Mechanical Systems I

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Mechanical Systems I

Section I

Purpose

The purpose of this course is to provide practitioners with the theoretical as well as current practical knowledge in the area of mechanical systems and various mechanical components and to strengthen the relationships of materials science to manufacturing techniques.

Brief history of Machines

Archimedes (287 BC-212 BC)

Archimedes was recognized because of his studies on levers and pulleys which in turn produced the block and tackle, compound pulley system known as mechanical advantage. He invented the Archimedes Screw which was a large spiral screw inside an enclosed revolving cylinder used to transport water to a higher elevation, primarily for irrigation at that time. This was the precursor to the grain elevator, the auger in a snow blower, and is used to transport solid waste in sewage plants.

Archimedes greatly improved the power and accuracy of the catapult. This mechanical system was useful in warfare to accurately deliver projectiles over great distances. The crossbow is another example.

Hero of Alexandria (10 AD-70 AD)

Hero of Alexandria was credited with inventing the windwheel or today referred to as windmill. This was the first approach to wind harnessing. This device has been used for centuries for irrigation, grain mills, sawmills, supplying energy during the beginning of the Industrial Revolution (although later being replaced by steam and internal combustion engines), and today used as wind turbines to produce renewable energy or green energy. Everything old is new again!

Hero is believed to have invented the aeolipile. This was the first-recorded steam engine. This device was mounted on a stand, which consisted of a covered cauldron or pot with water inside. On the lid were two tubes extending upward then turned at right angles to support a rotating sphere or drum on its' axis. The sphere would have two right angle nozzles bent in opposite directions. Steam was produced when the pot was heated from beneath. The sphere would become pressurized causing the steam to escape through the nozzles generating thrust. Since the nozzles were pointed in opposite the directions, the sphere would begin to rotate. This demonstrates the 2^{nd} and 3^{rd} laws of motion by Newton, a mechanical couple or torque, and finally reaching steady state.

Al-Jazari (1136-1206)

Al-Jazari gained tremendous knowledge from those who preceded him, improving their ideas to produce better mechanical devices. His own ideas laid the frame work for the future development of mechanical technology and modern mechanical engineering.

Al-Jazari's invention of a double-action suction pump consisting of valves and reciprocating pistons gave rise to the first known irrigation system. This device consisted of suction pipes and pump with a crankshaft-connecting rod mechanism to raise water to a higher elevation. This pump had three significant engineering features. It was the first use of suction in a pipe, it converted rotary into reciprocating motion through the use of the crankshaft-connecting rod mechanism, and utilized the double-acting principle.

Al-Jazari invented an astronomical clock which contained many complex devices. It kept time, displayed the zodiac, solar and lunar orbits, had automatic doors and figurines playing music.

Leonardo da Vinci (1452-1519)

Leonardo da Vinci wrote his Codex Atlanticus which contained over 1,000 pages of sketches and writings concerning his vast scope of knowledge. His great gift of observation and persistence left us with invaluable information laying a future foundation for machines that were produced many centuries later.

He was valued as an engineer and master of mechanisms. Leonardo's breadth of knowledge covers art, anatomy and engineering. His interest in flight led to the design of a vertical flying machine, which today we consider the helicopter.

Leonardo used and understood pulleys and gears, cantilevering and leverage, rack and pinion, linkages, aerofoil, centripetal force, cranks, hydraulics and momentum. His inventions consisted of a lens-grinding machine, musical instruments, and a vast array of complicated war apparatus'. He invented Leonardo's robot, which when later built to his exact specifications, proved to function as a humanoid automation. His relentless desire to create and invent has left us with a lasting legacy which continues in research and development.

James Watt (1736-1819)

Watt's steam engine was developed 50 years after the Newcomen engine. This greatly improved engine consumed 75% less coal making it less expensive to operate. The new breed of steam engine gave incredible mobility to factories since they did not need to be located on rivers or even cities. Watt continued to develop his engine to provide rotary motion (invention of the sun and planet gear mechanism) for driving production machinery, in turn, the Industrial Revolution took off.

Watt continued to redesign and improve his engines by making his steam engine where steam acted alternately on both sides of the piston. He developed the indicator diagram which improved the efficiency of engines. This diagram was a chart of the pressure of stream in the cylinder versus the steam's volume. This enabled him to calculate the work done by the steam, ensuring that all useful energy had been extracted.

As boilers were improved to produce steam at higher pressures more safely, the efficiency of the steam engine increased. With this increased performance and more accurate millwork, the Industrial Revolution was able to produce a vast variety of precision machine tools never before experienced. The technology brought about the advent of the steam ship and the locomotive with the benefit of faster and reliable mobility.

James Watt as an inventor and mechanical engineer gave the world an incredible opportunity launching mechanical systems into an irreversible technological advance.

Mechanical Components and Mechanisms

A mechanism is a set of resistant elements which when connected as a unit will give desired motion when a force is applied. These connections are referred to as kinematic pairs. Kinematic pairs are either a lower or higher pair. Examples of lower pairs are screw and cylindrical joints, hinges, and ball and sockets. Higher pairs are cams and gears, linkages in gimbals and robots.

Mechanical linkages:

A mechanical linkage is a combination of rigid connected links which form a closed chain by two or more joints. A linkage is designed to produce a desired output different from its input by altering the motion, velocity, acceleration, and by applying mechanical advantage. To allow motion between links, these joints have degrees of freedom. Most links are planar, those which take place in one plane. They usually have one degree of freedom. Non-planar or spatial linkages are more difficult to design and are less common.

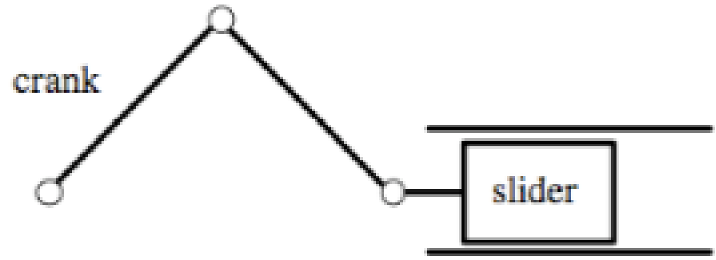
To calculate degrees of freedom for planar (motion confined to a plane) linkages use the Kutzbach-Gruebler equation shown below in its most simplistic terms:

$$m = 3(n-1) - 2j$$

m= mobility = degrees of freedom
n= number of links (including a single ground link)
j= number of one-degree-of-freedom kinematic pairs (pin or slider joints)

There are three types of joints associated with linkages. The **revolute** or pin type which has one degree of freedom. They are rivets, hinges, bolted joints, bearings and bushings. Second is the **prismatic** or slider. The prismatic has one or two degrees of freedom with linear motion. These are comprised of hydraulic cylinders, linear bearings, rollers and pistons. Third is the **spherical** or ball and socket type which has three degrees of freedom rotation.

When each link is treated as a vector and joined into a system of equations forming a loop, they are arranged in a matrix where an equation relates the input and output motion.



The above figure is a slider-crank linkage.

Using the Kutzbach-Gruebler equation, n= 4, j=4 and m=1 This linkage has one degree of freedom.

An **extending linkage** is utilized in equipment such as excavators (backhoes), booms and cranes. These linkages have three main advantages:

1. Compact for transport
2. Quick set-up features
3. Long reach capability during operations

These extending linkages are also referred to as aerial work platforms (AWP), elevated work platforms (EWP) and mobile elevated work platforms (MEWP) and have important uses in plant and facilities maintenance, construction and fire and medical emergencies. They are flexible and mobile and are considered temporary units for as needed bases only. The drive mechanisms can be either hydraulic or pneumatic.

When an aerial work platform can move up and down and also have rotational movement, it is considered to be an **articulated lift**. The name 'cherry picker' comes to mind. Originally used in orchards to prevent damage to fruit trees, these articulated lifts are used in cultivation and management of forest trees, construction, fire emergencies, power transmission and telecommunications industries.

Often the articulated lifts are equipped with tilt and weight sensing devices. When triggered, they can halt or stop the lift from operating. Some AWP's are equipped with a counterweight to maintain stability. These combinations of sensors help to prevent tilting and over loading the platform.

A **scissor linkage** is used in scissor lifts (jack or platform) design. When combined with hydraulics (a hydraulic cylinder), you have a lot of horizontal lift in a relatively small package. When a scissor lift is fully retracted, it occupies a fraction of its extended height. These lifts extend out relatively quickly.

The scissor lift can consist of more than one tier (linkage units) of folding supports arranged in a crisscross ('X') pattern. Remember that the bottom tier must be able to support all the above members, live and dead loads. Limits on the height–to–width ratio when fully extended and any movements in the system **must** be considered.

Cams and cam design:

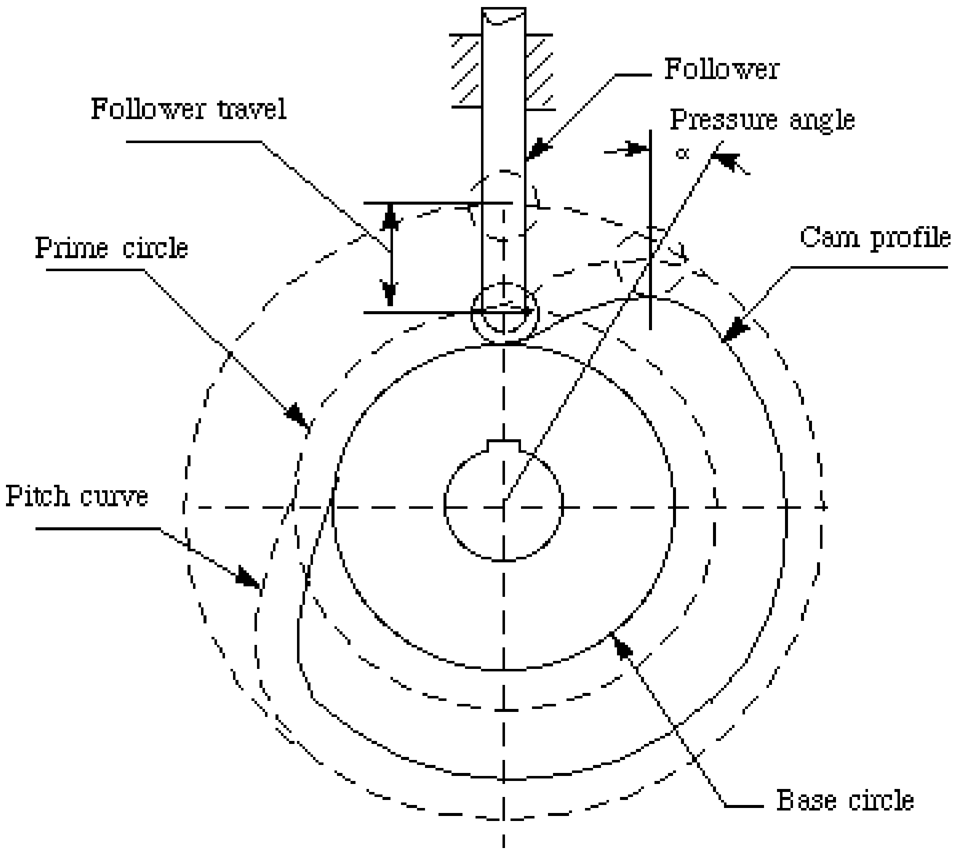
A cam is a rotating plate or cylinder made to specific dimensions which translates motion to a **follower** (converts the angular motion transmitted by the cam) by means of its edge or a groove cut on its surface. The follower must assume a defined series of positions relative to the **driver's** (cam) corresponding series of positions. Or the **follower** must arrive at the proper destination by the time the **driver** arrives at a particular rotational position. One must note the relationship between the follower and cam positions in terms of follower displacement vs. time. As the cam turns or rotates through a cycle, the follower will **rise** (the motion away from the cam center), **dwell** (no motion) or **return** (motion towards the cam center).

There are three basic forms of cam motion. 1.) **Uniform motion**: In constant velocity motion there is equal displacement for equal time (uniform velocity throughout the cycle or stroke). This type of motion causes large forces to occur at the beginning and end of each stroke. Cams rotating at high velocities make this approach unfavorable.

2.)**Harmonic motion**, as the term implies, the follower undergoes simple harmonic motion and the motion is smooth. The initial acceleration is maximum, the mid-point position acceleration is zero and negative maximum at the end.

3.) **Uniformly accelerated and retarded motion**: In constant acceleration motion the velocity increases at a uniform rate during the first half of the stroke and decreases at the same rate during the second half, in turn the follower experiences the smallest value of maximum acceleration along its path. In high- speed machinery, it should be noted that the cam design should avoid sudden shocks at the beginning of the motion and when reversing the direction of the motion of the follower.

Below is a diagram of a cam with references made to cam nomenclature.



* Trace point: A theoretical point on the follower, corresponding to the point of a fictitious knife-edge follower. It is used to generate the pitch curve. In the case of a roller follower, the trace point is at the center of the roller.

* Pitch curve: The path generated by the trace point at the follower is rotated about a stationary cam.

* Working curve: The working surface of a cam in contact with the follower. For the knife-edge follower of the plate cam, the pitch curve and the working curves coincide. In a close or grooved cam there is an inner profile and an outer working curve.

* Pitch circle: A circle from the cam center through the pitch point. The pitch circle radius is used to calculate a cam of minimum size for a given pressure angle.

