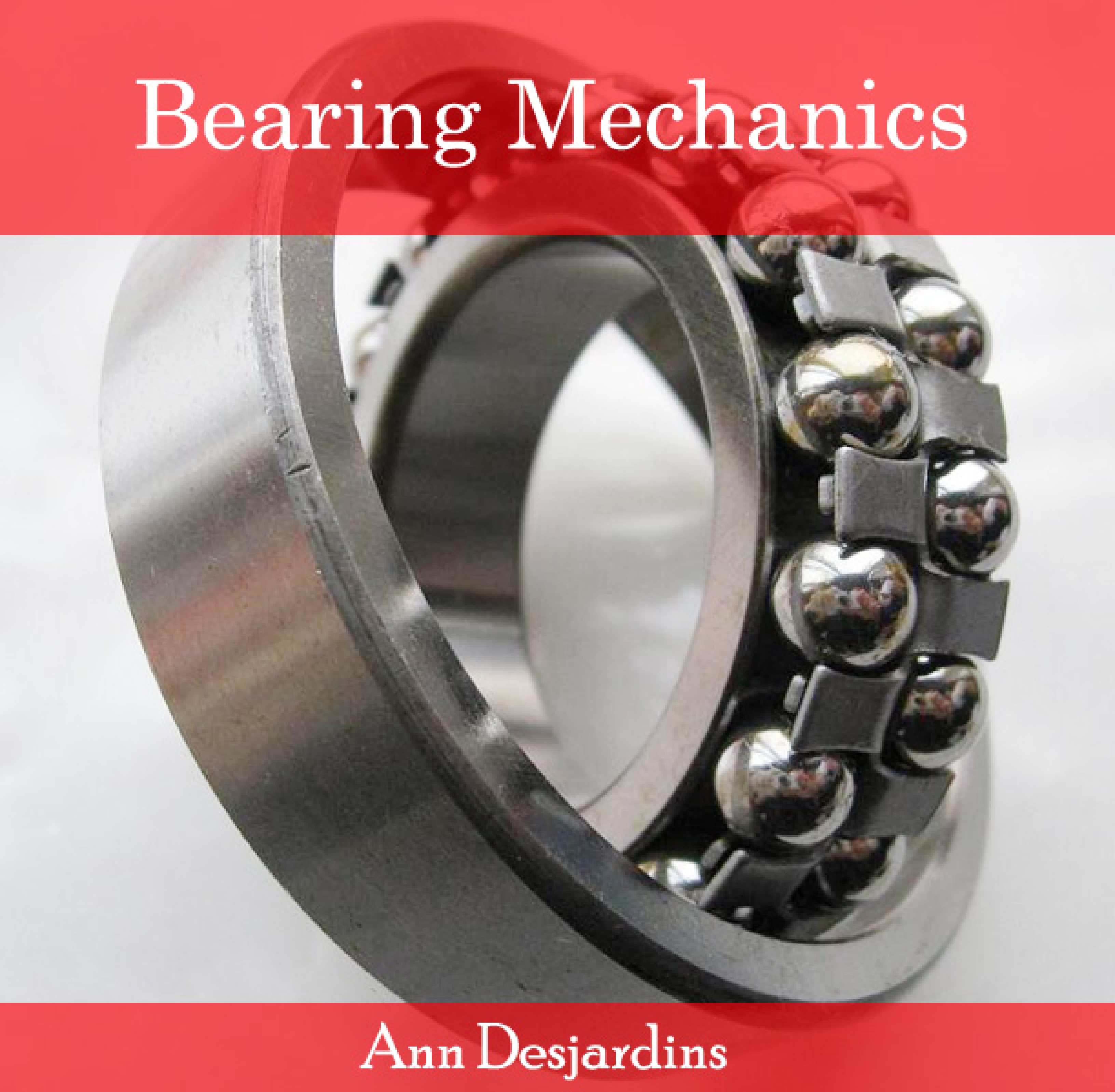
Bearing Mechanics



Ann Desjardins

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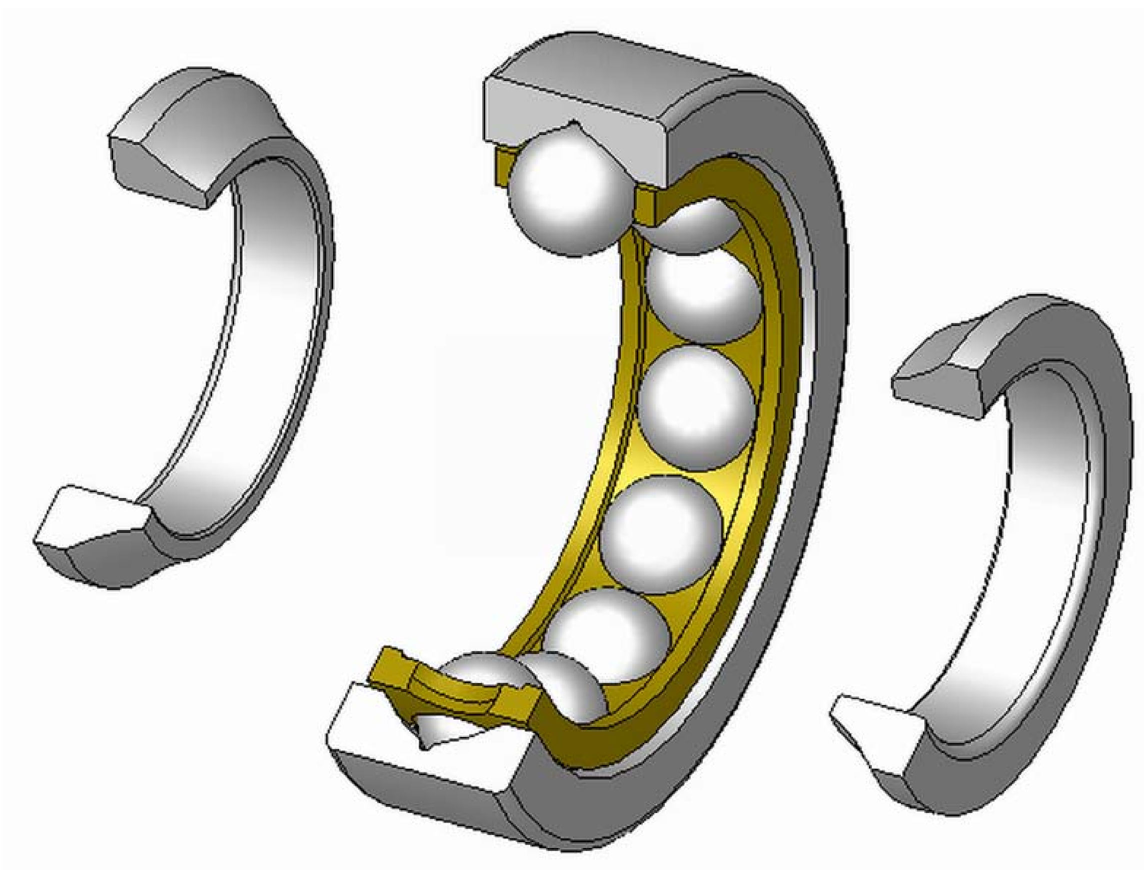
Bearing (Mechanical)



A cutaway example of a four-point contact ball bearing

A **bearing** is a device to allow constrained relative motion between two or more parts, typically rotation or linear movement. Bearings may be classified broadly according to the motions they allow and according to their principle of operation as well as by the directions of applied loads they can handle.

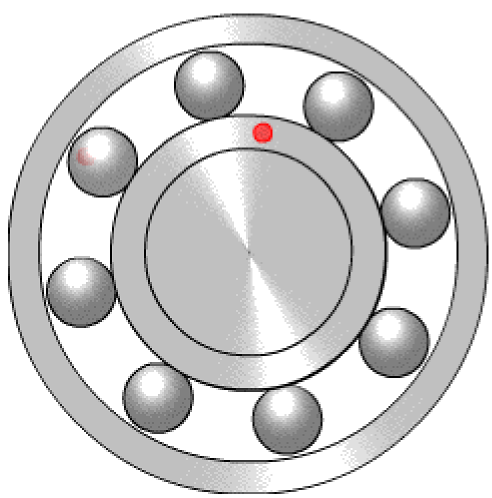
Overview

Plain bearings use surfaces in rubbing contact, often with a lubricant such as oil or graphite. A plain bearing may or may not be a discrete device. It may be nothing more than the bearing surface of a hole with a shaft passing through it, or of a planar surface that bears another (in these cases, not a discrete device); or it may be a layer of bearing metal either fused to the substrate (semi-discrete) or in the form of a separable sleeve (discrete). With suitable lubrication, plain bearings often give entirely acceptable accuracy, life, and friction at minimal cost. Therefore, they are very widely used.

However, there are many applications where a more suitable bearing can improve efficiency, accuracy, service intervals, reliability, speed of operation, size, weight, and costs of purchasing and operating machinery.

Thus, there are many types of bearings, with varying shape, material, lubrication, principle of operation, and so on. For example, rolling-element bearings use spheres or drums rolling between the parts to reduce friction; reduced friction allows tighter tolerances and thus higher precision than a plain bearing, and reduced wear extends the time over which the machine stays accurate. Plain bearings are commonly made of varying types of metal or plastic depending on the load, how corrosive or dirty the environment is, and so on. In addition, bearing friction and life may be altered dramatically by the type and application of lubricants. For example, a lubricant may improve bearing friction and life, but for food processing a bearing may be lubricated by an inferior food-safe lubricant to avoid food contamination; in other situations a bearing may be run without lubricant because continuous lubrication is not feasible, and lubricants attract dirt that damages the bearings.

Principles of operation



Ball bearing

There are at least six common principles of operation:

- plain bearing, also known by the specific styles: bushings, journal bearings, sleeve bearings, rifle bearings
- rolling-element bearings such as ball bearings and roller bearings
- jewel bearings, in which the load is carried by rolling the axle slightly off-center
- fluid bearings, in which the load is carried by a gas or liquid
- magnetic bearings, in which the load is carried by a magnetic field
- flexure bearings, in which the motion is supported by a load element which bends.

Motions

Common motions permitted by bearings are:

- Axial rotation e.g. shaft rotation
- Linear motion e.g. drawer
- spherical rotation e.g. ball and socket joint
- hinge motion e.g. door, elbow, knee

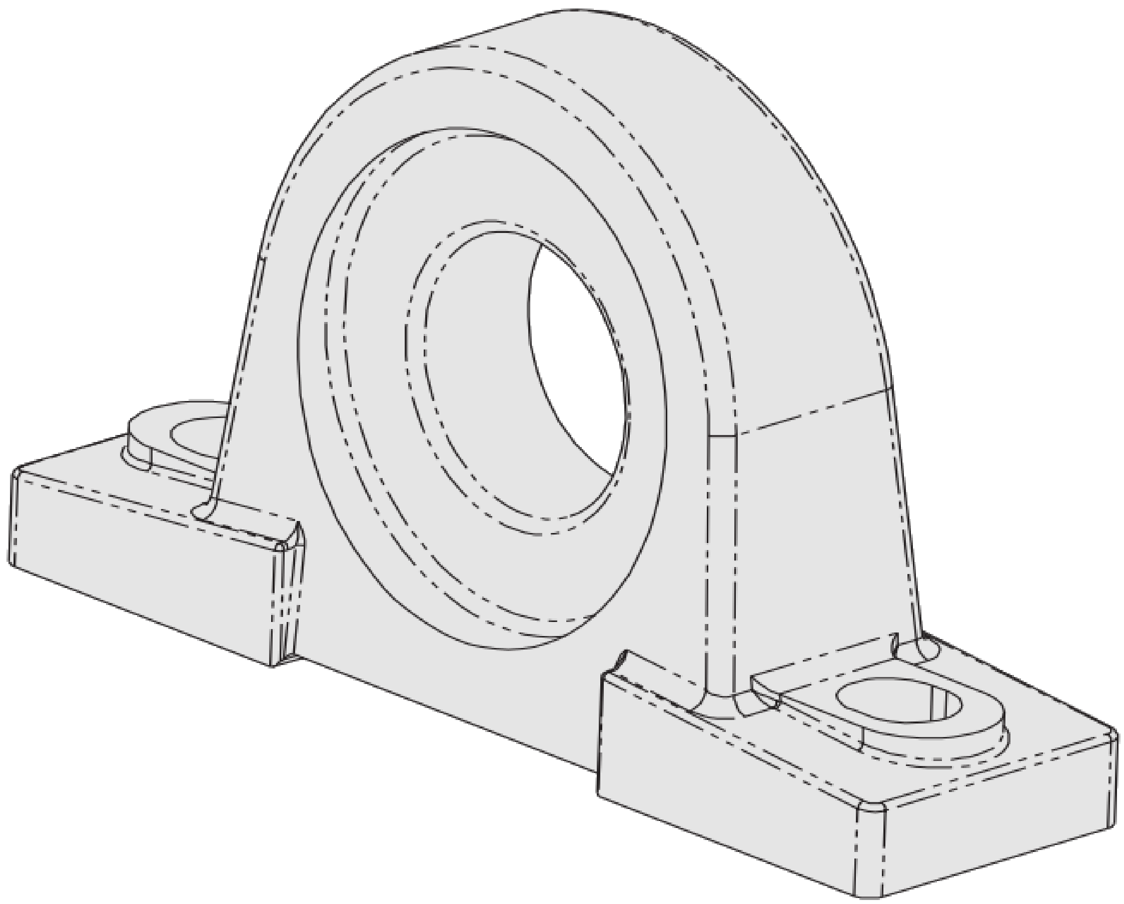
Friction

Reducing friction in bearings is often important for efficiency, to reduce wear and to facilitate extended use at high speeds and to avoid overheating and premature failure of the bearing. Essentially, a bearing can reduce friction by virtue of its shape, by its material, or by introducing and containing a fluid between surfaces or by separating the surfaces with an electromagnetic field.

- **By shape**, gains advantage usually by using spheres or rollers, or by forming flexure bearings.
- **By material**, exploits the nature of the bearing material used. (An example would be using plastics that have low surface friction.)
- **By fluid**, exploits the low viscosity of a layer of fluid, such as a lubricant or as a pressurized medium to keep the two solid parts from touching, or by reducing the normal force between them.
- **By fields**, exploits electromagnetic fields, such as magnetic fields, to keep solid parts from touching.

Combinations of these can even be employed within the same bearing. An example of this is where the cage is made of plastic, and it separates the rollers/balls, which reduce friction by their shape and finish.

Loads



A block bearing with provisions for fixing it

Bearings vary greatly over the size and directions of forces that they can support.

Forces can be predominately radial, axial (thrust bearings) or Bending moments perpendicular to the main axis.

Speeds

Different bearing types have different operating speed limits. Speed is typically specified as maximum relative surface speeds, often specified ft/s or m/s. Rotational bearings typically describe performance in terms of the product *DN* where *D* is the diameter (often in mm) of the bearing and *N* is the rotation rate in revolutions per minute.

Generally there is considerable speed range overlap between bearing types. Plain bearings typically handle only lower speeds, rolling element bearings are faster, followed by fluid bearings and finally magnetic bearings which are limited ultimately by centripetal force overcoming material strength.

Play

Some applications apply bearing loads from varying directions and accept only limited play or "slop" as the applied load changes. One source of motion is gaps or "play" in the bearing. For example, a 10 mm shaft in a 12 mm hole has 2 mm play.

Allowable play varies greatly depending on the use. As example, a wheelbarrow wheel supports radial and axial loads. Axial loads may be hundreds of newtons force left or right, and it is typically acceptable for the wheel to wobble by as much as 10 mm under the varying load. In contrast, a lathe may position a cutting tool to ±0.02 mm using a ball lead screw held by rotating bearings. The bearings support axial loads of thousands of newtons in either direction, and must hold the ball lead screw to ±0.002 mm across that range of loads.

Stiffness

A second source of motion is elasticity in the bearing itself. For example, the balls in a ball bearing are like stiff rubber, and under load deform from round to a slightly flattened shape. The race is also elastic and develops a slight dent where the ball presses on it.

The stiffness of a bearing is how the distance between the parts which are separated by the bearing varies with applied load. With rolling element bearings this is due to the strain of the ball and race. With fluid bearings it is due to how the pressure of the fluid varies with the gap (when correctly loaded, fluid bearings are typically stiffer than rolling element bearings).

Service life

Fluid and magnetic bearings can have practically indefinite service lives. In practice, there are fluid bearings supporting high loads in hydroelectric plants that have been in nearly continuous service since about 1900 and which show no signs of wear.

Rolling element bearing life is determined by load, temperature, maintenance, lubrication, material defects, contamination, handling, installation and other factors. These factors can all have a significant effect on bearing life. For example, the service life of bearings in one application was extended dramatically by changing how the bearings were stored before installation and use, as vibrations during storage caused lubricant failure even when the only load on the bearing was its own weight; the resulting damage is often false brinelling. Bearing life is statistical: several samples of a given bearing will often exhibit a bell curve of service life, with a few samples showing significantly better or worse life. Bearing life varies because microscopic structure and contamination vary greatly even where macroscopically they seem identical.

For plain bearings some materials give much longer life than others. Some of the John Harrison clocks still operate after hundreds of years because of the *lignum vitae* wood employed in their construction, whereas his metal clocks are seldom run due to potential wear.

Flexure bearings bend a piece of material repeatedly. Some materials fail after repeated bending, even at low loads, but careful material selection and bearing design can make flexure bearing life indefinite.

Although long bearing life is often desirable, it is sometimes not necessary. Harris describes a bearing for a rocket motor oxygen pump that gave several hours life, far in excess of the several tens of minutes life needed.

Bearings are often manufactured to what is called an "L10" life factor.

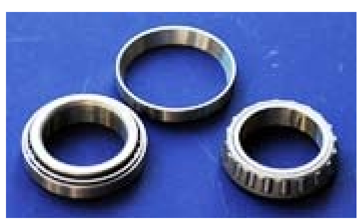
Maintenance

Many bearings require periodic maintenance to prevent premature failure, although some such as fluid or magnetic bearings may require little maintenance.

Most bearings in high cycle operations need periodic lubrication and cleaning, and may require adjustment to minimise the effects of wear.

Bearing life is often much better when the bearing is kept clean and well-lubricated. However, many applications make good maintenance difficult. For example bearings in the conveyor of a rock crusher are exposed continually to hard abrasive particles. Cleaning is of little use because cleaning is expensive, yet the bearing is contaminated again as soon as the conveyor resumes operation. Thus, a good maintenance program might lubricate the bearings frequently but clean them never.

History



Tapered bearings

Early Timken tapered roller bearing with notched rollers

The oldest instance of the bearing principle dates to the Egyptians when they used tree trunks under sleds. There are also Egyptian drawings of bearings used with hand drills.

The earliest recovered example of a bearing is a wooden ball bearing supporting a rotating table from the remains of the Roman Nemi ships in Lake Nemi, Italy. The wrecks were dated to 40 AD.

Leonardo da Vinci is often credited with drawing the first roller bearing around the year 1500. However, Agostino Ramelli is the first to have published sketches of roller and thrust bearings. An issue with ball and roller bearings is that the balls or rollers rub against each other causing additional friction which can be prevented by enclosing the balls or rollers in a cage. The captured, or caged, ball bearing was originally described by

Galileo in the 17th century. The mounting of bearings into a set was not accomplished for many years after that. The first patent for a ball race was by Philip Vaughan of Carmarthen in 1794.

Bearings saw use for holding wheel and axles. The bearings used there were plain bearings that were used to greatly reduce friction over that of dragging an object by making the friction act over a shorter distance as the wheel turned.

The first plain and rolling-element bearings were wood closely followed by bronze. Over their history bearings have been made of many materials including ceramic, sapphire, glass, steel, bronze, other metals and plastic (e.g., nylon, polyoxymethylene, polytetrafluoroethylene, and UHMWPE) which are all used today.

Watch makers produced "jeweled" pocket watches using sapphire plain bearings to reduce friction thus allowing more precise time keeping.

Even basic materials can have good durability. As examples, wooden bearings can still be seen today in old clocks or in water mills where the water provides cooling and lubrication.

The first practical caged-roller bearing was invented in the mid-1740s by horologist John Harrison for his H3 marine timekeeper. This uses the bearing for a very limited oscillating motion but Harrison also used a similar bearing in a truly rotary application in a contemporaneous regulator clock.

A patent on ball bearings, reportedly the first, was awarded to Jules Suriray, a Parisian bicycle mechanic, on 3 August 1869. The bearings were then fitted to the winning bicycle ridden by James Moore in the world's first bicycle road race, Paris-Rouen, in November 1869.

Friedrich Fischer's idea from the year 1883 for milling and grinding balls of equal size and exact roundness by means of a suitable production machine formed the foundation for creation of an independent bearing industry.

The modern, self-aligning design of ball bearing is attributed to Sven Wingquist of the SKF ball-bearing manufacturer in 1907, when he was awarded Swedish patent No. 25406 on its design.

Henry Timken, a 19th century visionary and innovator in carriage manufacturing, patented the tapered roller bearing, in 1898. The following year, he formed a company to produce his innovation. Through a century, the company grew to make bearings of all types, specialty steel and an array of related products and services.

Erich Franke invented and patented the wire race bearing in 1934. His focus was on a bearing design with a cross section as small as possible and which could be integrated into the enclosing design. After World War II he founded together with Gerhard

Heydrich the company Franke & Heydrich KG (today Franke GmbH) to push the development and production of wire race bearings.

Richard Stribeck's extensive research on ball bearing steels identified the metallurgy of the commonly used 100Cr6 (AISI 52100) showing coefficient of friction as a function of pressure.

Designed in 1968 and later patented in 1972, Bishop-Wisecarver's co-founder Bud Wisecarver created vee groove bearing guide wheels, a type of linear motion bearing consisting of both an external and internal 90 degree vee angle.

In the early 1980s, Pacific Bearing's founder, Robert Schroeder, invented the first bi-material plain bearing which was size interchangeable with linear ball bearings. This bearing had a metal shell (aluminum, steel or stainless steel) and a layer of Teflon-based material connected by a thin adhesive layer.

Today ball and roller bearings are used in many applications which include a rotating component. Examples include ultra high speed bearings in dental drills, aerospace bearings in the Mars Rover, gearbox and wheel bearings on automobiles, flexure bearings in optical alignment systems and bicycle wheel hubs.

Types

There are many different types of bearings.

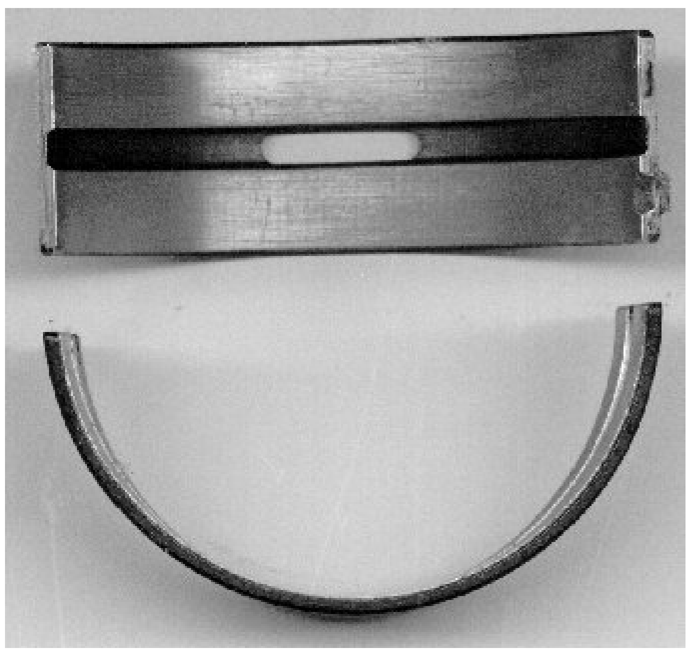
Type	Description	Friction	Stiffness†	Speed	Life	Notes
Plain bearing	Rubbing surfaces, usually with lubricant; some bearings use pumped lubrication and behave similarly to fluid bearings.	Depends on materials and construction, PTFE has coefficient of friction ~0.05-0.35, depending upon fillers added	Good, provided wear is low, but some slack is normally present	Low to very high	Low to very high - depends upon application and lubrication	Widely used, relatively high friction, suffers from stiction in some applications. Depending upon the application, lifetime can be higher or lower than rolling element bearings.
Rolling element bearing	Ball or rollers are used to prevent or minimise rubbing	Rolling coefficient of friction with steel can be ~0.005 (adding resistance due to seals, packed grease, preload and misalignment can increase friction to as much as 0.125)	Good, but some slack is usually present	Moderate to high (often requires cooling)	Moderate to high (depends on lubrication, often requires maintenance)	Used for higher moment loads than plain bearings with lower friction
Jewel bearing	Off-center bearing rolls in seating	Low	Low due to flexing	Low	Adequate (requires maintenance)	Mainly used in low-load, high precision work such as clocks. Jewel bearings may be very small.

Fluid bearing	Fluid is forced between two faces and held in by edge seal	Zero friction at zero speed, low	Very high	Very high (usually limited to a few hundred feet per second at/by seal)	Virtually infinite in some applications, may wear at startup/shutdown in some cases. Often negligible maintenance.	Can fail quickly due to grit or dust or other contaminants. Maintenance free in continuous use. Can handle very large loads with low friction.
Magnetic bearings	Faces of bearing are kept separate by magnets (electromagnets or eddy currents)	Zero friction at zero speed, but constant power for levitation, eddy currents are often induced when movement occurs, but may be negligible if magnetic field is quasi-static	Low	No practical limit	Indefinite. Maintenance free. (with electromagnets)	Active magnetic bearings (AMB) need considerable power. Electrodynamic bearings (EDB) do not require external power.
Flexure bearing	Material flexes to give and constrain movement	Very low	Low	Very high.	Very high or low depending on materials and strain in application. Usually maintenance free.	Limited range of movement, no backlash, extremely smooth motion

†Stiffness is the amount that the gap varies when the load on the bearing changes, it is distinct from the friction of the bearing.

Chapter 2

Plain Bearing

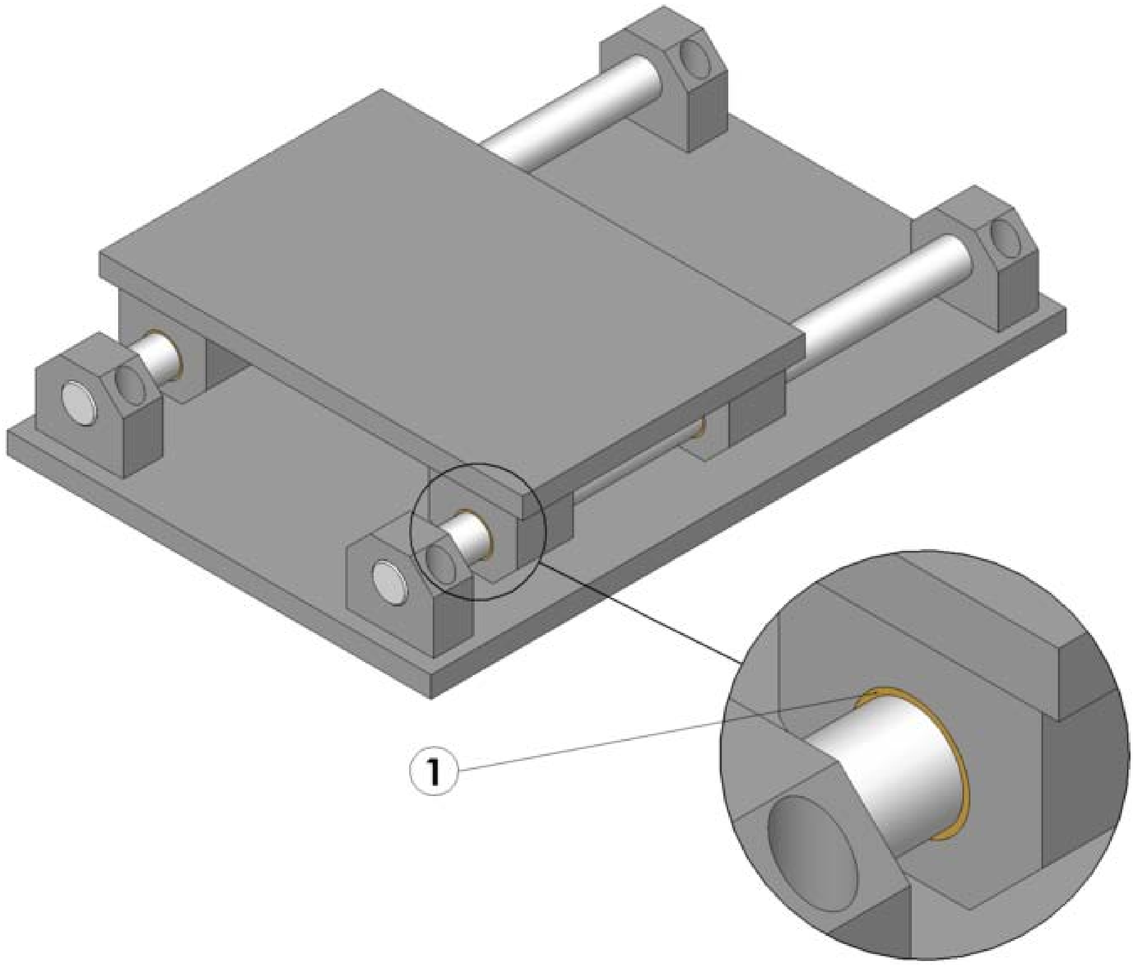


Crankshaft plain bearing shells

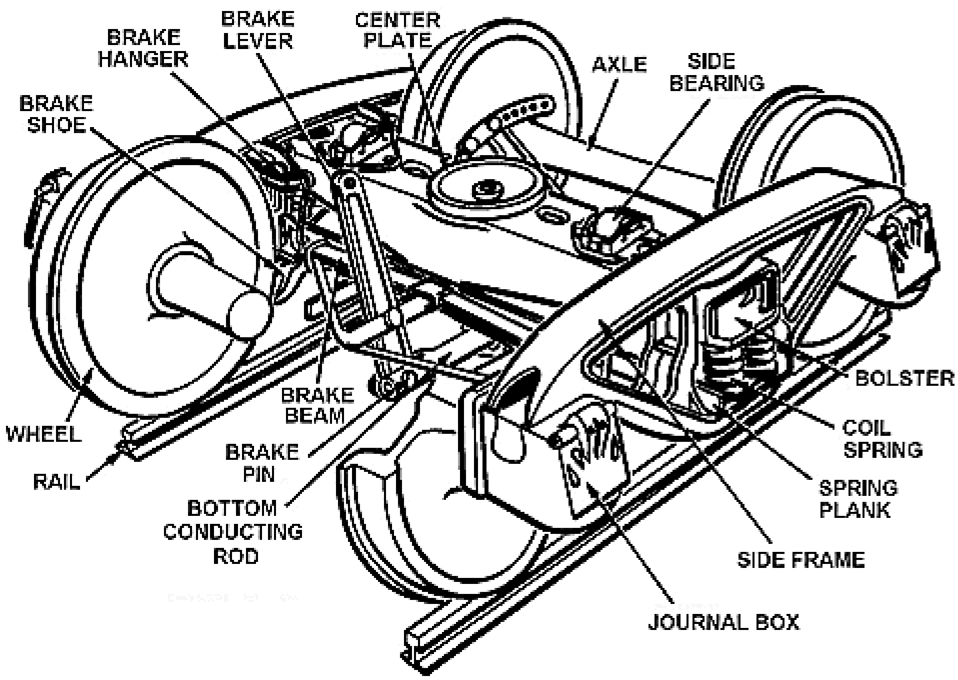
A **plain bearing**, also known as a **plane bearing**, is the simplest type of bearing, comprising just a bearing surface and no rolling elements. Therefore the journal (*i.e.*, the part of the shaft in contact with the bearing) slides over the bearing surface. The simplest example of a plain bearing is a shaft rotating in a hole. A simple linear bearing can be a pair of flat surfaces designed to allow motion; *e.g.*, a drawer and the slides it rests on or the ways on the bed of a lathe.

Plain bearings, in general, are the least expensive type of bearing. They are also compact, light weight, and have a high load-carrying capacity.

