficient size to keep the operating deflections within the limits outlined in Table 4.

2. **tolerance** – the recommended shaft diameter tolerances for mounting inner rings are indicated on the tabular pages.

3. **taper and out-of-roundness** – the taper and out-of-roundness should not exceed one-half the shaft diameter tolerance.

4. **surface finish** – the surface finish should not exceed 125 micro-inches, aa (arithmetic average) or 3.2 µm (on the Ra scale).

5. **locating shoulders or steps** – locating shoulders or steps in the shaft must be held to close concentricity with the bearing seat to prevent imbalance and resultant vibrations.

Housing Design

BEARINGS WITH OUTER RINGS

For bearings with outer rings the function of the housing is to locate and support the outer ring. The following specifications must be met:

1. strength – housings should be designed so that the radial loads which will be placed on the bearings will cause a minimum of deflection or distortion of the housing.

2. tolerance – the recommended housing bore tolerances for each type of needle roller bearing are indicated on the tabular pages.

3. taper – the taper within the length of the outer ring should not exceed .0005 inch (0.013 mm).

4.out-of-roundness – the housing bore should be round within one-half the housing bore diameter tolerance.

5. parallelism – when possible, line bore housings which are common to one shaft to obtain parallelism of the housing bores and the shaft axis.

6. surface finish – the surface finish of housing bore should not exceed 125 microinches, aa (arithmetic average) or 3.2 µm (on the Ra scale).

7. end chamfer – to permit easy introduction of the bearing into the housing, the end of the housing should have a generous chamfer.

Heavy duty roller bearings can be installed into housings with a transition fit or a clearance fit. The outer ring should be a transition fit in the housing when it rotates relative with the load. The outer ring may be a clearance fit in the housing when it is stationary relative to the load.In either case, locate the bearings by shoulders, or other locating devises, to prevent axial movement.

Since the heavy duty roller bearing does not require an interference fit in the housing to round and size it properly, a split housing may be used if desired. Dowels should be used to maintain proper register of the housing sections.

Drawn cup bearings have a thin case-hardened outer ring which is out-of-round from the hardening operation. For proper mounting it must always be pressed into the housing. Split housing will not round and size a drawn cup bearing. When split housings must be used, the bearing should first be mounted in a cylindrical sleeve.

The housing should be of sufficient tensile strength and section to round and size the bearing. It must be designed for minimum distortion under load. Steel or cast iron housings are preferred. Housing bores in low tensile strength materials such as aluminum, magnesium, phenolics, etc., should be reduced to provide more interference fit. Thin section cast iron and steel housings may also require reduced bores. Consult the Torrington engineering department for recommendations when working with these lower strength housings.

The housing should be through bored if possible. When shouldered housing bores are unavoidable, the bearing should be located far enough from the shoulder to avoid the danger of crushing the end of the drawn cup during installation.

When the drawn cup bearing is mounted close to the housing face, care should be taken to mount the bearing at least 0.008 inch (0.20 mm) within the housing face to protect the bearing lip.

BEARINGS WITHOUT OUTER RINGS

In many cases, such as with gear bores, it is desirable to have the housing bore serve as the outer raceway for caged needle roller assemblies or loose needle roller complements. In those instances, the following specifications must be met:

1. strength – the housing must be of sufficient cross section to maintain proper roundness and running clearance under the maximum load.

2. metallurgical – material selection, hardness and case depth should be consistent with the requirements for inner raceways given in the SHAFT DESIGN recommendations on page E77.

3. taper and out-of-roundness – the raceway out-of-roundness and taper should not exceed 0.0003 inch (0.008 mm) or one-half the bore tolerance, whichever is smaller. In addition, the bore diameter must never be smaller at both ends than in the center (sway-back).

4. surface finish – the raceway surface finish should not exceed 32 microinches, aa (arithmetic average) or 0.8 µm (on the Ra scale). In addition, the surface must be free of nicks, dents and scratches.

5. grind reliefs – care must be exercised to ensure that grind reliefes, fillets, etc. do not extend to the raceway. Oil holes in the raceway area are permissible but the edges must be blended smoothly with the raceway.