* Prime circle (reference circle): The smallest circle from the cam center through the pitch curve.

* Base circle: The smallest circle from the cam center through the cam profile curve.

* Stroke: The greatest distance or angle through which the follower moves or rotates.

* Follower displacement: The position of the follower from a specific zero or rest position (usually its the position when the follower contacts with the base circle of the cam) in relation to time or the rotary angle of the cam.

* Pressure angle: The angle at any point between the normal to the pitch curve and the instantaneous direction of the follower motion. This angle is important in cam design because it represents the steepness of the cam profile.

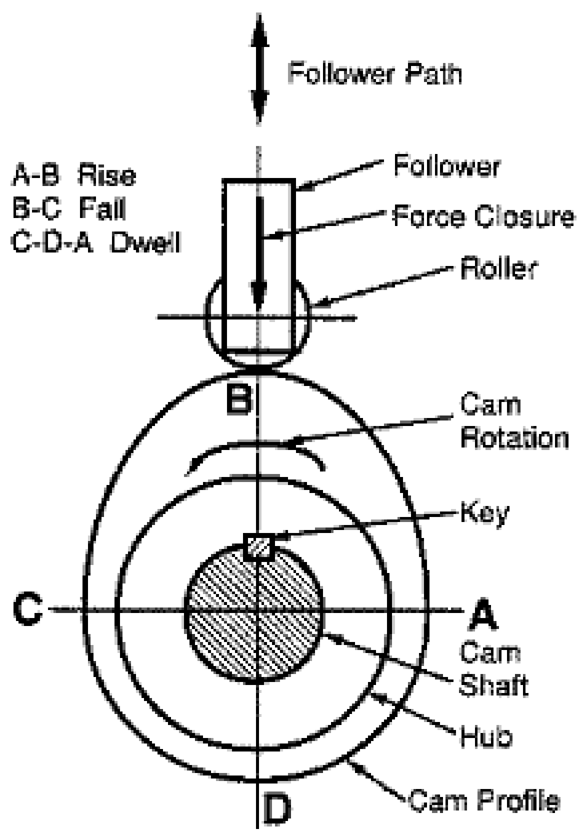
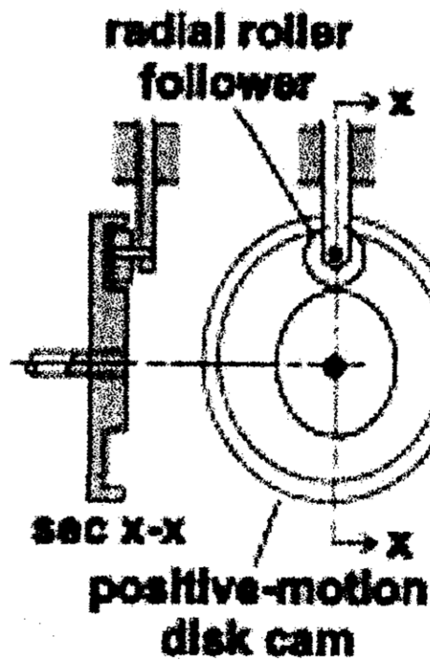
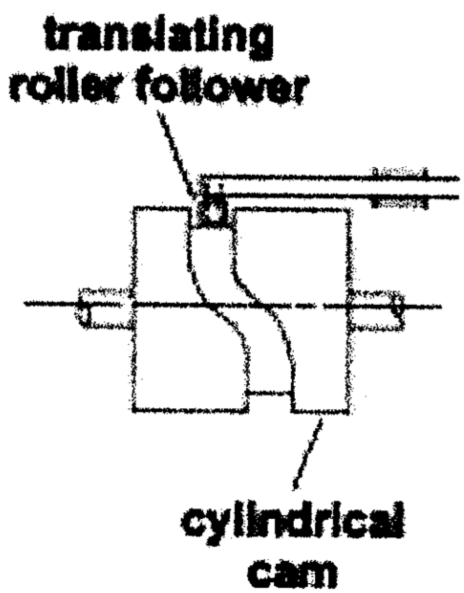


Figure shows cam follower rise, fall and dwell relationship to the rotation of cam.

Typical follower motions are dwell-rise-dwell-fall **(DRDF)** in which the follower is held stationary at both limits of travel and dwell-rise-fall-dwell **(DRFD)** in which the follower velocity reverses immediately when it reaches the maximum displacement. You should identify alternative motions such as ones including a constant-velocity motion or requiring a specific displacement at a given cam angle.



The figure above shows a cam with a groove cut on its surface. The follower will move relative to this cut out or profile.



The last figure is a cylindrical cam. The groove is machined on the outer surface with a contour for the follower to transfer the rotational motion of the rotating cam into a horizontal reciprocating motion. Remember that the **follower** is the output link which remains in constant contact with the geometry of the cam profile.

A common cam mechanism is the cam configuration in an automobile engine which lifts and lowers (opens and closes) the intake and exhaust values. As the camshaft (the **cam** being part of the shaft) rotates, it controls the reciprocating (up and down) motion controlling the **follower**. Cams are also used in packaging equipment, compressors and machine tool equipment.

Gears:

A gear is a component in a mechanism such as a power train, transmission or power reducer which transmits rotational torque by applying a force to the teeth of another gear. When designed and assembled properly, a geared arrangement can transmit forces at various velocities and torques and can change direction from the power source. The gear can mesh with any other gear or device which has compatible teeth. When two or more gears of unequal size are joined, they produce a mechanical advantage. That is, the rotational speed and torque of the second gear is different from that of the first . The velocity, also known as, rotational speed is expressed in revolutions per second, revolutions per minute or radians per second.

Gear ratio can be expressed by using the following approaches. The rotational speed and the number of teeth of one gear relates to the speed and number of teeth on the other. The rotational speed is directly proportional to the gear's circumferential speed divided by its radius. When two gears are meshed, the smaller gear turns more revolutions in a given period of time (faster) than the larger wheel. Equating the two, the speed ratio is the reciprocal ratio of the number of teeth on the two gears.

Speed 1 x number of teeth in 1 = Speed 2 x number of teeth in 2

Since the number of teeth is proportional to the circumference of the gears, the gear ration can be expressed as follows:

$$\text{Gear Ratio} = d/D = r/R = \omega_D / \omega_d$$

Where D= large gear diameter, d= small gear diameter, r= small radius, R= large radius, ω_D = angular velocity in large gear and ω_d = angular velocity in smaller gear

The gear ratio can also be referred to as the mesh ratio, m_G. The mesh ratio is the number of teeth in a meshed pair, expressed as a number greater than 1.

$$m_G = N_G / N_p$$

Where: m_G = the mesh ratio, N_G = number of teeth in gear, N_P = number of teeth in pinion.

Example: A pinion on an electric motor shaft has 20 teeth and the gear has 30 teeth. Since $m_G = N_G / N_p$ = Number of Teeth in Gear/ Number of Teeth in Pinion

= 30/20 = 1.5

One can see that the mesh ratio is 1.5

The pinion is the component having the lesser number of teeth. For spur and parallel-shaft helical gears, the base circle ratio must be identical to the gear ratio. The speed ratio of gears is inversely proportionate to the numbers of teeth. Only for standard spur and parallel-shaft helical gears is the pitch diameter equal to the gear ratio and inversely proportionate to the speed ratio. The speed (velocity) ratio:

$$V_r = N_{driver} / N_{driven}$$

(V_r) velocity ratio = (N_{driver}) number of teeth in driver or pinion gear divided by (N_{driven}) number of teeth in driven gear

Gears are grouped according to tooth form, shaft arrangement, pitch and quality. Below, gears are arranged by tooth form and shaft arrangements.

Tooth Form	Shaft Arrangement
Spur	Parallel
Helical	Parallel or skew
Worm	Skew
Bevel	Intersecting
Hypoid	Skew

Gear definitions and nomenclature:

P_d = diametral pitch

p = circular pitch

p_b = base pitch

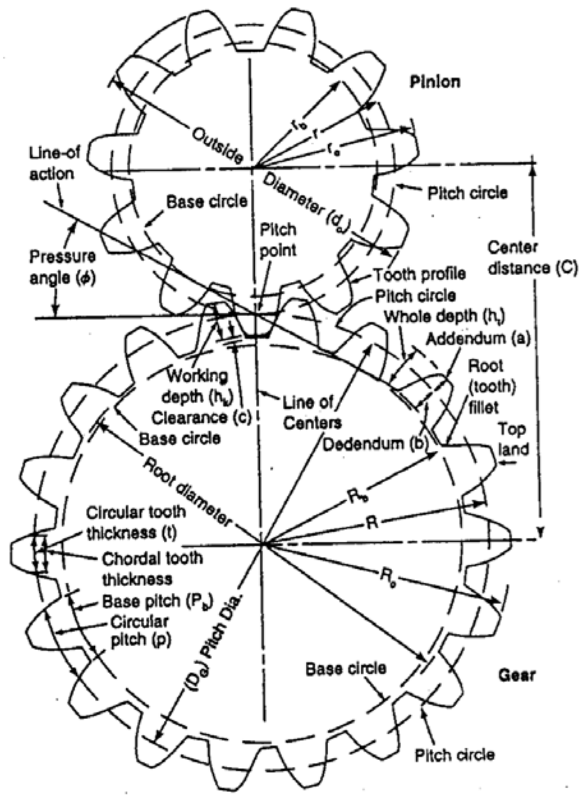
a = addendum, b = dedenum, B = backlash

C = center distance

d_0 = outside diameter of pinion, D_O = outside diameter of gear

ϕ = pressure angle

The following diagram will further illustrate gear terminology.



The Diametral pitch P_d is the ratio of number of teeth in the gear to the diameter of the pitch circle D measured in inches, P_d = N/D. The circular pitch p is the linear measure in inches along the pitch circle between corresponding points of adjacent teeth. P_d p = π The base pitch p_b is the distance along the line of action between successive involute tooth surfaces. The base and circular pitches are related as:

p_b = p cos ϕ where ϕ = the pressure angle

The pitch circle is the imaginary circle that rolls without slippage with a pitch circle of a mating gear. The pitch (circle) diameter equals D = N/ P_d = Np/ π

The base circle is the circle from which the involute tooth profiles are generated. The relationship between the base circle and pitch circle diameter is:

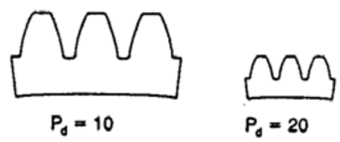
$$D_b = D \cos \phi$$

The tooth size is related to pitch. A large P_d implies small teeth and small P_d implies large teeth because the relationship with the diametral pitch is inverse. The opposite exists with tooth size and circular pitch where a small tooth has a small p but a large tooth has a large pitch. A gear tooth configuration is considered coarse when the diametral pitch P_d is less than 20 and fine when the P_d is 20 or greater.

The figure below shows a comparison of two different pitch and tooth sizes:

The gear to the left has a P_d (Diametral pitch) equal to 10

The gear to the right has a P_d equal to 20



Looking at a previous relationship of D_p and p, circular pitch which is:

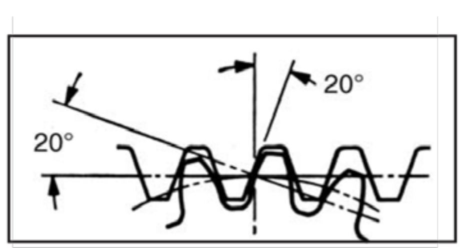
$$P_d \, p = \pi$$

Then the gear with the P_d = 10 and the other gear where P_d = 20

The p (circular pitch) = π / P_d = $\pi/10$ =3.14/10 = 0.3142

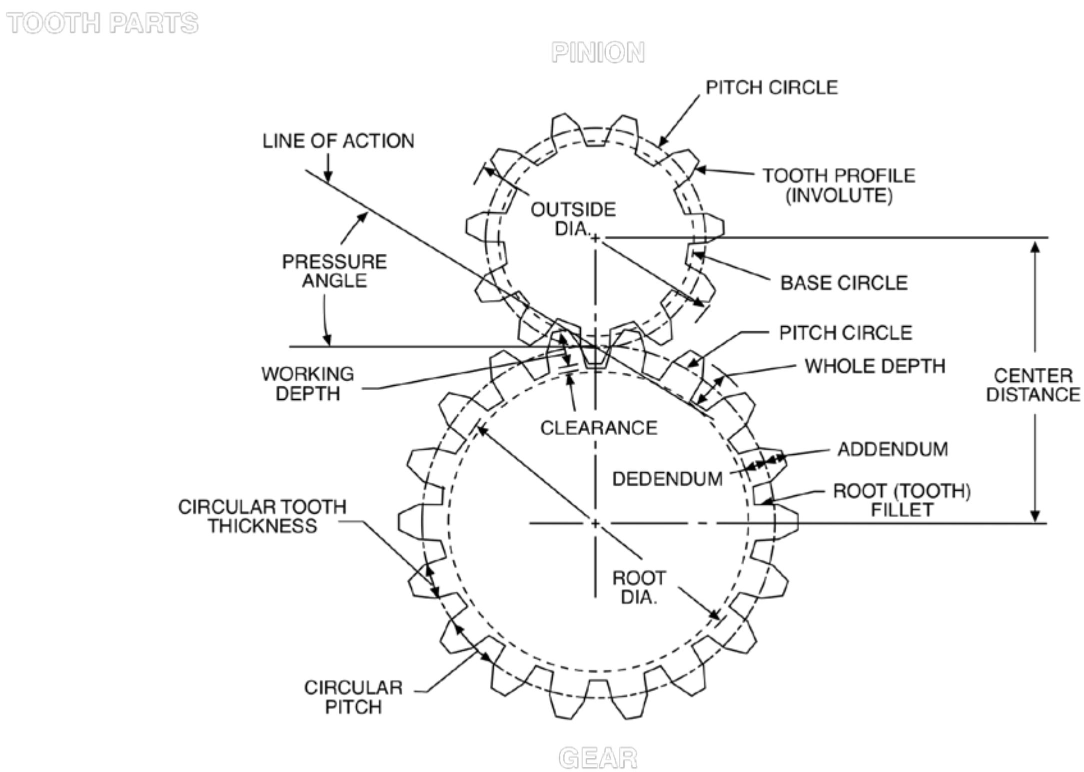
For the finer gear tooth configuration the p = 3.14/20 = 0.1571

The pressure angle ϕ for most standard gears which have been adopted by the ANSI and gear manufacturers are pressure angles of 14 1/2°, 20° and 25°. The 20° gear being the most versatile is used more frequently.