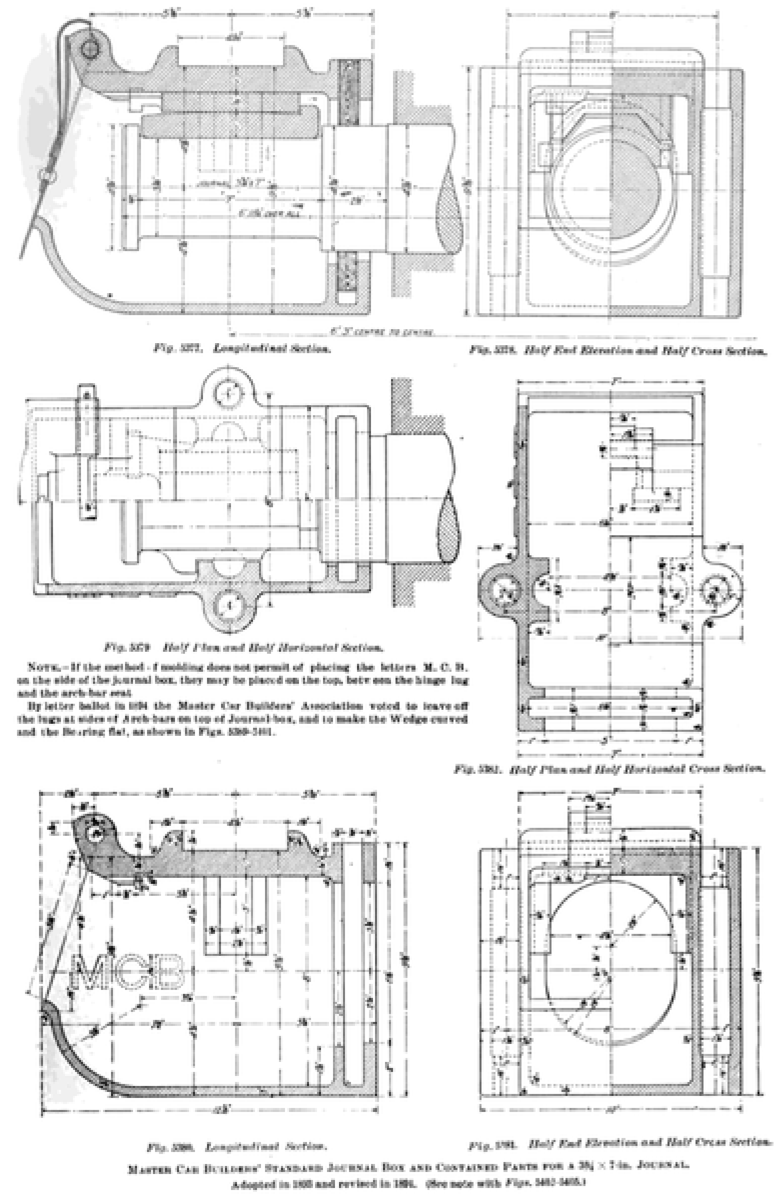
Design



A linear table with four linear bearings (1)



Journal bearing and journal box found on a US-style railroad truck



Journal box

The design of a plain bearing depends on the type of motion the bearing must provide. The three types of motions possible are:

- *Journal* (*friction*, *radial* or *rotary*) *bearing*: This is the most common type of plain bearing; it is simply a shaft rotating in a bearing.
    - In locomotive applications a *journal bearing* specifically refers to the plain bearing found at the ends of the axles of railroad wheel sets, which are enclosed by *journal boxes*.
- *Linear bearing*: This bearing provides linear motion; it may take the form of a circular bearing and shaft or two matching surfaces (*e.g.*, a slide plate).

- *Thrust bearing*: A thrust bearing provides a bearing surface for forces acting axial to the shaft.

Integral

Integral plain bearings are built into the object of use. It is a hole that has been prepared into a bearing surface. Industrial integral bearings are usually made from cast iron or babbitt and a hardened steel shaft is used in the bearing.

Integral bearings are not as common because bushings are easy to accommodate and if they wear out then they are just replaced. Depending on the material an integral bearing may be less expensive but it cannot be replaced. If an integral bearing wears out then the item may be replaced or reworked to accept a bushing. Integral bearings were very common in 19th-century machinery but became progressively less common as interchangeable manufacture permeated the industry.

An example of a common integral plain bearing is the hinge, which is both a thrust bearing and a journal bearing.

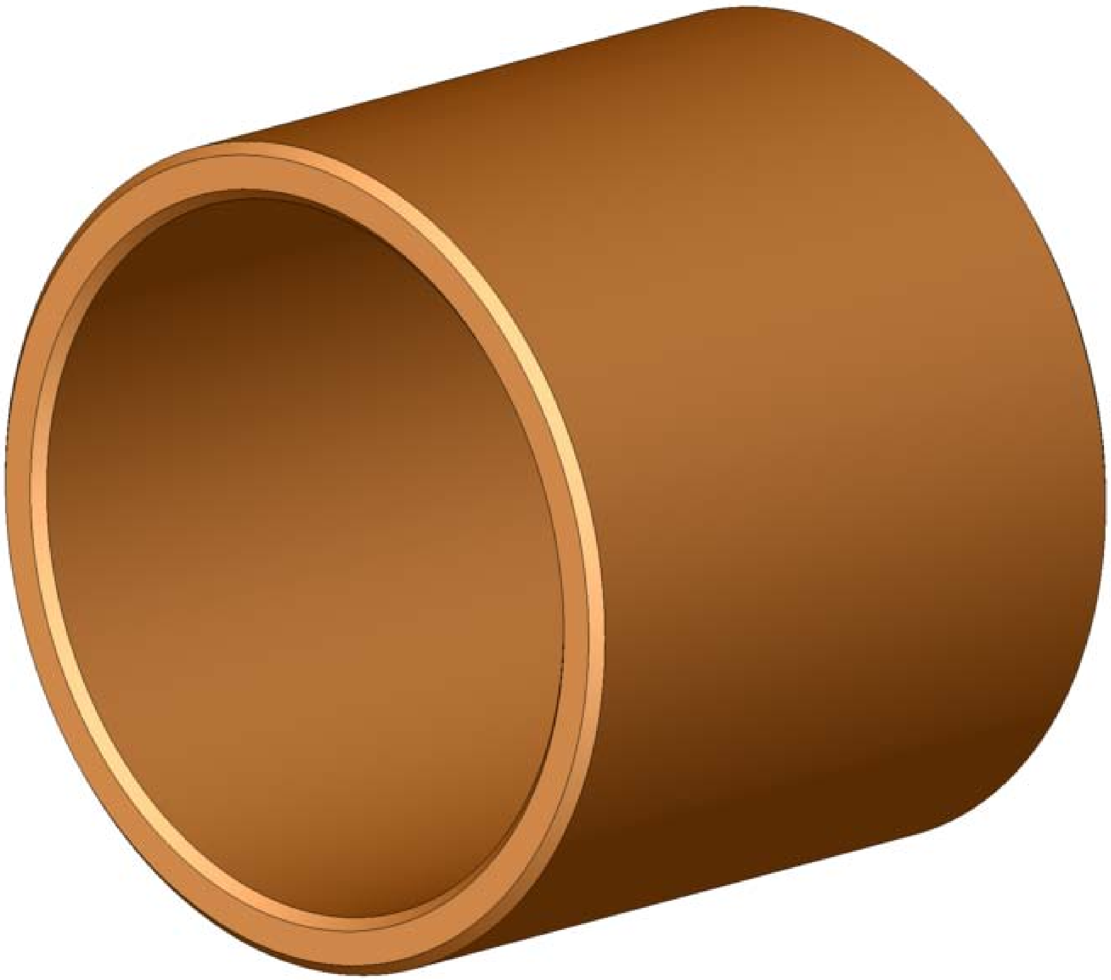
Bushing

A *bushing*, also known as a *bush*, is an independent plain bearing that is inserted into a housing to provide a bearing surface for rotary applications; this is the most common form of a plain bearing. Common designs include *solid* (*sleeve* and *flanged*), *split*, and *clenched* bushings. A sleeve, split, or clenched bushing is only a "sleeve" of material with an inner diameter (ID), outer diameter (OD), and length. The difference between the three types is that a solid sleeved bushing is solid all the way around, a split bushing has a cut along its length, and a clenched bearing is similar to a split bushing but with a clench across the cut. A flanged bushing is a sleeve bushing with a flange extending radially outward from the ID. The flange is used to positively locate the bushing when it is installed or to provide a thrust bearing surface.

Sleeve bearings of inch dimensions are almost exclusively dimensioned using the SAE numbering system. The numbering system uses the format -XXYY-ZZ, where XX is the ID in sixteenths of an inch, YY is the OD in sixteenths of an inch, and ZZ is the length in eights of an inch. Metric sizes also exist.

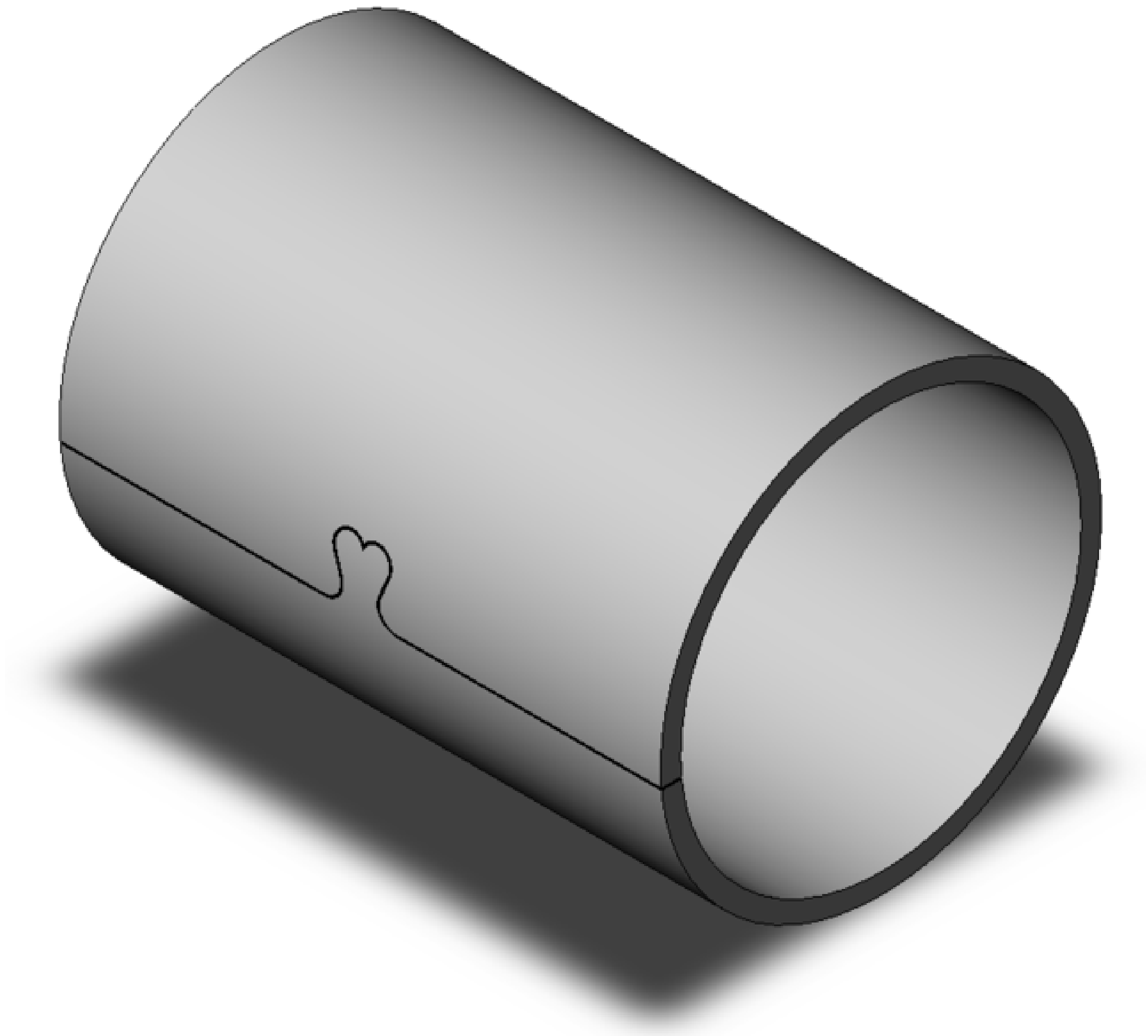
A linear bushing is not usually pressed into a housing, but rather secured with a radial feature. Two such examples include two retaining rings, or a ring that is molded onto the OD of the bushing that matches with a groove in the housing. This is usually a more durable way to retain the bushing, because the forces acting on the bushing could press it out.

The thrust form of a bushing is conventionally called a *thrust washer*.



A solid sleeve bushing

A flanged bushing



A clenched bushing

Two-piece

Two-piece plain bearings, known as *full bearings* in industrial machinery, are commonly used for larger diameters, such as crankshaft bearings. The two halves are called *shells*. There are various system used to keep the shells located. The most common method is a tab on the parting line edge that correlates with a notch in the housing to prevent axial movement after installation. For large, thick shells a button stop or dowel pin is used. The button stop is screwed to the housing, while the dowel pin keys the two shells together. Another less common method uses a dowel pin that keys the shell to the housing through a hole or slot in the shell.

The distance from one parting edge to the other is slightly larger than the corresponding distance in the housing so that a light amount of pressure is required to install the bearing. This keeps the bearing in place as the two halves of the housing are installed. Finally, the shell's circumference is also slightly larger than the housing circumference so that when the two halves are bolted together the bearing *crushes* slightly. This creates a large amount of radial force around the entire bearing which keeps it from *spinning*. It also forms a good interface for heat to travel out of the bearings into the housing.

Materials

Plain bearings must be made from a material that is durable, low friction, low wear to the bearing and shaft, resistant to elevated temperatures, and corrosion resistant. Often the bearing is made up of at least two constituents, where one is soft and the other is hard. The hard constituent supports the load while the soft constituent supports the hard constituent. In general, the harder the surfaces in contact the lower the coefficient of friction and the greater the pressure required for the two to seize.

Babbitt

Babbitt is usually used in integral bearings. It is coated over the bore, usually to a thickness of 1 to 100 thou (0.025 to 2.5 mm), depending on the diameter. Babbitt bearings are designed to not damage the journal during direct contact and to collect any contaminants in the lubrication.

Bi-material

Split bi-material bushings: a metal exterior with an inner plastic coating

Bi-material bearings consist of two materials, a metal shell and a plastic bearing surface. Common combinations include a steel-backed PTFE-coated bronze and aluminum-backed Frelon. Steel-backed PTFE-coated bronze bearings are rated for more load than most other bi-metal bearings and are used for rotary and oscillating motions. Aluminum-backed frelon are commonly used in corrosive environments because the Frelon is chemically inert.

Bearing properties of various bi-material bearings

	Temperature range	P (max.) [psi (MPa)]	V (max.) [sfm (m/s)]	PV (max.) [psi sfm (MPa m/s)]
Steel-backed PTFE-coated bronze	-328–536 °F / -200–280 °C	36,000 psi/248 MPa	390 (2.0 m/s)	51,000 (1.79 MPa m/s)
Aluminum-backed frelon	-400–400 °F / -240–204 °C	3,000 psi/21 MPa	300 (1.52 m/s)	20,000 (0.70 MPa m/s)

Bronze

A common plain bearing design utilizes a hardened and polished steel shaft and a softer bronze bushing. The bushing is replaced whenever it has worn too much.

Common bronze alloys used for bearings include: SAE 841, SAE 660 (CDA 932), SAE 863, and CDA 954.

Bearing properties of various bronze alloys

	Temperature range	P (max.) [psi (MPa)]	V (max.) [sfm (m/s)]	PV (max.) [psi sfm (MPa m/s)]
SAE 841	10–220 °F (-12–104 °C)	2,000 psi (14 MPa)	1,200 (6.1 m/s)	50,000 (1.75 MPa m/s)
SAE 660	10–450 °F (-12–232 °C)	4,000 psi (28 MPa)	750 (3.8 m/s)	75,000 (2.63 MPa m/s)
SAE 863	10–220 °F (-12–104 °C)	4,000 psi (28 MPa)	225 (1.14 m/s)	35,000 (1.23 MPa m/s)
CDA 954	Less than 500 °F (260 °C)	4,500 psi (31 MPa)	225 (1.14 m/s)	125,000 (4.38 MPa m/s)

Cast iron

A cast iron bearing is commonly used with a hardened steel shaft because the coefficient of friction is relatively low. The cast iron glazes over therefore wear becomes negligible.

Graphite

In harsh environments, such as ovens and dryers, a copper and graphite alloy, commonly known by the trademarked name graphalloy, is used. The graphite is a dry lubricant, therefore it is low friction and low maintenance. The copper adds strength, durability, and provides heat dissipation characteristics.

Bearing properties of graphitic materials

	Temperature range	P (max.) [psi (MPa)]	V (max.) [sfm (m/s)]	PV (max.) [psi sfm (MPa m/s)]
Graphalloy	-450–750 °F / -268–399 °C	750 psi/5 MPa	75 (0.38 m/s)	12,000 (0.42 MPa m/s)
Graphite	?	?	?	?

Unalloyed graphite bearings are used in special applications, such as locations that are submerged in water.

Jewels

Known as *jewel bearings*, these bearings use jewels, such as sapphire, ruby, and garnet.

Plastic

Solid plastic plain bearings are now increasingly popular due to dry-running lubrication-free behavior. Solid polymer plain bearings are low weight, corrosion resistant, and maintenance free. After research spanning decades, an accurate calculation of the service life of polymer plain bearings is possible today. Designing with solid polymer plain bearings is complicated by the wide range, and non-linearity, of coefficient of thermal expansion. These materials can heat rapidly when loaded.

Solid polymer type bearings are limited by the injection molding process. Not all shapes are possible with this process and the shapes which are possible are limited to what is considered good design practice for injection molding. Plastic bearings are subject to the same design cautions as all other plastic parts: creep, high thermal expansion, softening (increased wear/reduced life) at elevated temperature, brittle fractures at cold temperatures, swelling due to moisture absorption. While most bearing-grade plastics/polymers are designed to reduce these design cautions, they still exist and should be carefully considered before specifying an a solid polymer (plastic) type.

Plastic bearings are now everywhere from photocopy machines to the tills in the supermarket. Other applications include farm equipment, textile machinery, medical devices, food and packaging machines, car seating, marine equipment and many more.

Common plastics include nylon, polyacetal, polytetrafluoroethylene (PTFE), ultra high molecular weight polyethylene (UHMWPE), rulon, PEEK, urethane and vespel (a high-performance polyimide).

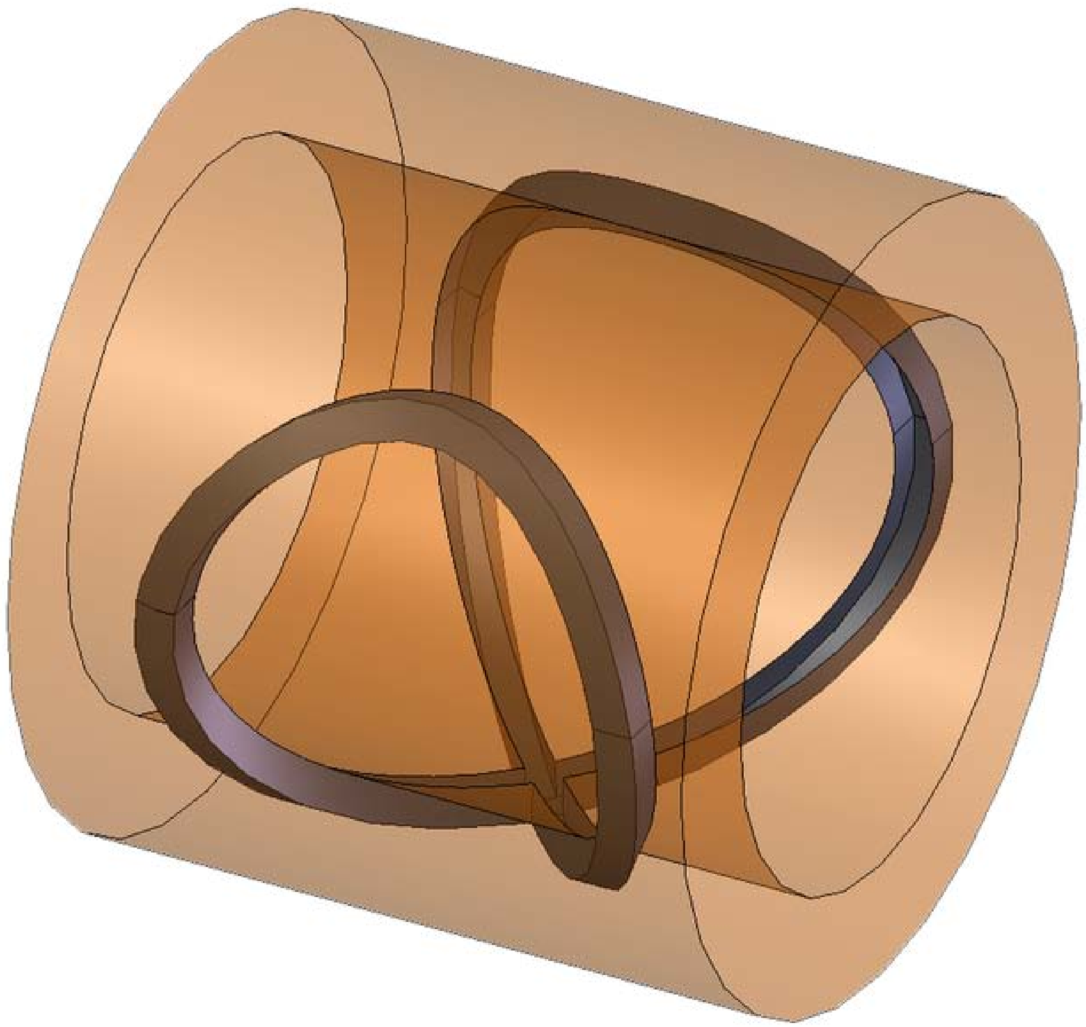
Bearing properties of various plastics

	Temperature range	P (max.) [psi (MPa)]	V (max.) [sfm (m/s)]	PV (max.) [psi sfm (MPa m/s)]
Frelon	-400–500 °F (-240–260 °C)	1,500 (10)	140 (0.71) (dry)	10,000 (0.35)
Nylon	-20–250 °F (-29–121 °C)	400 psi (3 MPa)	360 (1.83 m/s)	3,000 (0.11 MPa m/s)
MDS-filled nylon blend 1	-40–176 °F (-40–80 °C)	2,000 psi (14 MPa)	393 (2.0 m/s)	3,400 (0.12 MPa m/s)
MDS-filled nylon blend 2	-40–230 °F (-40–110 °C)	300 psi (2 MPa)	60 (0.30 m/s)	3,000 (0.11 MPa m/s)
PEEK blend 1	-148–480 °F (-100–249 °C)	8,500 psi (59 MPa)	400 (2.0 m/s)	3,500 (0.12 MPa m/s)
PEEK blend 2	-148–480 °F (-100–249 °C)	21,750 psi (150 MPa)	295 (1.50 m/s)	37,700 (1.32 MPa m/s)
Polyacetal	-20–180 °F (-29–82 °C)	1,000 psi (7 MPa)	1,000 (5.1 m/s)	2,700 (0.09 MPa m/s)
PTFE	-350–500 °F (-212–260 °C)	500 psi (3 MPa)	100 (0.51 m/s)	1,000 (0.04 MPa m/s)
Glass-filled PTFE	-350–500 °F (-212–260 °C)	1,000 psi (7 MPa)	400 (2.0 m/s)	11,000 (0.39 MPa m/s)
Rulon 641	-400–500 °F (-240–260 °C)	1,000 psi (7 MPa)	400 (2.0 m/s)	10,000 (0.35 MPa m/s)
Rulon J	-400–500 °F (-240–260 °C)	750 psi (5 MPa)	400 (2.0 m/s)	7,500 (0.26 MPa m/s)
Rulon LR	-400–500 °F (-240–260 °C)	1,000 psi (7 MPa)	400 (2.0 m/s)	10,000 (0.35 MPa m/s)
UHMWPE	-200–180 °F (-129–82 °C)	1,000 psi (7 MPa)	100 (0.51 m/s)	2,000 (0.07 MPa m/s)
MDS-filled urethane	-40–180 °F (-40–82 °C)	700 psi (5 MPa)	200 (1.02 m/s)	11,000 (0.39 MPa m/s)
Vespel	-400–550 °F (-240–288 °C)	4,900 psi (34 MPa)	3,000 (15.2 m/s)	300,000 (10.5 MPa m/s)

Others

- Ceramic bearings are very hard and sand and other grit which enter the bearing are simply ground to a fine powder which does not inhibit the operation of the bearing.
- Lubrite
- Lignum vitae is a self lubricating wood and in clocks it gives extremely long life.

Lubrication



A graphite filled groove bushing

The types of lubrication system can be categorized into three groups:

- **Class I** — bearings that require the application of a lubricant from an external source
- **Class II** — Bearings that contain a lubricant within the walls of the bearing
- **Class III** — bearings made of materials that are the lubricant

Examples of the second type of bearing are Oilites and plastic bearings made from polyacetal; examples of the third type are metalized graphite bearings and PTFE bearings.

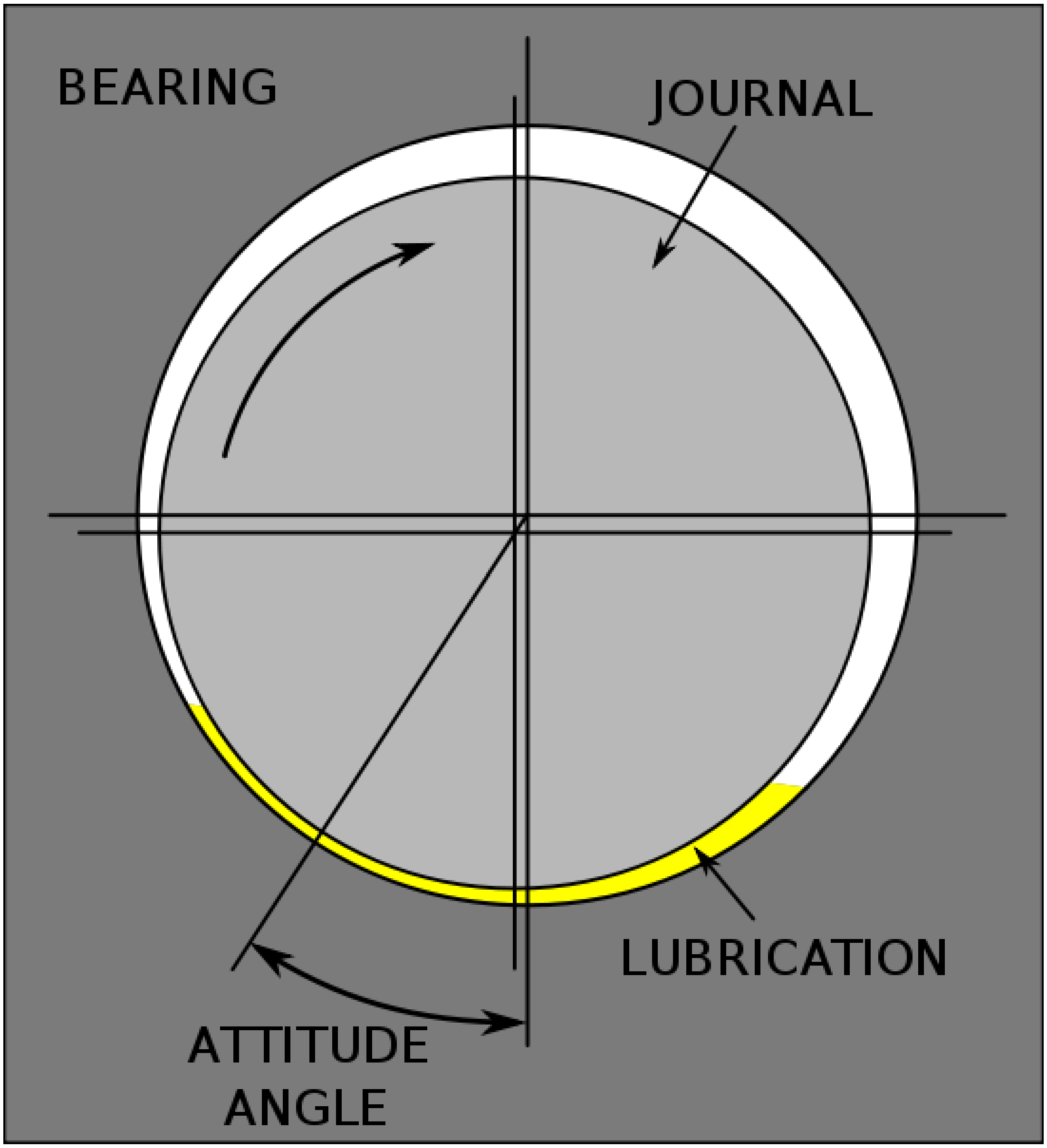
Most plain bearings have a plain inner surface, however some are grooved. The grooves help lubrication enter the bearing and cover the whole journal.

Self-lubricating plain bearings have a lubricant contained within the bearing walls. There are many forms of self-lubricating bearings. The first, and most common, are sintered metal bearings, which have porous walls. The porous walls draw oil in via capillary action and release the oil when pressure or heat are applied. Another form is a solid one-piece metal bushing with a figure eight groove channel on the ID that is filled with graphite. A similar bearing replaces the figure eight groove with holes that are plugged with graphite; this allows the bearing to be lubricated inside and out. The last form is a plastic bearing, which has the lubricant molded into the bearing. The lubricant is released as the bearing is run in.

There are three main types of lubrication: *full-film condition*, *boundary condition*, and *dry condition*. Full-film conditions are when the bearing's load is carried solely by a film of fluid lubricant and there is no contact between the two bearing surfaces. In mix or boundary conditions, load is carried partly by direct surface contact and partly by a film forming between the two. In a dry condition, the full load is carried by surface-to-surface contact.

Bearings that are made from bearing grade materials always run in the dry condition. The other two classes of plain bearings can run in all three conditions; the condition in which a bearing runs is dependent on the operating conditions, load, relative surface speed, clearance within the bearing, quality and quantity of lubricant, and temperature (affecting lubricant viscosity). If the plain bearing is not designed to run in the dry or boundary condition it will wear out and have a high coefficient of friction. Dry and boundary conditions may be experienced even in a fluid bearing when operating outside of its normal operating conditions; *e.g.*, at startup and shutdown.

Fluid lubrication



A schematic of a journal bearing under a hydrodynamic lubrication state showing how the journal centerline shifts from the bearing centerline.

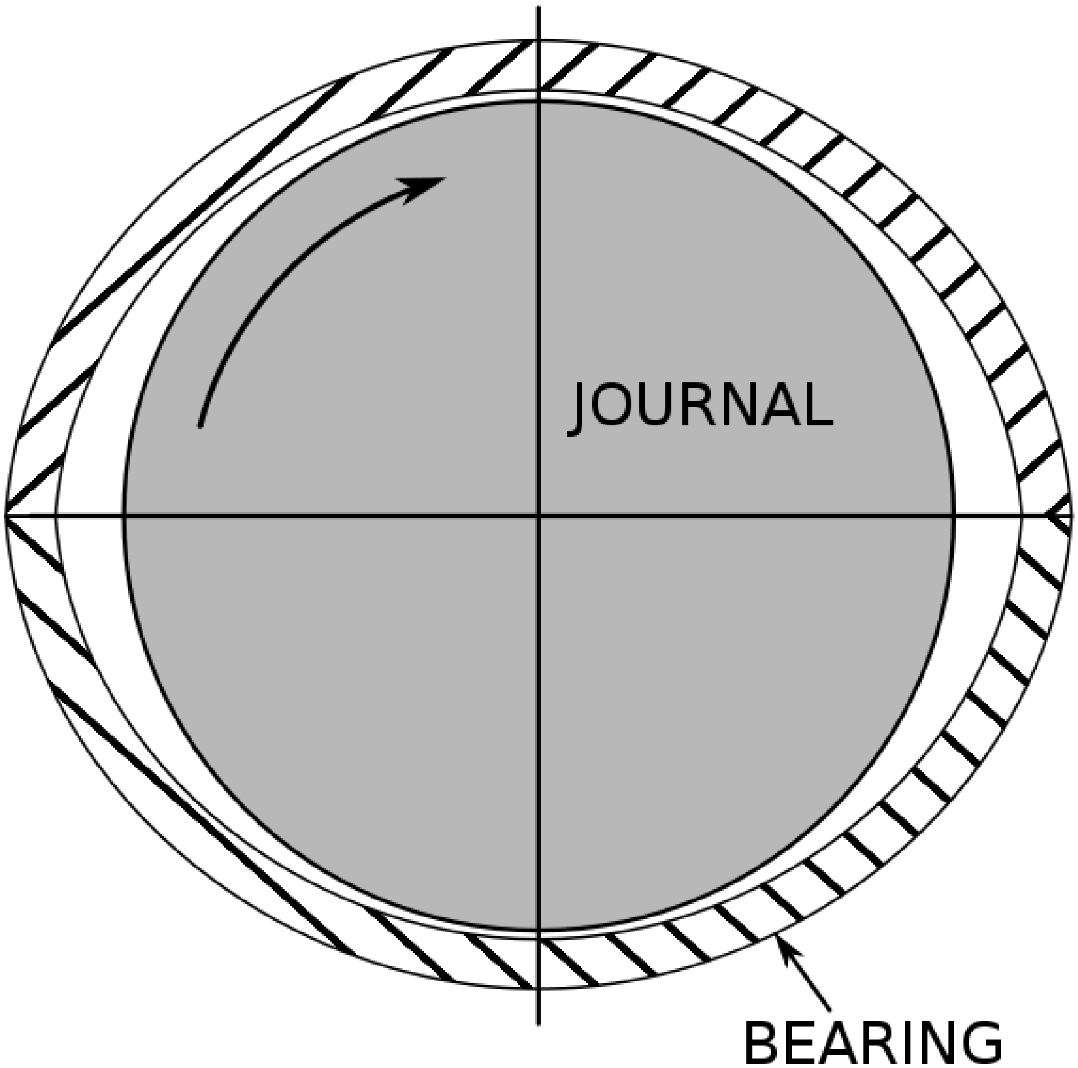
Fluid lubrication results in a full-film or a boundary condition lubrication mode. A properly designed bearing system reduces friction by eliminating surface-to-surface contact between the journal and bearing through fluid dynamic effects.