CONTROLLED CONTOUR ROLLERS

Roller bearing life is affected by the distribution of contact stress between roller and raceways. Even when cylindrical rollers are loaded under conditions of ideal alignment, the contact stress is not uniform along the length of the rollers, but rather is concentrated towards the ends. Misalignment causes even greater roller contact stress. This effect is illustrated in Figure 1.

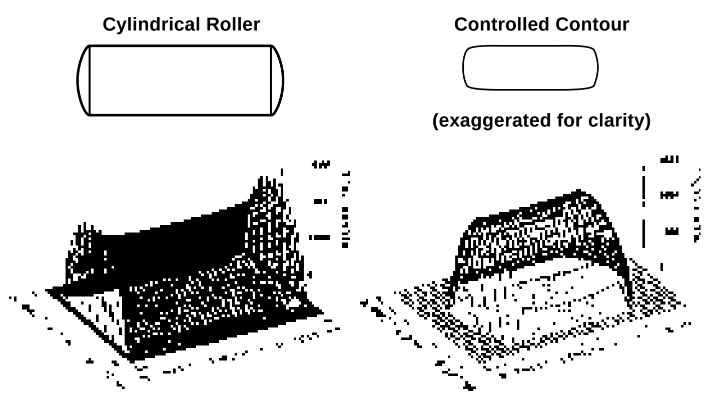


Figure 1: Comparative Stress Patterns

The use of Controlled Contour rollers in certain types and sizes of Torrington Bearings helps to reduce stress concentration at the ends of rollers, both under misalignment or ideal alignment, and results in more uniform stress distribution and optimum bearing performance.

ISO BORE TOLERANCES-MILLIMETERS

Nominal Diameters		F7		G7		H8		N6		N7		R6		R7	
Over	Incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low
6	10	+0.028	+0.013	+0.020	+0.005	+0.022	0	-0.007	-0.016	-0.004	-0.019	-0.016	-0.025	-0.013	-0.028
10	18	+0.034	+0.016	+0.024	+0.006	+0.027	0	-0.009	-0.02	-0.005	-0.023	-0.020	-0.031	-0.016	-0.034
18	30	+0.041	+0.020	+0.028	+0.007	+0.033	0	-0.011	-0.024	-0.007	-0.028	-0.024	-0.037	-0.020	-0.041
30	50	+0.050	+0.025	+0.034	+0.009	+0.039	0	-0.012	-0.028	-0.008	-0.033	-0.029	-0.045	-0.025	-0.050
50	65	+0.060	+0.030	+0.040	+0.010	+0.046	0	-0.014	-0.033	-0.009	-0.039	-0.035	-0.054	-0.030	-0.060
65	80	+0.060	+0.030	+0.040	+0.010	+0.046	0	-0.014	-0.033	-0.009	-0.039	-0.037	-0.056	-0.032	-0.062
80	100	+0.071	+0.036	+0.047	+0.012	+0.054	0	-0.016	-0.038	-0.010	-0.045	-0.044	-0.066	-0.038	-0.073
100	120	+0.071	+0.036	+0.047	+0.012	+0.054	0	-0.016	-0.038	-0.010	-0.045	-0.047	-0.069	-0.041	-0.076
120	140	+0.083	+0.043	+0.054	+0.014	+0.063	0	-0.020	-0.045	-0.012	-0.052	-0.056	-0.081	-0.048	-0.088
140	160	+0.083	+0.043	+0.054	+0.014	+0.063	0	-0.020	-0.045	-0.012	-0.052	-0.058	-0.083	-0.050	-0.090
160	180	+0.083	+0.043	+0.054	+0.014	+0.063	0	-0.020	-0.045	-0.012	-0.052	-0.061	-0.086	-0.053	-0.093
180	200	+0.096	+0.050	+0.061	+0.015	+0.072	0	-0.022	-0.051	-0.014	-0.060	-0.068	-0.097	-0.060	-0.106
200	225	+0.096	+0.050	+0.061	+0.015	+0.072	0	-0.022	-0.051	-0.014	-0.060	-0.071	-0.100	-0.063	-0.109
225	250	+0.096	+0.050	+0.061	+0.015	+0.072	0	-0.022	-0.051	-0.014	-0.060	-0.075	-0.104	-0.067	-0.113