The pressure angle ϕ for all gear types is the acute angle between the common normal to the profiles at the contact point and the normal pitch plane.

Focusing more closely on the pressure angle ϕ and with the aid of the next figure in larger detail, one can address **tooth proportion**.



Tooth proportions are established by addendum (the height by which a tooth projects beyond the pitch circle), dedendum (the depth of a tooth space below the pitch line). For gear clearance in mating parts, the dedendum is usually greater than the addendum. Tooth proportions are also established by working depth, clearance, tooth circular thickness and pressure angle.

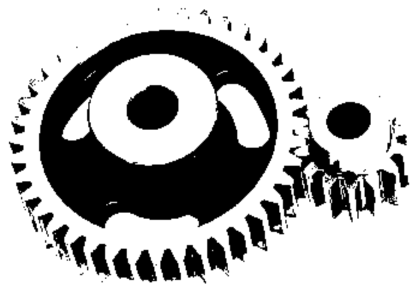
The next table shows tooth proportions of various standard diametral pitches for spur gears.

Diametral Pitch	Circular Pitch (Inches)	Thickness of Tooth on Pitch Line (Inches)	Depth to be Cut in Gear (Inches) (Hobbed Gears)	Addendum (Inches)
3	1.0472	.5236	.7190	.3333
4	.7854	.3927	.5393	.2500
5	.6283	.3142	.4314	.2000
6	.5236	.2618	.3565	.1667
8	.3927	.1963	.2696	.1250
10	.3142	.1571	.2157	.1000
12	.2618	.1309	.1798	.0833
16	.1963	.0982	.1348	.0625
20	.1571	.0785	.1120	.0500
24	.1309	.0654	.0937	.0417
32	.0982	.0491	.0708	.0312
48	.0654	.0327	.0478	.0208
64	.0491	.0245	.0364	.0156

The **quality** of a gear is classified as commercial, precision and ultraprecision.

Spur Gear

The most basic and common gear is the spur gear. The teeth project radially and are straight and align parallel to the axis of rotation. They are mounted on parallel shafts and can run at high speeds and are efficient. All meshing of gears have **backlash**. Backlash allows clearance between gears to help prevent noise, abnormal wear, heat and allows for better lubrication. Backlash does not adversely affect proper gear function except for loss of motion when reversing gear rotation. Backlash is also a result of design tolerances and fabrication. Spur gears are in many mechanical systems such as automatic packing machines, film– cutting machines and spindle and feed controls in milling and turning machinery.



Spur Gear

Helical Gear

Helical gear teeth are cut at a 45° angle to the bore (axis hole). This type of gear is used to connect non-intersecting shafts. Because of the helix angle, the meshing teeth have more surface contact. This permits the gears to run more quietly and smoothly giving it an advantage over the spur gear configuration. When helical gears mesh, due to this angle, a thrust load is created. Bearings are used to support this thrust load. Due to their complex shape, they are more expensive than spur gears and slightly less efficient.

Helical gears are used to connect parallel shafts or skew shafts at 90°. When used in a parallel configuration, the helical gears the same pitch, pressure angle and helix angle. They must connect with opposite hands (a left and a right hand gear). When connected as skew shafts, they are referred to as crossed-axis helical gears. They are used in conveyers, elevators, compressors, blowers, textile, plastic and rubber industries. They are used in transmission because of their durability and perform quieter at higher speeds.

Helical Gear

Charts, tables and gear design formulas are available for various gear types in mechanical engineering handbooks, from gear manufacturers, American National Standards Institute (ANSI) and American Gear Manufacturers' Association (AGMA)

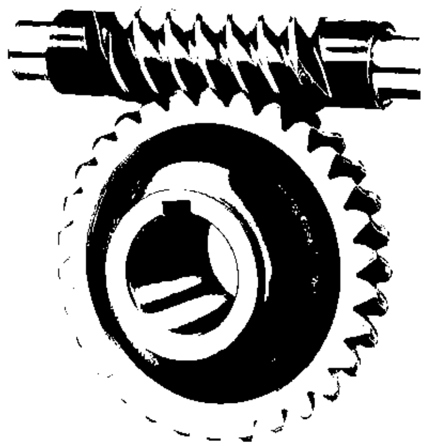
Worm Gear

Worm gears are used for obtaining large speed reductions or allow greater torque to be transmitted between non-intersecting shafts of 90° to each other. The worm portion is an ACME like thread cut around the shaft or a cut sleeve mounted to a shaft and a face width which is usually larger than its diameter. The meshing worm gear is also cylindrical and mounted on a shaft. It has a much larger diameter with a relatively smaller face. The worm usually drives the worm gear. Worm gears for power transmission purposes should have the worm (the **driver**) fabricated in steel and the worm wheel or gear (the **driven** part) fabricated in phosphor bronze. The worm should be hardened and ground to a fine surface finish. The contact between the worm and the gear is sliding, so the efficiency may be only 30 to 50%.

A unique feature in some designs is the worm mesh nonreversibility which occurs because of the large amount of sliding rather than rolling in this type of gearing. This is due to a more uniform distribution of tooth pressures on the assembly. For a given coefficient of friction, there is a critical value of lead angle below which the mesh is nonreversible. This is generally 10° and lower but is related to the materials and lubrication. This locking feature is helpful in conveyor systems where it acts as a brake when the motor is not turning.

Worm gears are used in material handling, packaging and food processing applications, conveyors and elevators. When using the duplex worm drives with adjustable backlash, this high–precision application is useful in precision Quality

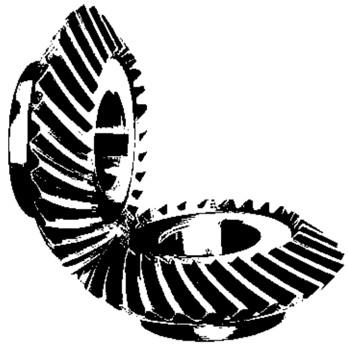
Control measuring systems, CNC turning, vertical and horizontal machining centers and other machine tools which require precision accuracies.



Worm Gear

Bevel Gear

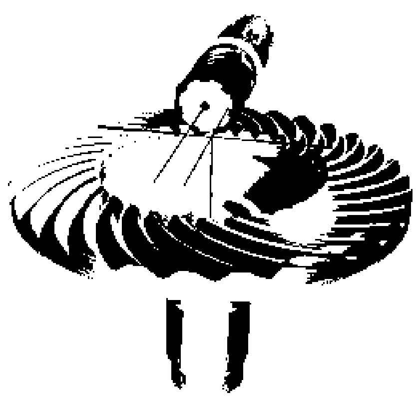
Bevel gears are used to connect two intersecting shafts in any given speed ratio. When the mating gears are of equal size (equal number of teeth) and are positioned at 90°, they are called miter gears (1:1 ratio). The teeth on bevel gears can be cut as spirals as opposed to straight in order to run more smoothly and quiet. If the teeth are spiral, the pinion and gear must be of opposite hand. Because bevel gears are used to reduce speed, the pinion is always smaller (has fewer teeth) than the gear. Even though bevel gears usually intersect at 90°, those that do not are referred to as angular bevel gears. They are used in differential drives, the main mechanism in a hand drill and gas turbine engines.



Bevel Gear

Hypoid

The hypoid gear unlike other gears can engage with axes in different planes. The basic pitch rolling surfaces are hyperbolas of revolution (revolved hyperboloid). The hypoid gear places the pinion off-axis to the ring gear allowing the pinion to be larger in diameter having more surface area. The pinion and ring gear are typically arranged as opposite hands and are both hypoid. This arrangement allows the translation of torque of 90° and is used in many car differentials. Because of high tooth pressures and high rubbing speeds, hypoid gearing requires special lubrication.



Hypoid Gear

In conclusion, in order to prevent premature gear failure, careful attention must be given to gear tooth geometry, tooth action, tooth pressures, surface finishes and quality (machining and fabrication of parts), assembly (precise alignment of parts, clearances, fits and tolerances), contamination, operating environment and proper choice of lubricants and lubrication systems.

Gear Lubrication

A vast variety of gear lubricants have been developed due to the types of gear systems, applications and operating conditions. The main purpose of the lubricant is to lower friction and the heat generated by the mechanical system in operation. The proper selection of a lubricant comes under the following criteria:

1. Gear type and materials
2. Speed (rpm) and corresponding sliding velocities
3. Tooth contact and load
4. Environment
5. Temperatures
6. Type of lubrication system
7. Seals
8. Type of service

There are three methods of gear lubrication:

1. Grease lubrication
2. Splash lubrication, also referred to as oil bath lubrication
3. Forced oil circulation lubrication

The **grease** lubrication method is suited for low speed operations where the lubricant must remain on the gear teeth in order to reduce tooth contact friction. Remember, the grease application will not help in reducing heat being generated by the system. The grease method is not suitable for high load or continuous operation. Too much grease used in an open gear system is impractical but in a closed gear system, excessive grease can cause agitation, viscous drag and results in power loss.

The **splash** lubrication method is used in enclosed gear system. The gears are bathed in and supplied by the oil reservoir. As the gears rotate, they also lubricate (splash) on the adjacent mechanical parts (bearings, seals and mounts). The oil reservoir must be maintained at the proper level in order to insure there is adequate oil during system operation. Excessive oil in the gear system will cause agitation; not enough oil will cause a rise in temperature causing lower viscosity of the lubricant, accelerated degradation, deformation of the mechanical system and decreased backlash.

The **forced** (pressurized) oil circulation method utilizes a pressure system (pump system) to apply the lubricant. The enclosed gears must be kept dirt and dust free. The **drop** method is where the pumped lubricant drops directly on the gear mechanism. The **spray** method uses pressurized lubricant directly on the gear contact area. The **oil mist** method has the lubricant mixed with compressed air. This is applicable for high–speed and large gearbox gearing. The forced lubrication system is comprised of a pump, piping, oil reservoir, filter and other devices. This method is considered the best way to lubricate gears because filtering and circulating and in some circumstances cooling the lubricant can be maintained at a manageable cleanliness and viscosity.

Bearings

A bearing is a mechanical device which supports two or more moving parts. The primary purpose is to reduce friction. Bearings are made to support radial, thrust or a combination of radial–thrust loads.

Plain bearings are the simplest and have three classifications. The **journal** or radial bearing is round or cylindrical in shape and supports a rotating shaft. The **thrust** bearing supports axial loads on a rotating member and restricts the lengthwise motion of the shaft. The **guide** or slipper bearing is one which supports lengthwise or straight sliding motion.

Some applications take place where the relative motion or sliding contact between the parts is pure sliding, without the benefit of lubrication. This exists as **dry** lubrication in teflon and nylon bushings. Most plain bearing applications are a polished, hardened steel shaft rotating in a bronze (soft) or oil impregnated material supporting the hardened journal. A second motion is **hydrodynamic** lubrication which depends on a film build-up where there is a partial or full separation of the rotating surfaces, and third, **hydrostatic** lubrication where the lubricant is introduced under pressure between the joining surfaces causing a force in opposition to the applied load and a lifting or separation of these surfaces. The last type can be a combination of hydrodynamic and hydrostatic lubrication.

The advantages of plain bearings are:

- Require less space
- Run quieter
- Cost less
- Have greater rigidity
- Their life is generally not limited by fatigue

The disadvantages are:

- Higher frictional properties meaning less efficient
- Failure due to contamination of lubricant
- More stringent lubrication requirements
- Susceptible to damage from interrupted lubrication supply

Efficiency of mechanical parts as well as power output is drastically lowered when lubrication is not used. The table below shows how the coefficient of friction changes from a clean to a lubricated system.