Fluid bearings can be *hydrostatically* or *hydrodynamically* lubricated. Hydrostatically lubricated bearings are lubricated by an external pump which always keeps a *static* amount of pressure. In a hydrodynamic bearing the pressure in the oil film is maintained by the rotation of the journal. Hydrostatic bearings enter a *hydrodynamic state* when the journal is rotating. Hydrostatic bearings almost always use oil, while hydrodynamic bearings can use oil or grease. An example of a hydrostatic bearing is the heavily-loaded bearings (main, connecting rod big-end and camshaft) in an automobile engine, which are usually fed oil via a hole in the bearing.

Hydrodynamic bearings require greater care in design and operation than hydrostatic bearings. They are also more prone to initial wear because lubrication does not occur until there is rotation of the shaft. At low rotational speeds the lubrication may not attain complete separation between shaft and bushing. As a result, hydrodynamic bearings are often aided by secondary bearings which support the shaft during start and stop periods, protecting the fine tolerance machined surfaces of the journal bearing.

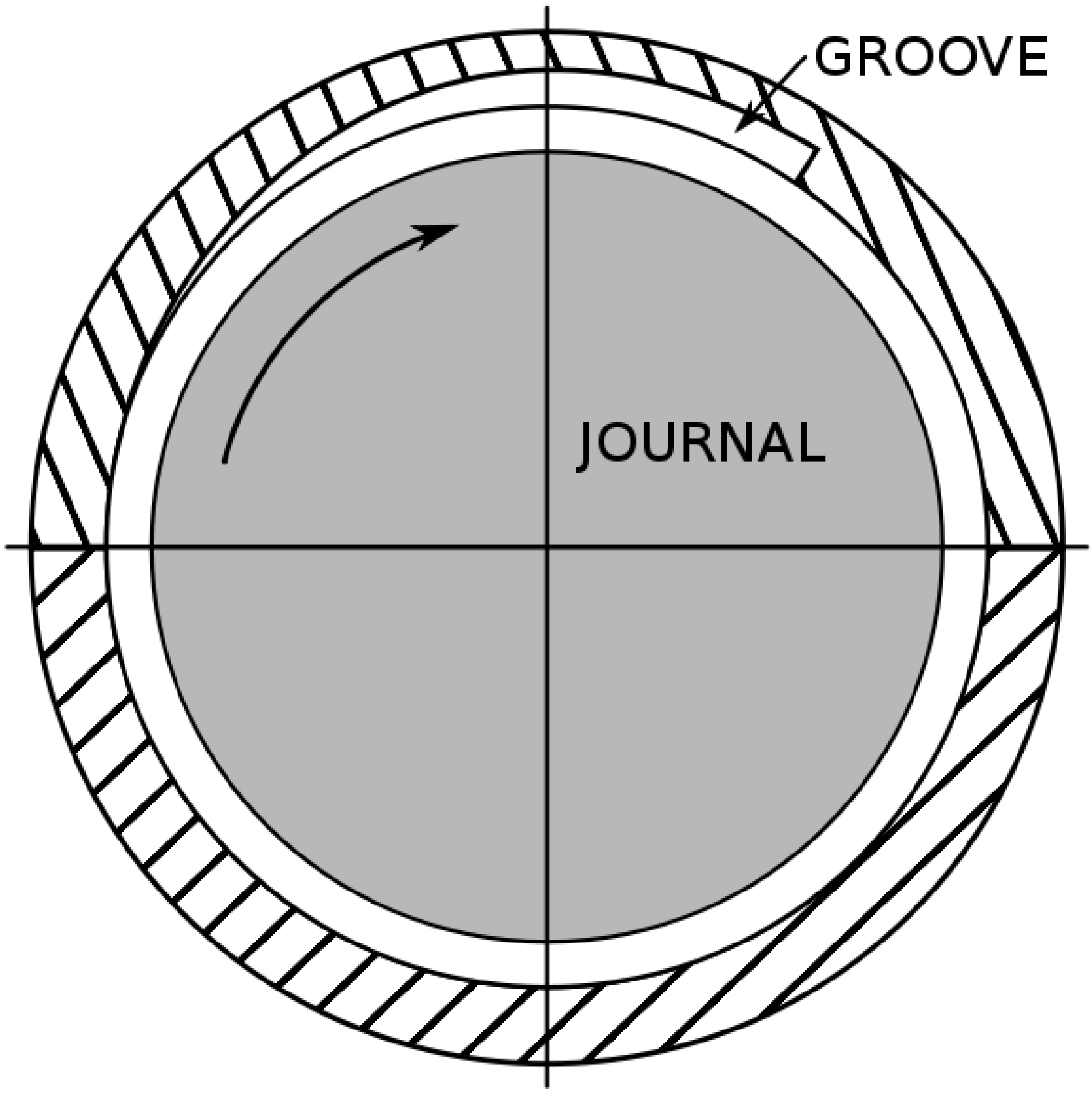
In the hydrodynamic state a lubrication "wedge" forms, which lifts the journal. The journal also slightly shifts horizontally in the direction of rotation. The location of the journal is measured by the *attitude angle*, which is angle formed between the horizontal and a line that crosses through the center of the journal and the center of the bearing. The attitude angle is dependent on the direction of rotation, oil pressure (in hydrostatic bearings), and electromagnetic forces (in electromagnetic equipment).

One disadvantage specific to fluid lubricated journal bearings is *oil whirl*, also known as *oil whip*. Oil whirl is when a lubrication wedge cannot form, but instead "whirls" around the bearing. This leads to direct contact between the journal and the bearing, which quickly wears out the bearing. Moreover, the journal precesses in the opposite direction of rotation causing the friction to increase.



A lemon bore

One design used to minimize this problem is called the *lemon bore* or *elliptical bore*. In this design shims are installed between the two halves of the bearing housing and then the bore is machined to size. After the shims are removed the bore resembles a lemon shape, which decreases the clearance in one direction of the bore and increases the pre-load in this direction. The disadvantage of this design is its lower load carrying capacity, as compared to typical journal bearings. It is also still susceptible to oil whirl at high speeds, however its cost is relatively low.



A pressure dam

Another design is the *pressure dam* or *dammed groove*, which has a shallow relief cut in the center of the bearing over the top half of the bearing. The groove abruptly stops in order to create a downward force to stabilize the journal. This design has a high load capacity and corrects most oil whirl situations. The disadvantage is that it only works in one direction. Offsetting the bearing halves does the same thing as the pressure dam. The only difference is the load capacity increases as the offset increases.

A more radical design is the tilting-pad design, which uses multiple pad that are designed to move with changing loads. It is usually used in very large applications.

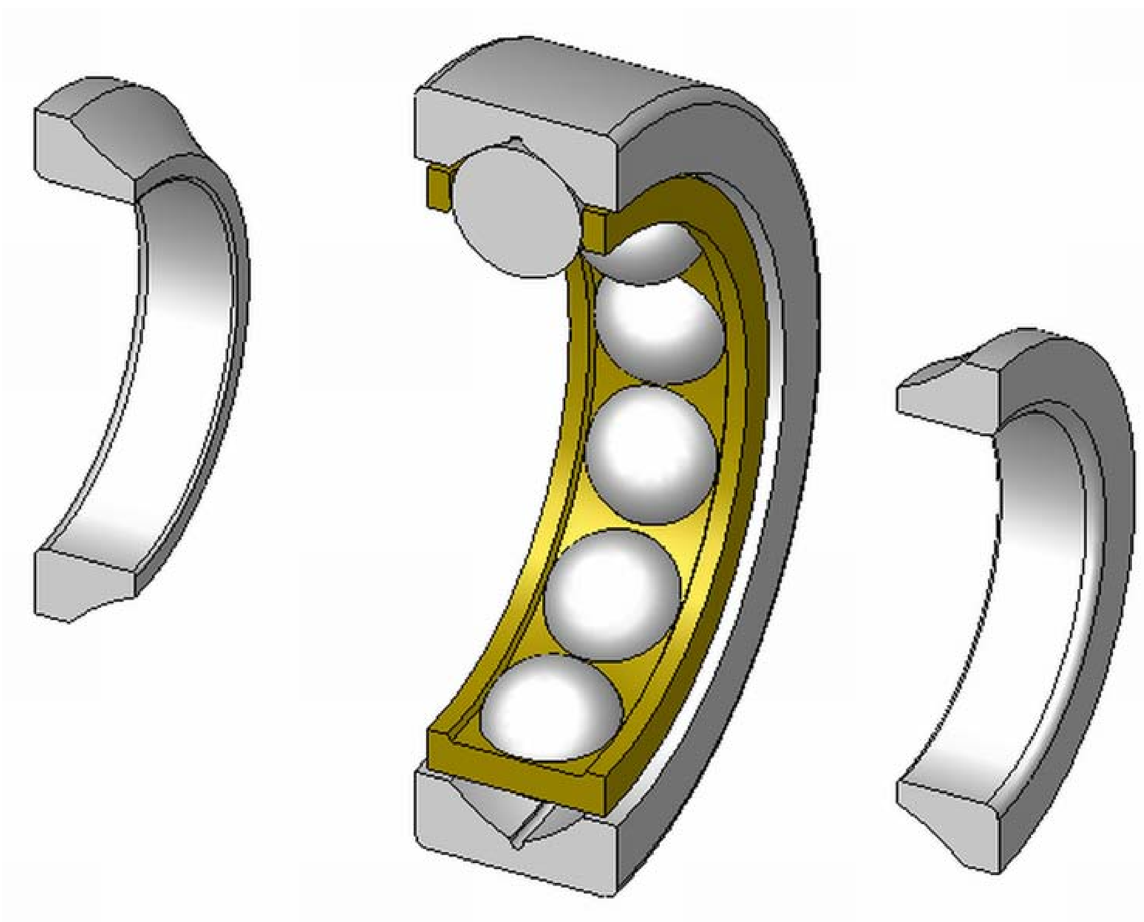
Related components

Other components that are commonly used with plain bearings include:

- *Pillow block*: These are standardized bearing mounts designed to accept plain bearings. They are designed to mount to a flat surface.
- *Ring oiler*: A lubricating mechanism used in the first half of the 20th century for medium speed applications.
- *Stuffing box*: A sealing system used to keep fluid from leaking out of a pressurized system through the plain bearing.

Chapter 3

Rolling-Element Bearing



Four-point-contact radial ball bearings

A **rolling-element bearing** is a bearing which carries a load by placing round elements between the two pieces. The relative motion of the pieces causes the round elements to roll with very little rolling resistance and with little sliding.

One of the earliest and best-known rolling-element bearings are sets of logs laid on the ground with a large stone block on top. As the stone is pulled, the logs roll along the ground with little sliding friction. As each log comes out the back, it is moved to the front where the block then rolls on to it. It is possible to imitate such a bearing by placing several pens or pencils on a table and placing an item on top of them.

A rolling-element rotary bearing uses a shaft in a much larger hole, and cylinders called "rollers" tightly fill the space between the shaft and hole. As the shaft turns, each roller acts as the logs in the above example. However, since the bearing is round, the rollers never fall out from under the load.

Rolling-element bearings have the advantage of a good tradeoff between cost, size, weight, carrying capacity, durability, accuracy, friction, and so on. Other bearing designs are often better on one specific attribute, but worse in most other attributes, although fluid bearings can sometimes simultaneously outperform on carrying capacity, durability, accuracy, friction, rotation rate and sometimes cost. Only plain bearings are used as widely as rolling-element bearings.

Design

Typical rolling-element bearings range in size from 10 mm diameter to a few metres diameter, and have load-carrying capacity from a few tens of grams to many thousands of tonnes.

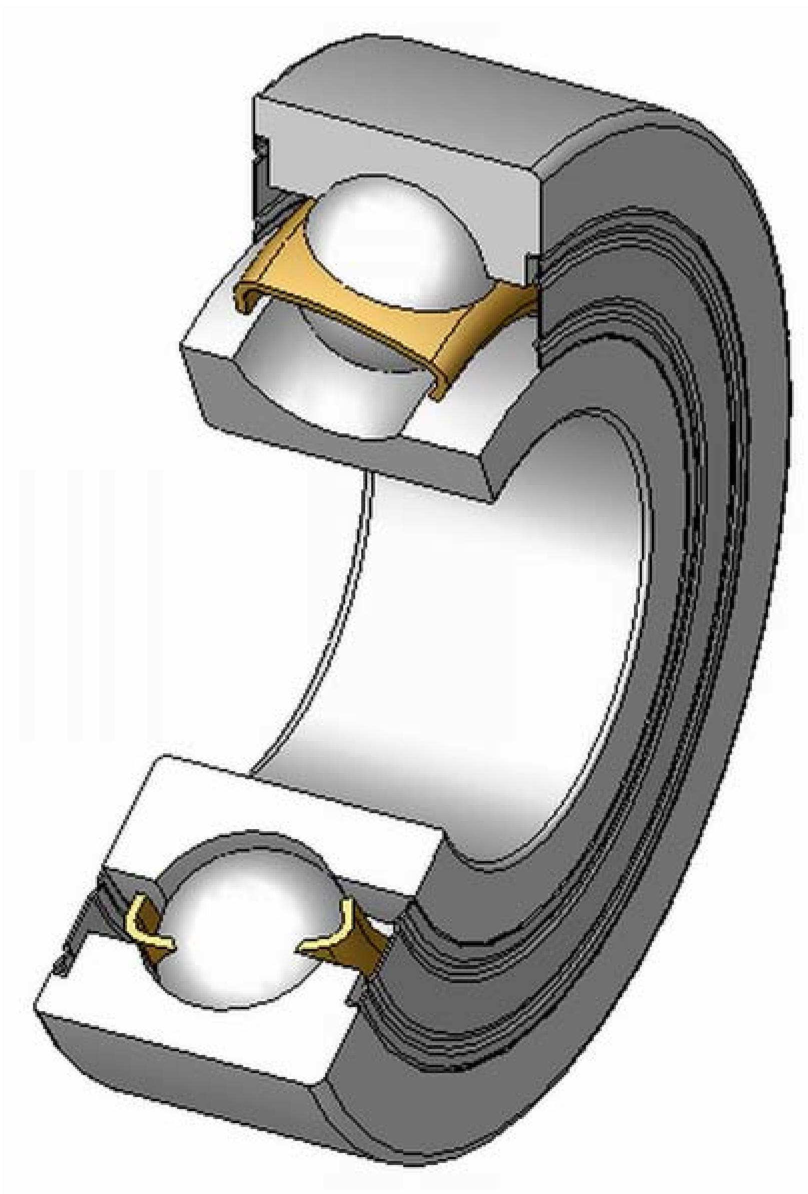
A particularly common kind of rolling-element bearing is the ball bearing. The bearing has inner and outer *races* and a set of balls. Each race is a ring with a groove where the balls rest. The groove is usually shaped so the ball is a slightly loose fit in the groove. Thus, in principle, the ball contacts each race at a single point. However, a load on an infinitely small point would cause infinitely high contact pressure. In practice, the ball deforms (flattens) slightly where it contacts each race, much as a tire flattens where it touches the road. The race also dents slightly where each ball presses on it. Thus, the contact between ball and race is of finite size and has finite pressure. Note also that the deformed ball and race do not roll entirely smoothly because different parts of the ball are moving at different speeds as it rolls. Thus, there are opposing forces and sliding motions at each ball/race contact. Overall, these cause bearing drag.

Most rolling element bearings use *cages* to keep the balls separate. This reduces wear and friction, since it avoids the balls rubbing against each other as they roll, and precludes them from jamming. Caged roller bearings were invented by John Harrison in the mid-18th century as part of his work on chronometers.

Types of rolling elements

There are five types of rolling-elements that are used in rolling element bearings: balls, cylindrical rollers, tapered rollers, spherical rollers, and needles.

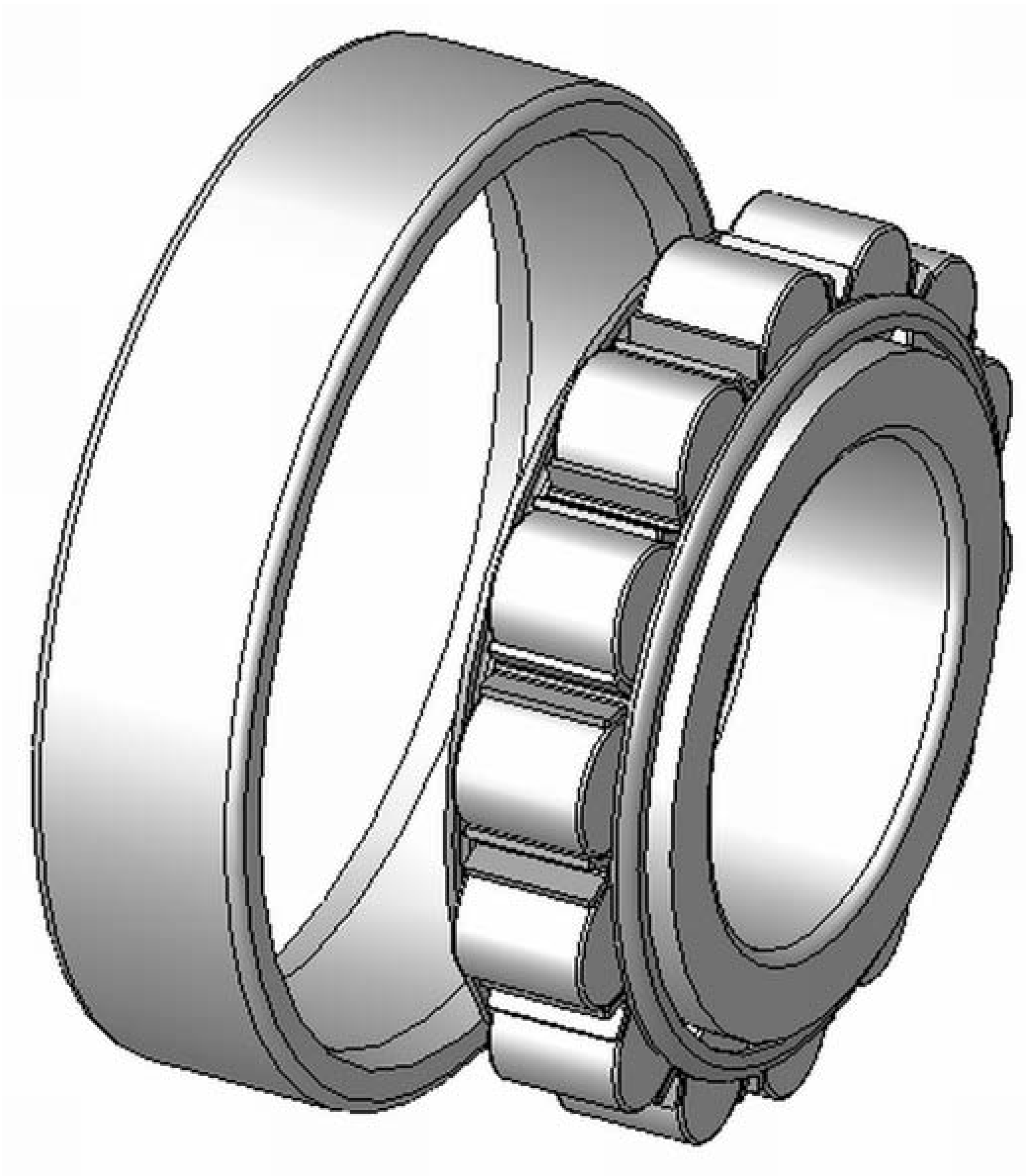
Ball



A ball bearing

Ball bearings use balls instead of cylinders. Ball bearings can support both radial (perpendicular to the shaft) and axial loads (parallel to the shaft). For lightly-loaded bearings, balls offer lower friction than rollers. Ball bearings can operate when the bearing races are misaligned. Precision balls are typically cheaper to produce than shapes such as rollers; combined with high-volume use, ball bearings are often much cheaper than other bearings of similar dimensions. Ball bearings may have high point loads, limiting total load capacity compared to other bearings of similar dimensions.

Cylindrical roller



A roller bearing

Common roller bearings use cylinders of slightly greater length than diameter. Roller bearings typically have higher load capacity than ball bearings, but a lower capacity and higher friction under loads perpendicular to the primary supported direction. If the inner and outer races are misaligned, the bearing capacity often drops quickly compared to either a ball bearing or a spherical roller bearing.

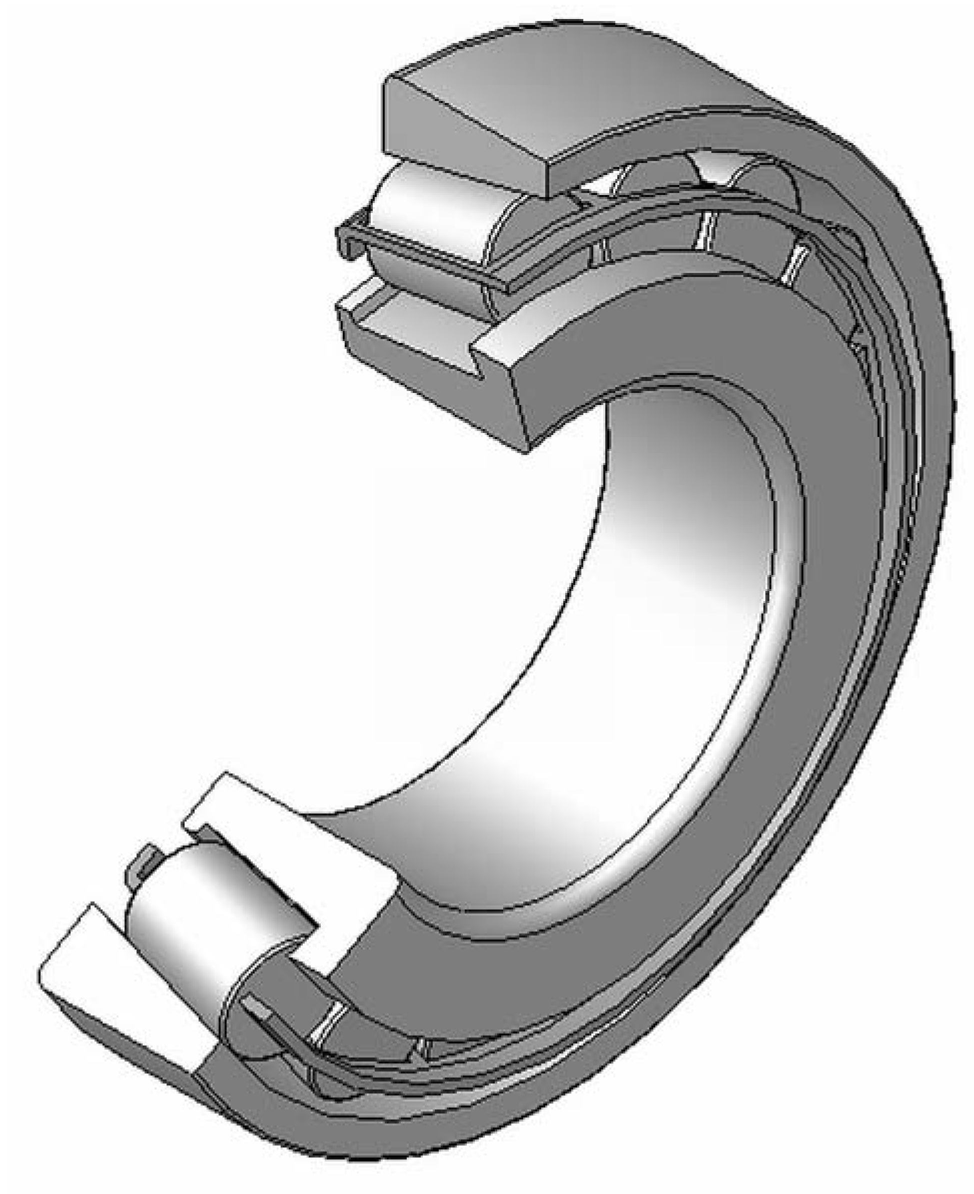
Roller bearings are the earliest known type of rolling-element-bearing, dating back to at least 40 BC.

Needle

A needle roller bearing

Needle roller bearings use very long and thin cylinders. Often the ends of the rollers taper to points, and these are used to keep the rollers captive, or they may be hemispherical and not captive but held by the shaft itself or a similar arrangement. Since the rollers are thin, the outside diameter of the bearing is only slightly larger than the hole in the middle. However, the small-diameter rollers must bend sharply where they contact the races, and thus the bearing fatigues relatively quickly.

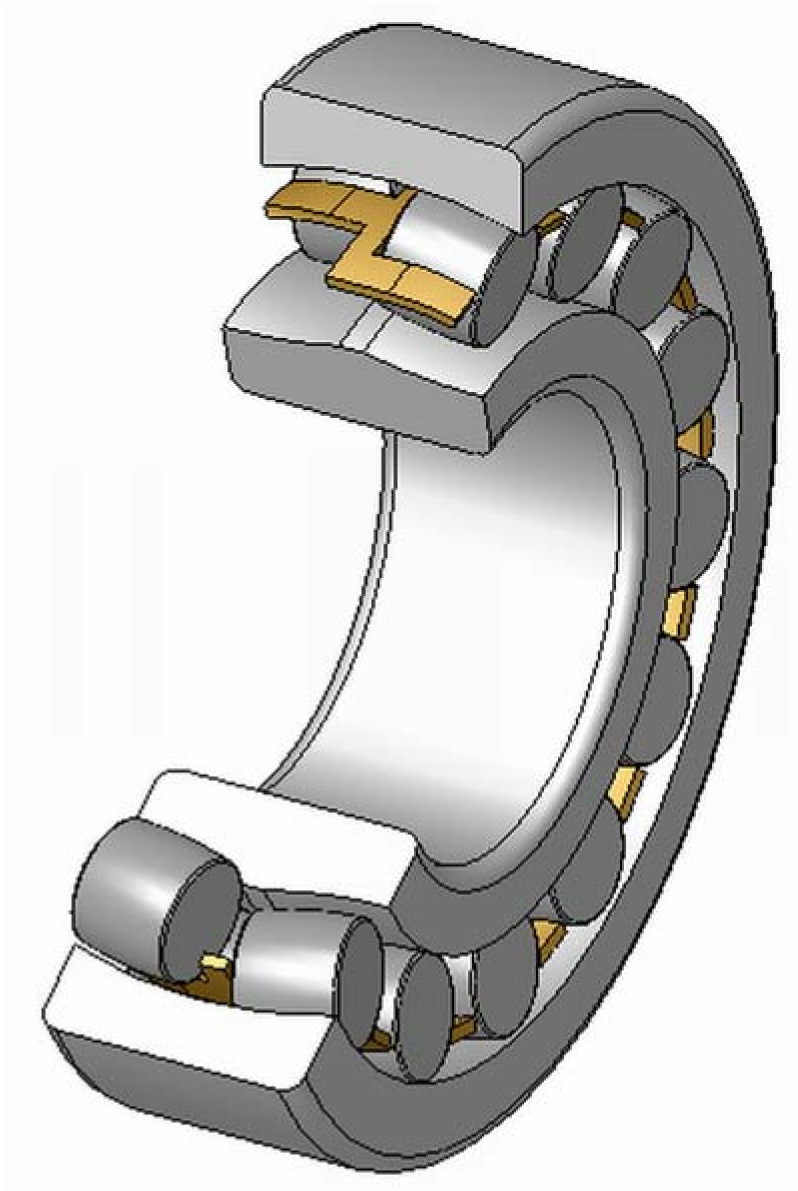
Tapered roller



Tapered roller bearings

Tapered roller bearings use conical rollers that run on conical races. Most roller bearings only take radial or axial loads, but tapered roller bearings support both radial and axial loads, and generally can carry higher loads than ball bearings due to greater contact area. Taper roller bearings are used, for example, as the wheel bearings of most cars, trucks, buses, and so on. The downsides to this bearing is that due to manufacturing complexities, tapered roller bearings are usually more expensive than ball bearings; and additionally under heavy loads the tapered roller is like a wedge and bearing loads tend to try to eject the roller; the force from the collar which keeps the roller in the bearing adds to bearing friction compared to ball bearings.

Spherical roller



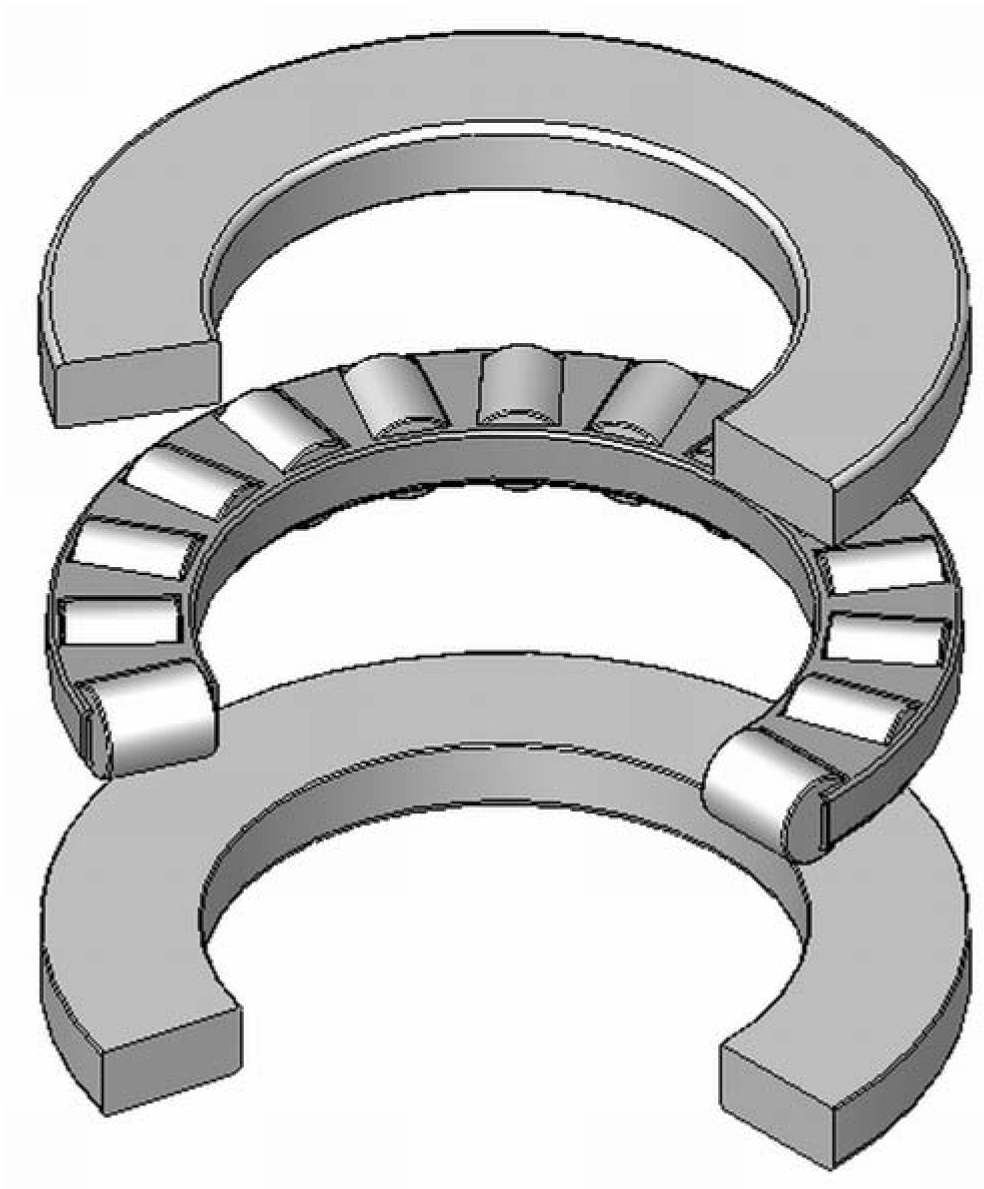
Spherical roller bearings

Spherical roller bearings use rollers that are thicker in the middle and thinner at the ends; the race is shaped to match. Spherical roller bearings can thus adjust to support misaligned loads. However, spherical rollers are difficult to produce and thus expensive, and the bearings have higher friction than a comparable ball bearing since different parts of the spherical rollers run at different speeds on the rounded race and thus there are opposing forces along the bearing/race contact.

Configurations

The configuration of the races determine the types of motions and loads that a bearing can best support. A given configuration can serve multiple of the following types of loading.

Thrust loadings



A thrust roller bearing

Thrust bearings are used to support axial loads, such as vertical shafts. Commonly spherical, conical or cylindrical rollers are used; but non-rolling element bearings such as hydrostatic or magnetic bearings see some use where particularly heavy loads or low friction is needed.