ISO SHAFT TOLERANCES-MILLIMETERS

Nominal Diameters		f5		f6		h5		h6		j6		m6	
Over	Incl.	high	low	high	low	high	low	high	low	high	low	high	low
3	6	-0.010	-0.015	-0.010	-0.018	0	-0.005	0	-0.008	+0.006	-0.002	+0.012	+0.004
6	10	-0.013	-0.019	-0.013	-0.022	0	-0.006	0	-0.009	+0.007	-0.002	+0.015	+0.006
10	18	-0.016	-0.024	-0.016	-0.027	0	-0.008	0	-0.011	+0.008	-0.003	+0.018	+0.007
18	30	-0.020	-0.029	-0.020	-0.033	0	-0.009	0	-0.013	+0.009	-0.004	+0.021	+0.008
30	50	-0.025	-0.036	-0.025	-0.041	0	-0.011	0	-0.016	+0.011	-0.005	+0.025	+0.009
50	80	-0.030	-0.043	-0.030	-0.049	0	-0.013	0	-0.019	+0.012	-0.007	+0.030	+0.011
80	120	-0.036	-0.051	-0.036	-0.058	0	-0.015	0	-0.022	+0.013	-0.009	+0.035	+0.013
120	180	-0.043	-0.061	-0.043	-0.068	0	-0.018	0	-0.025	+0.014	-0.011	+0.040	+0.015

ISO BORE TOLERANCES - INCH

Nominal Diameters		F7		G7		H8		N6		N7		R6		R7	
Over	Incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low
0.2362	0.3937	+0.0011	+0.0005	+0.0008	+0.0002	+0.0009	0	-0.0003	-0.0006	-0.0002	-0.0007	-0.0006	-0.001	-0.0005	-0.0011
0.3937	0.7087	+0.0013	+0.0006	+0.0009	+0.0002	+0.0011	0	-0.0004	-0.0008	-0.0002	-0.0009	-0.0008	-0.0012	-0.0006	-0.0013
0.7087	1.1811	+0.0016	+0.0008	+0.0011	+0.0030	+0.0013	0	-0.0004	-0.0009	-0.0003	-0.0011	-0.0009	-0.0015	-0.0008	-0.0016
1.1811	1.9685	+0.0020	+0.0010	+0.0013	+0.0004	+0.0015	0	-0.0005	-0.0011	-0.0003	-0.0013	-0.0012	-0.0018	-0.001	-0.002
1.9685	2.5591	+0.0024	+0.0012	+0.0016	+0.0004	+0.0018	0	-0.0006	-0.0013	-0.0004	-0.0015	-0.0014	-0.0021	-0.0012	-0.0024
2.5591	3.1496	+0.0024	+0.0012	+0.0016	+0.0004	+0.0018	0	-0.0006	-0.0013	-0.0004	-0.0015	-0.0015	-0.0022	-0.0013	-0.0024
3.1496	3.9370	+0.0028	+0.0014	+0.0018	+0.0005	+0.0021	0	-0.0006	-0.0015	-0.0004	-0.0018	-0.0017	-0.0026	-0.0015	-0.0029
3.9370	4.7244	+0.0028	+0.0014	+0.0018	+0.0005	+0.0021	0	-0.0006	-0.0015	-0.0004	-0.0018	-0.0018	-0.0027	-0.0016	-0.003
4.7244	5.5118	+0.0033	+0.0017	+0.0021	+0.0006	+0.0025	0	-0.0008	-0.0018	-0.0005	-0.002	-0.0022	-0.0032	-0.0019	-0.0035
5.5118	6.2992	+0.0033	+0.0017	+0.0021	+0.0006	+0.0025	0	-0.0008	-0.0018	-0.0005	-0.002	-0.0023	-0.0033	-0.002	-0.0035
6.2992	7.0866	+0.0033	+0.0017	+0.0021	+0.0006	+0.0025	0	-0.0008	-0.0018	-0.0005	-0.002	-0.0024	-0.0034	-0.0021	-0.0037
7.0866	7.8740	+0.0038	+0.0020	+0.0024	+0.0006	+0.0028	0	-0.0009	-0.002	-0.0006	-0.0024	-0.0027	-0.0038	-0.0024	-0.0042
7.8740	8.8583	+0.0038	+0.0020	+0.0024	+0.0006	+0.0028	0	-0.0009	-0.002	-0.0006	-0.0024	-0.0028	-0.0039	-0.0025	-0.0043
8.8583	9.8425	+0.0038	+0.0020	+0.0024	+0.0006	+0.0028	0	-0.0009	-0.002	-0.0006	-0.0024	-0.003	-0.0041	-0.0026	-0.0044