Coefficients of Static Friction for Steel on Various Materials

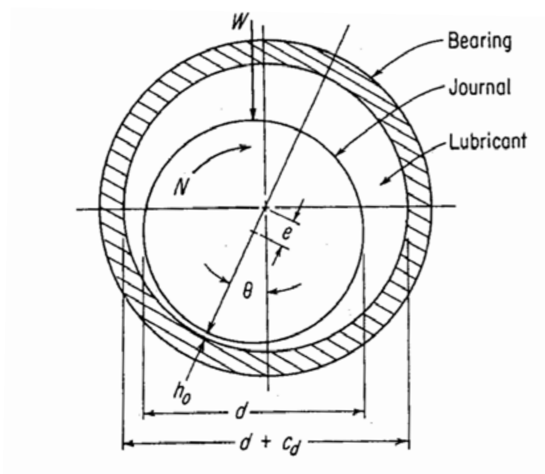
Material	Coefficient of Friction	
	Clean	Lubricated
Steel	0.8	0.16
Copper-lead alloy	0.22	
Phosphor bronze	0.35	
Aluminum bronze	0.45	
Brass	0.35	0.19
Cast iron	0.4	0.21
Bronze		0.16
Sintered bronze		0.13
Hard carbon	0.14	0.11–0.14
Graphite	0.1	0.1
Tungsten carbide	0.4–0.6	0.1–0.2
Plexiglas	0.4–0.5	0.4–0.5
Polystyrene	0.3–0.35	0.3–0.35
Polythene	0.2	0.2
Teflon	0.04	0.04

Plain bearings have a 95 to 98 percent efficiency when properly lubricated. The viscosity and adhesion of the lubricant to the journal and bearing are dependent upon proper lubrication. The journal and bearing form a radial clearance when assembled. This causes a wedge-shaped film between both elements. A hydrodynamic pressure created in the film is sufficient to float the journal and therefore, carry the applied load. Under perfect lubrication, the two surfaces are not in contact.

The three main components in this bearing system are the journal, bearing (bushing) and lubricant. The lubricant is considered incompressible. The entire system depends on the viscosity, speed of the bearing elements and the geometry of the film.

The minimum clearance of the lubricating film between the journal and bearing is the minimum film thickness h_0. This minimum clearance is dependent upon the surface finishes and rigidity of the journal and bearing. In most cases as in electric motors and generators operating at medium speed; h_0 = 0.003 inches to 0.005 inches. For large steel journals operating at high speed with pressure supplied lubrication; h_0 = 0.0001 inches to 0.0002 inches.

The following diagrams further describe the components of the plain bearing configuration and the hydrodynamic pressure profile.



W = applied load

N = revolution

e = eccentricity of journal center to bearing center

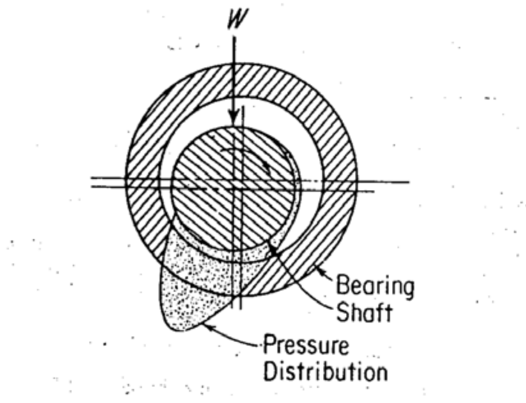
θ = attitude angle, angle between W(applied load) and point h_0

d = diameter of shaft

c_d = bearing clearance

d + c_d = diameter of bearing

h_0 = minimum film distance



Plain Journal Bearing (Side View)

Other design factors to consider are: the relationship between the eccentricity ratio (eccentricity to radial clearance) and the load–carrying coefficient, side leakage, operating temperatures, allowable mean bearing pressures, length–diameter ratios, clearance between journal and bearing, coefficient of friction, heat dissipation, materials, lubricant and lubrication systems.

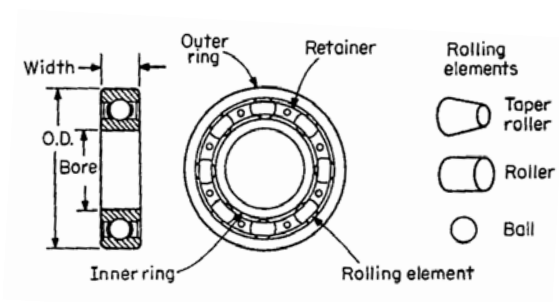
Ball and Roller Bearings

Ball bearings (rolling–contact) use hardened spheres (balls) to support both radial and axial loads in rotating and stationary members. With a minimum of friction, they permit relatively free rotation. The inner and outer ring of the bearing is usually made of 52100 AINI steel, hardened to Rockwell C 60 to 67 with a micro–finished ground raceway. The bearing consists of the following elements: the two rings previously stated, the rolling elements and the cage holding the individual rollers.

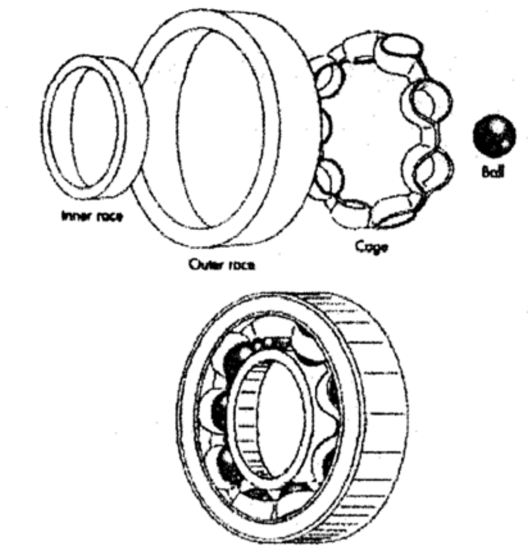
Most often the rollers (balls or rollers) are manufactured under the same requirements as the inner and outer rings. Special applications may call for materials like stainless steel, Monel, ceramics or plastics. Bearings which use ceramic balls and steel rings are classified as a hybrid. They last longer in extreme conditions.

Cages in the bearing retain the balls and rollers in place within the two rings. The cage separates and retains the rollers in a uniform position. The most common style cages are pressed–steel, riveted or clinched and filled nylon. When higher speed and greater strength is required, solid machined cages are made. Bronze or phenolic–type material is also used. The latter, when running at high speeds is quieter with a minimum amount of friction. Full–complement bearings have no cages.

The figure below describes the roller bearing terminology:



This next figure shows the exploded view and the finished ball bearing:



Shown is the inner race, outer race, cage and ball above finished bearing.

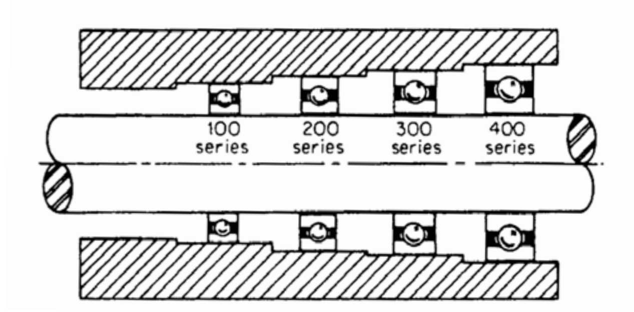
Manufacturers of rolling–contact bearings have to follow specific specifications in order to standardize production and have interchangeability. These boundary conditions consist of bore diameters, outside diameters, width and precise manufacturing tolerances. Originally, these standards were promulgated by the Anti–Friction Bearing Manufacturers Association (AFBMA). In 1993, the title was changed to the American Bearing Manufacturers Association (ABMA). The updated standards are coordinated with the American National Standards Institute (ANSI). The current standards are specified as the ABMA/ANSI standards list. These standards are represented below:

Standard Number	**Description**
1	Terminology of bearings and parts
4	Tolerances and gaging practices
7	Mounting dimensions
8.2	Mounting accessories

9	Load ratings and fatigue life (ball)
10	Metal ball standards
11	Load ratings and fatigue life (roller)
12	Instrument ball bearing (metric/inch)
13	Vibration and noise measuring
20	Basic boundary dimensions

The above standards were prepared by the Bearing Technical Committee which is sponsored by the ABMA with approval of the ANSI. For established requirements in progressive levels of precision of ball bearings, the designation of ABEC–1, ABEC–5, ABEC–7 and ABEC–9 are used to establish and specify standards for tolerances for bore (inside diameter), outside diameter, width (thickness) and radial runout (eccentricity). ABEC–9 is the most precise for ball bearings. RBEC–1 and RBEC–5 are for **roller** bearings. Precision levels above ABEC–1 and RBEC–1 are required due to precision fits on shafts and bearing supports, to reduce runout and to allow mechanical systems to operate at increasingly higher speeds.

The selection of rolling–contact bearings embodies many variables and design strategies since several types are available. Each type is manufactured in several standard "series". The following figure will show that the bore (inside diameter of the bearing) has the same idententical size, but that the differences are the outside diameter, width and ball size which become progressively larger. These requirements are due to various load–carrying capacities for various size shafts, which gives the designer much needed flexibility in selecting standard–size interchangeable bearings.



Types of standard bearings:

Ball Bearings

The **Single–Row Radia**l (Non–filling slot) is shown in a previous figure and is the simplest. This ball bearing is referred to as the deep groove or Conrad bearing and is commercially available with many variations. This bearing is symmetrical and is usually used for radial and moderate thrust loads (maximum two–thirds of radial). Precise alignment of shaft and bearing bore is essential because this type of bearing is not self–aligning.

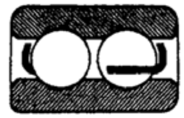
Single–row–radial

The **Single–Row Radial** (Filling slot) is mostly used for radial loads. This bearing is assembled with as many balls as possible, which can carry a heavier radial load. This type of bearing can withstand a thrust load as well as radial if the thrust load is 60 percent or less than the radial. This type of bearing is also referred to as a **maximum capacity** bearing.

The **Single–Row Angular–contact** is designed to carry radial as well as thrust loads, where the thrust component is large and axial deflection is light. The outer ring's one shoulder is made higher than the other to absorb the additional thrust. Except where used for a pure thrust load in one direction, this type of bearing is applied either in pairs or one at each end of a shaft, opposed.

Single–row–angular

The **Double–Row** bearing consists of two rows of balls. This type is designed for heavier radial loads without increasing the outside diameter. The width (thickness) is 60 to 80 percent larger than the comparable single–row. Thrust loads must be light due to the filling slot.



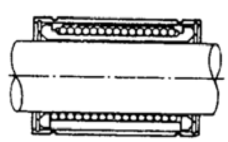
Double–row

The **Internal Self–Aligning Double–Row** is used basically for radial loading where self–alignment (± 4°) is required. Caution should be noted: Any **excessive misalignment** or thrust load (10 percent of radial) will cause early failure.

Internal Self–Aligning Double Row

The **Double–Row Angular–contact** is assembled as one unit consisting of two single–row angular–contact bearings assembled as one unit. They support combined radial and thrust loads or heavy thrust loads dependent on the contact–angle magnitude. They are referred to as duplex bearings when mounted in pairs. Pairs can be mounted back–to–back, face–to–face or in tandem. These bearings are usually preloaded which maximize resistance to deflection in the shaft and axial movement under combined loads with thrust from either direction.

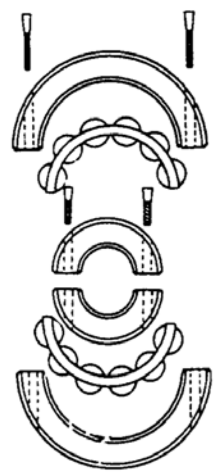
The **Ball Bushing** is a type of bushing with rows and lines of small balls set in a sleeve. These are primarily used for linear motion and mounted on hardened shafts (Rockwell C 58–64). They are not recommended for rotary motion. The figure below further clarifies the description.



Ball Bushing

The **Split–Type Ball Bearing** is cut in half or two symmetrical sets of pieces. It can be manufactured as a ball or roller bearing consisting of a split cage, split inner and outer rings. The bearing is held together as one unit by two screws in each half–cut inner and outer ring. This bearing is more expensive but is used when installing or removing a bearing may be difficult.

The Split–Type ball bearing is shown on the next page.

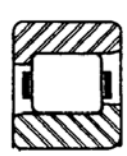


Split–Type Ball Bearing

Roller Bearings

In roller type bearings, the rolling element is designed in a roller (cylindrical shape) rather than a ball or sphere. Their rollers and raceway designs are suited for axial, combined axial and thrust, or thrust loads.

The **Cylindrical Roller** is by definition cylindrical (approximate length to diameter ratio ranging from 1:1 to 1:3) for the rolling element configuration. Used for heavy radial loads and for free axial movement in the shaft.



Cylindrical Roller

The **Barrel Roller** bearing's rolling element is barrel–shaped and cylindrical. They are made as single–row and double–row mountings. As in the cylindrical roller bearing, the single–row mount has a low thrust capacity. When made as an **angular** mount and as a double–row mount, this type can resist combined axial and thrust loads.

The **Spherical Roller** is made as a double–row, self–aligning mounting. It is barrel–shaped with one end of the roller smaller than the other. This provides small thrust, keeping the rollers in contact with the center guide flange. This type can carry high radial and thrust loads and even maintain a slight misalignment of the shaft and bearing housing. Because of their internal self–aligning feature, it is used in many HVAC fan applications.

Spherical–roller

The **Tapered–Roller** is designed for all the elements on the rolling surface and mating raceways to intersect at a common point on the axis. When adjusted for a pre–load, this will produce maximum system rigidity. They are available in double–row design and are excellent for carrying heavy radial and thrust loads.