Radial loadings

Rolling element bearings are often used for axles due to their low rolling friction. For light loads, such as bicycles, ball bearings are often used. For heavy loads and where the loads can greatly change during cornering, such as cars and trucks, tapered rolling bearings are used.

Linear motion

Linear motion roller-element bearings are typically designed for either shafts or flat surfaces. Flat surface bearings often consist of rollers and are mounted in a cage, which is then placed between the two flat surfaces; a common example is drawer-support hardware. Roller-element bearing for a shaft use bearing balls in a groove designed to recirculate them from one end to the other as the bearing moves; as such, they are called *linear ball bearings* or *recirculating bearings*.

Bearing failure

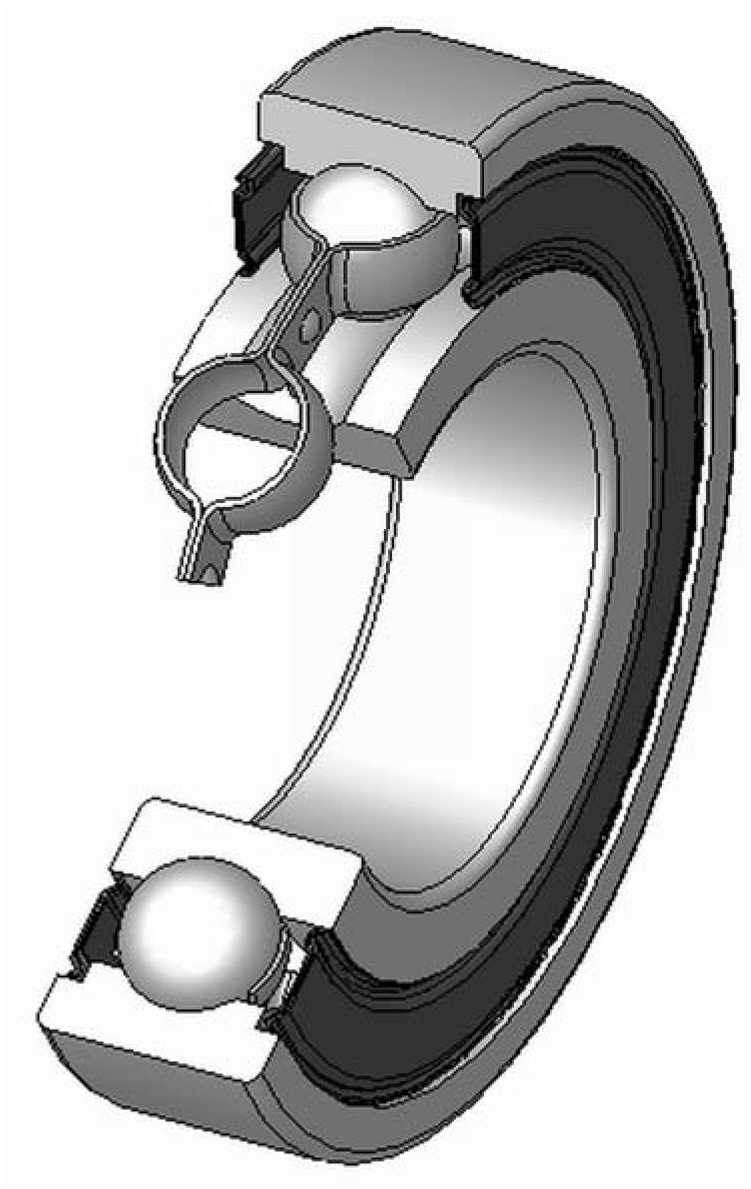
A prematurely failed rear bearing cone from a mountain bicycle, caused by a combination of pitting due to wet conditions, improper lubrication, and fatigue from frequent shock loading.

Rolling-element bearings often work well in non-ideal conditions, but sometimes minor problems cause bearings to fail quickly and mysteriously. For example, with a stationary (non-rotating) load, small vibrations can gradually press out the lubricant between the races and rollers or balls (false brinelling). Without lubricant the bearing fails, even though it is not rotating and thus is apparently not being used. For these sorts of reasons, much of bearing design is about failure analysis.

There are three usual limits to the lifetime or load capacity of a bearing: abrasion, fatigue and pressure-induced welding. Abrasion is when the surface is eroded by hard contaminants scraping at the bearing materials. Fatigue is when a material breaks after it is repeatedly loaded and released. Where the ball or roller touches the race there is always some deformation, and hence a risk of fatigue. Smaller balls or rollers deform more sharply, and so tend to fatigue faster. Pressure-induced welding is when two metal pieces are pressed together at very high pressure and they become one. Although balls, rollers and races may look smooth, they are microscopically rough. Thus, there are high-pressure spots which push away the bearing lubricant. Sometimes, the resulting metal-to-metal contact welds a microscopic part of the ball or roller to the race. As the bearing continues to rotate, the weld is then torn apart, but it may leave race welded to bearing or bearing welded to race.

Although there are many other apparent causes of bearing failure, most can be reduced to these three. For example, a bearing which is run dry of lubricant fails not because it is "without lubricant", but because lack of lubrication leads to fatigue and welding, and the resulting wear debris can cause abrasion. Similar events occur in false brinelling damage. In high speed applications, the oil flow also reduces the bearing metal temperature by convection. The oil becomes the heat sink for the friction losses generated by the bearing.

Constraints and trade-offs



Caged radial ball bearings

All parts of a bearing are subject to many design constraints. For example, the inner and outer races are often complex shapes, making them difficult to manufacture. Balls and rollers, though simpler in shape, are small; since they bend sharply where they run on the races, the bearings are prone to fatigue. The loads within a bearing assembly are also affected by the speed of operation: rolling-element bearings may spin over 100,000 rpm, and the principal load in such a bearing may be momentum rather than the applied load. Smaller rolling elements are lighter and thus have less momentum, but smaller elements also bend more sharply where they contact the race, causing them to fail more rapidly from fatigue. Maximum rolling element bearing speeds are often specified in 'DN', which is the product of the diameter (in mm) and the maximum RPM. For angular contact bearings DNs over 2.1 million have been found to be reliable in high performance rocketry applications.

There are also many material issues: a harder material may be more durable against abrasion but more likely to suffer fatigue fracture, so the material varies with the application, and while steel is most common for rolling-element bearings, plastics, glass, and ceramics are all in common use. A small defect (irregularity) in the material is often responsible for bearing failure; one of the biggest improvements in the life of common bearings during the second half of the 20th century was the use of more homogeneous materials, rather than better materials or lubricants (though both were also significant). Lubricant properties vary with temperature and load, so the best lubricant varies with application.

Although bearings tend to wear out with use, designers can make tradeoffs of bearing size and cost versus lifetime. A bearing can last indefinitely—longer than the rest of the machine—if it is kept cool, clean, lubricated, is run within the rated load, and if the bearing materials are sufficiently free of microscopic defects. Note that cooling, lubrication, and sealing are thus important parts of the bearing design.

The needed bearing lifetime also varies with the application. For example, Tedric A. Harris reports in his *Rolling Bearing Analysis* on an oxygen pump bearing in the U.S. Space Shuttle which could not be adequately isolated from the liquid oxygen being pumped. All lubricants reacted with the oxygen, leading to fires and other failures. The solution was to lubricate the bearing with the oxygen. Although liquid oxygen is a poor lubricant, it was adequate, since the service life of the pump was just a few hours.

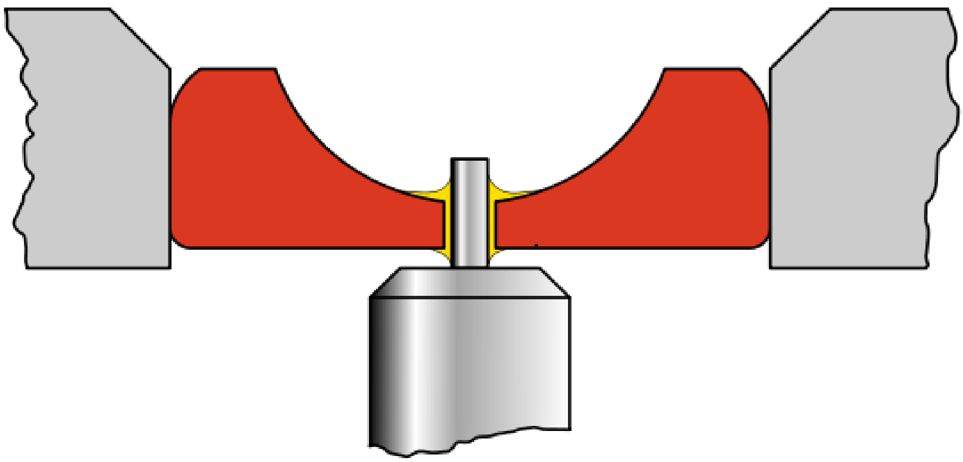
The operating environment and service needs are also important design considerations. Some bearing assemblies require routine addition of lubricants, while others are factory sealed, requiring no further maintenance for the life of the mechanical assembly. Although seals are appealing, they increase friction, and in a permanently-sealed bearing the lubricant may become contaminated by hard particles, such as steel chips from the race or bearing, sand, or grit that gets past the seal. Contamination in the lubricant is abrasive and greatly reduces the operating life of the bearing assembly. Another major cause of bearing failure is the presence of water in the lubrication oil. Online water-in-oil monitors have been introduced in recent years to monitor the effects of both particles and the presence of water in oil and their combined effect.

Chapter 4

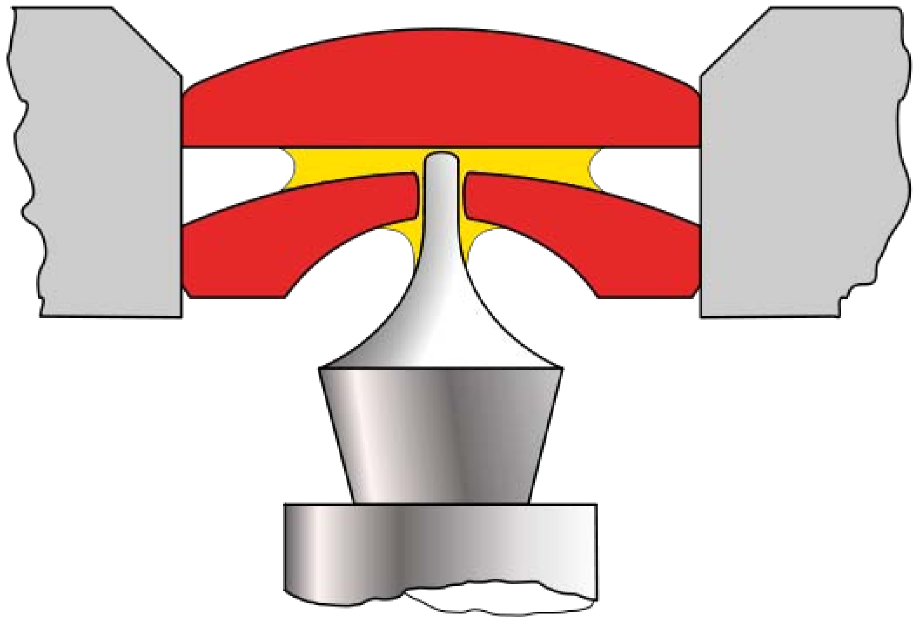
Jewel Bearing and Flexure Bearing

Jewel bearing

Ruby jewel bearings used for a balance wheel in a mechanical watch movement.



Cross section of a jewel bearing in a mechanical watch. This type of donut-shaped bearing (red) is called a *hole jewel*, used for most of the ordinary wheels in the gear train and usually made of synthetic sapphire (ruby). It is press-fitted into a hole in the movement's supporting plate (grey). The cup-shaped depression in the top of the jewel is the oil cup; its purpose is to hold the lubricating oil (yellow) in contact with the bearing shaft by capillary action.



In wheels where friction is critical, a *capstone* is added on the end to prevent the shoulder of the shaft from bearing against the face of the jewel.

A **jewel bearing** is a plain bearing in which a metal spindle turns in a jewel-lined pivot hole. The hole is typically shaped like a torus and is slightly larger than the shaft diameter. The jewel material is usually synthetic sapphire. Jewel bearings are used in precision instruments, but their largest use is in mechanical watches.

History

Jewel bearings were invented in 1704 for use in watches by Nicolas Fatio de Duillier, Peter Debaufre, and Jacob Debaufre, who received an English patent for the idea. Originally natural jewels were used, such as diamond, sapphire, ruby, and garnet. In 1902, a process to make synthetic sapphire and ruby (crystalline aluminum oxide, also known as corundum) was invented by Auguste Verneuil, making jewelled bearings much less expensive. Today most jewelled bearings are synthetic ruby or sapphire.

Historically, jewel pivots were made by grinding using diamond abrasive. Modern jewel pivots are often made using high-powered lasers, chemical etching, and ultrasonic milling.

Characteristics

The advantages of jewel bearings include high accuracy, very small size and weight, low and predictable friction, including good temperature stability, and the ability to operate without lubrication and in corrosive environments. They are known for their low static friction and highly consistent dynamic friction. The static coefficient of friction of brass-on-steel is 0.35, while that of sapphire-on-steel is 0.10–0.15. Sapphire surfaces are very hard and durable, with Mohs hardness of 9 and Knoop hardness of 2000, and can maintain smoothness over decades of use, thus reducing friction variability. Disadvantages include brittleness and fragility, limited availability/applicability in medium and large bearing sizes and capacities, and friction variations if the load is not axial.

Uses

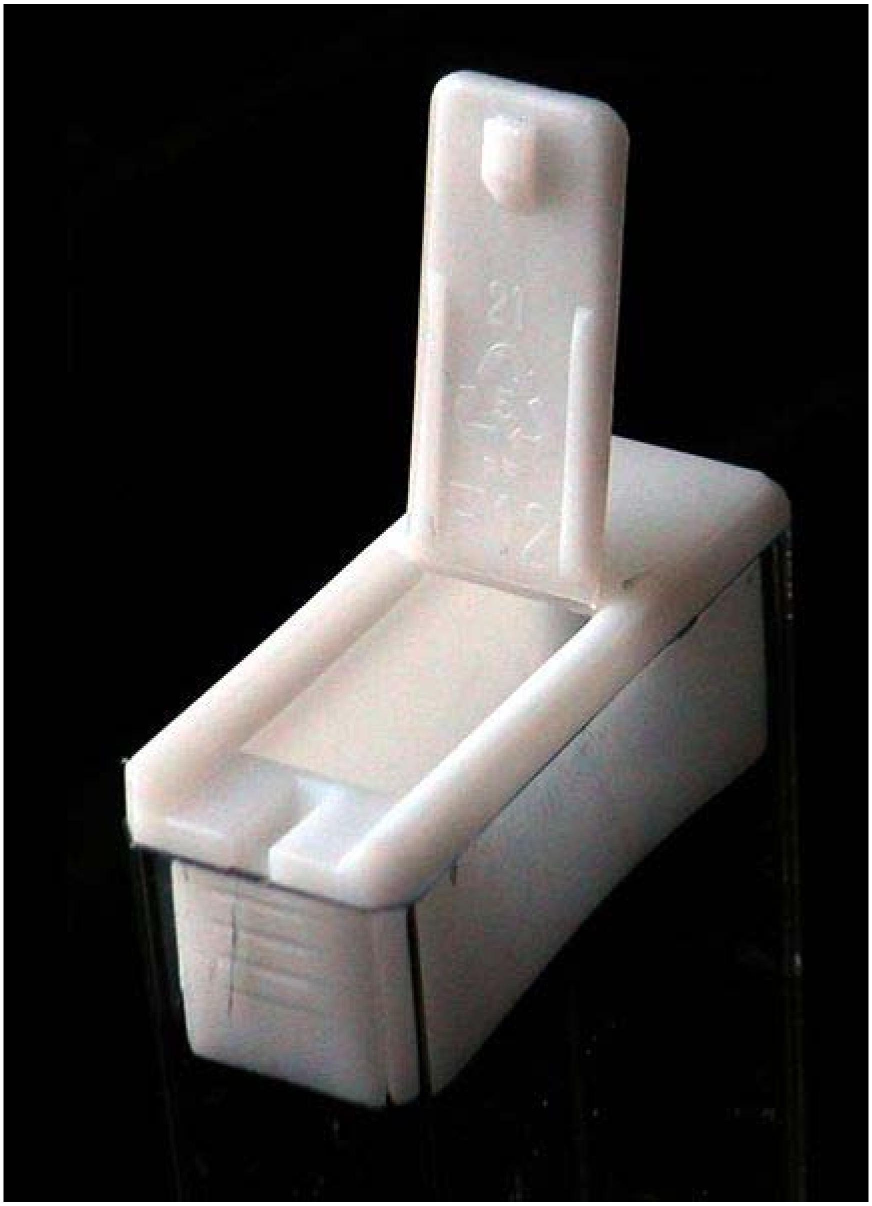
The largest use for jewel bearings is in mechanical watches, where their low and predictable friction improves watch accuracy. A typical mark of watch quality was a note such as *17 jewels*. More jewel bearings often meant better precision. Some makers added non-functional or unnecessary jewels to give the impression of accuracy. Some watches had as many as 100 jewels, most of them of no use. A typical *fully jeweled* time-only watch has 17 jewels: two cap jewels, two pivot jewels, an impulse jewel for the balance wheel, two pivot jewels, two pallet jewels for the pallet fork, and two pivot jewels each for the escape, fourth, third, and center wheels. Modern electronic watches achieve accuracy entirely separate from the friction of the mechanism, but early quartz watches used jewels to increase battery life, and high-grade quartz watches use jewels to reduce friction and wear.

Today, jewel bearings are also used widely in sensitive measuring equipment. They are typically used for very small applications, such as high-precision instruments; galvanometers, compasses, gimbals, and turbine flow meters. Bearing bores are typically less than 1 mm and typically support loads of under the weight of 1 gram, although they are made as large as 10 mm and support loads up to about the weight of 500 g.

Flexure bearing

A **flexure bearing** is a bearing which allows motion by bending a load element.

A typical flexure bearing is just one part, joining two other parts. For example, a hinge may be made by attaching a long strip of a flexible element to a door and to the door frame. Another example is a rope swing, where the rope is tied to a tree branch.



A living hinge (a type of flexure bearing), on the lid of a Tic Tac box.

Flexure bearings have the advantage over most other bearings that they are simple and thus inexpensive. They are also often compact, light weight, have very low friction, and are easier to repair without specialized equipment. Flexure bearings have the disadvantages that the range of motion is limited, and often very limited for bearings that support high loads.

A flexure bearing relies on the bearing element being made of a material which can be repeatedly flexed without disintegrating. However, most materials fall apart if flexed a lot. For example, most metals will fatigue with repeated flexing, and will eventually snap. Thus, one part of flexure bearing design is avoiding fatigue. Note, however, that fatigue is important in other bearings. For example, the rollers and races in a rolling-element bearing fatigue as they flatten against each other.

Flexure bearings can give very low friction and also give very predictable friction. Many other bearings rely on sliding or rolling motions, which are necessarily uneven because the bearing surfaces are never perfectly flat. A flexure bearing operates by bending of materials, which causes motion at microscopic level, so friction is very uniform. For this reason, flexure bearings are often used in sensitive precision measuring equipment.

Flexure bearings are not limited to low loads, however. For example, the drive shafts of some sports cars replace cardan universal joints with an equivalent joint called a rag joint which works by bending rubberized fabric. The resulting joint is lighter yet is capable of carrying hundreds of kilowatts, with adequate durability for a sports car.

Many flexure bearings are combined with other elements. For example, many motor vehicles use leaf springs. The spring both holds the position of the axle as the axle moves (flexure bearing) and provides force to support the vehicle (springing). In many cases it is not clear where flexure bearing leaves off and something else takes up. For example, turbines are often supported on flexible shafts so an imperfectly-balanced turbine can find its own center and run with reduced vibration. Seen one way, the flexible shaft includes the function of a flexure bearing; seen another, the shaft is not a "bearing".

Chapter 5

Fluid Bearing

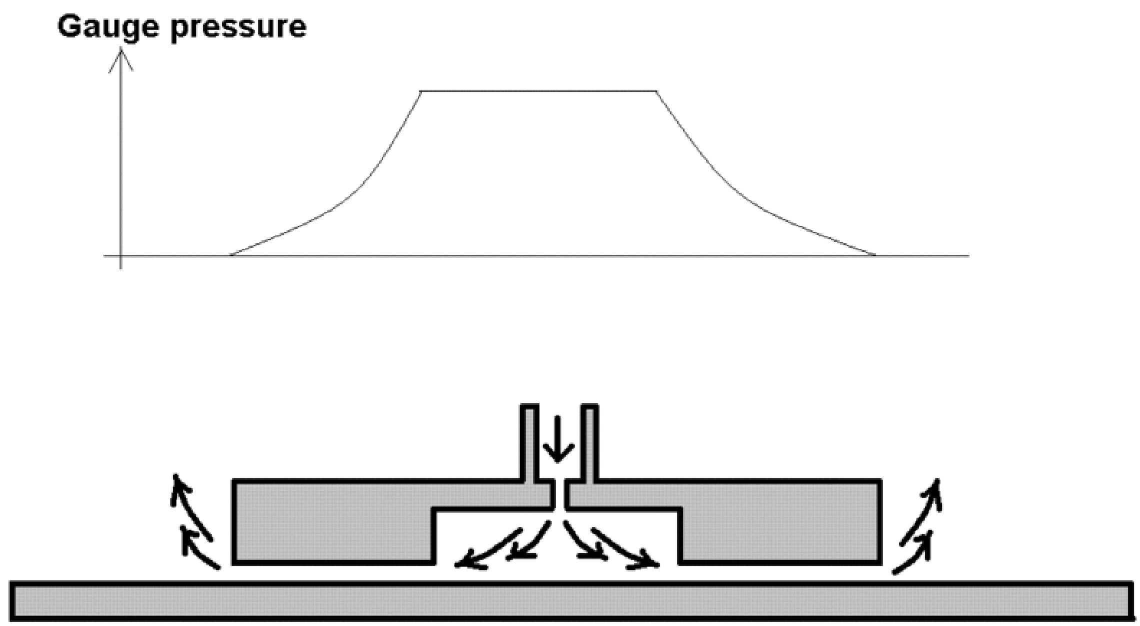
Hydrodynamic bearing demonstration rig.

Fluid bearings are bearings which solely support the bearing's loads on a thin layer of liquid or gas.

They can be broadly classified as **fluid dynamic bearings** or **hydrostatic bearings**. Hydrostatic bearings are externally pressurized fluid bearings, where the fluid is usually oil, water or air, and the pressurization is done by a pump. Hydrodynamic bearings rely on the high speed of the journal self-pressurizing the fluid in a wedge between the faces.

Fluid bearings are frequently used in high load, high speed or high precision applications where ordinary ball bearings have short life or high noise and vibration. They are also used increasingly to reduce cost. For example, hard disk drive motor fluid bearings are both quieter and cheaper than the ball bearings they replace.

Operation



A hydrostatic bearing has two surfaces which has a fluid forced, via a restrictive orifice, in between the surfaces so that it keeps them apart. If the gap between the surfaces reduces then the outflow via the edges of the bearing is reduced and the pressure goes up, forcing the surfaces apart again very strongly, giving excellent control of the gap and giving low friction

Fluid bearings use a thin layer of liquid or gas fluid between the bearing faces, typically sealed around or under the rotating shaft.

There are two principal ways of getting the fluid into the bearing:

- In **fluid static**, **hydrostatic** and many **gas** or **air bearings**, the fluid is pumped in through an orifice or through a porous material.
- In **fluid-dynamic bearings**, the bearing rotation sucks the fluid on to the inner surface of the bearing, forming a lubricating wedge under or around the shaft.

Hydrostatic bearings rely on an external pump. The power required by that pump contributes to system energy loss just as bearing friction otherwise would. Better seals can reduce leak rates and pumping power, but may increase friction.

Hydrodynamic bearings rely on bearing motion to suck fluid into the bearing and may have high friction and short life at speeds lower than design or during starts and stops. An external pump or secondary bearing may be used for startup and shutdown to prevent damage to the hydrodynamic bearing. A secondary bearing may have high friction and short operating life, but good overall service life if bearing starts and stops are infrequent.

Hydrodynamic lubrication

Hydrodynamic (HD) *lubrication*, also known as *fluid film lubrication* has essential elements:

1. A lubricant, which must be a viscous fluid.
2. Hydrodynamic flow behavior of fluid between bearing and journal.
3. The surfaces between which the fluid films move must be convergent.

Hydrodynamic (Full Film) Lubrication is obtained when two mating surfaces are completely separated by a cohesive film of lubricant.

The thickness of the film thus exceeds the combined roughness of the surfaces. The coefficient of friction is lower than with boundary-layer lubrication. Hydrodynamic lubrication prevents wear in moving parts, and metal to metal contact is prevented.

Hydrodynamic lubrication requires thin, converging fluid films. These fluids can be liquid or gas, so long as they exhibit viscosity. In computer components, like a hard disk, heads are supported by hydrodynamic lubrication in which the fluid film is the atmosphere.

The scale of these films are on the order of micrometers. Their convergence creates pressures normal to the surfaces they contact, forcing them apart.

3 Types of bearings include:

- Self-acting: Film exists due to relative motion.
- Squeeze film: Film exists due to relative normal motion.
- Externally pressurized: Film exists due to external pressurization.

Conceptually the bearings can be thought of as two major geometric classes: bearing-journal(Anti Friction), and plane-slider(Friction).

The Reynolds equations can be used to derive the governing principles for the fluids. Note that when gases are used, their derivation is much more involved.

The thin films can be thought to have pressure and viscous forces acting on them. Because there is a difference in velocity there will be a difference in the surface traction vectors. Because of mass conservation we can also assume an increase in pressure, making the body forces different.

Characteristics and principles of operation

Fluid bearings can be relatively cheap compared to other bearings with a similar load rating. The bearing can be as simple as two smooth surfaces with seals to keep in the working fluid. In contrast, a conventional rolling-element bearing may require many high-precision rollers with complicated shapes. Hydrostatic and many gas bearings do have the complication and expense of external pumps.

Most fluid bearings require little or no maintenance, and have almost unlimited life. Conventional rolling-element bearings usually have shorter life and require regular maintenance. Pumped hydrostatic and aerostatic (gas) bearing designs retain low friction down to zero speed and need not suffer start/stop wear, provided the pump does not fail.

Fluid bearings generally have very low friction—far better than mechanical bearings. One source of friction in a fluid bearing is the viscosity of the fluid. Hydrostatic gas bearings are among the lowest friction bearings. However, lower fluid viscosity also typically means fluid leaks faster from the bearing surfaces, thus requiring increased power for pumps or seals.

When a roller or ball is heavily loaded, fluid bearings have clearances that change less under load (are "stiffer") than mechanical bearings. It might seem that bearing stiffness, as with maximum design load, would be a simple function of average fluid pressure and the bearing surface area. In practice, when bearing surfaces are pressed together, the fluid outflow is constricted. This significantly increases the pressure of the fluid between the bearing faces. As fluid bearing faces can be comparatively larger than rolling surfaces, even small fluid pressure differences cause large restoring forces, maintaining the gap.

However, in lightly loaded bearings, such as disk drives, the typical ball bearing stiffnesses are ~10^7 MN/m. Comparable fluid bearings have stiffness of ~10^6 MN/m. Because of this, some fluid bearings, particularly hydrostatic bearings, are deliberately designed to pre-load the bearing to increase the stiffness.

Fluid bearings often inherently add significant damping. This helps attenuate resonances at the gyroscopic frequencies of journal bearings (sometimes called conical or rocking modes).

It is very difficult to make a mechanical bearing which is atomically smooth and round; and mechanical bearings deform in high-speed operation due to centripetal force. In contrast, fluid bearings self-correct for minor imperfections.

Fluid bearings are typically quieter and smoother (more consistent friction) than rolling-element bearings. For example, hard disks manufactured with fluid bearings have noise ratings for bearings/motors on the order of 20-24 dB, which is a little more than the background noise of a quiet room. Drives based on rolling-element bearings are typically at least 4 dB noisier.

Tilting pad bearings are used as radial bearings for supporting and locating shafts in compressors.

Disadvantages

Overall power consumption is typically higher compared to ball bearings.

Power consumption and stiffness or damping greatly vary with temperature, which complicates the design and operation of a fluid bearing in wide temperature range situations.

Fluid bearings can catastrophically seize under shock situations. Ball bearings deteriorate more gradually and provide acoustic symptoms.

Like cage frequency vibration in a ball bearing, the half frequency whirl is a bearing instability that generates eccentric precession which can lead to poor performance and reduced life.

Fluid leakage; keeping fluid in the bearing can be a challenge.

Oil fluid bearings are impractical in environments where oil leakage can be destructive or where maintenance is not economical.

Fluid bearing "pads" often have to be used in pairs or triples to avoid the bearing from tilting and losing the fluid from one side.

Some fluid bearings

Foil bearings

Foil bearings are a type of fluid dynamic air bearing that was introduced in high speed turbine applications in the 1960s by Garrett AiResearch. They use a gas as the working fluid, usually air and require no external pressurisation system.

Journal bearings

Pressure-oiled journal bearings appear to be plain bearings but are arguably fluid bearings. For example, journal bearings in gasoline (petrol) and diesel engines pump oil at low pressure into a large-gap area of the bearing. As the bearing rotates, oil is carried into the working part of the bearing, where it is compressed, with oil viscosity preventing the oil's escape. As a result, the bearing hydroplanes on a layer of oil, rather than on metal-on-metal contact as it may appear.

This is an example of a fluid bearing which does not use a secondary bearing for start/stop. In this application, a large part of the bearing wear occurs during start-up and shutdown, though in engine use, substantial wear is also caused by hard combustion contaminants that bridge the oil film.

Air bearings

Unlike contact-roller bearings, air bearings utilize a thin film of pressurized air to provide an exceedingly low friction load-bearing interface between surfaces. The two surfaces don't touch. Being non-contact, air bearings avoid the traditional bearing-related problems of friction, wear, particulates, and lubricant handling, and offer distinct advantages in precision positioning, such as lacking backlash and stiction, as well as in high-speed applications.

The fluid film of the bearing is air that flows through the bearing itself to the bearing surface. The design of the air bearing is such that, although the air constantly escapes from the bearing gap, the pressure between the faces of the bearing is enough to support the working loads.

Examples

Air hockey is a game based on an aerostatic bearing which suspends the puck and player's paddles to provide low friction and thus fast motion. The bearing uses a flat plane with periodic orifices which deliver air just over ambient pressure. The puck and paddles rest on air.

Another example of a fluid bearing is ice skating. Ice skates form a hydrodynamic fluid bearing where the skate and ice are separated by a layer of water caused by entropy.

Michell/Kingsbury tilting-pad fluid bearings

Michell/Kingsbury fluid dynamic tilting-pad bearings were invented independently and almost simultaneously by both British-born Australian, Anthony George Maldon Michell and American tribologist Albert Kingsbury. Michell's patent was granted in 1905, while Kingsbury's first patent attempt was 1907. Kingsbury's patent was eventually granted in 1911 after he demonstrated that he had been working on the concept for many years. As stated by Sydney Walker, a long-time employee of Michell's, the granting of Kingsbury's patent was "a blow which Michell found hard to accept".

The bearing has sectional *shoes*, or *pads* on pivots. When the bearing is in operation, the rotating part of the bearing carries fresh oil in to the pad area through viscous drag. Fluid pressure causes the pad to tilt slightly, creating a narrow constriction between the shoe and the other bearing surface. A wedge of pressurised fluid builds behind this constriction, separating the moving parts. The tilt of the pad adaptively changes with bearing load and speed. Various design details ensure continued replenishment of the oil to avoid overheating and pad damage.

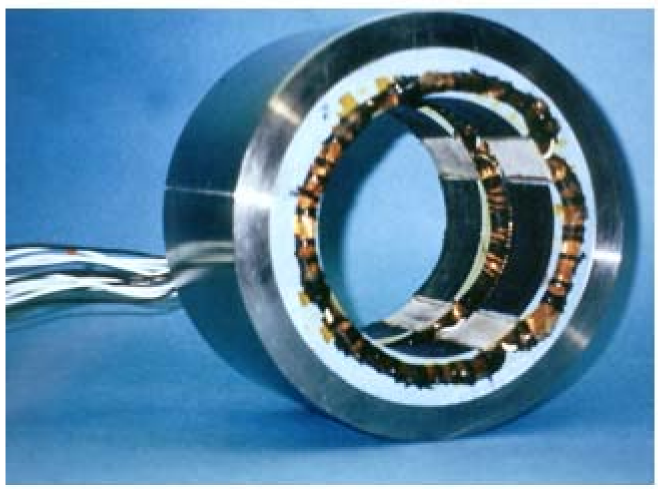
Michell/Kingsbury fluid bearings are used in a wider variety of heavy-duty rotating equipment, including in hydroelectric plants to support turbines and generators weighing hundreds of tons. They are also used in very heavy machinery, such as marine propeller shafts.