ISO SHAFT TOLERANCES - INCH

Nominal Diameters		f5		f6		h5		h6		j6		m6	
Over	Incl.	high	low	high	low	high	low	high	low	high	low	high	low
0.1181	0.2362	-0.0004	-0.0006	-0.0004	-0.0007	0	-0.0002	0	-0.0003	+0.0002	-0.0001	+0.0005	+0.0002
0.2362	0.3937	-0.0005	-0.0007	-0.0005	-0.0009	0	-0.0002	0	-0.0004	+0.0003	-0.0001	+0.0006	+0.0002
0.3937	0.7087	-0.0006	-0.0009	-0.0006	-0.0011	0	-0.0003	0	-0.0004	+0.0003	-0.0001	+0.0007	+0.0003
0.7087	1.1811	-0.0008	-0.0011	-0.0008	-0.0013	0	-0.0004	0	-0.0005	+0.0004	-0.0002	+0.0008	+0.0003
1.1811	1.9685	-0.001	-0.0014	-0.001	-0.0016	0	-0.0004	0	-0.0006	+0.0004	-0.0002	+0.0010	+0.0004
1.9685	3.1496	-0.0012	-0.0017	-0.0012	-0.0019	0	-0.0005	0	-0.0007	+0.0004	-0.0003	+0.0012	+0.0004
3.1496	4.7244	-0.0014	-0.002	-0.0014	-0.0023	0	-0.0006	0	-0.0009	+0.0005	-0.0004	+0.0014	+0.0005
4.7244	7.0866	-0.0017	-0.0024	-0.0017	-0.0027	0	-0.0007	0	-0.0010	+0.0006	-0.0004	+0.0016	+0.0006

Conversion Tables

TO CONVERT FROM	TO		MULTIPLY BY
Acceleration			
foot/second^2	meter/second^2	m/s^2	0.3048
inch/second^2	meter/second^2	m/s^2	0.0254
Area			
foot^2	meter^2	m^2	0.09290304
inch^2	meter^2	m^2	0.00064516
inch^2	millimeter^2	mm^2	645.16
yard^2	meter^2	m^2	0.836127
mile^2 (U S. statute)	meter^2	m^2	2589988
Bending Moment or Torque			
dyne-centimeter	newton-meter	N • m	0.0000001
kilogram-force-meter	newton-meter	N • m	9.806650
pound-force-inch	newton-meter	N • m	0.1129848
pound-force-foot	newton-meter	N • m	1.355818
Energy			
B.T.U. (International Table)	joule	J	1055.056
foot-pound-force	joule	J	1.355818
kilowatt-hour	megajoule	MJ	3.6
Force			
kilogram-force	newton	N	9.806650
kilopond-force	newton	N	9.806650
pound-force (lbf avoirdupois)	newton	N	4.448222
Length			
fathom	meter	m	1.8288
foot	meter	m	0.3048
inch	millimeter	mm	25.4
microinch	micrometer	um	0.0254
micron (µn)	millimeter	mm	0.0010
mile (U.S. statute)	meter	m	1609.344
yard	meter	m	0.9144
nautical mile (UK)	meter	m	1853.18
Mass			
kilogram-force-second^2/meter (mass)	kilogram	kg	9.806650
kilogram-mass	kilogram	kg	1.0
pound-mass (Ibm avoirdupois)	kilogram	kg	0.4535924
ton (long, 2240 Ibm)	kilogram	kg	1016.047
ton (short, 2000 Ibm)	kilogram	kg	907.1847
tonne	kilogram	kg	1000.000
Power			
BTU (International Table)/hour	watt	W	0.293071
8TU (International Table)/minute	watt	W	17.58426
horsepower (550 ft lbf/s)	kilowatt	kW	0.745700
BTU (therrnochemical)/minute	watt	W	17.57250
Pressure or Stress IForceIAreal			
newton/meter^2	pascal	Pa	1 .0000
kilogram-force/centimeter^2	pascal	Pa	98066.50
kilogram-force/meter^2	pascal	Pa	9.806650
kilogram-force/millirneter^2	pascal	Pa	9806650
pound-force/foot^2	pascal	Pa	47.88026
pound-force/inch^2 (psi)	megapascal	MPa	0.006894757
Temperature			
degree Celsius	degree Kelvin	°K	$t_k = t_c + 273.15$
degree Fahrenheit	degree Kelvin	°K	$k = \frac{5}{9}(t_f + 459.67)$
degree Fahrenheit	degree Celsius	°C	$t_c = \frac{5}{9}(t_f - 32)$
Velocity			
foot/minute	meter/second	m/s	0.00508
foot/second	meter/second	m/s	0.3048
inch/second	meter/second	m/s	0.0254
kilometer/hour	meter/second	m/s	0.27778
mile/hour (U.S. statute)	meter/second	m/s	0.44704
mile/hour (U.S. statute)	kilometer/hour	km/h	1.609344
Volume			
foot^3	meter^3	m^3	0.02831685
gallon (U.S. liquid)	liter	*l*	3.785412
liter	meter^3	m^3	0.001
inch^3	meter^3	m^3	0.00001638706
inch^3	centimeter^3	cm^3	16.38706
inch^3	millimeter^3	mm^3	16387.06
ounce (U.S. fluid)	centimeter^3	cm^3	29.57353
yard^3	meter^3	m^3	0.7645549