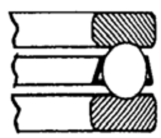
Tapered–roller

The **Needle Bearing** has relatively small rollers, usually not larger than ¼ of an inch in diameter and usually a high length to diameter ratio ranging from 6:1 to 10:1. They are made with or without an inner raceway and do well when space is an issue. If the shaft is used as the inner race, it must be hardened and the shaft must have a ground finish. These bearings **cannot** support thrust loads but in the full–compliment type can be used for high loads, oscillating or slow speeds. The cage type should be used for rotational motion.

Needle Bearing

Thrust Bearings

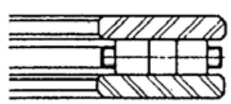
Thrust bearings are designed to carry thrust loads only. In some cases they can take thrust and radial loads combined. They can carry radial loads only in combination with other bearings taking the radial load. The **Ball Thrust Bearing,** also referred to as a one–direction ball thrust, consists of a shaft ring and a flat or spherical housing ring with a single row of balls between. They can carry only thrust loads in one direction and cannot carry any radial loads. The ball thrust bearing is used for low–speed applications.



Ball Thrust Bearing

The **Two-Direction Ball Thrust** has a shaft ring with grooves and balls on both sides and two housing rings; therefore, thrust loads in both directions can be supported. This type bearing cannot carry radial loads.

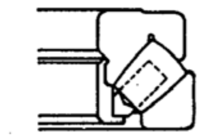
The **Straight– Roller Thrust Bearing** is made with short rollers banked together in a series which minimizes skidding which would cause the rollers to twist. These bearings are used for moderate speeds and loads.



Straight–Roller Thrust Bearing

The **Spherical–Roller Thrust** bearing is similar in design to the radial spherical roller except this has a larger contact angle. The roller shape is barrel and one end of the roller is smaller than the other. This design is good for very high thrust loads and can accommodate radial loads as well.

The **Tapered–Roller Thrust** bearing has straight tapered rollers and several different arrangements of housing and shaft are used. This design also eliminates the problem of skidding but causes a thrust load between the ends of the rollers and the shoulders on the race. Because of sliding contact between the roller end and flange, speeds are limited.

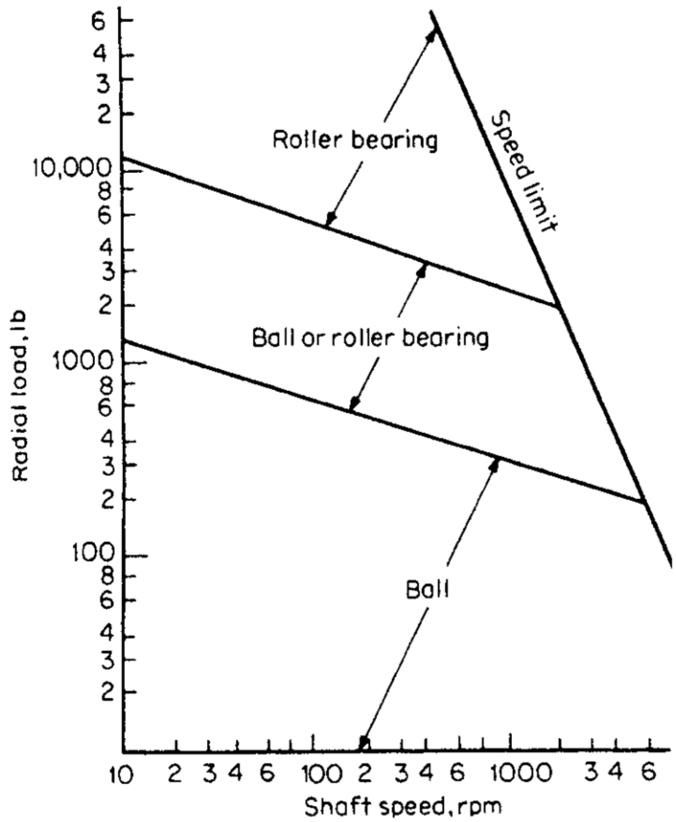


Tapered–Roller Thrust Bearing

There are several important factors in choosing the proper type of roller bearing to use. These factors consist of types of loads, speed, environment (space constraints), misalignment issues and precise shaft positioning. Choices between using a ball or roller as the rolling bearing come under certain general rules:

1. Contact points are the theoretical basis in ball function; therefore, round or spherical members will be suited for higher speeds and lighter loads than roller bearings.
2. Roller bearings function theoretically on a line contact. Since loads are distributed, they have a heavier load capacity and better shock tolerance, but are limited in speed. They are more expensive than roller bearings.

The figure below will further aid in the selection process of ball versus roller and is based on a rated life of 30,000 hours.



The **bearing rated life** formula is used to predict the statistical life of a bearing and is based upon the number of revolutions or hours at a given constant speed. The statistic shows that 10 out of 100 bearings will fail before rated life. These formulas are based on an exponential relationship of load to life. The **Rated Life** L_{10} is also referred to as **Rating Life** L_{10} which is the life associated with 90 percent reliability.

$$L_{10} = \left(\frac{C}{P}\right)^K \times 10^6$$

where L_{10} = rated life, revolutions
C = basic load rating, lb
P = equivalent radial load, lb
K = constant, 3 for ball bearings, 10/3 for roller bearings

To convert to hours of life (L_{10}), this formula becomes

$$L_{10} = \frac{16{,}700}{N}\left(\frac{C}{P}\right)^K$$

where N = rotational speed, r/min

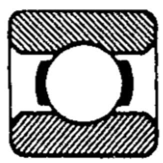
The following table will guide you in an overview of design lives versus the application.

Application	Design life, h, L_{10}	Application	Design life, h, L_{10}
Agricultural equipment	3,000–6,000	Domestic appliances	1,000–2,000
Aircraft engines	1,000–3,000	Electric motors:	
Aircraft jet engines	1,500–4,000	Domestic	1,000–2,000
Automotive:		Industrial	20,000–30,000
Bus, car	2,000–5,000	Elevator	8,000–15,000
Trucks	1,500–2,500	Fans:	
Blowers:	20,000–30,000	Industrial	8,000–15,000
Continuous 8-h service	20,000–40,000	Mine ventilation	40,000–50,000
Continuous 24-h service	40,000–60,000	Gearing units (multipurpose)	8,000–15,000
Continuous 24-h service (extreme reliability)	100,000–200,000	Intermittent service	8,000–15,000
Compressors	40,000–60,000	Paper machines	50,000–60,000
Conveyors	20,000–40,000	Pumps	40,000–60,000

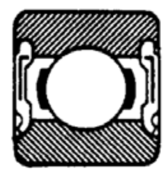
Bearing Closures

Bearings come in three types of closures:

1. Open
2. Shielded
3. Sealed



Open Bearing



Shielded and Sealed Bearing

Shielded and sealed bearings are used to retain lubricant and to prevent contamination from entering the bearing. In the shielded bearing the clearance is small between the stationary shield and rotating ring. This is effective in keeping dirt out without an increase in friction.

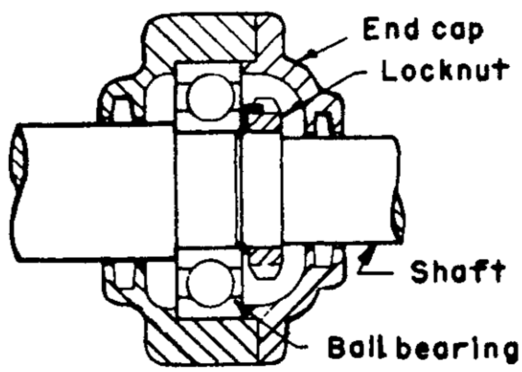
The sealed bearing has a flexible lip (usually rubber) that comes in contact with the inner ring. Friction is increased by this but the sealed type is more effective in retaining lubricant and further protecting the bearing from dirt intrusion.

Mounting of Bearings

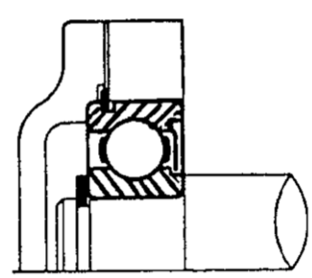
In order to ensure proper bearing life (rated life), correct mounting of the roller unit is essential. To choose the proper mounting application these factors must be considered:

1. Accuracy (Tolerance requirements)
2. Load
3. Speed
4. Cost (Level of desired quality)

The most recommended method for properly securing a bearing is a press fit against a shaft shoulder held in place by a locknut. This fit tends to be more of a slip fit versus a tight press fit (allowances for thermal expansion). The shoulder is where the bearing seats should have an undercut. The bearing should be set in place by the locknut with end caps to secure the bearing against the housing shoulder. Refer to the figure on the next page as a reference:



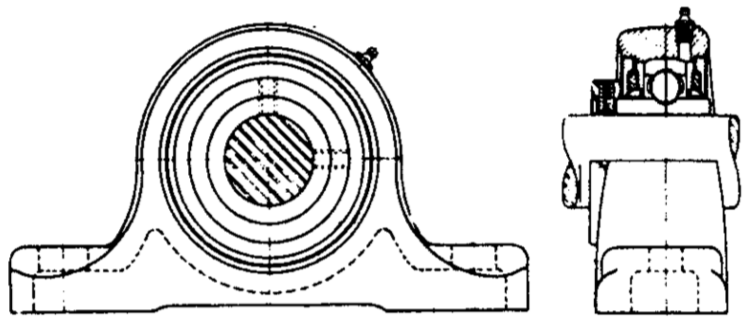
Retaining rings are used to secure a bearing on a shaft or in a housing. In the figure below, the retaining ring (also referred to as a snap ring) is located on the lower left side of the bearing which snaps into the cut groove in the shaft.



Ball bearing being secured by a retaining ring

Each shaft assembly is considered a unit and must allow for expansion by permitting one end to float.

A convenient method for securing a bearing for fan and conveyor applications is the use of the ball–bearing pillow blocks illustrated on the next page, which are commercially available.



Pillow block assembly (Front and Side View)

Bearing Lubrication

Lubricants serve several functions:

1. To obtain or exceed the bearing's rated life
2. Provide lubrication to the rolling and sliding elements
3. Provide cooling
4. Prevent rusting of highly finished bearing surfaces
5. Prevent scuffing and wear
6. Lubricant provides a seal against moisture and rust

Choices between using oil or grease as the lubricant depends upon operating temperature, speed (RPM), load, type of application, sealing and bearing size. Oil lubrication should be used in high temperature environments and high speeds and where the oil can circulate and be cooled. The use of grease simplifies the lubrication process. Some bearings are shipped with grease sealed in them (sealed–for–life) designs. They require little or no maintenance. For temperatures up to 180°F, a high–quality lithium–based NLGI 2 grease is recommended.

Bearing lubrication is analogous to gear lubrication methods. They can be grease, splash or forced oil. Splash oil lubrication or forced oil lubrication should be used when the bearing system is running at high temperatures and high speed.

When **high–temperature** does **not** occur, only a small amount of lubricant is required between the rolling surfaces with **no** sacrifice in performance. Optimum lubrication for rolling–contact bearings can be predicted by elastohydrodynamic theory (EHD). The film thickness is dependent on bearing speed and lubricant viscosity but not affected by load.

Bearing Failure

1. **Fatigue:** Fatigue is a given in any mechanical system. Using the optimum performance of the Rated Life, L_{10} formula, fatigue is a natural by–product of wear.
2. **Overheating:** Excessive or inadequate lubrication, grease liquefaction, oil foaming, entry of abrasive or corrosive contaminants, bearing housing warp age or running out–of–round, seal failure, inadequate bearing clearance or preload, race turning, cage wear or shaft expansion.
3. **Vibration:** Contamination or steel chips in bearing, fatigued race or rolling elements, race turning, rotor out–of–balance, shaft out–of–round, race misalignment, housing resonance, cage wear, flats on races or rolling elements, large clearances, corrosion, false–brinelling, mixed rolling element diameters or races running out–of–square.
4. **Turning on shaft:** Growth of race due to overheating, fretting wear, initial assembly improper or rough (coarse) finish on shaft, excessive shaft deflection or seal rubbing on inner race.
5. **Binding of shaft:** Lubricant leakage, breakdown or contamination, housing distortion or uneven shimming, tight rubbing seals, preloaded bearings, cocked races, excessive tightening of assembly, thermal expansion of shaft or bearing housing or cage failure.
6. **Displaced shaft:** Bearing wear, improper assembly, bearing shifted or overheated, inadequate shaft or housing shoulder, lubrication, cage failure or loosened assembly.
7. **Noisy bearings:** Lubrication issues, contamination, pinched bearing, rubbing seals, clearances, unsecured bearing or housing, damaged rollers or unequal sizes, brinelling due to assembly abuse, handling, or shock loads, out–of–round issues, housing not flat or chips or scoring under race ways.
8. **Lubrication leakage:** Overfilling, improper grease, grease breakdown due to high operating temperatures or operating life is longer than grease life, seal wear or failure, improper shaft attitude, O–ring failure or flow rate of lubricant.