The first tilting pad bearing in service was probably that built under A.G.M. Michell's guidance by George Weymoth (Pty) Ltd, for a centrifugal pump at Cohuna on the Murray River, Victoria, Australia, in 1907, just two years after Michell had published and patented his three-dimensional solution to Reynold's equation. By 1913, the great merits of the tilting-pad bearing had been recognised for marine applications. The first English ship to be fitted out with the bearing was the cross-channel steamboat the *Paris*, but many naval vessels were similarly equipped during the First World War. The practical results were spectacular - the troublesome thrust block became dramatically smaller and lighter, significantly more efficient, and remarkably free from maintenance troubles. It was estimated that the Royal Navy saved coal to a value of £500,000 in 1918 alone as a result of fitting Michell's tilting-pad bearings.

According to the ASME, the first Michell/Kingsbury fluid bearing in the USA was installed in the Holtwood Hydroelectric Power Plant (on the Susquehanna River, near Lancaster, Pennsylvania, USA) in 1912. The 2.25-tonne bearing supports a water turbine and electric generator with a rotating mass of about 165 tonnes and water turbine pressure adding another 40 tonnes. The bearing has been in nearly continuous service since 1912, with no parts replaced. The ASME reported it was still in service as of 2000. As of 2002, the manufacturer estimated the bearings at Holtwood should have a maintenance-free life of about 1,300 years.

Chapter 6

Magnetic Bearing



A magnetic bearing

A **magnetic bearing** is a bearing which supports a load using magnetic levitation. Magnetic bearings support moving machinery without physical contact, for example, they can levitate a rotating shaft and permit relative motion with very low friction and no mechanical wear. They are in service in such industrial applications as electric power generation, petroleum refining, machine tool operation and natural gas pipelines. They are also used in the Zippe-type centrifuge used for uranium enrichment. Magnetic bearings are used in turbomolecular pumps where oil-lubricated bearings are a source of contamination. Magnetic bearings support the highest speeds of any kind of bearing; they have no known maximum relative speed.

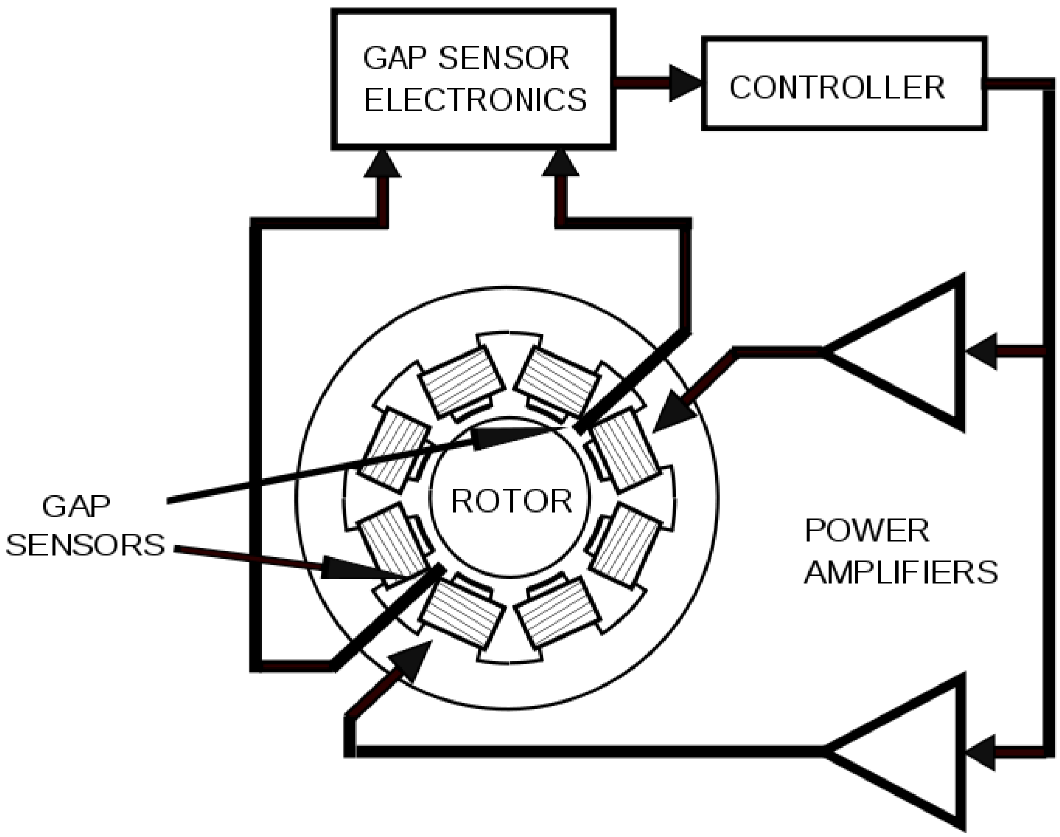
Description

It is difficult to build a magnetic bearing using permanent magnets due to the limitations described by Earnshaw's theorem, and techniques using diamagnetic materials are relatively undeveloped. As a result, most magnetic bearings require continuous power input and an active control system to hold the load stable. Many bearings can use permanent magnets to carry the static load, and then only use power when the levitated object deviates from its optimum position. Magnetic bearings also typically require some kind of back-up bearing in case of power or control system failure and during initial start-up conditions.

Two sorts of instabilities are very typically present with magnetic bearings. Firstly attractive magnets give an unstable static force, decreasing with greater distance, and increasing at close distances. Secondly since magnetism is a conservative force, in and of itself it gives little if any damping, and oscillations may cause loss of successful suspension if any driving forces are present, which they very typically are.

With the use of an induction-based levitation system present in maglev technologies such as the Inductrack system, magnetic bearings could do away with complex control systems by using Halbach Arrays and simple closed loop coils. These systems gain in simplicity, but are less advantageous when it comes to eddy current losses. For rotating systems it is possible to use homopolar magnet designs instead of multipole halbach structures, which reduces losses considerably. An example of this - that has solved the Earnshaws theorem - is the homopolar electrodynamic bearings invented by Dr Torbjörn Lembke.

Active magnetic bearing



Basic operation for a single axis

An active magnetic bearing (AMB) works on the principle of electromagnetic suspension and consists of an electromagnet assembly, a set of power amplifiers which supply current to the electromagnets, a controller, and gap sensors with associated electronics to provide the feedback required to control the position of the rotor within the gap. These elements are shown in the diagram. The power amplifiers supply equal bias current to two pairs of electromagnets on opposite sides of a rotor. This constant tug-of-war is mediated by the controller which offsets the bias current by equal but opposite perturbations of current as the rotor deviates by a small amount from its center position.

The gap sensors are usually inductive in nature and sense in a differential mode. The power amplifiers in a modern commercial application are solid state devices which operate in a pulse width modulation (PWM) configuration. The controller is usually a microprocessor or DSP.

Active bearings have several advantages, they do not suffer from wear, they have low friction, and they can often accommodate irregularities in the mass distribution automatically, allowing it to spin around its centre of mass with very low vibration.

History

The evolution of active magnetic bearings may be traced through the patents issued in this field. The table below lists several early patents for active magnetic bearings. Earlier patents for magnetic suspensions can be found but are excluded here because they consist of assemblies of permanent magnets of problematic stability per Earnshaw's Theorem.

Early active magnetic bearing patents were assigned to Jesse Beams at the University of Virginia during World War II and are concerned with ultracentrifuges for purification of the isotopes of various elements for the manufacture of the first nuclear bombs, but the technology did not mature until the advances of solid-state electronics and modern computer-based control technology with the work of Habermann and Schweitzer. Extensive modern work in magnetic bearings has continued at the University of Virginia in the Rotating Machinery and Controls Industrial Research Program. The first international symposium for active magnetic bearing technology was held in 1988 with the founding of the International Society of Magnetic Bearings by Prof. Schweitzer (ETHZ), Prof. Allaire (University of Virginia), and Prof. Okada (Ibaraki University).

In 1987 further improved AMB designs were created in Australia by E.Croot but these designs were not manufactured due to expensive costs of production. However, some of those designs have since been used by Japanese electronics companies, they remain a specialty item: where extremely high RPM is required.

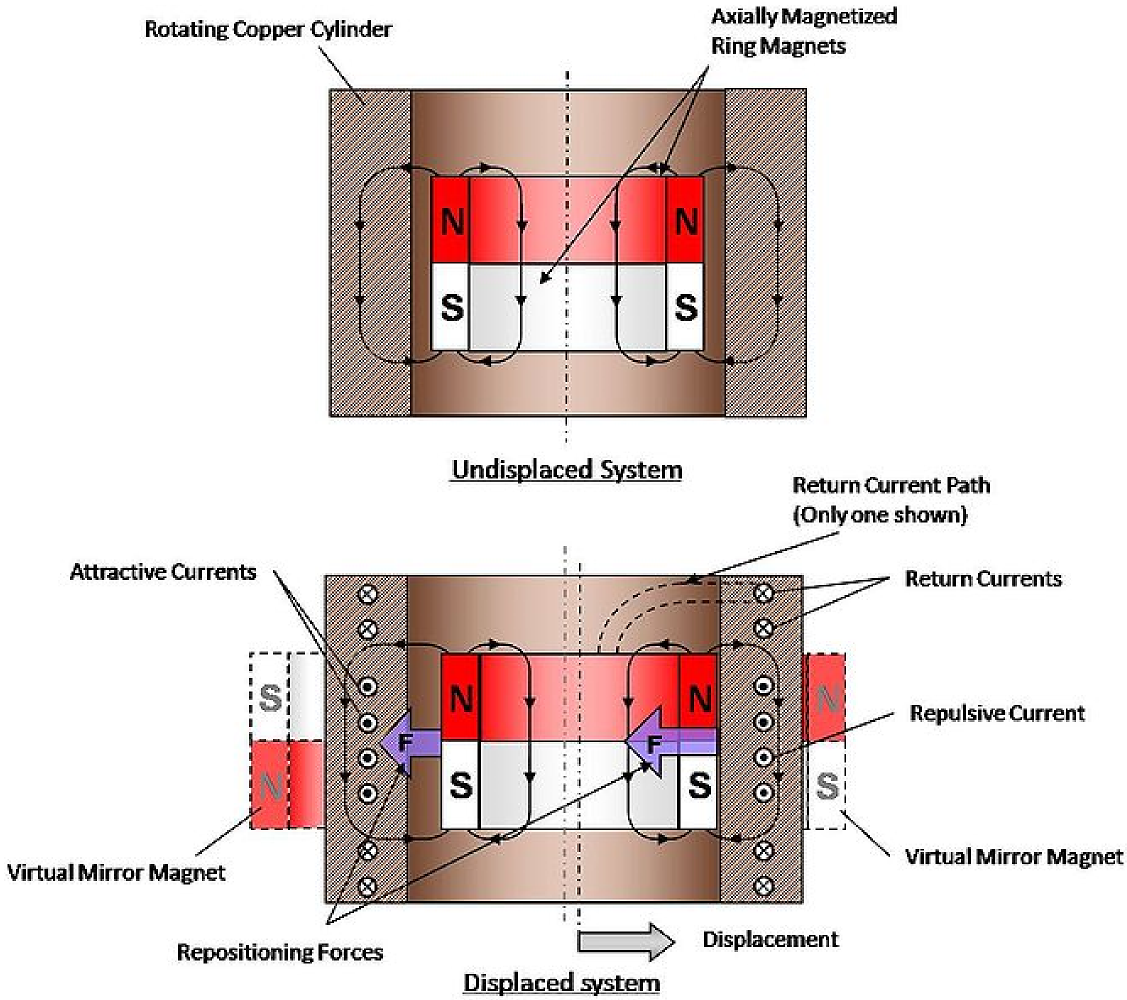
Since then there have been nine succeeding symposia. Kasarda reviews the history of AMB in depth. She notes that the first commercial application of AMB’s was with turbomachinery. The AMB allowed the elimination of oil reservoirs on compressors for the NOVA Gas Transmission Ltd. (NGTL) gas pipelines in Alberta, Canada. This reduced the fire hazard allowing a substantial reduction in insurance costs. The success of these magnetic bearing installations led NGTL to pioneer the research and development of a digital magnetic bearing control system as a replacement for the analog control systems supplied by the American company Magnetic Bearings Inc. (MBI). In 1992, NGTL's magnetic bearing research group formed the company Revolve Technologies Inc . to commercialize the digital magnetic bearing technology. This firm was later purchased by SKF of Sweden. The French company S2M, founded in 1976, was the first to commercially market AMB’s. Extensive research on magnetic bearings continues at the University of Virginia in the Rotating Machinery and Controls Industrial Research Program .

Starting from 1996 the Dutch oil and gas company NAM installed over a period of 10 years 20 large E-motor driven (with variable speed drive) gas compressors of 23 MW fully equipped with AMB's on both the E-motor and the compressor. These compressors are used in the Groningen gas field to deplete the remaining gas from this large gas field and to increase the field capacity. The motor - compressor design is done by Siemens and the AMB are delivered by Waukesha (owned by Dover). (Originally these bearings were designed by Glacier, this company is later on taken over by Federal Mogul and now part of Waukesha) By using AMB's and a direct drive between motor and compressor (so no the gearbox in between) and applying dry gas seals a full so called dry-dry system (=fully oil free) has been installed. A few of the main advantages by applying AMB's in the driver as well as in the compressor (compared to the traditional configuration with a gearbox, plain bearings and a gasturbine-driver) is a relative simple system with a very wide operating envelope, high efficiencies (particularly at partial load) and also, as done in the Groningen field, to install the full installation outdoors (no large compressor building needed.

Early U.S. Patents in AMB

Inventor(s)	Year	Patent No.	Invention Title
Beams, Holmes	1941	2,256,937	Suspension of Rotatable Bodies
Beams	1954	2,691,306	Magnetically Supported Rotating Bodies
Beams	1962	3,041,482	Apparatus for Rotating Freely Suspended Bodies
Beams	1965	3,196,694	Magnetic Suspension System
Wolf	1967	3,316,032	Poly-Phase Magnetic Suspension Transformer
Lyman	1971	3,565,495	Magnetic Suspension Apparatus
Habermann	1973	3,731,984	Magnetic Bearing Block Device for Supporting a Vertical Shaft Adapted for Rotating at High Speed
Habermann, Loyen, Joli, Aubert	1974	3,787,100	Devices Including Rotating Members Supported by Magnetic Bearings
Habermann, Brunet	1977	4,012,083	Magnetic Bearings
Habermann, Brunet, LeClére	1978	4,114,960	Radial Displacement Detector Device for a Magnetic Bearings

Electrodynamic bearing



An axial homopolar electrodynamic bearing

Electrodynamic bearings (EDB) are a novel type of bearing that is a passive magnetic technology. EDBs do not require any control electronics to operate. They work by the electrical currents generated by motion causing a restoring force.

Applications

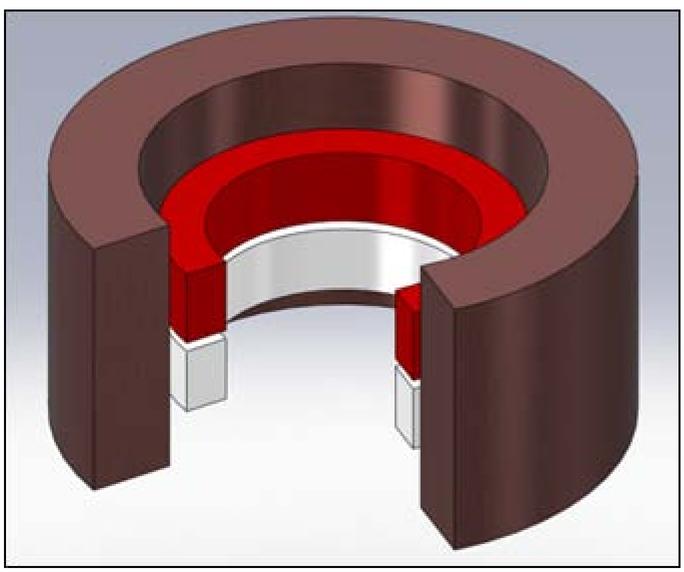
Magnetic bearing advantages include very low and predictable friction, ability to run without lubrication and in a vacuum. Magnetic bearings are increasingly used in industrial machines such as compressors, turbines, pumps, motors and generators. Magnetic bearings are commonly used in watt-hour meters by electric utilities to measure home power consumption. Magnetic bearings are also used in high-precision instruments and to support equipment in a vacuum, for example in flywheel energy storage systems. A flywheel in a vacuum has very low windage losses, but conventional bearings usually fail quickly in a vacuum due to poor lubrication. Magnetic bearings are also used to support maglev trains in order to get low noise and smooth ride by eliminating physical contact surfaces. Disadvantages include high cost, and relatively large size.

A new application of magnetic bearings is their use in artificial hearts. The use of magnetic suspension in ventricular assist devices was pioneered by Prof. Paul Allaire and Prof. Houston Wood at the University of Virginia culminating in the first magnetically suspended ventricular assist centrifugal pump (VAD) in 1999.

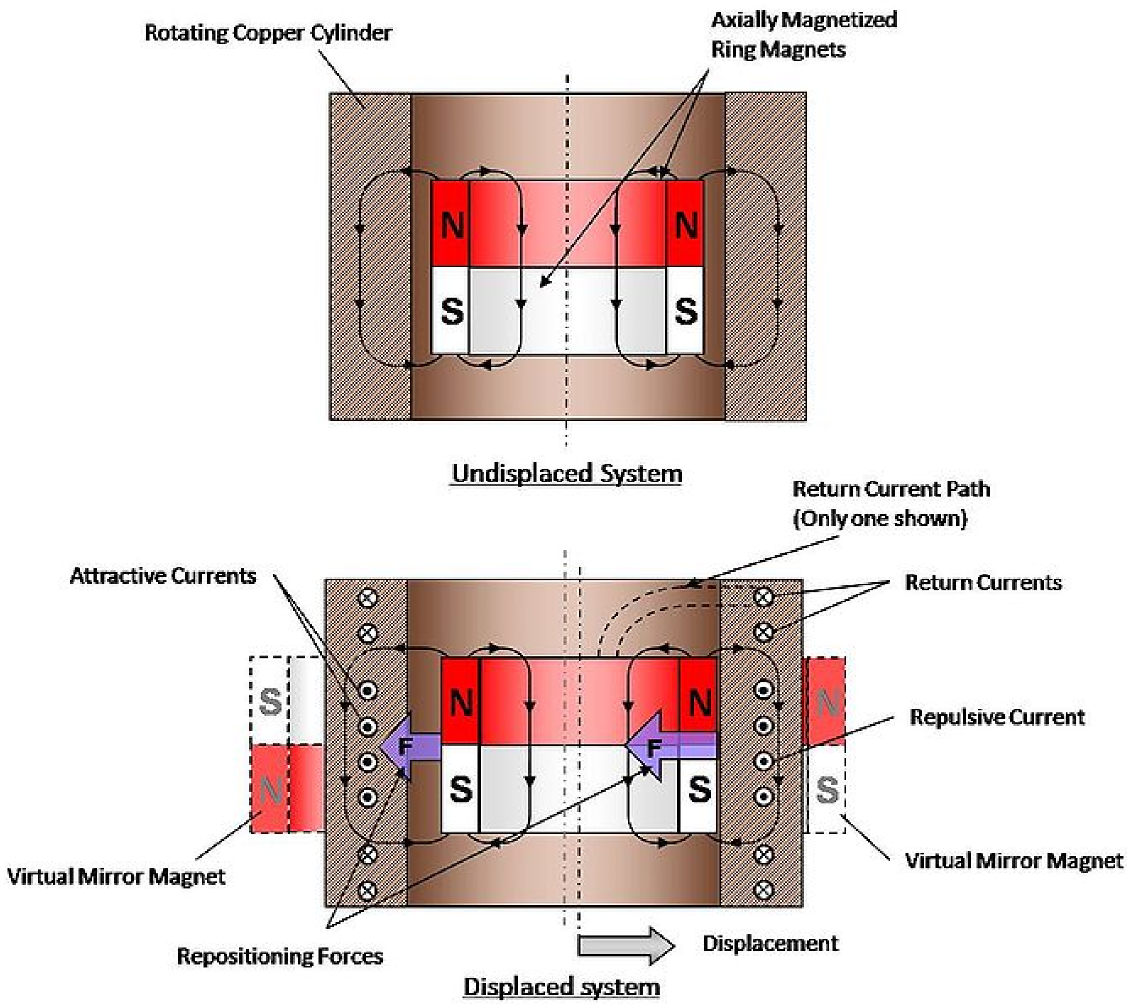
Chapter 7

Electrodynamic Bearing and Foil Bearing

Electrodynamic bearing



3D-image of an axially magnetized ring magnet surrounded by a copper cylinder. The working principle is shown in a 2D-image below.



Magnetic mirroring - the working principle for electrodynamic bearing.

Electrodynamic bearings (EDBs) are novel, promising systems that can be used to realize contactless electrodynamic suspension of rotating shafts. Relative to active magnetic bearings (AMB) the passive nature of the levitation achieved by EDBs allows a simpler, more reliable and cheaper solution, opening the field of application to medium and large-scale production.

The working principle is based on the induction of eddy currents in a rotating conductor. When an electrically conducting material is moving in a magnetic field, a current will be generated in the material that counters the change in the magnetic field (known as Lenz' Law). This generates a current that will result in a magnetic field that is oriented opposite to the one from the magnet. The electrically conducting material is thus acting as a magnetic mirror.

Radial magnetic bearing

Avoiding eddy current losses

Before the mid 1990s the eddy currents damping was problematic, but eddy currents and associated power dissipation can be reduced to very low values. The principle of operation is as follows:

A bearing must (1) support a loading force (for example, the weight of a rotor) and (2) provide a force gradient (a restoring force) to hold the rotor in position. Permanent magnets can support weight (in a conventional way, without eddy currents), and without creating destabilizing force gradients, but the Earnshaw theorem precludes achieving stability by this means. Eddy currents can provide a stabilizing *force gradient* without applying a *force* at the operating position (for example, when a shaft is centered). Creating this force *gradient* does not require eddy currents (which are induced in proportion to shaft offset). In practice, eddy currents, and hence resistive losses, can be reduced to small values in normal operation. Dynamic bearings of this class, using permanent magnets and ordinary, resistive conductors, can support load and apply restoring force while dissipating little power (and in principle, none).

An improved design approach for bearings of this class was described and analyzed by Dr. Torbjörn Lembke in his PhD thesis at the Royal Institute of Technology, KTH, in Stockholm, Sweden.

Linear magnetic bearing

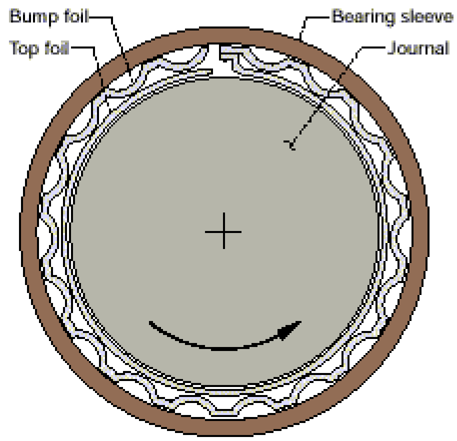
Linear dynamic magnetic bearings also exist. For example inductrack which uses halbach arrays and litz wire loops.

Foil bearing

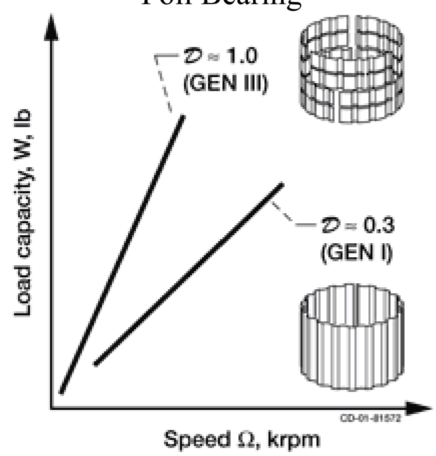
A typical foil-air bearing for the core rotor shaft of an aircraft turbine engine.

Foil or **foil-air bearings** are a type of air bearing. A shaft is supported by a compliant, spring loaded foil journal lining. Once the shaft is spinning quickly enough, the working fluid (usually air) pushes the foil away from the shaft so that there is no more contact. The shaft and foil are separated by the air's high pressure which is generated by the rotation which pulls gas into the bearing via viscosity effects. A high speed of the shaft with respect to the foil is required to initiate the air gap, and once this has been achieved, no wear occurs. Unlike aero or hydrostatic bearings, foil bearings require no external pressurisation system for the working fluid, so the hydrodynamic bearing is self-starting.

Development



Foil Bearing



$$W = \mathcal{D}\,(L \times D)\,(D \times \Omega)$$

Where

W is the maximum steady-state load that can be supported, N, lb
$\mathcal{D}$ is the bearing load capacity coefficient, N/(mm³•krpm) (lb/(in.³•krpm))
L is the bearing axial length, mm, in.
D is the shaft diameter, mm, in.
Ω is the shaft speed in 1000 rpm, krpm

Load capacity against rotation speed, for Gen I and Gen III bearings

Foil bearings were first developed in the late 1950s by AiResearch Mfg. Co. of the Garrett Corporation using independent R&D funds to serve military and space applications. They were first tested for commercial use in United Airlines Boeing 727 and Boeing 737 cooling turbines in the early- and mid-1960s. Garrett AiResearch air cycle machine foil bearings were first installed as original equipment in 1969 in the DC-10's environmental control systems. Garrett AiResearch foil bearings were installed on all U.S. military aircraft to replace existing oil-lubricated rolling-contact bearings. The ability to operate at cryogenic gas temperatures as well as at very high temperatures gave foil bearings many other potential applications.

Current generation foil bearings with advanced coatings have greatly exceeded the limitations of earlier designs. Anti-wear coatings exist that allow over 100,000 start/stop cycles for typical applications. New third generation bearings can hold over 9,000 times their weight, at extremely high speeds.

Applications

Turbomachinery is the most common application because foil bearings operate at high speed. The main advantage of foil bearings is the elimination of the oil systems required by traditional bearing designs. Other advantages are:

- Higher efficiency, due to a lower heat loss to friction; instead of fluid friction, the main source of heat is parasitic drag
- Increased reliability
- Higher speed capability
- Quieter operation
- Wider operating temperature range (40 K to 2500 K)
- High vibration and shock load capacity
- No scheduled maintenance
- No external support system
- Truly oil free where contamination is an issue
- Capable of operating above critical speed

Areas of current research are:

- Higher load capacity
- Improved damping
- Improved coatings

The main disadvantages are:

- Lower capacity than roller or oil bearings
- Wear during start-up and stopping
- High speed required for operation

Chapter 8

Linear-Motion Bearing and Race (Bearing)

Linear-motion bearing

A **linear-motion bearing** or **linear slide** is a bearing designed to provide free motion in one dimension. There are many different types of linear motion bearings and this family of products is generally broken down into two sub-categories: *rolling-element* and *plane*.

Motorized linear slides such as machine slides, XY tables, roller tables and some dovetail slides are bearings moved by drive mechanisms. Not all linear slides are motorized, and non-motorized dovetail slides, ball bearing slides and roller slides provide low-friction linear movement for equipment powered by inertia or by hand. All linear slides provide linear motion based on bearings, whether they are ball bearings, dovetail bearings or linear roller bearings. XY Tables, linear stages, machine slides and other advanced slides use linear motion bearings to provide movement along both X and Y multiple axis.

Rolling-element bearing

A rolling-element bearing is generally composed of a sleeve-like outer ring and several rows of balls retained by cages. The cages were originally machined from solid metal and were quickly replaced by stampings. It features smooth motion, low friction, high rigidity and long life. They are economical, and easy to maintain and replace. Thomson (currently owned by Danaher) is generally given credit for first producing [what is now known as] a linear ball bearing.

- Rolling-element bearings can only run on hardened steel or stainless steel shafting (raceways).
- Rolling-element bearings are more rigid than plane bearings.
- Rolling-element bearings do not handle contamination well and require seals.
- Rolling-element bearings require lubrication.

Rolling-element bearings are manufactured in two forms: ball bearing slides and roller slides.

Ball Bearing Slides

Also called "ball slides", ball bearing slides are the most common type of linear slide. Ball bearing slides offer smooth precision motion along a single-axis linear design, aided by ball bearings housed in the linear base, with self-lubrication properties that increase reliability. Ball bearing slide applications include delicate instrumentation, robotic assembly, cabinetry, high-end appliances and clean room environments, which primarily serve the manufacturing industry but also the furniture, electronics and construction industries. For example, a widely used ball bearing slide in the furniture industry is a ball bearing drawer slide.

Commonly constructed from materials such as aluminum, hardened cold rolled steel and galvanized steel, ball bearing slides consist of two linear rows of ball bearings contained by four rods and located on differing sides of the base, which support the carriage for smooth linear movement along the ball bearings. This low-friction linear movement can be powered by either a drive mechanism, inertia or by hand. Ball bearing slides tend to have a lower load capacity for their size compared to other linear slides because the balls are less resistant to wear and abrasions. In addition, ball bearing slides are limited by the need to fit into housing or drive systems.

Roller Slides

Also known as crossed roller slides, roller slides are non-motorized linear slides that provide low-friction linear movement for equipment powered by inertia or by hand. Roller slides are based on linear roller bearings, which are frequently criss-crossed to provide heavier load capabilities and better movement control. Serving industries such as manufacturing, photonics, medical and telecommunications, roller slides are versatile and can be adjusted to meet numerous applications which typically include clean rooms, vacuum environments, material handling and automation machinery.

Consisting of a stationary linear base and a moving carriage, roller slides work similarly to ball bearing slides, except that the bearings housed within the carriage are cylinder-shaped instead of ball shaped. The rollers crisscross each other at a 90° angle and move between the four semi-flat and parallel rods that surround the rollers. The rollers are between "V" grooved bearing races, one being on the top carriage and the other on the base. The travel of the carriage ends when it meets the end cap, a limiting component. Typically, carriages are constructed from aluminum and the rods and rollers are constructed from steel, while the end caps are constructed from stainless steel.

Although roller slides are not self-cleaning, they are suitable for environments with low levels of airborne contaminants such as dirt and dust. As one of the more expensive types of linear slides, roller slides are capable of providing linear motion on more than one axis through stackable slides and double carriages. Roller slides offers line contact versus point contact as with ball bearings, creating a broader contact surface due to the consistency of contact between the carriage and the base and resulting in less erosion.

Plain bearing

Plain bearings are very similar in design to rolling-element bearings, except they slide without the use of ball bearings.

- Plain bearings can run on hardened steel or stainless steel shafting (raceways), *or* can be run on hard-anodized aluminum or soft steel or aluminum. The specific type of polymer/fluoro-polymer will determine what hardness is allowed.
- Plain bearings are less rigid than rolling-element bearings.
- Plain bearings handle contamination well and often do *not* need seals/scrapers.
- Plain bearings generally handle a wider temperature range than rolling-element bearings
- Plain bearings (plastic versions) do not require oil or lubrication (often it can be used to increase performance characteristics)

Dovetail slides

Dovetail slides, or dovetail way slides are typically constructed from cast iron, but can also be constructed from hard-coat aluminum, acetal or stainless steel. Like any bearing, a dovetail slide is composed of a stationary linear base and a moving carriage. a Dovetail carriage has a v-shaped, or dovetail-shaped protruding channel which locks into the linear base's correspondingly shaped groove. Once the dovetail carriage is fitted into its base's channel, the carriage is locked into the channel's linear axis and allows free linear movement. When a platform is attached to the carriage of a dovetail slide, a dovetail table is created, offering extended load carrying capabilities.

Since dovetail slides have such a large surface contact area, a greater force is required to move the saddle than other linear slides, which results in slower acceleration rates. Additionally, dovetail slides have difficulties with high-friction but are advantageous when it comes to load capacity, affordability and durability. Capable of long travel, dovetail slides are more resistant to shock than other bearings, and they are mostly immune to chemical, dust and dirt contamination. Dovetail slides can be motorized, mechanical or electromechanical. Electric dovetail slides are driven by a number of different devices, such as ball screws, belts and cables, which are powered by functional motors such as stepper motors, linear motors and handwheels. Dovetail slides are direct contact systems, making them fitting for heavy load applications including CNC machines, shuttle devices, special machines and work holding devices. Mainly used in the manufacturing and laboratory science industries, dovetail slides are not ideal for high-precision applications.

Race (bearing)

The rolling-elements of a rolling-element bearing ride on **races**. The large race that goes into a bore is called the *outer race*, and the small race that the shaft rides in is called the *inner race*.

Manufacture

The outer diameter (OD) of the races are often centerless ground using the throughfeed process. Centerless grinding can achieve a very high degree of accuracy, especially when done in stages. These stages are: rough, semi-finish and finish. Each grinding stage is designed to remove enough stock material from the casing so that the next stage does not encounter any problems such as burning or surface chatter, the finish stage achieves the final dimension.

Each grinding wheel at all of the aforementioned stages has a varying degree of abrasive quality (finish being the finest grade) to achieve the appropriate stock removal for the next stage and final surface finish required.