VISCOSITY CONVERSION TABLE

SUS Saybolt (sec.)	R' Redwood (sec.)	E Engler (deg.)	cSt Centistokes
35	32.2	1.18	27
40	36.2	1.32	4.3
45	40.6	1.46	59
50	44.9	1.60	7.4
55	49.1	1.75	8.9
60	53.5	1.88	10.4
65	57.9	2.02	11.8
70	62.3	2.15	13.1
75	67.6	2.31	14.5
80	71.0	2.42	15.8
85	75.1	2.55	17.0
90	79.6	2.68	18.2
95	84.2	2.81	19.4
100	88.4	2.95	20.6
110	97.1	3.21	23.0
120	105.9	3.49	25.0
130	114.8	3.77	27.5
140	123.6	4.04	29.8
150	132.4	4.32	32.1
160	141.1	4.59	34.3
170	150.0	4.88	36.5
180	158.8	5.15	38.8
190	167.5	5.44	41.0
200	176.4	5.72	43.2
220	194.0	6.28	47.5
240	212	6.85	51.9
260	229	7.38	56.5
280	247	7.95	60.5
300	265	8.51	64.9
325	287	9.24	70.3
350	309	9.95	75.8
375	331	10.7	81.2
400	353	11.4	86.8
425	375	12.1	92.0
450	397	12.8	97.4
475	419	13.5	103
500	441	14.2	108
550	485	15.6	119
600	529	17.0	130
650	573	18.5	141
700	617	19.9	152
750	661	21.3	163
800	705	22.7	173
850	749	24.2	184
900	793	25.6	195
950	837	27.0	206
1000	882	28.4	217
1200	1058	34.1	260
1400	1234	39.8	302
1600	1411	45.5	347
1800	1587	51	390
2000	1763	57	433
2500	2204	71	542
3000	2646	85	650
3500	3087	99	758
4000	3526	114	867
4500	3967	128	974
5000	4408	142	1082
5500	4849	156	1150
6000	5290	170	1300
6500	5730	185	1400
7000	6171	199	1510
7500	6612	213	1630
8000	7053	227	1740
8500	7494	242	1850
9000	7934	256	1960
9500	8375	270	2070
10000	8816	284	2200