Section II

Materials and processes

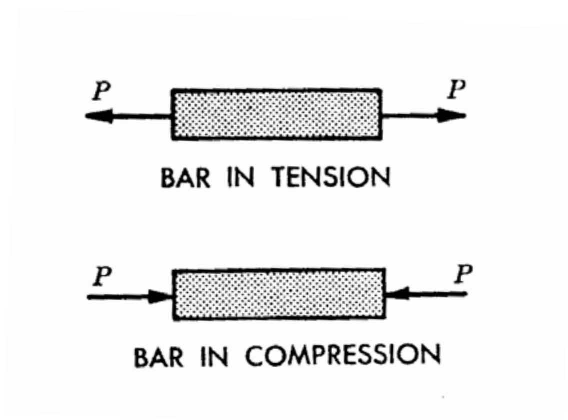
In the first section of this course we dealt with machine components (machine elements) and different linkages and mechanisms. Their relationship as individual working members and combined as units was considered. Having said this, the importance of conveying the dimension of strength of materials (Materials Science and Engineering) and having the ability to choose the proper material for the right application is also paramount. One cannot or must not separate these two sciences; together, they form a bond that unites the mechanical and physical properties of materials. The study of mechanics and the physical properties of materials must be considered as equals.

A basic knowledge and understanding of how materials act (perform) under various loads and the proper criteria to choose them will be covered in this overview section.

Strength of materials is how materials react to external forces being applied to them and their resulting stresses and deformation. Forces can work in combination or as individual forces such as gravity, acceleration, impact, environmental pressures from gas or liquid, effects of power transmission, temperature variations, corrosive materials (liquids and gasses) and degrees of impact. In order to properly evaluate materials and to record their reactions, a test specimen (test sample) of a particular material is chosen. This is known as a tension test and whose procedure is strictly verified and defined by the American Society of Testing and Materials (ASTM).

The specimen is clamped in a testing machine which gradually pulls it apart. The gradually applied force is an axial tensile load. This applied force is deliberately slow so that all parts of the specimen are considered in equilibrium at each instance. When the load increases, the length (deformation or elongation) increases. A transducer connected in a series on the specimen provides an electronic reading of the load corresponding to the displacement. The data derived from this type of test is tabulated and a stress–strain curve is plotted.

Materials testing determines numerous mechanical properties of materials. This testing incorporates the effects of stresses in ferrous materials (iron, steel, mild steel, carbon steel, tool steel, cast iron and stainless steel) and are magnetic, except for certain stainless steels, non–ferrous materials (aluminum, zinc, brass, lead and tin) which are not magnetic and more corrosion resistant, plastics and composite materials. The following equations will aid in understanding how the stress–strain curve is plotted and shows the relationship between stress and strain and other important mechanical properties of materials.



The bar in tension is considered **tensile stress**

The bar in compression is considered **compressive stress**

A third type of stress is known as **shear stress**.

P

P

Shearing Stress

Shear Stress, $\tau = \frac{P}{A}$ where P = force, expressed in pounds (lb)

A = Cross–sectional area expressed in square inches (in^2)

Tensile and compressive stress work at right–angles to the area and shearing stress are always in the same plane of the area and is at right–angles to the tensile and compressive stresses.

Stress, $\sigma = \frac{P}{A}$ where P = force, expressed in pounds (lb)

A = area (original cross–sectional), expressed in square inches (in^2)

Strain, $\varepsilon = \frac{\delta}{L}$ where δ = change in length of bar expressed in inches

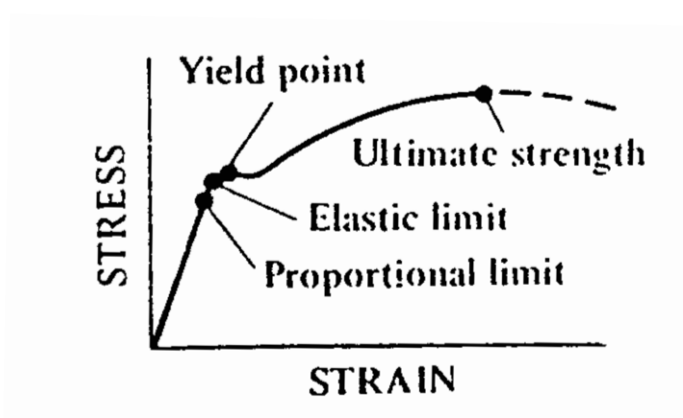
δ is also is referred to as elongation or deformation.

L = original length of bar in inches

Stress is expressed in pounds per square inch (psi)

Strain is expressed in inch per inch (in/in)

Below is a stress–strain curve showing the proportional limit, elastic limit, yield point and ultimate strength:



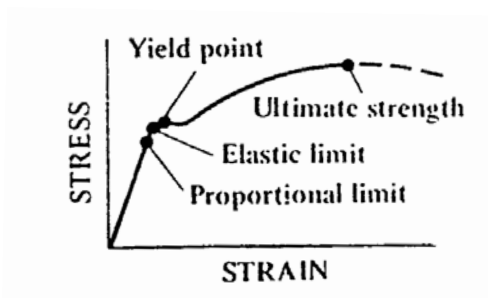
The **proportional limit** is the point where the relationship between stress and strain is no longer linear (stress as a linear function of strain) and at this point the stress begins to deviate from the straight–line.

Hooke's Law: Hooke's Law is only valid within the range of elastic behavior.

$\sigma = E\varepsilon$ or $\varepsilon = \dfrac{\sigma}{E}$ where E = Modulus of Elasticity

The Modulus of Elasticity is expressed in pounds per square inch (psi).

The Modulus of Elasticity is also referred to as **Young's Modulus.**

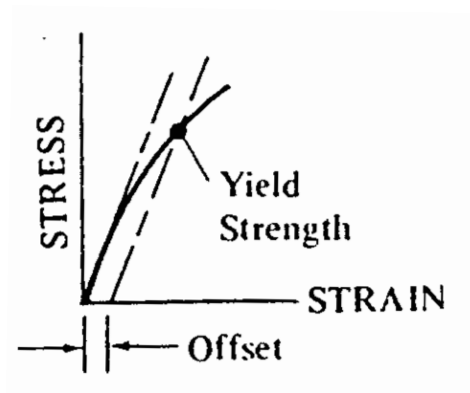


The **elastic limit** is when the test specimen is stressed to the point that when released it will return to its original shape. This is the maximum stress where there is **no permanent damage** to the specimen. The elastic limit for steel is considered the same as its proportional limit.

The **elastic range** (elastic region) is the region of the stress–strain curve extending from the origin to the proportional limit. This is when the working stress does not exceed the elastic limit. The **plastic range** (plastic region) extends from the top of the proportional limit to the point of rupture. This is when the working stress exceeds the elastic limit.

The **yield point** is a point on the stress–strain curve where there is a sudden increase in strain without any corresponding increase in stress. There are some materials that show two points on the curve where there is an increase in strain but not in stress. These are called upper and lower yield points. Some materials **do not** have a yield point.

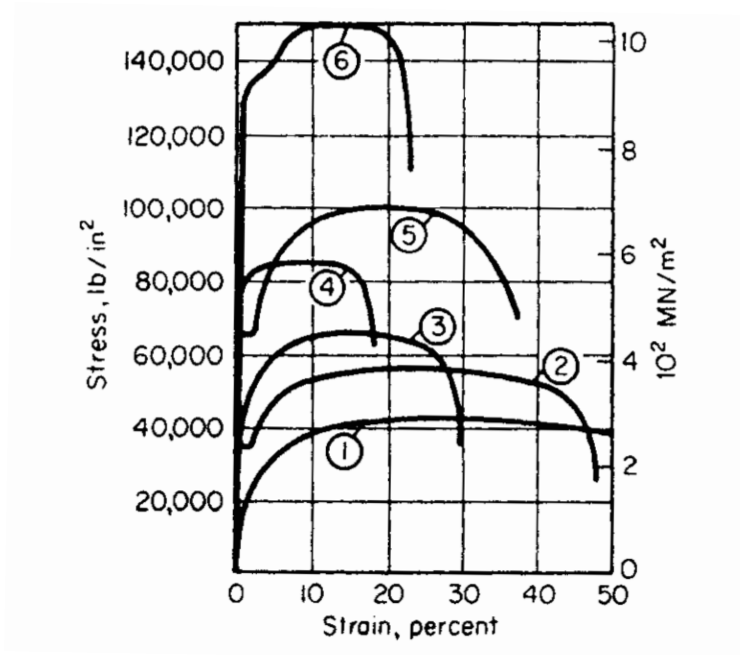
The **ultimate strength**, also referred to as the tensile strength, is the highest or maximum stress value reached on the stress–strain curve.



This stress-strain figure shows the yield strength point of a material.

The **yield strength** is the maximum stress that can be applied without permanent deformation to the material. This applies only to materials which have an elastic limit. Determining the elastic limits in some materials is difficult because they do not have an elastic limit. **Yield strength** is determined by the offset method shown in the figure above. In this case, the stress value on the curve corresponds to a definite amount of permanent set or strain. This can be 0.1 to 0.2 per cent (0.001 to 0.002 inch per inch) of its original size (dimension).

Below is a comparison stress–strain curve for various metals:



1. Soft Brass, 2. Low Carbon Steel, 3. Hard Bronze, 4. Cold Rolled Steel, 5. Medium Carbon Steel annealed and 6. Medium Carbon Steel, heat treated.

Finally: Since , $\sigma = \frac{P}{A}$ and $\sigma = E\varepsilon$ and $\varepsilon = \frac{\sigma}{E}$

Then $\frac{P}{A} = E\frac{\delta}{L}$ therefore: $\delta = \frac{PL}{AE}$

Where δ = change in original length (elongation)

The test data taken from a specimen, in this case, material AISI 1020 cold–rolled steel has been tabulated and a stress–strain curve is developed. Below is the data transferred to the curve:

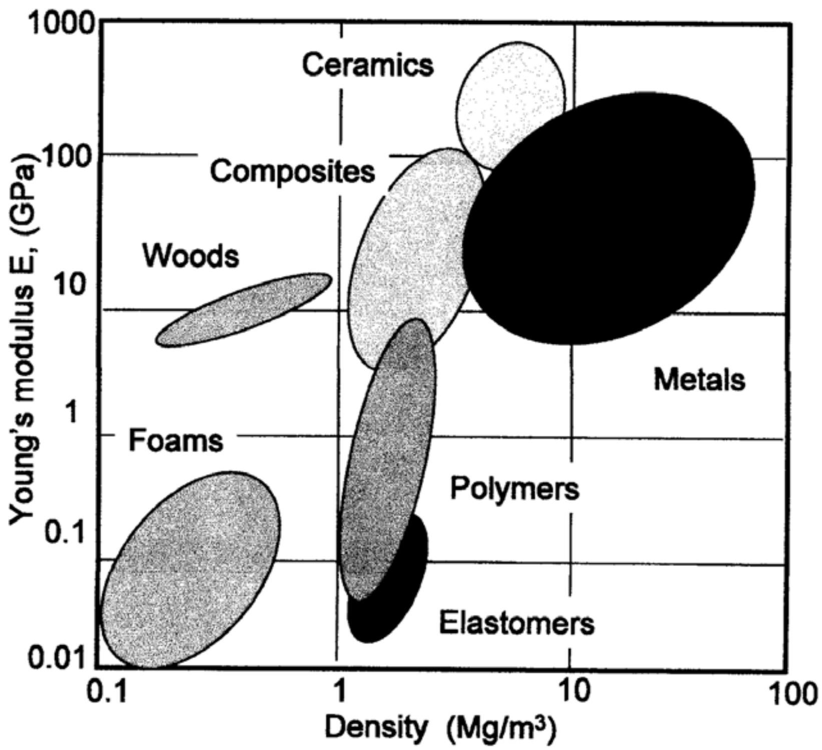
Laboratory Data			Calculated Data	
Initial data:	Load, kips	Elongation in 2 in., in. × 10^{-4}	Stress, ksi	Strain, in. per in. × 10^{-4}
Material AISI 1020 cold-rolled steel	F	δ	s	ϵ
Diameter: 0.503 in.	0.5	0	2.5	0
Gage length: 2.000 in.	1.5	3	7.5	1.5
Hardness: Rockwell B75	2.5	6	12.5	3.0
Speed of testing: 0.025 in. per min	3.5	10	17.5	5.0
	4.5	13	22.5	6.5
	5.5	16	27.5	8.0
	6.5	20	32.5	10.0
	7.5	23	37.5	11.5
	8.5	26	42.5	13.0
	9.5	30	47.5	15.0
	*10.0	32	50.0	16.0
	10.5	34	52.5	17.0
	11.0	36	55.0	18.0
	11.5	39	57.5	19.5
	12.0	43	60.0	21.5
	12.5	46	62.5	23.0
	13.0	51	65.0	25.5
	13.5	60	67.5	30.0
	14.0	73	70.0	36.5
	14.5	88	72.5	44.0
	15.0	118	75.0	59.0
Ultimate	15.83		79.15	
Breaking	12.50		62.50	
Final data:				
Diameter: 0.312 in.			Elastic-limit stress: 52,500 psi	
Gage length: 2.328 in.			Yield stress: 72,500 psi	
Hardness: Rockwell B76			Ultimate stress: 79,150 psi	
Character of fracture: ¾ cup-cone			Percent elongation in 2 in.: 16.4%	
			Percent reduction in area: 61.7%	
			Modulus of elasticity $E = 29.7 \times 10^6$ psi	