Bearing casings are introduced to the grinding action via means of a transfer from the delivery system to a pair of infeed rollers, these infeed rollers are tapered to a certain angle so that the casings are driven forward until the regulating wheel and grinding wheel catch them and slow them to their grinding speed which can be altered by speed control of the regulating wheel. The casings are constantly rotating and are fed into the grinding area to prevent separation which can cause finish/size problems or even a "bump" that can potentially crack or destroy casings and will damage the grinding and regulating wheels. Whilst grinding, the bearing cases run through the grinding stages in one long tube of casings that is showered with a cutting fluid. The 'tube' rests on a hardened steel blade with an angled, highly ground surface held on a horizontal plane between the grinding wheel and regulating wheel, often named a Work Rest Blade, the tube causes wear on the working surface of the blade so it must be reground at regular intervals. The height of the work rest blade perfectly aligns the bearing casing with the horizontal centreline of the grinding wheel creating a flawless ground finish, the work rest blade height can be altered using packing bars placed underneath the blade, height adjustments must be made depending on the diameter of the casings being ground. Each casing exits the grinding zone onto a high speed conveyor that delivers them to whatever storage and/or inspection arrangement a manufacturer may have, inspection is also carried out by the operator of the centreless line, by checking finish appearance, diameter, squareness and roundness by use of a dial test indicator in varying configurations, size allowances are permitted but are extremely tight depending on the customers requirements and can vary plus or minus within micrometres of finish diameter, Sizes can be adjusted on all grinding stages via a compensation button which can be pushed to remove extra material in varying micrometre units, the grinding wheel can move away at the same compensation to make the casings bigger if so required if the casing size moves from the operators target, and as the grinding wheel wears. Because a centerless grinding line has typically three grinding machines the operator must be in complete control and must prevent blockages in transfers, grinding exits and packing areas, also size and quality must constantly be checked, so the operator is always alert while operating the line and checking for problems and quality issues. Safety features include an emergency stop button which immediately moves the grinding wheel away from the ground rings on its revolutionary axis, because of the wheel's momentum, it cannot be stopped but the power is cut and the wheel slows naturally, it cannot be reactivated until the emergency stop is reset.

Chapter 9

Spherical Bearing and Thrust Bearing

Spherical bearing

A **spherical bearing** is a bearing that permits angular rotation about a central point in two orthogonal directions (usually within a specified angular limit based on the bearing geometry). Typically these bearings support a rotating shaft in the [bore] of the inner ring that must move not only rotationally, but also at an angle.

Construction

Construction of spherical bearings can be hydrostatic or strictly mechanical. A spherical bearing by itself can consist of an outer ring and an inner ring and a locking feature that makes the inner ring captive within the outer ring in the axial direction only. The outer surface of the inner ring and the inner surface of the outer ring are collectively considered the raceway and they slide against each other, either with a lubricant or a maintenance-free polytetrafluoroethylene (PTFE) based liner. Some spherical bearings incorporate a rolling element such as a race of ball-bearings, allowing lower friction.

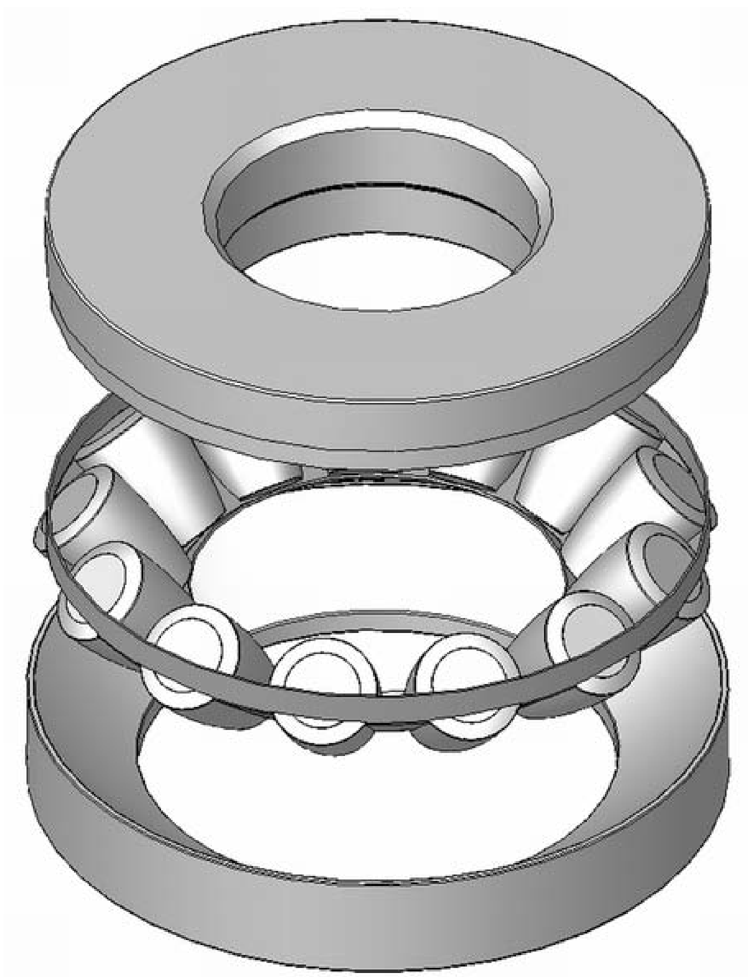
History

The Swede Sven Wingquist (1876–1953) invented the spherical bearing in 1907. He founded a global company, AB Svenska Kullagerfabriken, still the world's leading producer of industrial bearings.

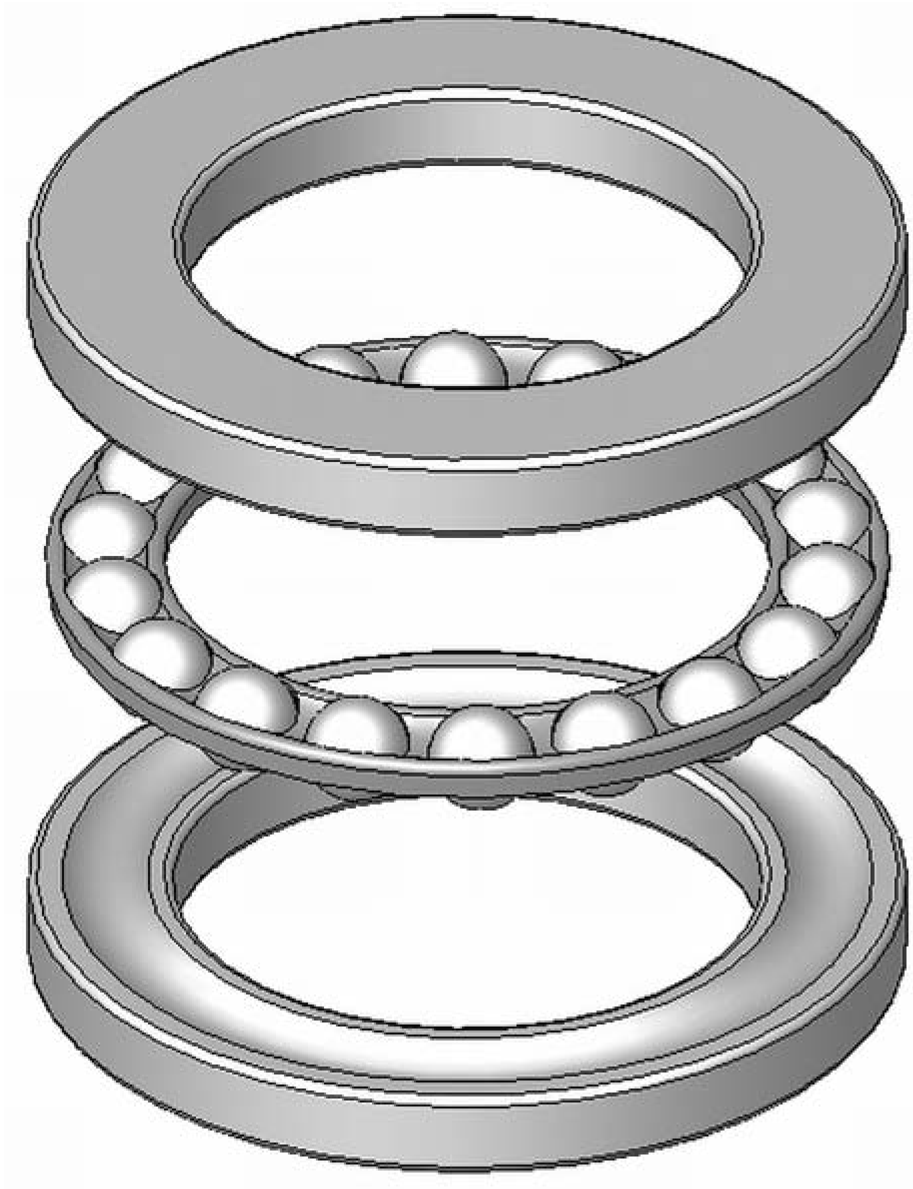
Application

Spherical bearings are used in countless applications, wherever rotational motion must be allowed to change the alignment of its rotation axis. A prime example is a tie rod on a vehicle suspension. The mechanics of the suspension allow the axle to move up and down, but the linkages are designed to control that motion in one direction only and they must allow motion in the other directions. Spherical bearings have been used in car suspensions, driveshafts, heavy machinery, sewing machines, and many other applications.

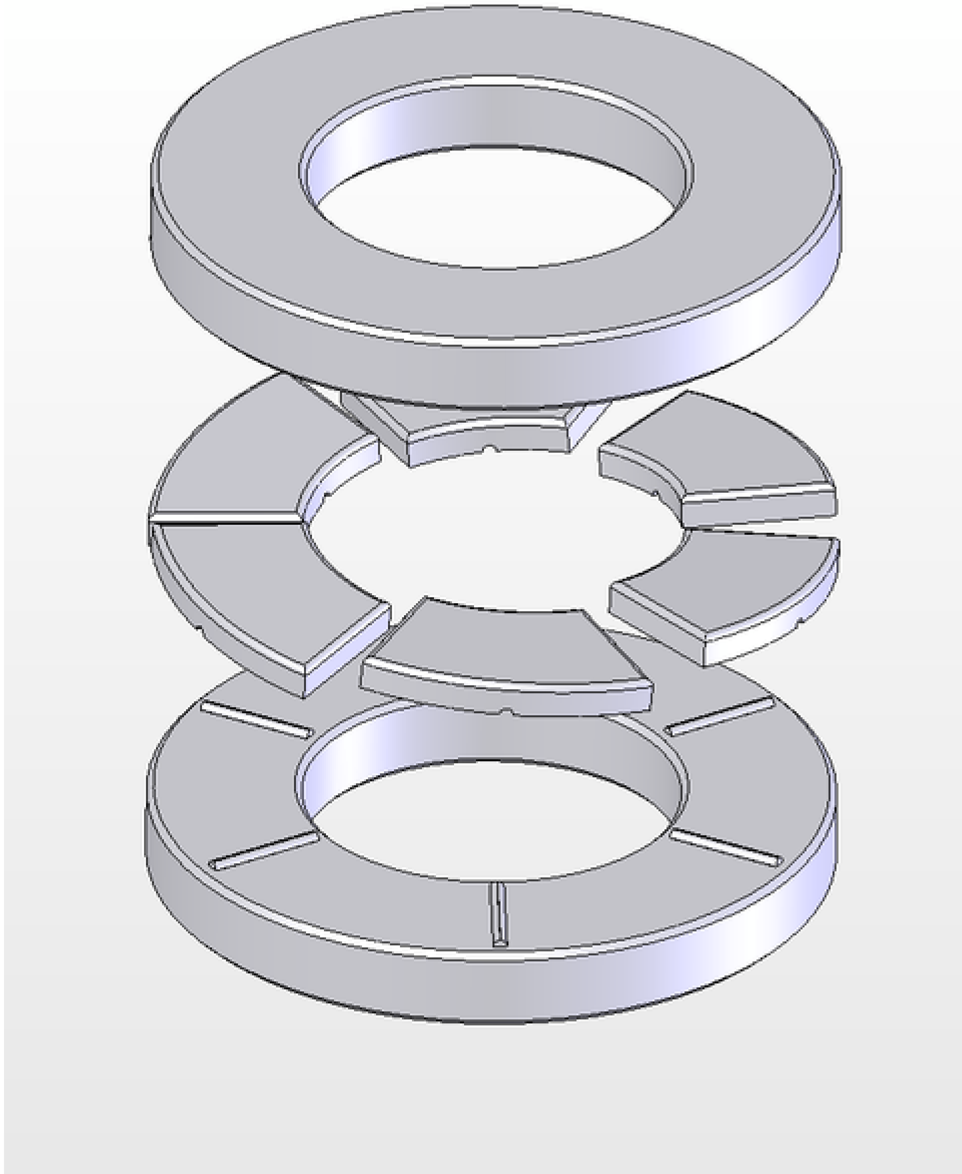
Thrust bearing



A self-aligning roller thrust bearing



A thrust ball bearing



A fluid film thrust bearing

A **thrust bearing** is a particular type of rotary bearing. Like other rotary bearings they permit rotation between parts, but they are designed to support a high axial load while doing this.

Thrust bearings come in several varieties.

- *Ball thrust bearings*, composed of ball bearings supported in a ring, can be used in low thrust applications where there is little radial load.
- *Roller thrust bearing*s consist of small cylindrical rollers arranged flat with their axes pointing to the axis of the bearing. They give very good carrying capacity

and are cheap, but tend to wear due to the differences in radial speed and friction is higher than with ball bearings.

- *Tapered roller bearings* consist of small tapered rollers arranged so that their axes all converge at a point on the axis of the bearing. The length of the roller and the diameter of the wide and the narrow ends and the angle of rollers need to be carefully calculated to provide the correct taper so that each end of the roller rolls smoothly on the bearing face without skidding. These are the type most commonly used in automotive applications (to support the wheels of a motor car for example), where they are used in pairs to accommodate axial thrust in either direction, as well as radial loads. They can support rather larger thrust loads than the ball type due to the larger contact area, but are more expensive to manufacture.
- *Fluid bearings*, where the axial thrust is supported on a thin layer of pressurized liquid—these give low drag.
- *Magnetic bearings*, where the axial thrust is supported on a magnetic field. This is used where very high speeds or very low drag is needed, for example the Zippe-type centrifuge.

They are commonly used in automotive, marine, and aerospace applications.

Thrust bearings are used in cars because the forward gears in modern car gearboxes use helical gears which, while aiding in smoothness and noise reduction, cause axial forces that need to be dealt with. The double helical or herringbone gear balances the thrust caused by normal helical gears.

One specific thrust bearing in an automobile is the clutch "throw out" bearing, sometimes called the *clutch release bearing*.

Fluid-film thrust bearings were invented by Australian engineer George Michell (pronounced Mitchell) who patented his invention in 1905. Michell bearings contain a number of sector-shaped pads, arranged in a circle around the shaft, and which are free to pivot. These create wedge-shaped regions of oil inside the bearing between the pads and a rotating disk, which support the applied thrust and eliminate metal-on-metal contact.

Michell's invention was notably applied to the thrust block in ships. The small size (one-tenth the size of old bearing designs), low friction and long life of Michell's invention made possible the development of more powerful engines and propellers. They were used extensively in ships built during World War I, and have become the standard bearing used on turbine shafts in ships and power plants worldwide.

Thrust ball bearing

Thrust ball bearings consist of two precision chrome steel washers (ring) and a ball complement spaced by bronze retainer. They can be supplied with or without radius ball grooves in the rings. Thrust bearings are used under purely axial loads.

Chapter 10

Ball (Bearing), Ball Joint and Ball Spline

Ball (bearing)

Bearing **balls** are special highly spherical and smooth balls, most commonly used in ball bearings. The balls come in many different *grades*, as defined by the American Bearing Manufacturers Association (ABMA), which defines the precision of the balls. They are manufactured in specially designed machines for the job.

In 2008, the United States produced 5,778 million balls.

Grade

Bearing balls are manufactured to a specific grade, which defines its geometric tolerances. The grades range from 2000 to 3, where the smaller the number the higher the precision. Grades are written "GXXXX", i.e. grade 100 would be "G100". The grades are divided into two categories: *semi-precision* and *precision*. Grades 100 and greater are semi-precision balls and lower than that are precision balls.

The specification defines three parameters: surface integrity, size, and sphericity. The surface integrity refers to surface smoothness, hardness, and lack of defects, such as flats, pits, soft spots, and cuts. The surface smoothness is measured in two ways. surface roughness and waviness.

Size refers to how tight the tolerances are on the size, as measured by two parallel plates in contact with the ball surface. The starting size is the *nominal ball diameter*, which is the nominal, or theoretical, ball diameter. The ball size is then determined by measuring the *ball diameter variation*, which is the difference between the largest and smallest diameter measurement. For a given lot there is a *lot diameter variation*, which is the difference between the mean diameter of the largest ball and the smallest ball of the lot.

Sphericity, or *deviation from spherical form*, refers to how much the ball deviates from a true spherical form (out of roundness). This is measured by rotating a ball against a linear transducer with a gauge force of less than 4 grams (0.14 oz). The resulting polar graph is then circumscribed with the smallest circle possible and the difference between this circumscribed circle and the nominal ball diameter is the variation.

Grade tolerances for inch sizes

Grade	Size range [in]	Sphericity [in]	Lot diameter variation [in]	Nominal ball diameter tolerance [in]	Maximum surface roughness (Ra) [µin]
3	0.006–2	0.000003	0.000003	±0.00003	0.5
5	0.006–6	0.000005	0.000005	±0.00005	0.8
10	0.006–10	0.00001	0.00001	±0.00005	1.0
25	0.006–10	0.000025	0.000025	±0.0001	2.0
50	0.006–10	0.00005	0.00005	±0.0002	3.0
100	0.006–10	0.0001	0.0001	±0.0005	5.0
200	0.006–10	0.0002	0.0002	±0.001	8.0
1000	0.006–10	0.001	0.001	±0.005	

Grade tolerances for metric sizes

Grade	Sphericity [mm]	Lot diameter variation [mm]	Nominal ball diameter tolerance [mm]	Maximum surface roughness (Ra) [mm]
3	0.00008	0.00008	±0.0008	0.012
5	0.00013	0.00013	±0.0013	0.02
10	0.00025	0.00025	±0.0013	0.025
25	0.0006	0.0006	±0.0025	0.051
50	0.0012	0.0012	±0.0051	0.076
100	0.0025	0.0025	±0.0381	0.127
200	0.005	0.005	±0.025	0.203
1000	0.025	0.025	±0.127	

Manufacture

The manufacture of bearing balls depends on the type of material the balls are being made from.

Metal

Metal balls start as a wire. The wire is sheared to give a pellet with a length approximately the size of the desired ball outer diameter (OD). This pellet is then headed into a rough spherical shape. Next, the balls are then fed into a machine that de-flashes them. The machine does this by feeding the balls between two heavy cast iron or hardened steel plates, called *rill plates*. One of the plates is held stationary while the other rotates. The top plate has a section an opening to allow balls to enter and exit the rill plates. These plates have fine circumferential grooves that the balls track in. The balls are run through the machine long enough so that each ball passes through multiple of these grooves, which ensures each ball is the same size, even if a particular groove is out of specification. The controllable machine variables are the amount of pressure applied, the speed of the plates, and how long the balls are left in the machine.

During the operation coolant is pumped between the rill plates because the high pressure between the plates and friction create excess heat. The high pressure applied to the balls also induces cold working, which strengthens the balls.

Sometimes the balls are then run through a *soft grinding* process afterward to improve precision. This is done in the same type of machine, but the rill plates are replaced with grinding stones.

If the balls are steel they are then heat treated. After heat treatment they are descaled to remove any residue or by-products.

The balls are then *hard ground*. They are ground in the same type of machine as used before, but either an abrasive is introduced into coolant or the rotating plate is replaced with a very hard fine-grain grinding wheel. This step can get the balls within ±0.0001 in (0.0025 mm). If the balls need more precision then they are lapped, again in the same type of machine. However, this time the rill plates are made of a softer material, usually cast iron, less pressure is applied, the plate is rotated slowly. This step is what gives bearing balls their shiny appearance and can bring the balls between grades 10 and 48.

If even more precision is needed then proprietary chemical and mechanical processes are usually used.

The inspection of bearing balls was one of the case studies in Frederick Winslow Taylor's classic *Principles of Scientific Management*.

Plastic

Plastic bearing balls are made in the same manner as described above.

Ceramic

Ceramic bearing balls are made of sintered materials that are then ground to size and shape as above. Common materials include: silicon nitride and zirconium oxide.

Materials

Common materials include carbon steel, stainless steel, chrome steel, brass, aluminium, tungsten carbide, platinum, gold, titanium, plastic. Other less common materials include copper, monel, k-monel, lead, silver, glass, and niobium.

Material comparison for common bearing balls

Material	UNS 52100	Stainless steel 440C	M50	BG-42	REX-20	440NDUR	Haynes 25	S13N4	BECU	455	C276
Hardness [HRC]	60	58	62	66	60	50	70	40	50	40	
Temperature limit [°F]	300	300	400	600	300	1200	1500	400	500	1000	
Corrosion resistance	1	3	1	2	1	4	5	5	1	4	5
Cost	1	1	1	2	3	1	5	5	3	2	4
Availability	1	1	2	2	2	4	5	3	3	2	4
Magnetic	Magnetic	Magnetic	Magnetic	Magnetic	Magnetic	Magnetic	Non-magnetic	Non-magnetic	Non-magnetic	Magnetic	Magnetic
Size limit	None	None	None	None	None	None	1.5 in (38 mm)		None	None	5 in (130 mm)
Relative load capacity	3	2	4	4	5	3	1	5	1	1	1
Relative fatigue life	3	2	4	4	5	3	1	5	1	1	1

Atypical uses

One interesting atypical use for bearing balls is at San Francisco International Airport. The building is supported by 267 columns, each of which rests on a steel ball with a diameter of 5 feet (1.5 m). The ball sits in a concave foundation. If an earthquake occurs, the ground can move up to 20 inches (0.51 m) in any direction, as the columns roll on their bases. This is an effective way to separate the building from the movement of the ground. After the earthquake has ended, the columns are re-centered on their bases by the force of gravity.

Ball joint

A VW ball joint

In an automobile, **ball joints** are spherical bearings that connect the control arms to the steering knuckles. More specifically, a ball joint is a steel bearing stud and socket enclosed in a steel casing. The bearing stud is tapered and threaded. It fits into a tapered hole in the steering knuckle. A protective encasing prevents dirt from getting into the joint assembly. Motion control ball joints tend to be retained with an internal spring, which helps to prevent vibration problems in the linkage. Commonly found in automotive throttle linkages, throttle body set ups, these are also widely used on construction equipment, the end of gas springs and in children's toys.

Theory

A ball joint is used for allowing three rotations. It fixes the three possible translations (X,Y,Z).

Purpose

Ball joints are the pivot between the wheels and the suspension of an automobile. Ball joints play a critical role in the safe operation of an automobile's steering and suspension. Ball joints can also be found in most linkage systems for motion control applications, and should not be confused with spherical rod end bearings, which are a different design.

Maintenance

Sealed ball joints do not require lubrication as they are "lubed for life" but standard ball joints must be lubed from time to time. It's best to inspect standard ball joints once a year. Generally speaking, standard ball joints will outlive sealed ones because eventually the seal will break, causing the joint to dry out and rust. While there is no exact lifespan that can be put on a sealed ball joint, they can fail as early as 80,000 miles. Signs of a failing ball joint start with a clicking or snapping sound when the wheel is turned and eventually turns into a squeaking sound at the end of a stop, when the gas pedal is used and/or also when hitting bumps. Another symptom could be 'thud' noises coming from front suspension when going over bumps.

If a ball joint fails, the results can be dangerous as the wheel's angle becomes unconstrained, causing loss of control. Because the tire will be at an unintended angle, the vehicle will come to an abrupt halt damaging the tires. Also, during failure, debris can damage other parts of the vehicle.

Ball spline

Ball splines (Ball Spline bearings) are a special type of linear motion bearing that are used to provide nearly frictionless linear motion while allowing the member to transmit torque simultaneously. There are grooves ground along the length of the shaft (thus forming splines) for the recirculating ground balls to run inside. The outer shell that houses the balls is called a nut rather than a bushing, but is not a nut in the traditional sense—it is not free to rotate about the shaft, but is free to travel up and down the shaft.

By increasing the contact area of the ball bearings on the shaft to approximately 45 degrees, the side load and direct load carrying capabilities are greatly increased. Each nut can be individually preloaded at the factory to decrease the available radial play to ensure rigidity. This process not only increases the contact area, increasing direct loading capabilities, but it also restricts any radial movement, increasing the overhung moment capabilities. This creates a sturdier structure that can handle a very strenuous working environment.

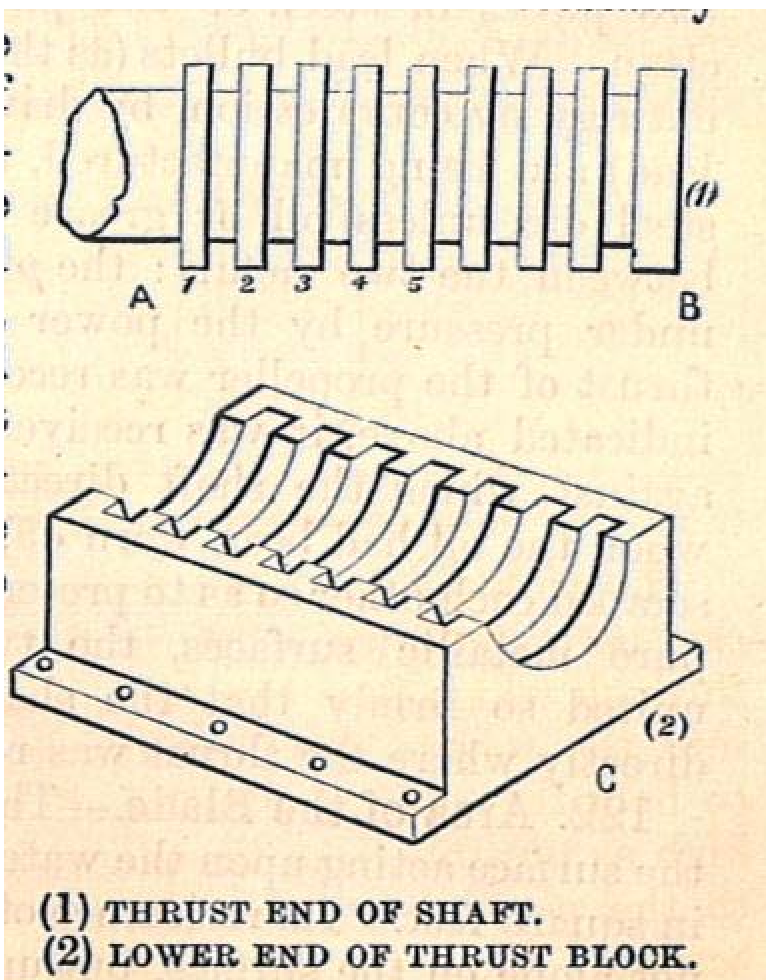
Chapter 11

Thrust Block and Cam Follower

Thrust block

A **thrust block**, also known as a **thrust box**, is a specialised form of thrust bearing used in ships, to resist the thrust of the propellor shaft and transmit it to the hull.

Early thrust boxes



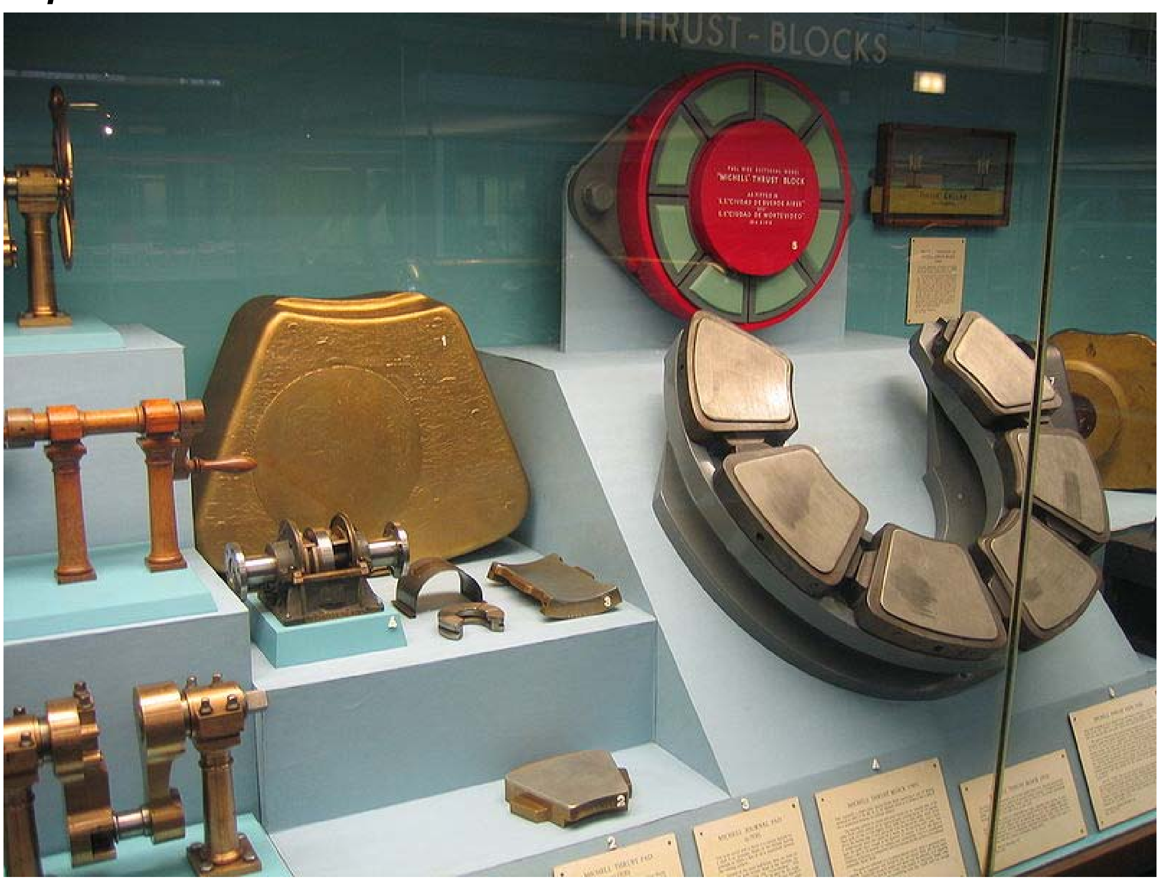
Multi-collar thrust box, with shaft

Early screw-propelled steamships used a thrust block or *thrust box* composed of perhaps a dozen lower-rated plain thrust journal bearings stacked on the same shaft. These were problematic in service: they were bulky, difficult to dismantle, wasted power through friction and they had a tendency to overheat. The thrust box was built of a box-like cast iron housing with a radial bearing at each end and a number of collars formed on the shaft between them. This shaft was often a short section of removable shaft called the *thrust shaft*, linking the engine ahead to the propellor shaft astern. A series of iron horseshoe-shaped collars fitted over the small diameter of the shaft and bore against the forward face of the shaft's collars. Each horseshoe was faced with a low-friction pad of babbitt metal. Lubrication was by an oil bath in the box and a plentiful volume was important for cooling purposes too.

Although lignum vitae wood was used for the radial stave bearings in the stuffing box, cooled directly by seawater itself, this material wasn't capable of withstanding the force needed for the thrust blocks of any but the earliest screw vessels.

Each horseshoe was independently adjustable forwards and back, by either wedged gibs or a screwed adjustment. A particular problem with these thrust boxes was in adjusting them so that the force was shared equally between all the collars. Adjustment was often done on the basis of their operating temperature, gauged with the engineer's hand.

Improved Michell thrust blocks



Michell thrust blocks at the London Science Museum

Improved understanding of the *theory* of lubrication films (initially by Reynolds) allowed the development of much more efficient bearing surfaces. This allowed the replacement of multiple collars in a thrust box by a single **thrust block**.

Fluid-film thrust bearings were invented by Australian engineer George Michell who patented his invention in 1905. Michell bearings contain a number of sector-shaped pads, arranged in a circle around the shaft, and which are free to pivot. These create wedge-shaped films of oil between the pads and a rotating disk on the shaft. Each lubricant "wedge" can only be of a limited length (in the direction of travel, i.e. circumferential) so multiple pads are needed rather than a single ring. No lubrication pump is needed, the rotation of the shaft itself is sufficient.

The need for an efficient thrust block became even more important with the advent of steam turbines and their higher propellor speeds. Despite this, there was some reluctance to adopt them in their homeland, until the discovery that World War I U-boats were using them. After this they were soon adopted widely. The large single pad illustrated is a model of one used in the battlecruiser HMS Hood, the pride of the Royal Navy.