Conversion Table: Inches/Millimeters

INCHES TO MILLIMETERS — UNITS

inches		0	1	2	3	4	5	6	7	8	9
0	0.0000	0.000	25.400	50.800	76.200	101.600	127.000	152.400	177.800	203.200	228.600
1⁄16	0.0625	1.588	26.988	52.388	77.788	103.188	128.588	153.988	179.388	204.788	230.188
1⁄8	0.1250	3.175	28.575	53.975	79.375	104.775	130.175	155.575	180.975	206.375	231.775
3⁄16	0.1875	4.763	30.162	55.562	80.962	106.362	131.762	157.162	182.562	207.962	233.362
1⁄4	0.2500	6.350	31.750	57.150	82.550	107.950	133.350	158.750	184.150	209.550	234.950
5⁄16	0.3125	7.938	33.338	58.738	84.138	109.538	134.938	160.338	185.735	211.138	236.538
3⁄8	0.3750	9.525	34.925	60.325	85.725	111.125	136.525	161.925	187.325	212.725	238.125
7⁄16	0.4375	11.112	36.512	61.912	87.312	112.712	138.112	163.512	188.912	214.312	239.712
1⁄2	0.5000	12.700	38.100	63.500	88.900	114.300	139.700	165.100	190.500	215.900	241.300
9⁄16	0.5625	14.288	39.688	65.088	90.488	115.888	141.288	166.688	192.088	217.488	242.888
5⁄8	0.6250	15.875	41.275	66.675	92.075	117.475	142.875	168.275	193.675	219.075	244.475
11⁄16	0.6875	17.462	42.862	68.262	93.662	119.062	144.462	169.862	195.262	220.662	246.062
3⁄4	0.7500	19.050	44.450	69.850	95.250	120.650	146.050	171.450	196.850	222.250	247.650
13⁄16	0.8125	20.638	46.038	71.438	96.838	122.238	147.638	173.038	198.438	223.838	249.238
7⁄8	0.8750	22.225	47.625	73.025	98.425	123.825	149.225	174.625	200.025	225.425	250.825
15⁄16	0.9375	23.812	49.212	74.612	100.012	125.412	150.812	176.212	201.612	227.012	252.412

inches		10	11	12	13	14	15
0	0.0000	254.000	279.400	304.800	330.200	355.600	381.000
1/16	0.0625	255.588	280.988	306.388	331.788	357.188	382.588
1/8	0.1250	257.175	282.575	307.975	333.375	358.775	384.175
3/16	0.1875	258.762	284.162	309.562	334.962	360.362	385.762
1/4	0.2500	260.350	285.750	311.150	336.550	361.950	387.350
5/16	0.3125	261.938	287.338	312.738	338.138	363.538	388.938
3/8	0.3750	263.525	288.925	314.325	339.725	365.125	390.525
7/16	0.4375	265.112	290.512	315.912	341.312	366.712	392.112
1/2	0.5000	266.700	292.100	317.500	342.900	368.300	393.700
9/16	0.5625	268.288	293.688	319.088	344.488	369.888	395.288
5/8	0.6250	269.875	295.275	320.675	346.075	371.475	396.875
11/16	0.6875	271.462	296.862	322.262	347.662	373.062	398.462
3/4	0.7500	273.050	298.450	323.850	349.250	374.650	400.050
13/16	0.8125	274.638	300.038	325.438	350.838	376.238	401.638
7/8	0.8750	276.225	301.625	327.025	352.425	377.825	403.225
15/16	0.9375	277.812	303.212	328.612	354.012	379.412	404.812

B.S.I. Norm No. 350
A.S.A. Norm No. B48.1 } 1 inch = 25.400 mm (exact)

DIN 4890, 1 mm = $\frac{1}{25.4}$ inches

UNITS

inches		10
0	—	254
1	25.4	279.4
2	50.8	304.8
3	76.2	330.2
4	101.6	355.6
5	127	381
6	152.4	406.4
7	177.8	431.8
8	203.2	457.2
9	228.6	482.6

FRACTIONS

1/10"

Inch	mm
0.1	2.54
0.2	5.08
0.3	7.62
0.4	10.16
0.5	12.70
0.6	15.24
0.7	17.78
0.8	20.32
0.9	22.86

1/100"

inches	mm
0.01	0.254
0.02	0.508
0.03	0.762
0.04	1.016
0.05	1.270
0.06	1.524
0.07	1.778
0.08	2.032
0.09	2.286

1/1000"

inches	mm
0.001	0.0254
0.002	0.0508
0.003	0.0762
0.004	0.1016
0.005	0.1270
0.006	0.1524
0.007	0.1778
0.008	0.2032
0.009	0.2286