Materials Selection

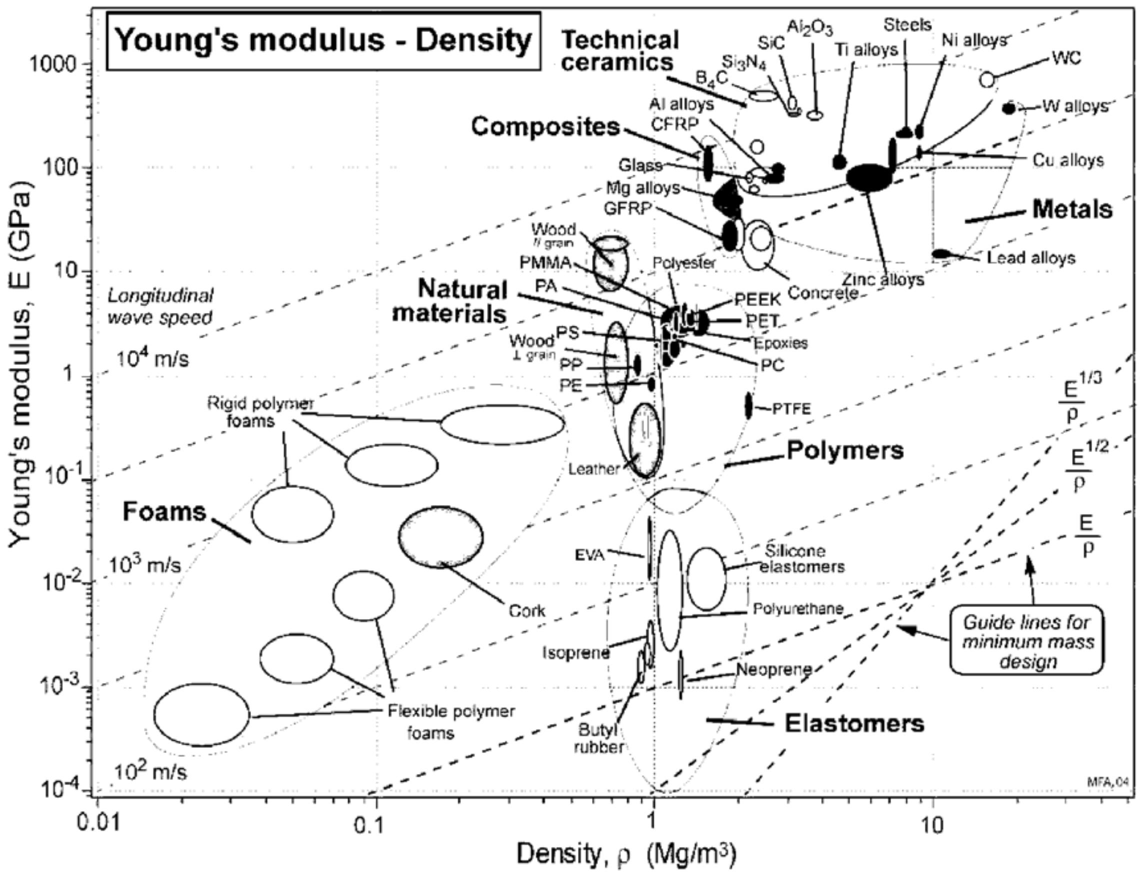
When preparing a design application some preliminary material choices will be investigated. You want the material to meet specific critical criteria such as load requirements, strength, endurance, working environment, constraints, availability of material and costs. Professor Michael Ashby of Cambridge University, United Kingdom, devised several useful material selection charts which groups materials stiffness, strength, density and unit cost. These charts are designed to be used as a preliminary guide in narrowing down and choosing a material. This does not negate the need and necessity for the engineer to do a thorough and proper theoretical as well as practical approach and use sound engineering judgment.

The **Ashby Charts,** also referred to as the Ashby Material Selection Charts, groups engineering materials together with relevant importance because materials are not chosen for one specific property but for a combination of properties. For an example, if you are looking for a material that is lightweight but strength and density is also important.

Ashby has grouped materials together in each chart allowing the engineer to investigate which material will fit the desired criteria. Metal, ceramics, polymers, elastomers, glass, foams, composites and natural materials are placed in groups. Below is a chart which begins to show how Ashby developed his charts in order to be used effectively in evaluating large ranges of material classes and mechanical properties:



This next diagram will be all-inclusive and left for discussion.



Each chart contains a set of guidelines and materials which when materials intersect them will perform equally well under the given conditions. Materials above the guideline will show superior performance and materials falling below the line will perform poorly and must be eliminated during the preliminary process.

This chart displays the Young's Modulus, E, versus Density. Returning to our original premise, lightweight materials, we can use this particular Ashby Chart because of density. Ashby ingeniously groups all the engineering materials in one chart making it easier to evaluate.

Observation shows that the materials have a tendency of clustering together and metals with high Young's moduli have high densities. Polymers have both low moduli and density, while ceramics and glass have high moduli but lower densities than metals. There are still other materials which have both high moduli and densities. An expansive amount of knowledge and information can be gleaned from this chart.

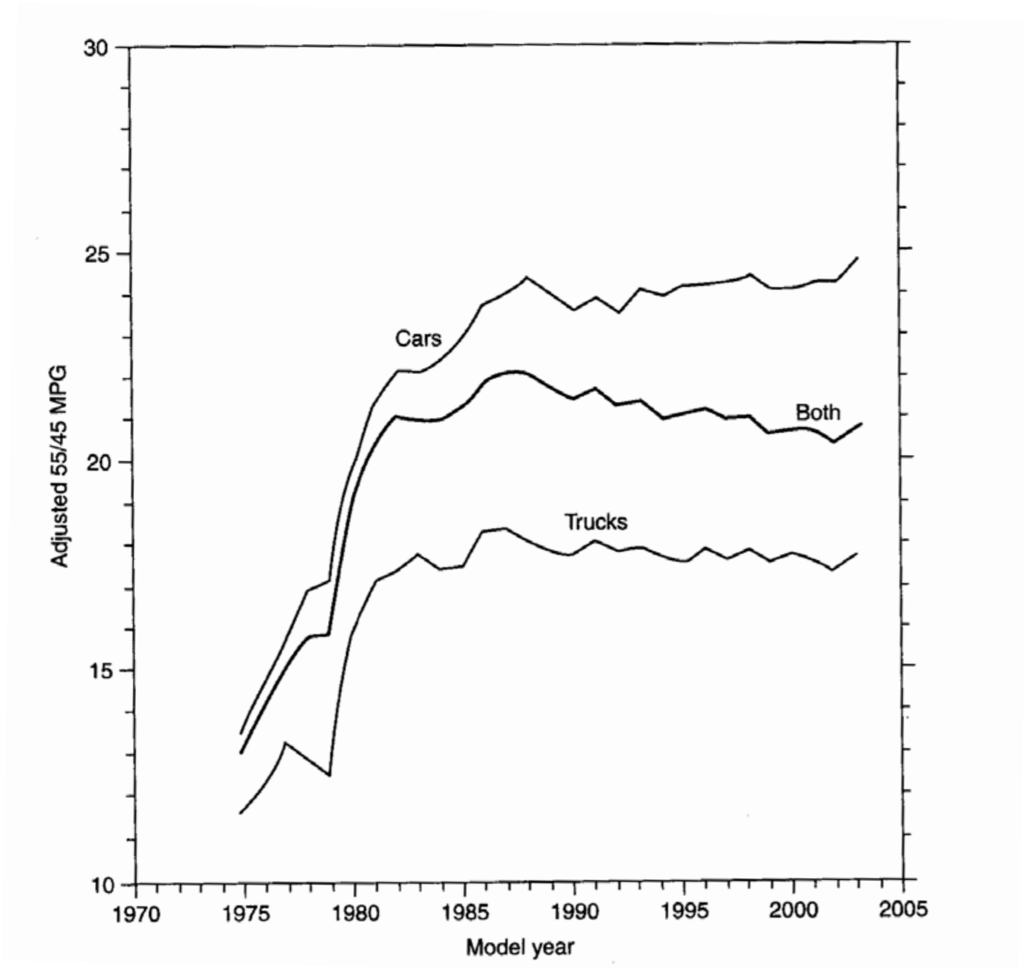
Other useful Ashby Material Selection Charts include:

- Young's Modulus vs. Strength
- Young's Modulus vs. Relative Cost per Unit Volume
- Young's Modulus vs. Energy Content
- Strength vs. Density
- Strength vs. Relative Cost per Unit Volume
- Strength vs. Energy Content
- Fracture Toughness vs. Density
- Fracture Toughness vs. Young's Modulus
- Fracture Toughness vs. Strength
- Specific Modulus vs. Specific Strength
- Loss Coefficients vs. Young's Modulus
- Thermal Conductivity vs. Thermal Diffusivity
- Linear Expansion Coefficient vs. Thermal Conductivity
- Linear Expansion Coefficient vs. Young's Modulus
- Normalized Strength vs. Linear Expansion Coefficient
- Strength at Temperature vs. Temperature
- Wear Rate Constant vs. Hardness

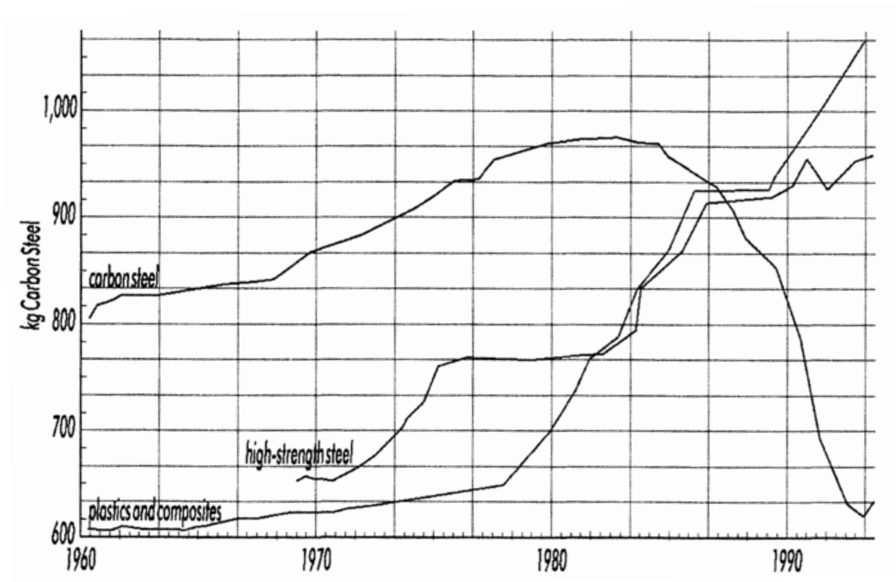
As seen, the Ashby Charts cover a broad spectrum of material content. When utilized properly they offer immense sources of information in a condensed version. This system and resource makes material analysis and procurement much easier.

Engineers are constantly challenged not only by mechanical system requirements, constraints and cost but also by sociological and economic constraints as well.

One such example presents itself in the following two graphs and table. These exhibits show how the design of automobiles has evolved due to the mentioned constraints. Fuel efficiency and economy needed to be addressed and evaluated and will remain a **constant constraint**. Weight, strength, durability and costs are competing factors. Upon review, see how new materials and better use of existing materials have changed the manufacturing landscape. All of these changes and challenges exist in other areas as well. Design scenarios must adjust to current demands.



Above, the trend is making automobiles and trucks more fuel–efficient.



This second figure displays the average material composition of automobiles from 1960 to 1993. The use of carbon steel has decreased dramatically where as high–strength steel, plastics and composite material have risen in order to manufacture more fuel–efficient vehicles.

The table below also shows the percentage of materials used in automobiles from 1978 to 2001. Again, there are significant changes to lighter materials and material strength demands.

Material	1978 Pounds	1978 Percent	1985 Pounds	1985 Percent	2001 Pounds	2001 Percent
Ferrous materials						
Conventional steel	1880.0	53.8%	1481.5	46.5%	1349.0	40.8%
High-strength steel	127.5	3.6	217.5	6.8	351.5	10.6
Stainless steel	25.0	0.7	29.0	0.9	54.5	1.6
Other steels	56.0	1.6	54.5	1.7	25.5	0.8
Iron	503.0	14.4	468.0	14.7	345.0	10.4
Total ferrous	2591.5	74.1%	2250.5	70.6%	2125.5	64.2%
Nonferrous materials						
Powdered metal[1]	16.0	0.5	19.0	0.6	37.5	1.1
Aluminum	112.0	3.2	138.0	4.3	256.5	7.8
Copper	39.5	1.1	44.0	1.4	46.0	1.4
Zinc die castings	28.0	0.8	18.0	0.5	11.0	0.3
Plastics/composites	176.0	5.0	211.5	6.6	253.0	7.6
Rubber	141.5	4.1	136.0	4.3	145.5	4.4
Glass	88.0	2.5	85.0	2.7	98.5	3.0
Fluids & lubricants	189.0	5.4	184.0	5.8	196.0	5.9
Other materials[2]	112.5	3.2	101.5	3.2	139.5	4.2
Total nonferrous	902.5	25.8%	937.02	29.4%	1183.5	35.7%
Total	3494.0 lb		3187.5 lb		3309.0 lb	

Table of Average Materials Consumption for U. S. Automobiles

Factors of Safety

When a machine or mechanical component is chosen with appropriate criteria, and various loads considered, there still remains the possibility permissible maximum stress in the design may not allow for possible uncertainties which may exist. This maximum stress is referred to as the allowable stress, working stress and also called the design stress.

The **design stress** should not have redundancies, over design or wasted material but should be designed properly to prevent failure. With caution, the machine design unit may incur certain unforeseen or unexpected loads. There is always the risk that the **working stress** will exceed the strength of the material. The **factor of safety** tries to minimize this risk.