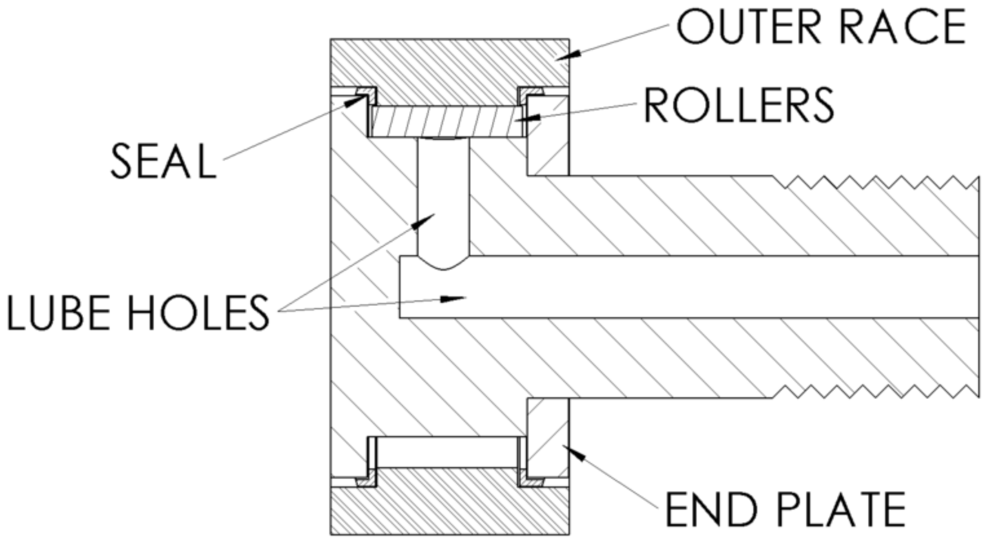
Cam follower

In automotive terms a *cam follower* may also refer to a tappet (or lifter) or rocker arm.

A **cam follower**, also known as a **track follower**, is a specialized type of roller or needle bearing designed to follow cams. Cam followers come in a vast array of different configurations, however the most defining characteristic is how the cam follower mounts to its mating part; *stud* style cam followers use a stud while the *yoke* style has a hole through the middle.

The first cam follower was invented and patented in 1937 by Thomas L. Robinson of the McGill Manufacturing Company. It replaced using just a standard bearing and bolt. The new cam followers were easier to use because the stud was already included and they could also handle higher loads.

Construction



A cross-sectional view of a stud type cam follower

While cam followers appear to be very similar to roller bearings in construction they have quite a few differences. Standard ball and roller bearings are designed to be pressed into a rigid housing, which provides circumferential support. This keeps the outer race from deforming, so the race cross-section is relatively thin. In the case of cam followers the outer race is loaded at a single point, so the outer race needs a thicker cross-section to reduce deformation. However, in order to facilitate this the roller diameter must be decreased, which also decreases the dynamic bearing capacity.

End plates are used to contain the needles or bearing axially. On stud style followers one of the end plates is integrated into the inner race/stud; the other is pressed onto the stud up to a shoulder on the inner race. The inner race is induction hardened so that the stud remains soft if modifications need to be made. On yoke style followers the end plates are peened or pressed onto the inner race or liquid metal injected onto the inner race. The inner race is either induction hardened or through hardened.

Another difference is that a lubrication hole is provided to relubricate the follower periodically. A hole is provided at both ends of the stud for lubrication. They also usually they have a black oxide finish to help reduce corrosion.

Types

There are many different types of cam followers available.

Anti-friction element

The most common anti-friction element employed is a *full complement* of needle rollers. This design can withstand high radial loads but no thrust loads. A similar design is the *caged needle roller* design, which also uses needle rollers, but uses a cage to keep them separated. This design allows for higher speeds but decreases the load capacity. The cage also increases internal space so it can hold more lubrication, which increases the time between relubrications. Depending on the exact design sometimes two rollers are put in each pocket of the cage.

For heavy-duty applications a *roller* design can be used. This employs two rows of larger rollers to increase the dynamic load capacity and provide some thrust capabilities. This design can support higher speeds than the full complement design.

For light-duty applications a *bushing* type follower can be used. Instead of using a type of a roller a plastic bushing is used to reduce friction, which provides a maintenance free follower. The disadvantage is that is can only support light loads, slow speeds, no thrust loads, and the temperature limit is 200 °F (93 °C). A bushing type stud follower can only support approximately 25% of the load of a roller type stud follower, while the heavy and yoke followers can handle 50%.

Shape

The outer diameter (OD) of the cam follower (stud or yoke) can be the standard cylindrical shape or be crowned. Crowned cam followers are used to keep the load evenly distributed if it deflects or if there is any misalignment between the follower and the followed surface. They are also used in turntable type applications to reduce skidding. Crowned followers can compensate for up to 0.5° of misalignment, while a cylindrical OD can only tolerate 0.06°. The only disadvantage is that they cannot bear as much load because of higher stresses.

Stud

Stud style cam followers usually have a *standard* sized stud, but a *heavy* stud is available for increased static load capacity.

Drives

The standard driving system for a stud type cam follower is a slot, for use with a flat head screwdriver. However, hex sockets are available for higher torquing ability, which is especially useful for eccentric cam followers and those used in blind holes. The only problem with hex sockets is that it eliminates relubrication capabilities on that end of the cam follower.

Eccentricity

Stud type cam followers are available with an eccentric stud. The stud has a bushing pushed onto it that has an eccentric outer diameter. This allows for adjustability during installation to eliminate any backlash. The adjustable range for an eccentric bearing is twice that of the eccentricity.

Yoke

Yoke type cam followers are usually used in applications where minimal deflection is required, as they can be support on both sides. They can support the same static load as a heavy stud follower.

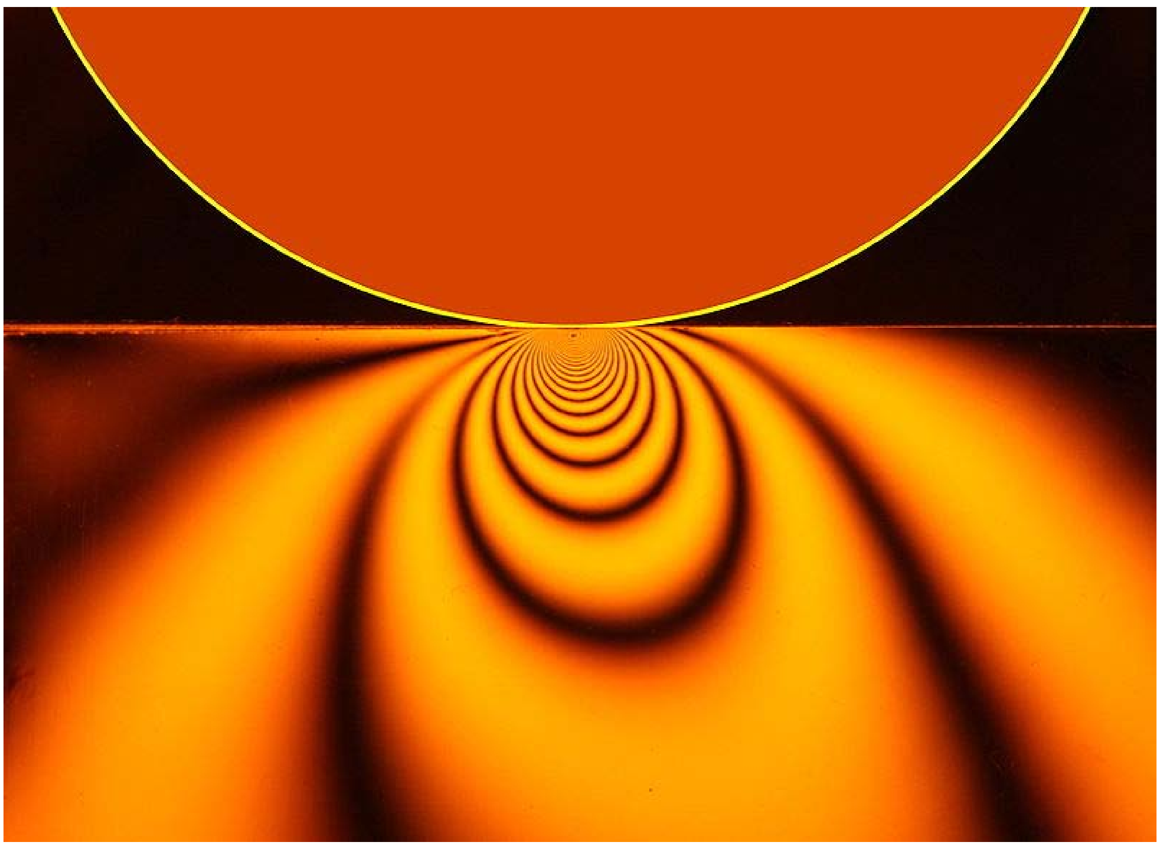
Track followers

All cam followers can be track followers, but not all track followers are cam followers. Some track followers have specially shaped outer diameters (OD) to follow tracks. For example, track followers are available with a V-groove for following a V-track, or the OD can have a flange to follow the lip of the track.

Specialized track followers are also designed to withstand thrust loads so the anti-friction elements are usually bearing balls or of a tapered roller bearing construction.

Chapter 12

Contact Mechanics



Stresses in a contact area loaded simultaneously with a normal and a tangential force. Stresses were made visible using photoelasticity.

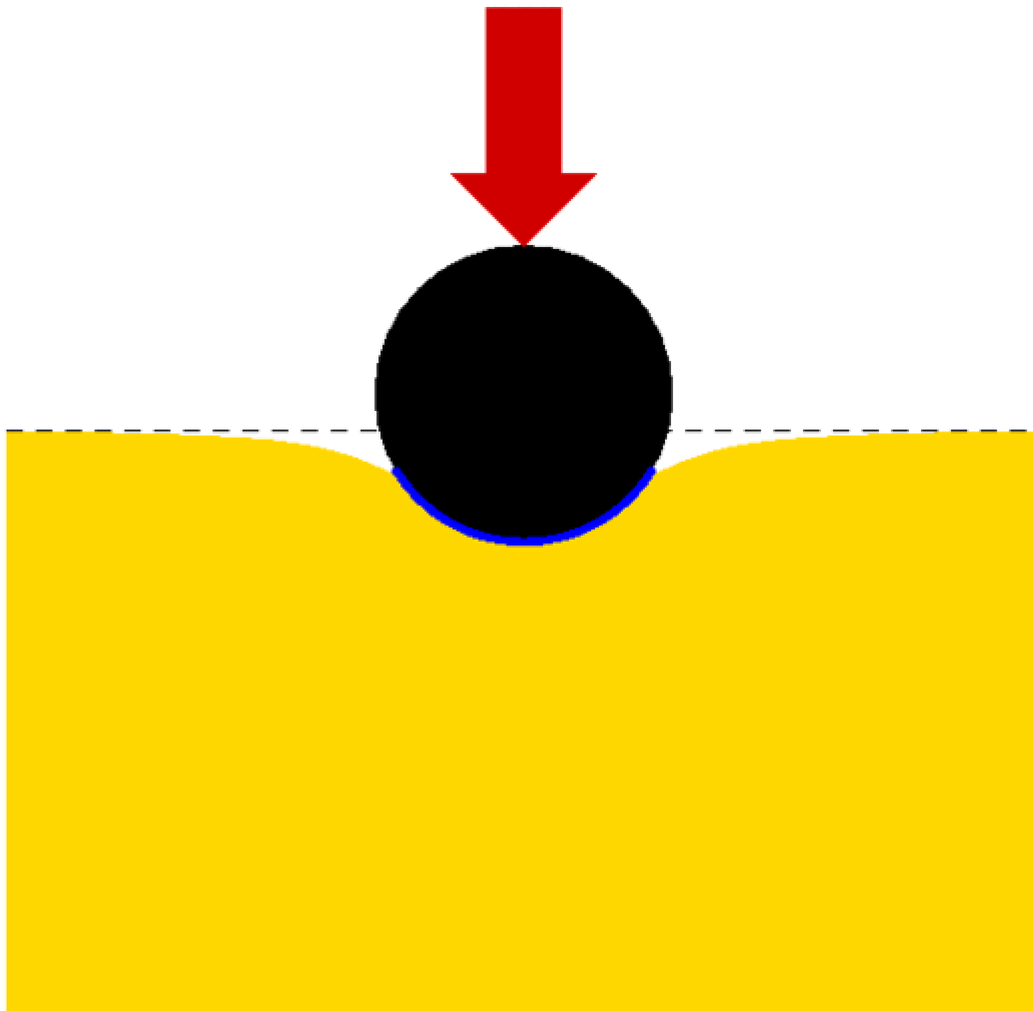
Contact mechanics is the study of the deformation of solids that touch each other at one or more points. The physical and mathematical formulation of the subject is built upon the mechanics of materials and continuum mechanics and focuses on computations involving elastic, viscoelastic, and plastic bodies in static or dynamic contact. Contact mechanics is foundational to the field of mechanical engineering; it provides necessary information for the safe and energy efficient design of technical systems.

The original work in contact mechanics dates back to 1882 with the publication of the paper "On the contact of elastic solids" ("Ueber die Berührung fester elastischer Körper") by Heinrich Hertz. Hertz was attempting to understand how the optical properties of multiple, stacked lenses might change with the force holding them together. Results in this field have since been extended to all branches of engineering, but are most essential in the study of tribology and indentation hardness. Hertzian contact stress refers to the localized stresses that develop as two curved surfaces come in contact and deform slightly under the imposed loads. This amount of deformation is dependent on the modulus of elasticity of the material in contact. It gives the contact stress as a function of the normal contact force, the radii of curvature of both bodies and the modulus of elasticity of both bodies. In gears and bearings in operation, these contact stresses are cyclic in nature and over time lead to sub-surface fatigue cracks. Hertzian contact stress forms the foundation for the equations for load bearing capabilities in bearings, gears, and any other bodies where two surfaces are in contact.

Principles of contacts mechanics can be applied in areas such as locomotive wheel-rail contact, coupling devices, braking systems, tires, bearings, combustion engines, mechanical linkages, gasket seals, metalworking, metal forming, ultrasonic welding, electrical contacts, and many others. Current challenges faced in the field may include stress analysis of contact and coupling members and the influence of lubrication and material design on friction and wear. Applications of contact mechanics further extend into the micro- and nanotechnological realm.

The motion of a single body in space is described by the governing equations of continuum mechanics. The approach used in contact mechanics is to restrict the motion of two or more bodies in space by additional constraints. These *unilateral* constraints ensure that bodies do not penetrate each other after coming into contact. Once the general equations for a contact problem are set up, different solution schemes can be used to simulate the behaviour of bodies in contact and to compute displacement and stress fields. A distinction is usually drawn between contact with and without friction.

History



When a sphere is pressed against an elastic material, the contact area increases.

Classical contact mechanics is most notably associated with Heinrich Hertz. In 1882 Hertz solved the problem involving contact between two elastic bodies with curved surfaces. This still-relevant classical solution provides a foundation for modern problems in contact mechanics. For example, in mechanical engineering and tribology, **Hertzian contact stress**, is a description of the stress within mating parts. In general, the Hertzian contact stress usually refers to the stress close to the area of contact between two spheres of different radii.

It was not until nearly one hundred years later that Johnson, Kendall, and Roberts found a similar solution for the case of adhesive contact . This theory was rejected by Boris Derjaguin and co-workers who proposed a different theory of adhesion in the 1970s. The Derjaguin model came to be known as the DMT (after Derjaguin, Muller and Toporov) model, and the Johnson et al. model came to be known as the JKR (after Johnson, Kendall and Roberts) model for adhesive elastic contact. This rejection proved to be instrumental in the development of the Tabor and later Maugis parameters that quantify which contact model (of the JKR and DMT models) represent adhesive contact better for specific materials.

Further advancement in the field of contact mechanics in the mid-twentieth century may be attributed to names such as Bowden and Tabor. Bowden and Tabor were the first to emphasize the importance of surface roughness for bodies in contact. Through investigation of the surface roughness, the true contact area between friction partners is found to be less than the apparent contact area. Such understanding also drastically changed the direction of undertakings in tribology. The works of Bowden and Tabor yielded several theories in contact mechanics of rough surfaces.

The contributions of Archard (1957) must also be mentioned in discussion of pioneering works in this field. Archard concluded that, even for rough elastic surfaces, the contact area is approximately proportional to the normal force. Further important insights along these lines were provided by Greenwood and Williamson (1966), Bush (1975), and Persson (2002). The main findings of these works were that the true contact surface in rough materials is generally proportional to the normal force, while the parameters of individual micro-contacts (i.e. pressure, size of the micro-contact) are only weakly dependent upon the load.

Non-adhesive contact

The classical theory of contact focused primarily on non-adhesive contact where no tension force is allowed to occur within the contact area, i.e., contacting bodies can be separated without adhesion forces. Several analytical and numerical approaches have been used to solve contact problems that satisfy the no-adhesion condition. Complex forces and moments are transmitted between the bodies where they touch, so problems in contact mechanics can become quite sophisticated. In addition, the contact stresses are usually a nonlinear function of the deformation. To simplify the solution procedure, a frame of reference is usually defined in which the objects (possibly in motion relative to one another) are static. They interact through surface tractions (or pressures/stresses) at their interface.

As an example, consider two objects which meet at some surface S in the (x,y)-plane with the z-axis assumed normal to the surface. One of the bodies will experience a normally-directed pressure distribution $p_z = p(x,y) = q_z(x,y)$ and in-plane surface traction distributions $q_x = q_x(x,y)$ and $q_y = q_y(x,y)$ over the region S. In terms of a Newtonian force balance, the forces:

$$P_z = \int_S p(x,y)\ \mathrm{d}A\ ;\quad Q_x = \int_S q_x(x,y)\ \mathrm{d}A\ ;\quad Q_y = \int_S q_y(x,y)\ \mathrm{d}A$$

must be equal and opposite to the forces established in the other body. The moments corresponding to these forces:

$$M_x = \int_S y\, p(x,y)\, \mathrm{d}A\,;\ M_y = \int_S x\, p(x,y)\, \mathrm{d}A\,;\ M_z = \int_S [x\, q_y(x,y) - y\, q_x(x,y)]\, \mathrm{d}A$$

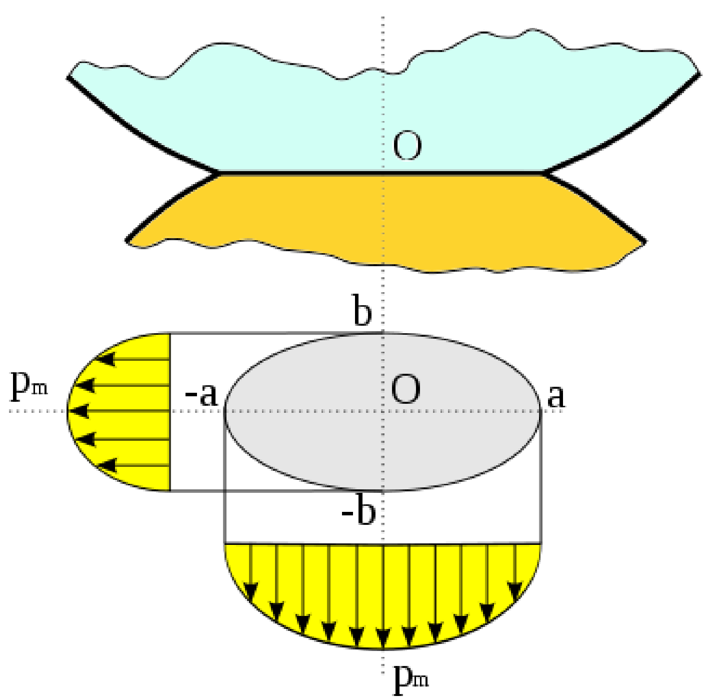
are also required to cancel between bodies so that they are kinematically immobile.

The following assumptions are made in determining the solutions of **Hertzian** contact problems:

- the strains are small and within the elastic limit,
- each body can be considered an elastic half-space, i.e.,, the area of contact is much smaller than the characteristic radius of the body,
- the surfaces are continuous and non-conforming, and
- the surfaces are frictionless.

Additional complications arise when some or all these assumptions are violated and such contact problems are usually called **non-Hertzian**.

Analytical solution techniques



Contact between two spheres.

Analytical solution methods for non-adhesive contact problem can be classified into two types based on the geometry of the area of contact. A **conforming contact** is one in which the two bodies touch at multiple points before any deformation takes place (i.e., they just "fit together"). A **non-conforming contact** is one in which the shapes of the bodies are dissimilar enough that, under zero load, they only touch at a point (or possibly along a line). In the non-conforming case, the contact area is small compared to the sizes of the objects and the stresses are highly concentrated in this area.

A common approach in linear elasticity is to superpose a number of solutions each of which corresponds to a point load acting over the area of contact. For example, in the case of loading of a half-plane, the Flamant solution is often used as a starting point and then generalized to various shapes of the area of contact. The force and moment balances between the two bodies in contact act as additional constraints to the solution.

Numerical solution techniques

Distinctions between conforming and non-conforming contact do not have to be made when numerical solution schemes are employed to solve contact problems. These methods do not rely on further assumptions within the solution process since they base solely on the general formulation of the underlying equations . Besides the standard equations describing the deformation and motion of bodies two additional inequalities can be formulated. The first simply restricts the motion and deformation of the bodies by the assumption that no penetration can occur. Hence the gap g_N between two bodies can only be positive or zero

$$g_N \geq 0$$

where $g_N = 0$ denotes contact. The second assumption in contact mechanics is related to the fact, that no tension force is allowed to occur within the contact area (contacting bodies can be lifted up without adhesion forces). This leads to an inequality which the stresses have to obey at the contact interface. It is formulated for the contact pressure $p_N = \mathbf{t} \cdot \mathbf{n}$

$$p_N \leq 0 \,.$$

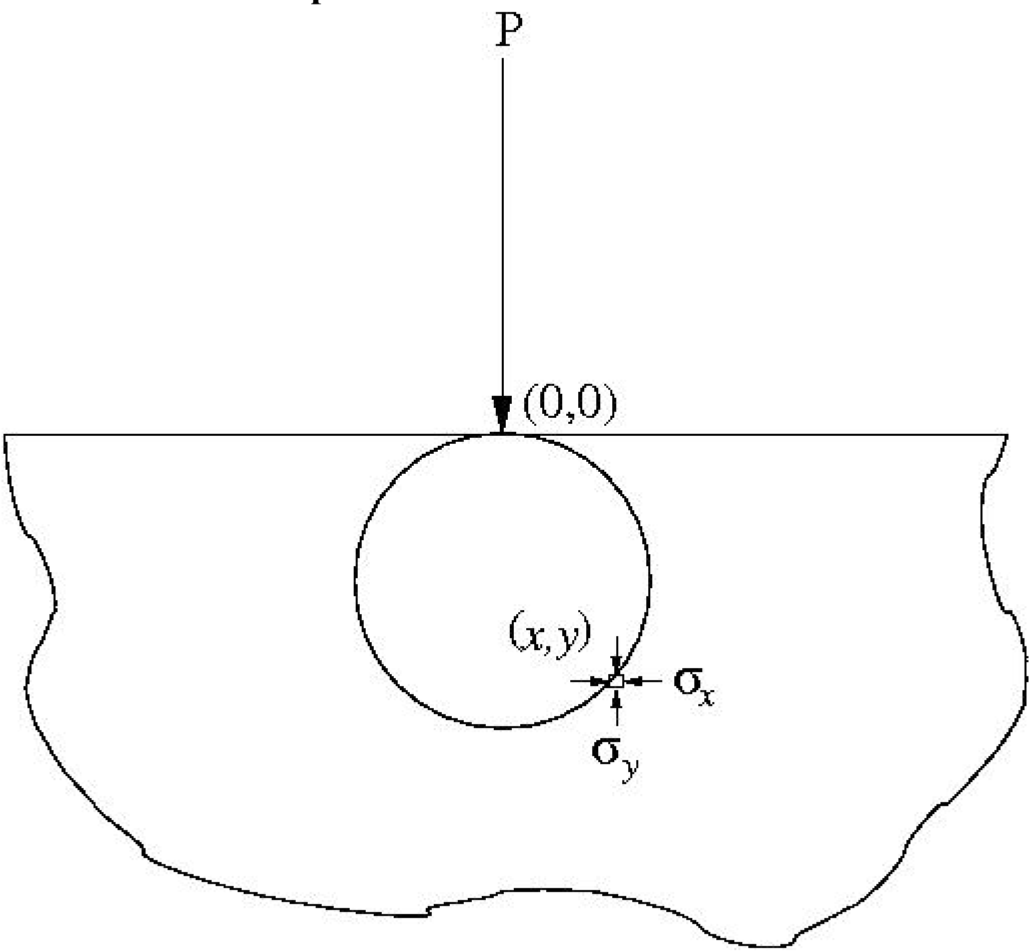
Since for contact, $g_N = 0$, the contact pressure is always negative, $p_N < 0$, and further for non contact the gap is open, $g_N > 0$, and the contact pressure is zero, $p_N = 0$, the so called Kuhn–Tucker form of the contact constraints can be written as

$$g_N \geq 0 \,, \quad p_N \leq 0 \,, \quad p_N \, g_N = 0 \,.$$

These conditions are valid in a general way. The mathematical formulation of the gap depends upon the kinematics of the underlying theory of the solid (e.g., linear or nonlinear solid in two- or three dimensions, beam or shell model).

Classical solutions

Point contact on a half-plane



Schematic of the loading on a plane by force P at a point (0,0).

A starting point for solving contact problems is to understand the effect of a "point-load" applied to an isotropic, homogeneous, and linear elastic half-plane, shown in the figure to the right. The problem may be either be plane stress or plane strain. This is a boundary value problem of linear elasticity subject to the traction boundary conditions:

$$\sigma_{xz}(x,0) = 0 \ ; \quad \sigma_z(x,z) = -P\delta(x,z)$$

where δ(*x*,*z*) is the Dirac delta function. The boundary conditions state that are no shear stresses on the surface and a singular normal force P is applied at (0,0). Applying these conditions to the governing equations of elasticity produces the result

$$\sigma_{xx} = -\frac{2P}{\pi}\frac{x^2 z}{(x^2+z^2)^2}\ ;\quad \sigma_{zz} = -\frac{2P}{\pi}\frac{z^3}{(x^2+z^2)^2}$$
$$\sigma_{xz} = -\frac{2P}{\pi}\frac{xz^2}{(x^2+z^2)^2}$$

for some point, (x,y), in the half-plane. The circle shown in the figure indicates a surface on which the maximum shear stress is constant. From this stress field, the strain components and thus the displacements of all material points may be determined.

Line contact on a half-plane

*Normal loading over a region (*a,b*)*

Suppose, rather than a point load P, a distributed load $p(x)$ is applied to the surface instead, over the range $a < x < b$. The principle of linear superposition can be applied to determine the resulting stress field as the solution to the integral equations:

$$\sigma_{xx} = -\frac{2z}{\pi}\int_a^b \frac{p(x')(x-x')^2\,dx'}{[(x-x')^2+z^2]^2}\ ;\quad \sigma_{zz} = -\frac{2z^3}{\pi}\int_a^b \frac{p(x')(x-x')^2\,dx'}{[(x-x')^2+z^2]^2}$$
$$\sigma_{xz} = -\frac{2z^2}{\pi}\int_a^b \frac{p(x')(x-x')\,dx'}{[(x-x')^2+z^2]^2}$$

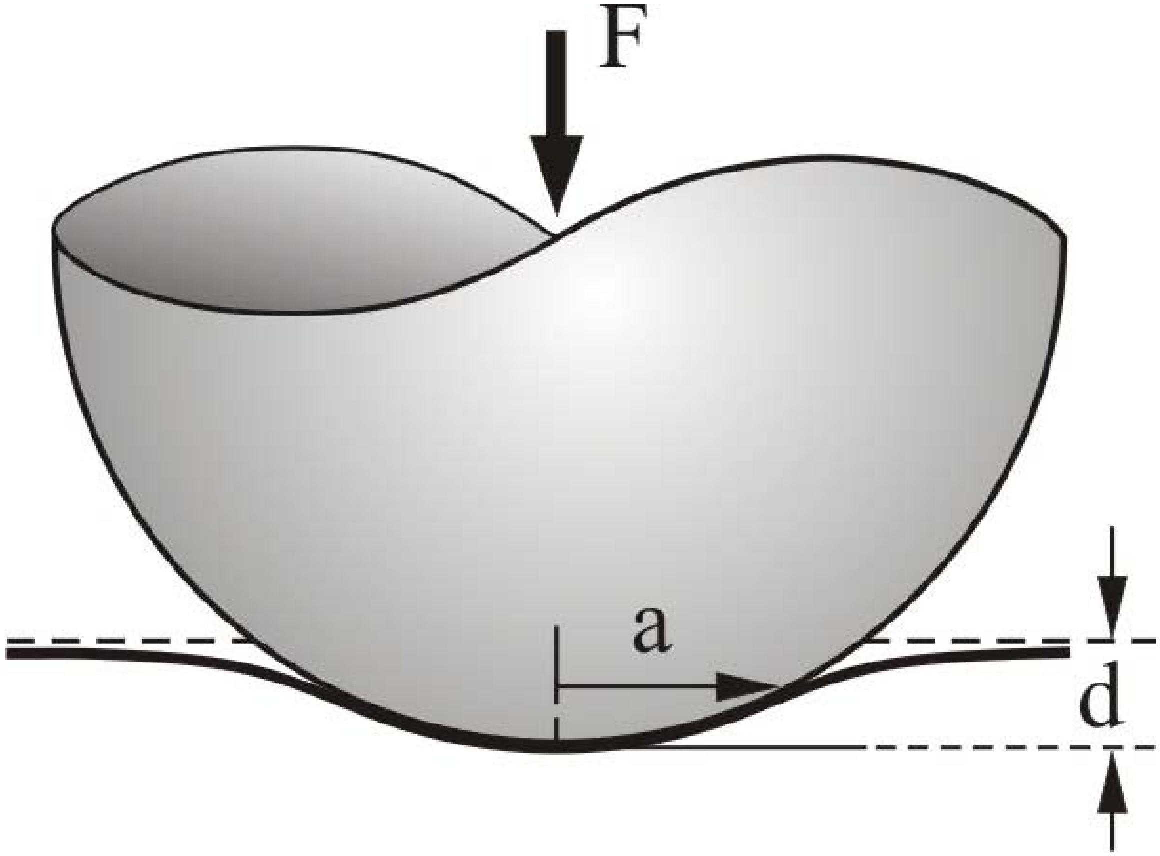
*Shear loading over a region (*a,b*)*

The same principle applies for loading on the surface in the plane of the surface. These kinds of tractions would tend to arise as a result of friction. The solution is similar the above (for both singular loads Q and distributed loads $q(x)$) but altered slightly:

$$\sigma_{xx} = -\frac{2}{\pi}\int_a^b \frac{q(x')(x-x')^3\,dx'}{[(x-x')^2+z^2]^2}\ ;\quad \sigma_{zz} = -\frac{2z^2}{\pi}\int_a^b \frac{q(x')(x-x')\,dx'}{[(x-x')^2+z^2]^2}$$
$$\sigma_{xz} = -\frac{2z}{\pi}\int_a^b \frac{q(x')(x-x')^2\,dx'}{[(x-x')^2+z^2]^2}$$

These results may themselves be superposed onto those given above for normal loading to deal with more complex loads.