1/10000"

inches	mm
0.0001	0.00254
0.0002	0.00508
0.0003	0.00762
0.0004	0.01016
0.0005	0.01270
0.0006	0.01524
0.0007	0.01778
0.0008	0.02032
0.0009	0.02286

MILLIMETERS TO INCHES — UNITS

mm		10	20	30	40	50	60	70	80	90
0	—	0.39370	0.78740	1.18110	1.57480	1.96850	2.36220	2.75591	3.14961	3.54331
1	0.03937	0.43307	0.82677	1.22047	1.61417	2.00787	2.40157	2.79528	3.18898	3.58268
2	0.07874	0.47244	0.86614	1.25984	1.65354	2.04724	2.44094	2.83465	3.22835	3.62205
3	0.11811	0.51181	0.90551	1.29921	1.69291	2.08661	2.48031	2.87402	3.26772	3.66142
4	0.15748	0.55118	0.94488	1.33858	1.73228	2.12598	2.51969	2.91339	3.30709	3.70079
5	0.19685	0.59055	0.98425	1.37795	1.77165	2.16535	2.55906	2.95276	3.34646	3.74016
6	0.23622	0.62992	1.02362	1.41732	1.71102	2.20472	2.59843	2.99213	3.38583	3.77953
7	0.27559	0.66929	1.06299	1.45669	1.85039	2.24409	2.63780	3.03150	3.42520	3.81890
8	0.31496	0.70866	1.10236	1.49606	1.88976	2.28346	2.67717	3.07087	3.46457	3.85827
9	0.35433	0.74803	1.14173	1.53543	1.92913	2.32283	2.71654	3.11024	3.50394	3.89764

mm		100	200	300
0	—	3.93701	7.87402	11.81100
10	0.39370	4.33071	8.26772	12.20470
20	0.78740	4.72441	8.66142	12.59840
30	1.18110	5.11811	9.05512	12.99210
40	1.57480	5.51181	9.44882	13.38580
50	1.96850	5.90551	9.84252	13.77950
60	2.36220	6.29921	10.23620	14.17320
70	2.75591	6.69291	10.62990	14.56690
80	3.14961	7.08661	11.02360	14.96060
90	3.54331	7.48031	11.41730	15.35430

FRACTIONS

1/10 mm

mm	inches
0.1	0.00394
0.2	0.00787
0.3	0.01181
0.4	0.01575
0.5	0.01969
0.6	0.02362
0.7	0.02756
0.8	0.03150
0.9	0.03543

1/100 mm

mm	inches
0.01	0.00039
0.02	0.00079
0.03	0.00118
0.04	0.00157
0.05	0.00197
0.06	0.00236
0.07	0.00276
0.08	0.00315
0.09	0.00354

1/1000 mm

mm	inches
0.001	0.000039
0.002	0.000079
0.003	0.000118
0.004	0.000157
0.005	0.000197
0.006	0.000236
0.007	0.000276
0.008	0.000315
0.009	0.000354

STEEL HARDNESS NUMBERS*

APPROX. HARDNESS CONVERSION NUMBERS FOR STEEL, BASED ON ROCKWELL C

Rockwell C-Scale Hardness Number	Diamond Pyramid Hardness Number Vickers	Brinell Hardness Number 10 mm ball 3000 kg load			Rockwell Hardness Number			Rockwell Superficial Hardness Number Superficial Brale Penetrator			Shore Scleroscope Hardness Number	Tensile Strength (approx.) 1000 psi	Rockwell C-Scale Hardness Number
		Standard Ball	Hultgren Ball	Tungsten Carbide Ball	A-Scale 60 kg load Brale Penetrator	B-Scale 100 kg load 1/16" Dia. Ball	D-Scale 100 kg Brale Penetrator	15-N Scale 15 kg load	30-N Scale 30 kg load	45-N Scale 45 kg load			
68	940	—	—	—	85.6	—	76.9	93.2	84.4	75.4	97	—	68
67	900	—	—	—	85	—	76.1	92.9	83.6	74.2	95	—	67
66	865	—	—	—	84.5	—	75.4	92.5	82.8	73.3	92	—	66
65	832	—	—	739	83.9	—	74.5	92.2	81.9	72	91	—	65
64	800	—	—	722	83.4	—	73.8	91.8	81.1	71	88	—	64
63	772	—	—	705	82.8	—	73	91.4	80.1	69.9	87	—	63
62	746	—	—	688	82.3	—	72.2	91.1	79.3	68.8	85	—	62
61	720	—	—	670	81.8	—	71.5	90.7	78.4	67.7	83	—	61
60	697	—	613	654	81.2	—	70.7	90.2	77.5	66.6	81	—	60
59	674	—	599	634	80.7	—	69.9	89.8	76.6	65.5	80	326	59
58	653	—	587	615	80.1	—	69.2	89.3	75.7	64.3	78	315	58
57	633	—	575	595	79.6	—	68.5	88.9	74.8	63.2	76	305	57
56	613	—	561	577	79	—	67.7	88.3	73.9	62	75	295	56
55	595	—	546	560	78.5	—	66.9	87.9	73	60.9	74	287	55
54	577	—	534	543	78	—	66.1	87.4	72	59.8	72	278	54
53	560	—	519	525	77.4	—	65.4	86.9	71.2	58.6	71	269	53
52	544	500	508	512	76.8	—	64.6	86.4	70.2	57.4	69	262	52
51	528	487	494	496	76.3	—	63.8	85.9	69.4	56.1	68	253	51
50	513	475	481	481	75.9	—	63.1	85.5	68.5	55	67	245	50
49	498	464	469	469	75.2	—	62.1	85	67.6	53.8	66	239	49
48	484	451	455	455	74.7	—	61.4	84.5	66.7	52.5	64	232	48
47	471	442	443	443	74.1	—	60.8	83.9	65.8	51.4	63	225	47
46	458	432	432	432	73.6	—	60	83.5	64.8	50.3	62	219	46
45	446	421	421	421	73.1	—	59.2	83	64	49	60	212	45
44	434	409	409	409	72.5	—	58.5	82.5	63.1	47.8	58	206	44
43	423	400	400	400	72	—	57.7	82	62.2	46.7	57	201	43
42	412	390	390	390	71.5	—	56.9	81.5	61.3	45.5	56	196	42
41	402	381	381	381	70.9	—	56.2	80.9	60.4	44.3	55	191	41
40	392	371	371	371	70.4	—	55.4	80.4	59.5	43.1	54	186	40
39	382	362	362	362	69.9	—	54.6	79.9	58.6	41.9	52	181	39
38	372	353	353	353	69.4	—	53.8	79.4	57.7	40.8	51	176	38
37	363	344	344	344	68.9	—	53.1	78.8	56.8	39.6	50	172	37
36	354	336	336	336	68.4	(109)	52.3	78.3	55.9	38.4	49	168	36
35	345	327	327	327	67.9	(108.5)	51.5	77.7	55	37.2	48	163	35
34	336	319	319	319	67.4	(108)	50.8	77.2	54.2	36.1	47	159	34
33	327	311	311	311	66.8	(107.5)	50	76.6	53.3	34.9	46	154	33
32	318	301	301	301	66.3	(107)	49.2	76.1	52.1	33.7	44	150	32
31	310	294	294	294	65.8	(106)	48.4	75.6	51.3	32.5	43	146	31
30	302	286	286	286	65.3	(105.5)	47.7	75	50.4	31.3	42	142	30
29	294	279	279	279	64.7	(104.5)	47	74.5	49.5	30.1	41	138	29
28	286	271	271	271	64.3	(104)	46.1	73.9	48.6	28.9	41	134	28
27	279	264	264	264	63.8	(103)	45.2	73.3	47.7	27.8	40	131	27
26	272	258	258	258	63.3	(102.5)	44.6	72.8	46.8	26.7	38	127	26
25	266	253	253	253	62.8	(101.5)	43.8	72.2	45.9	25.5	38	124	25
24	260	247	247	247	62.4	(101)	43.1	71.6	45	24.3	37	121	24
23	254	243	243	243	62	100	42.1	71	44	23.1	36	118	23
22	248	237	237	237	61.5	99	41.6	70.5	43.2	22	35	115	22
21	243	231	231	231	61	98.5	40.9	69.9	42.3	20.7	35	113	21
20	238	226	226	226	60.5	97.8	40.1	69.4	41.5	19.6	34	110	20

* Source ASTM