Contact between a Sphere and an Elastic Half-Space



Contact between a sphere and an elastic half-space

An elastic sphere of radius R indents an elastic half-space to depth d, and thus creates a contact area of radius $a = \sqrt{Rd}$. The applied force F is related to the displacement d by

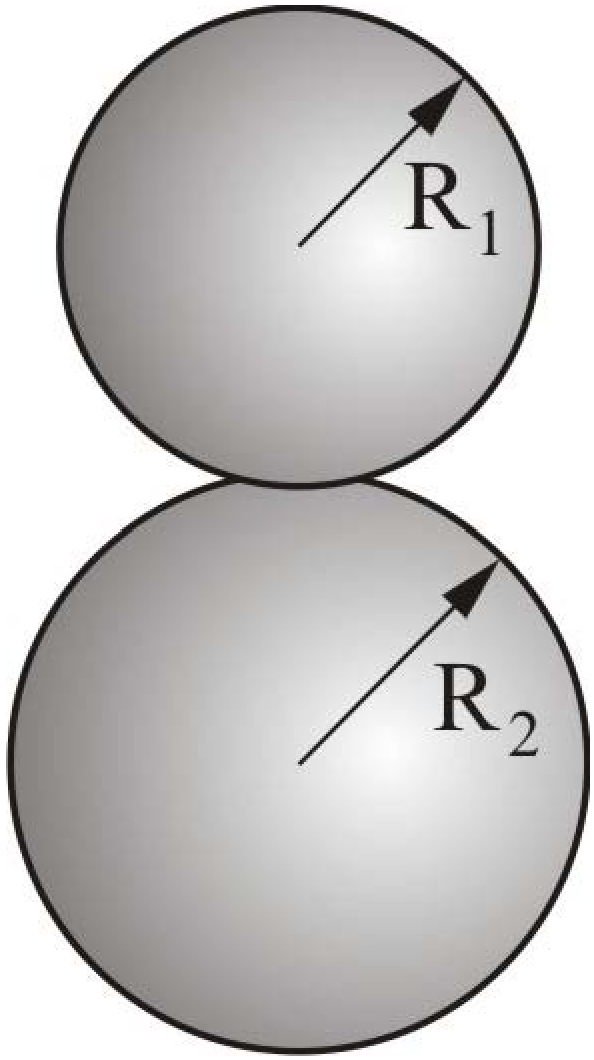
$$F = \frac{4}{3} E^* R^{1/2} d^{3/2}$$

where

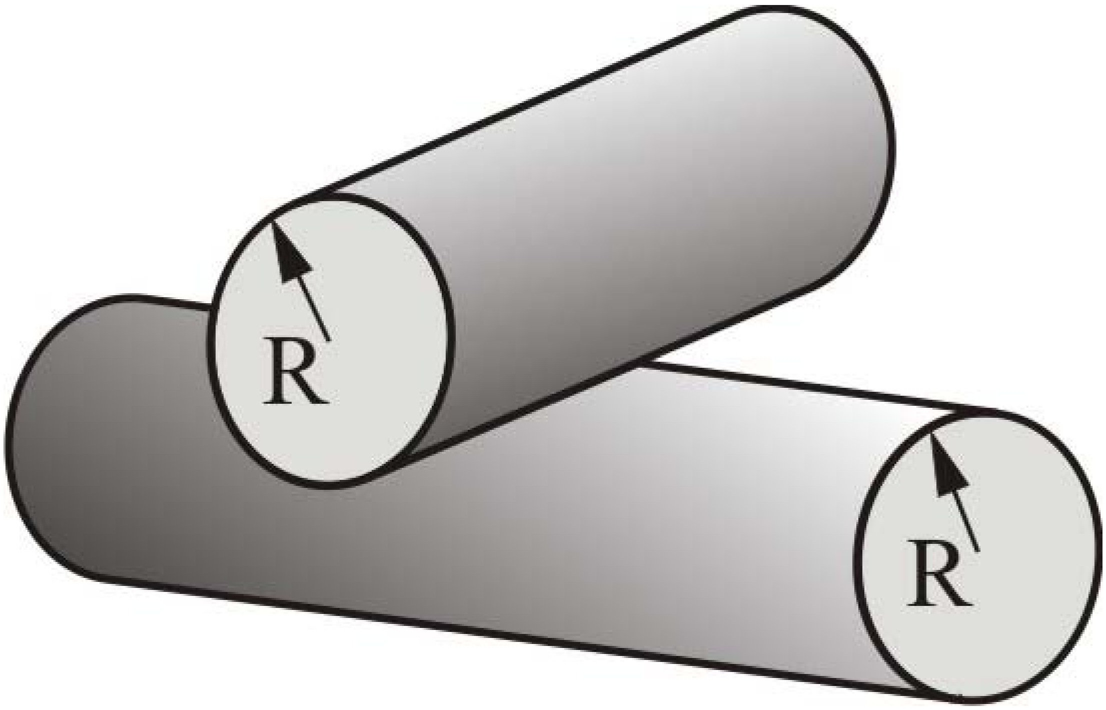
$$\frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}$$

and E_1,E_2 are the elastic moduli and ν_1,ν_2 the Poisson's ratios associated with each body.

Contact between two spheres



Contact between two spheres



Contact between two crossed cylinders of equal radius

For contact between two spheres of radii R_1 and R_2, the area of contact is a circle of radius a. The distribution of normal traction in the contact area as a function of distance from the center of the circle is

$$p(r) = p_0 \left(1 - \frac{r^2}{a^2}\right)^{1/2}$$

where p_0 is the maximum contact pressure given by

$$p_0 = \frac{3F}{2\pi a^2} = \frac{1}{\pi}\left(\frac{6FE^{*2}}{R^2}\right)^{1/3}$$

where the effective radius R is defined as

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$$

The area of contact is related to the applied load $F$by the equation

$$a^3 = \frac{3FR}{4E^*}$$

The depth of indentation *d* is related to the maximum contact pressure by

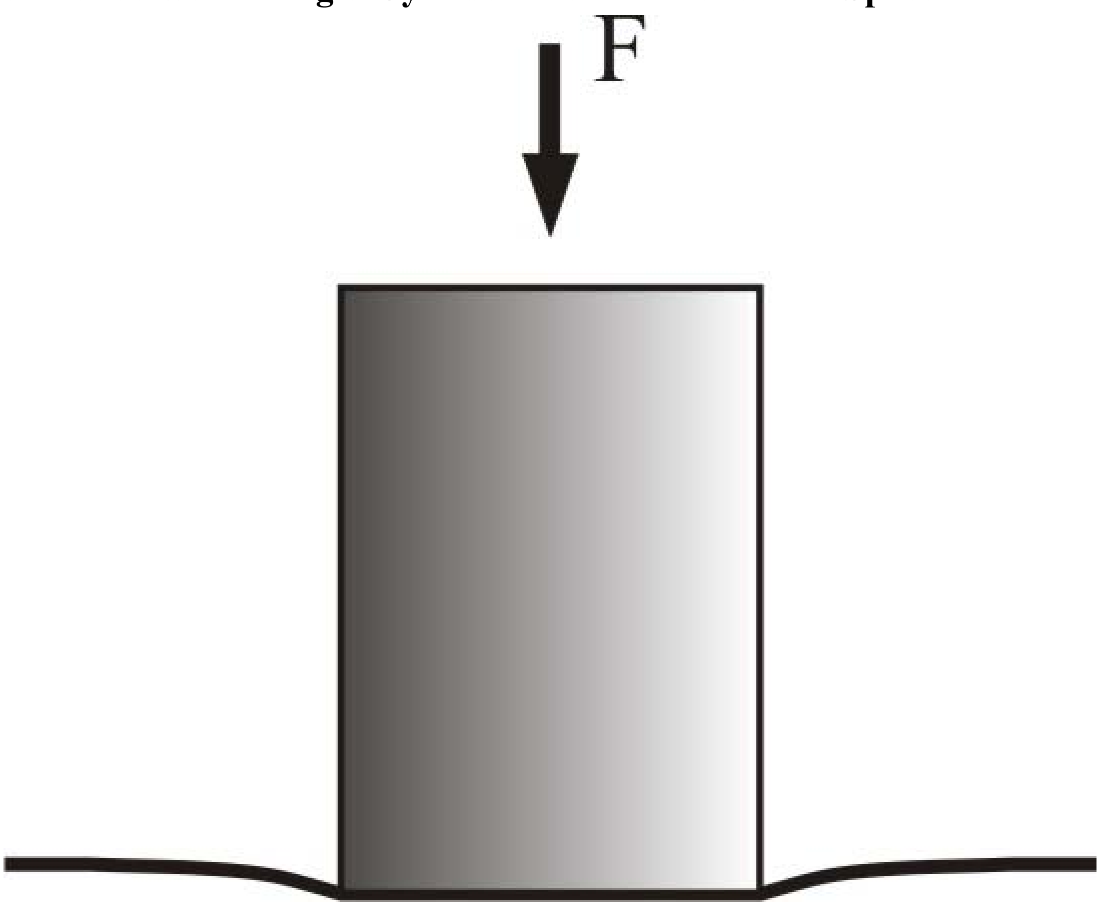
$$d = \frac{a^2}{R} = \left(\frac{9F^2}{16RE^{*2}}\right)^{1/3}$$

The maximum shear stress occurs in the interior at $z \approx 0.49a$for ν = 0.33.

Contact between Two Crossed Cylinders of Equal Radius *R*

This is equivalent to contact between a sphere of radius *R* and a plane.

Contact between a Rigid Cylinder and an Elastic Half-Space



Contact between a rigid cylindrical indenter and an elastic half-space

If a rigid cylinder is pressed into an elastic half-space, it creates a pressure distribution described by

$$p(r) = p_0 \left(1 - \frac{r^2}{a^2}\right)^{-1/2}$$

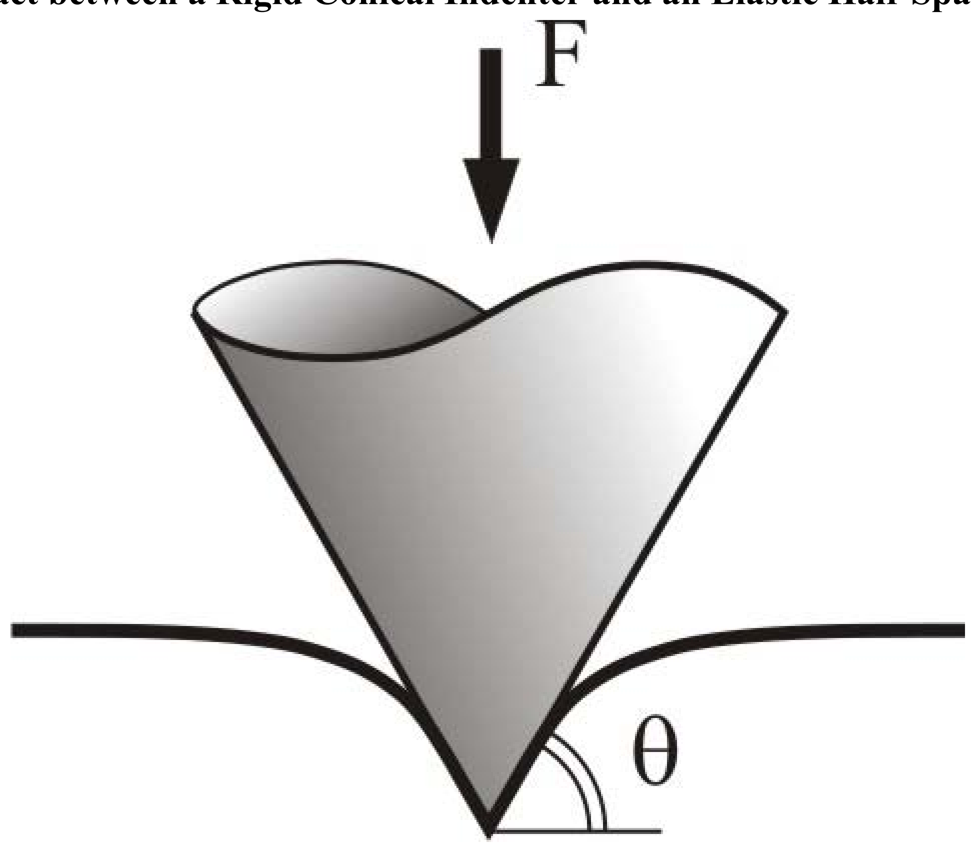
where a is the radius of the cylinder and

$$p_0 = \frac{1}{\pi} E^* \frac{d}{a}$$

The relationship between the indentation depth and the normal force is given by

$$F = 2aE^* d$$

Contact between a Rigid Conical Indenter and an Elastic Half-Space



Contact between a rigid conical indenter and an elastic half-space

In the case of indentation of an elastic half-space using a rigid conical indenter, the indentation depth and contact radius are related by

$$d = \frac{\pi}{2} a \tan\theta$$

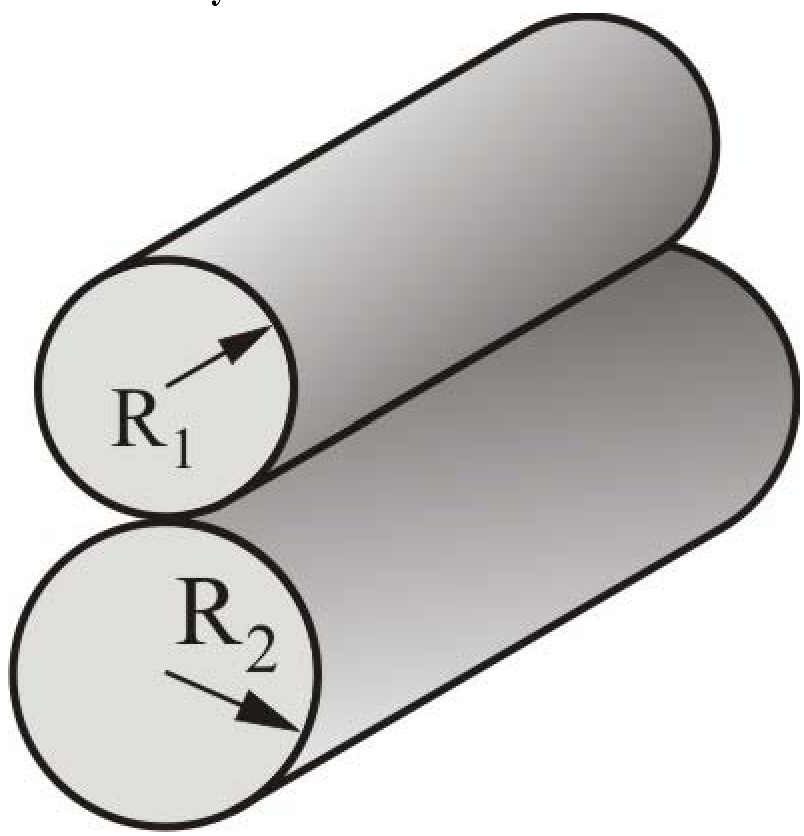
with θ defined as the angle between the plane and the side surface of the cone. The pressure distribution takes on the form

$$p(r) = -\frac{Ed}{\pi a\left(1-\nu^2\right)} \ln\left(\frac{a}{r} + \sqrt{\left(\frac{a}{r}\right)^2 - 1}\right)$$

The stress has a logarithmic singularity on the tip of the cone. The total force is

$$F_N = \frac{2}{\pi} E^* \frac{d^2}{\tan\theta}$$

Contact between Two Cylinders with Parallel Axes



Contact between two cylinders with parallel axes

In contact between two cylinders with parallel axes, the force is linearly proportional to the indentation depth:

$$F = \frac{\pi}{4} E^* L d$$

The radii of curvature are entirely absent from this relationship. The contact radius is described through the usual relationship

$$a = \sqrt{Rd}$$

with

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$$

as in contact between two spheres. The maximum pressure is equal to

$$p_0 = \left(\frac{E^* F}{\pi L R}\right)^{1/2}$$

Contact between Rough Surfaces

When two bodies with rough surfaces are pressed into each other, the true contact area A is much smaller than the apparent contact area A_0. In contact between a "random rough" surface and an elastic half-space, the true contact area is related to the normal force F by

$$A = \frac{\kappa}{E^* h'} F$$

with h' equal to the root mean square (also known as the quadratic mean) of the surface slope and $\kappa \approx 2$. The median pressure in the true contact surface

$$p_{\mathrm{av}} = \frac{F}{A} \approx \frac{1}{2} E^* h'$$

can be reasonably estimated as half of the effective elastic modulus E^* multiplied with the root mean square of the surface slope h' .

For the situation where the asperities on the two surfaces have a Gaussian height distribution and the peaks can be assumed to be spherical, the average contact pressure is sufficient to cause yield when $p_{\mathrm{av}} = 1.1\sigma_y \approx 0.39\sigma_0$ where σ_y is the uniaxial yield stress and σ_0 is the indentation hardness. Greenwood and Williamson defined a dimensionless parameter Ψ called the **plasticity index** that could be used to determine whether contact would be elastic or plastic.

The Greenwood-Williamson model requires knowledge of two statistically dependent quantities; the standard deviation of the surface roughness and the curvature of the asperity peaks. An alternative definition of the plasticity index has been given by Mikic. Yield occurs when the pressure is greater than the uniaxial yield stress. Since the yield stress is proportional to the indentation hardness σ_0, Micic defined the plasticity index for elastic-plastic contact to be

$$\Psi = \frac{E^* h'}{\sigma_0} > \tfrac{2}{3} \,.$$

In this definition Ψ represents the micro-roughness in a state of complete plasticity and only one statistical quantity, the rms slope, is needed which can be calculated from surface measurements. For $\Psi < \frac{2}{3}$, the surface behaves elastically during contact.

In both the Greenwood-Williamson and Mikic models the load is assumed to be proportional to the deformed area. Hence, whether the system behaves plastically or elastically is independent of the applied normal force.

Adhesive contact

When two solid surfaces are brought into close proximity to each other they experience attractive van der Waals forces. Bradley's van der Waals model provides a means of calculating the tensile force between two rigid spheres with perfectly smooth surfaces. The Hertzian model of contact does not consider adhesion possible. However, in the late 1960s, several contradictions were observed when the Hertz theory was compared with experiments involving contact between rubber and glass spheres.

It was observed that, though Hertz theory applied at large loads, at low loads

- the area of contact was larger than that predicted by Hertz theory,
- the area of contact had a non-zero value even when the load was removed, and
- there was strong adhesion if the contacting surfaces were clean and dry.

This indicated that adhesive forces were at work. The Johnson-Kendall-Roberts (JKR) model and the Derjaguin-Muller-Toporov (DMT) models were the first to incorporate adhesion into Hertzian contact.

Bradley model for rigid contact

It is commonly assumed that the surface force between two atomic planes at a distance *z* from each other can be derived from the Lennard-Jones potential. With that assumption we can write

$$f(z) = \frac{16\gamma}{3z_0}\left[\left(\frac{z}{z_0}\right)^{-9} - \left(\frac{z}{z_0}\right)^{-3}\right]$$

where f is the force (positive in compression), 2γ is the is the total surface energy of **both** surfaces per unit area, and z_0 is the equilibrium separation of the two atomic planes.

The Bradley model assumed applied the Lennard-Jones potential to find the force of adhesion between two rigid spheres. The total force between the spheres is found to be

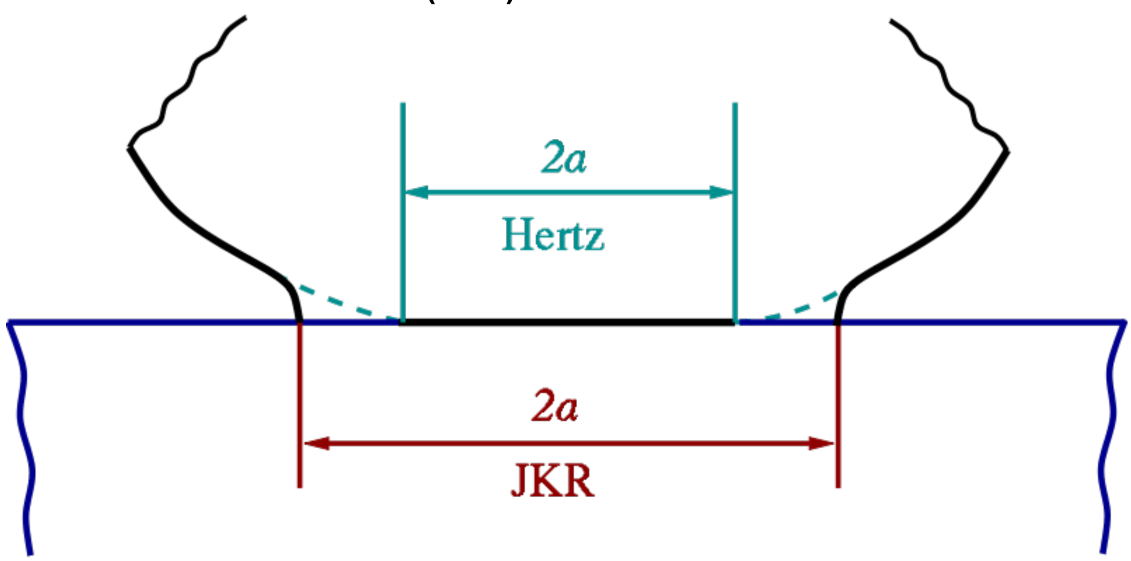
$$F_a = \frac{16\gamma\pi R}{3}\left[\frac{1}{4}\left(\frac{z}{z_0}\right)^{-8} - \left(\frac{z}{z_0}\right)^{-2}\right] \; ; \; \frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_1}$$

where R_1,R_2 are the radii of the two spheres.

The two spheres separate completely when the **pull-off force** is achieved at $z = z_0$ at which point

$$F_a = F_c = -4\gamma\pi R$$

Johnson-Kendall-Roberts (JKR) model for elastic contact



Schematic of contact area for the JKR model.

To incorporate the effect of adhesion in Hertzian contact, Johnson, Kendall, and Roberts formulated the JKR theory of adhesive contact using a balance between the stored elastic energy and the loss in surface energy. The JKR model considers the effect of contact pressure and adhesion only inside the area of contact. The general solution for the pressure distribution in the contact area in the JKR model is

$$p(r) = p_0\left(1 - \frac{r^2}{a^2}\right)^{1/2} + p_0'\left(1 - \frac{r^2}{a^2}\right)^{-1/2}$$

Note that in the original Hertz theory, the term containing p_0' was neglected on the ground that tension could not be sustained in the contact zone. For contact between two spheres

$$p_0 = \frac{2aE^*}{\pi R} ~;~~ p_0' = -\left(\frac{4\gamma E^*}{\pi a}\right)^{1/2}$$

where a is the radius of the area of contact, F is the applied force, 2γ is the total surface energy of **both** surfaces per unit contact area, $R_i,\, E_i,\, \nu_i,~~ i = 1, 2$ are the radii, Young's moduli, and Poisson's ratios of the two spheres, and

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} ~;~~ \frac{1}{E^*} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}$$

The approach distance between the two spheres is given by

$$d = \frac{\pi a}{2E^*}(p_0 + 2p_0') = \frac{a^2}{R}$$

The Hertz equation for the area of contact between two spheres, modified to take into account the surface energy, has the form

$$a^3 = \frac{3R}{4E^*}\left(F + 6\gamma\pi R + \sqrt{12\gamma\pi R F + (6\gamma\pi R)^2}\right)$$

When the surface energy is zero, $\gamma = 0$, the Hertz equation for contact between two spheres is recovered. When the applied load is zero, the contact radius is

$$a^3 = \frac{9R^2\gamma\pi}{E^*}$$

The tensile load at which the spheres are separated, i.e., $a = 0$, is predicted to be

$$F_c = -3\gamma\pi R$$

This force is also called the **pull-off force**. Note that this force is independent of the moduli of the two spheres. However, there is another possible solution for the value of a at this load. This is the critical contact area a_c, given by

$$a_c^3 = \frac{9R^2\gamma\pi}{4E^*}$$

If we define the **work of adhesion** as

$$\Delta\gamma = \gamma_1 + \gamma_2 - \gamma_{12}$$

where γ_1,γ_2 are the adhesive energies of the two surfaces and γ_{12} is an interaction term, we can write the JKR contact radius as

$$a^3 = \frac{3R}{4E^*}\left(F + 3\Delta\gamma\pi R + \sqrt{6\Delta\gamma\pi RF + (3\Delta\gamma\pi R)^2}\right)$$

The tensile load at separation is

$$F = -\frac{3}{2}\Delta\gamma\pi R$$

and the critical contact radius is given by

$$a_c^3 = \frac{9R^2\Delta\gamma\pi}{4E^*}$$

The critical depth of penetration is

$$d_c = \frac{a_c^2}{R} = \left(\frac{9}{4}\right)^{\frac{2}{3}}(\Delta\gamma)^{\frac{2}{3}}\left(\frac{\pi^{\frac{2}{3}}\ R^{\frac{1}{3}}}{E^{*\frac{2}{3}}}\right)$$

Derjaguin-Muller-Toporov (DMT) model for elastic contact

The Derjaguin-Muller-Toporov model developed an alternative model for adhesive contact. The DMT theory assumed that the contact profile remained the same as in Hertzian contact but will additional attractive interactions outside the area of contact.

The Hertz equation for the area of contact between two spheres from DMT theory is

$$a^3 = \frac{3R}{4E^*}(F + 4\gamma\pi R)$$

and the pull-off force is

$$F_c = -4\gamma\pi R$$

When the pull-off force is achieved the contact area becomes zero and there is no singularity in the contact stresses at the edge of the contact area.

In terms of the work of adhesion $\Delta\gamma$

$$a^3 = \frac{3R}{4E^*}(F + 2\Delta\gamma\pi R)$$

and

$$F_c = -2\Delta\gamma\pi R$$

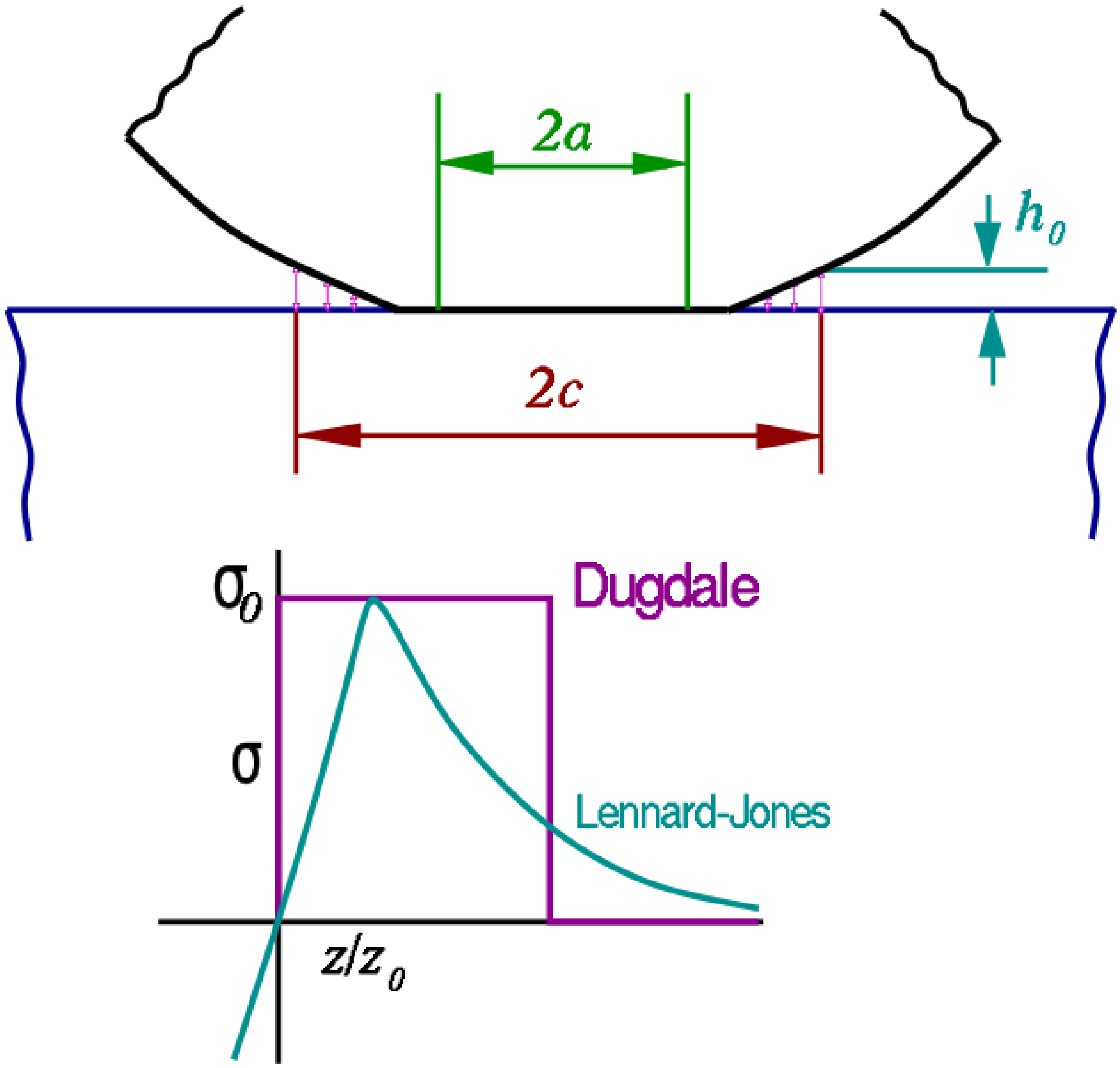
Tabor coefficient

In 1977, Tabor showed that the apparent contradiction between the JKR and DMT theories could be resolved by noting that the two theories were the extreme limits of a single theory parametrized by the **Tabor coefficient** (μ) defined as

$$\mu := \frac{d_c}{z_0} \approx \left[\frac{R(\Delta\gamma)^2}{E^{*2} z_0^3}\right]^{1/3}$$

where z_0 is the equilibrium separation between the two surfaces in contact. The JKR theory applies to large, compliant spheres for which μ is large. The DMT theory applies for small, stiff spheres with small values of μ.

Maugis-Dugdale model for elastic contact



Schematic of contact area for the Maugis-Dugdale model.

Further improvement to the Tabor idea was provided by Maugis who represented the surface force in terms of a Dugdale cohesive zone approximation such that the work of adhesion is given by

$$\Delta\gamma = \sigma_0 \; h_0$$

where σ_0 is the maximum force predicted by the Lennard-Jones potential and h_0 is the maximum separation obtained by matching the areas under the Dugdale and Lennard-Jones curves. This means that the attractive force is constant for $z_0 \leq z \leq z_0 + h_0$. There is not further penetration in compression. Perfect contact occurs in an area of radius a and adhesive forces of magnitude σ_0 extend to an area of radius $c > a$. In the region $a < r < c$, the two surfaces are separated by a distance $h(r)$ with $h(a) = 0$ and $h(c) = h_0$. The ratio m is defined as

$$m := \frac{c}{a.}$$

In the Maugis-Dugdale theory, the surface traction distribution is divided into two parts - one due to the Hertz contact pressure and the other from the Dugdale adhesive stress. Hertz contact is assumed in the region $-a < r < a$. The contribution to the surface traction from the Hertz pressure is given by

$$p^H(r) = \left(\frac{3F^H}{2\pi a^2}\right)\left(1 - \frac{r^2}{a^2}\right)^{1/2}$$

where the Hertz contact force F^H is given by

$$F^H = \frac{4E^* a^3}{3R}$$

The penetration due to elastic compression is

$$d^H = \frac{a^2}{R}$$

The vertical displacement at $r = c$ is

$$u^H(c) = \frac{1}{\pi R}\left[a^2(2 - m^2)\sin^{-1}\left(\frac{1}{m}\right) + a^2\sqrt{m^2 - 1}\right]$$

and the separation between the two surfaces at $r = c$ is

$$h^H(c) = \frac{c^2}{2R} - d^H + u^H(c)$$

The surface traction distribution due to the adhesive Dugdale stress is

$$p^D(r) = \begin{cases} -\frac{\sigma_0}{\pi}\cos^{-1}\left[\frac{2 - m^2 - \frac{r^2}{a^2}}{m^2\left(1 - \frac{r^2}{m^2 a^2}\right)}\right] & \text{for} \quad r \leq a \\ -\sigma_0 & \text{for} \quad a \leq r \leq c \end{cases}$$

The total adhesive force is then given by

$$F^D = -2\sigma_0 m^2 a^2 \left[\cos^{-1}\left(\frac{1}{m}\right) + a^2\sqrt{m^2-1}\right]$$

The compression due to Dugdale adhesion is

$$d^D = -\left(\frac{2\sigma_0 a}{E^*}\right)\sqrt{m^2-1}$$

and the gap at $r = c$ is

$$h^D(c) = \left(\frac{4\sigma_0 a}{\pi E^*}\right)\left[\sqrt{m^2-1}\cos^{-1}\left(\frac{1}{m}\right) + 1 - m\right]$$

The net traction on the contact area is then given by $p(r) = p^H(r) + p^D(r)$ and the net contact force is $F = F^H + F^D$. When $h(c) = h^H(c) + h^D(c) = h_0$ the adhesive traction drops to zero.

Non-dimensionalized values of a,c,F,d are introduced at this stage that are defied as

$$\bar{a} = \alpha a\,;\ \ \bar{c} := \alpha c\,;\ \ \bar{d} := \alpha^2 R d\,;\ \ \alpha := \left(\frac{4E^*}{3\pi\Delta\gamma R^2}\right)^{1/3}\ ;\ \ \bar{A} := \pi c^2\,;\ \ \bar{F} = \frac{F}{\pi\Delta\gamma R}$$

In addition, Maugis proposed a parameter λ which is equivalent to the Tabor coefficient. This parameter is defined as

$$\lambda := 1.16\mu = \sigma_0\left(\frac{9R}{2\pi\Delta\gamma {E^*}^2}\right)^{1/3}$$

Then the net contact force may be expressed as

$$\bar{F} = \bar{a}^3 - \frac{4}{3}\,\lambda\bar{a}^2\left[\sqrt{m^2-1} + m^2\sec^{-1} m\right]$$

and the elastic compression as

$$\bar{d} = \bar{a}^2 - \frac{4}{3}\,\lambda\bar{a}\sqrt{m^2-1}$$

The equation for the cohesive gap between the two bodies takes the form

$$\frac{\lambda \bar{a}^2}{2}\left[(m^2-2)\sec^{-1}m+\sqrt{m^2-1}\right]+\frac{4\lambda\bar{a}}{3}\left[\sqrt{m^2-1}\sec^{-1}m-m+1\right]=1$$

This equation can be solved to obtain values of *c* for various values of *a* and λ. For large values of λ, $m \rightarrow 1$ and the JKR model is obtained. For small values of λ the DMT model is retrieved.

Carpick-Ogletree-Salmeron (COS) model

The Maugis-Dugdale model can only be solved iteratively if the value of λ is not known a-priori. The Carpick-Ogletree-Salmeron approximate solution simplifies the process by using the following relation to determine the contact radius *a*:

$$a = a_0(\beta)\left(\frac{\beta+\sqrt{1-F/F_c(\beta)}}{1+\beta}\right)^{2/3}$$

where a_0 is the contact area at zero load, and β is a transition parameter that is related to λ by

$$\lambda = -0.924\ln(1-1.02\beta)$$

The case β = 1 corresponds exactly to JKR theory while β = 0 corresponds to DMT theory. For intermediate cases 0 < β < 1 the COS model corresponds closely to the Maugis-Dugdale solution for 0.1 < λ < 5.