One way to incorporate the **factor of safety** in designs is to use the following equation:

$$s_w = \frac{S_m}{f_s}$$ where, f_s = factor of safety

S_m = strength of the material (pounds per square inch)

s_w = **allowable working stress** (lbs. per square inch)

The factor of safety is greater than 1, the allowable working stress will be less than the strength of the material.

In most cases, S_m is based on yield strength for ductile materials, ultimate strength for brittle materials and fatigue strength for materials which are subject to cyclic stressing.

The following table gives factor of safety recommendations for various applications:

f_s	Application
1.3–1.5	For use with highly reliable materials where loading and environmental conditions are not severe, and where weight is an important consideration.
1.5–2	For applications using reliable materials where loading and environmental conditions are not severe.
2 –2.5	For use with ordinary materials where loading and environmental conditions are not severe.
2.5–3	For less tried and for brittle materials where loading and environmental conditions are not severe.
3 –4	For applications in which material properties are not reliable and where loading and environmental conditions are not severe, or where reliable materials are to be used under difficult loading and environmental conditions.

Working Stress

The following six equations are used to determine the allowable working stress when combined with the normal stress value (σ) and multiplied by the stress concentration factor (K). The nominal stress can be simple stress, combined stress or cyclic stress. Depending on the type of nominal stresses, one of the following equations will apply:

For: s_w = Kσ and s_w = Kτ K = stress concentration factor

σ = simple normal stress (tensile or compressive)

τ = shear stress

For: s_w = Kσ' and s_w = Kτ' K = stress concentration factor

σ' and τ' are combined normal and shear stresses

For: s_w = Kσ_{cy} and $s_w = K\tau_{cy}$ K = stress concentration factor

σ_{cy} and τ_{cy} are cyclic normal stress and strain

These equations only apply for evenly distributed stress.

Stress Concentration Factors

When a mechanical part has an abrupt change (no relief cut or radii formed) or the cross–section has a severe change in its shape, the maximum stress will occur at that section. In punches and dies, the die plate which supports the die part has drilled holes in each corner in order to eliminate fracture or cracking in that area. This is also important in heat–treated shafts and bearings where small relief cuts (fillet radii) are made. A fillet is a concave easing of an interior corner of a part.

Simplified stress calculations do not serve well in this particular circumstance. The ratio of this maximum stress to the nominal stress is the stress concentration factor, K. This is a constant for a particular shape and is independent of material as long as it is isotropic (homogeneous–having the same physical properties in all directions). The stress concentration factor is determined by either experimental or theoretical methods.

Stress concentration will cause failure of brittle materials if the concentrated stress is larger than the ultimate stress. Plastic deformation will occur in ductile material when the concentrated stress is higher than the yield strength. Even in ductile materials, when a machine component is cyclically loaded, it can encounter stress concentration which will lead to failure.

Stress concentration is related to the type of material that comprises the component, geometry of the component, type of loads and stresses, processing (heat-treated, annealed, stress relieved, etc), and environmental conditions. When stress concentration factors that specifically match all of the preceding conditions are not available, then the stress concentration factor equation can be altered as shown below:

$$K = 1 + q(K_t - 1)$$

K_t = Theoretical stress concentration factor

K_t is only function of geometry and the nature of stress

q = index of sensitivity

When there is no theoretical stress concentration available for a particular geometry then K_t = 1

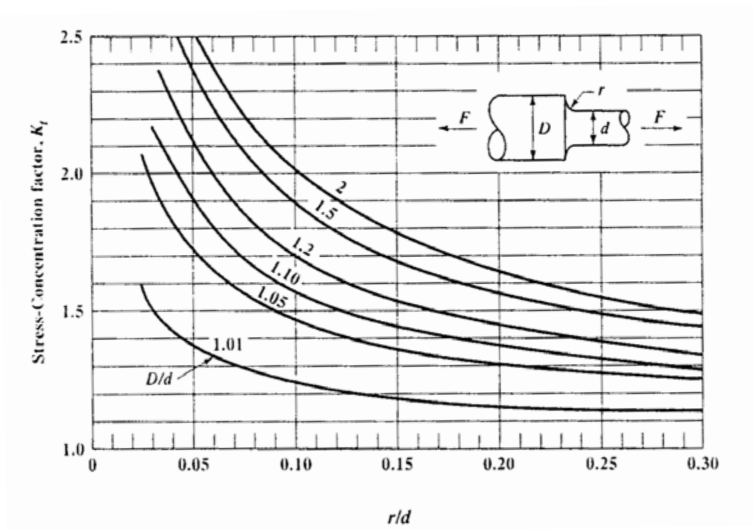
q = 0 and K = 1 For constant stresses in ductile and cast iron materials

q = 0.15 for constant stresses in brittle materials as in hardened steel

q = 0.25 for very brittle metals as steel that have been quenched but not drawn

With impact stresses (applied suddenly) q (index of sensitivity) ranges from 0.4 to 0.6 for ductile materials, for cast iron, q = 0.5 and q = 1.0 for brittle materials.

The following curve is just one of several which displays the Stress Concentration Factor for a filleted shaft in tension:



Section III

Geometric Dimensioning and Tolerances

Once you have finished your design work and have chosen your materials, all these elements must be presented for fabrication. The mechanical system (gear assemblies, linkages and other mechanisms) has to be detailed (drawn) in standard (uniform) specifications in engineering drawings which in turn are communicated to the manufacturer. Remember the tighter (held more precise) the tolerances required the more expensive the cost. Loose tolerances create another issue with fits, alignment and mating of parts.

These engineering drawings must convey the following critical instructions:

- Dimensions: Scale, relationships to datum's, degree of precision
- Geometry: Shape of an object, orientation of objects to critical dimensions
- Tolerances: How tolerances are represented in variations to basic dimensions and degrees of precision.
- Critical Views: References to areas on a drawing which need additional explanation or critical attention to detail.
- Material: Material specification, no substitutes unless reanalyzed

- Material Processes: Roughing and finishing specifications, heat–treatment, plating , materials inspection, or other critical manufacturing (fabricating) processes.
- Finish: Surface finish(quality) and desired final surface result. Included in final finishes may be hard–coats, sprayed ceramics and/or chrome plating.

Tolerances must not only be established on components and machine elements but on the housings in which they are assembled. A housing is a structure in which hole sizes, dimensions and centerlines are precisely determined and described in order to facilitate the proper alignments, functionality and assembly of parts and components. Engineering drawings must encompass individual parts as well as detailed assembly drawings.

Geometric Dimensioning and Tolerancing referred is referred to as GD&T is a comprehensive system of dimensioning to symbolically describe the geometry of a part, parts or assembly in geometrical tolerance zones. It defines individual tolerance features as well as tolerances between other allowable dimensional tolerances in other datums.

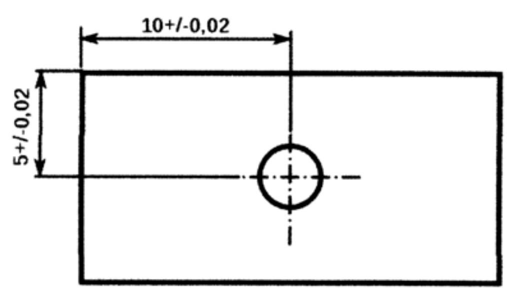
Because of design intent, the GD&T drawings display a geometrical tolerance zone within which features must be constrained. It is a technical vehicle which enables the intended design to be interrupted into symbolic dimensioning. The primary users of GD&T drawings are:

- Design
- Manufacturing
- Quality Assurance

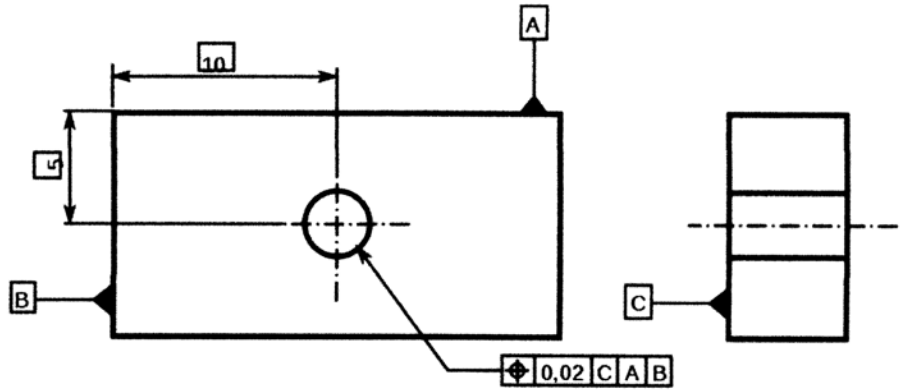
There are two organizations which best set, describe and establish the GD&T standards, the American Society of Mechanical Engineers (ASME) and the International Organization for Standardization (ISO). We shall discuss both systems. The ASME Y14.5 is more comprehensive and more concise than the ISO 8015 and ISO 26921 Standards. The ISO separates its standards into various categories, which makes maneuvering for information and details more difficult and time consuming.

Comparison of ASME Y14.5 and ISO Geometric Symbols

Symbol for	ANSI Y14.5M	ISO	Symbol for	ANSI Y14.5	ISO	Symbol for	ANSI Y14.5M	ISO
Straightness	⏤	⏤	Circular Runout*	↗	↗	Feature Control Frame	⌖ | Ø 0.5 Ⓜ | A | B | C	⌖ | Ø 0.5 Ⓜ | A | B | C
Flatness	▱	▱	Total Runout*	⌰	⌰	Datum Feature*	A	A OR A
Circularity	○	○	At Maximum Material Condition	Ⓜ	Ⓜ	All Around - Profile		(proposed)
Cylindricity	⌭	⌭	At Least Material Condition	Ⓛ	Ⓛ	Conical Taper	⌲	⌲
Profile of a Line	⌒	⌒	Regardless of Feature Size	NONE	NONE	Slope	⌳	⌳
Profile of a Surface	⌓	⌓	Projected Tolerance Zone	Ⓟ	Ⓟ	Counterbore/Spotface	⌴	⌴ (proposed)
Angularity	∠	∠	Diameter	Ø	Ø	Countersink	⌵	⌵ (proposed)
Perpendicularity	⊥	⊥	Basic Dimension	$\boxed{50}$	$\boxed{50}$	Depth/Deep	↧	↧ (proposed)
Parallelism	//	//	Reference Dimension	(50)	(50)	Square (Shape)	□	□
Position	⌖	⌖	Datum Target	Ø8 A1	Ø8 A1	Dimension Not to Scale	15	15
Concentricity/Coaxiality	◎	◎	Target Point	X	X	Number of Times/Places	8X	8X
Symmetry	⌯	⌯	Dimension Origin	⌱	⌱	Arc Length	⌒105	⌒105
Radius	R	R	Spherical Radius	SR	SR	Sperical Diameter	SØ	SØ
Between*	⟷	None	Controlled Radius	CR	None	Statical Tolerance	⟨ST⟩	None



Above is a **linear** representation of a hole location



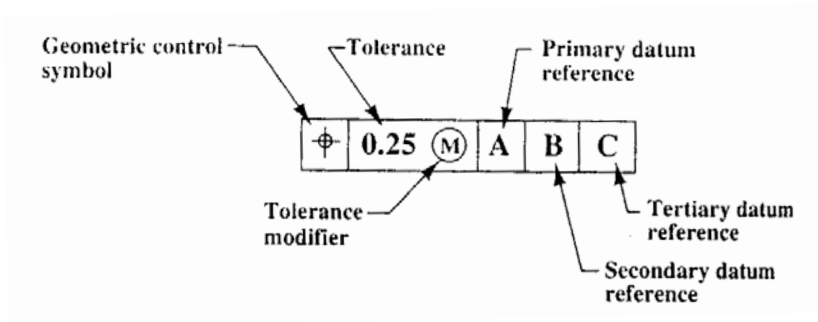
GD&T representation of a Feature Control Frame and Datum Reference

With the Geometric Dimensioning Tolerance standard, more information is transferred and displayed for a faster and more accurate interptation. There are other important design related criteria displayed below:

Type	Geometric Characteristics		Pertains To	Basic Dimensions	Feature Modifier	Datum Modifier
Form	—	Straightness	ONLY individual feature		Modifier not applicable	NO datum
	○	Circularity				
	⏥	Flatness				
	⌭	Cylindricity				
Profile	⌒	Profile (Line)	Individual or related	Yes if related		RFS implied unless MMC or LMC is stated
	⌓	Profile (Surface)				
Orientation	∠	Angularity	ALWAYS related fea-ture(s)	Yes	RFS implied unless MMC or LMC is stated	
	⊥	Perpendicularity				
	//	Parallelism				
Location	⌖	Position		Yes		
	◎	Concentricity			Only RFS	Only RFS
	⌯	Symmetry				
Runout	↗	Circular Runout				
	⌰	Total Runout				

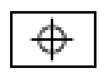
The previous table shows the meaning of various geometric control symbols. These symbols are incorporated into the drawing to give significant clarity and directives.

Below is the **Feature Control Frame** which is a powerful tool:



The Feature Control Frame specified on the engineering drawing indicates the type of geometric control for the features within the rectangular box. It governs the tolerance control, datum control hierarchy, geometric control and other applicable controls such as tolerance modifiers.

In the figure above, the first symbol shown is the geometric control symbol.

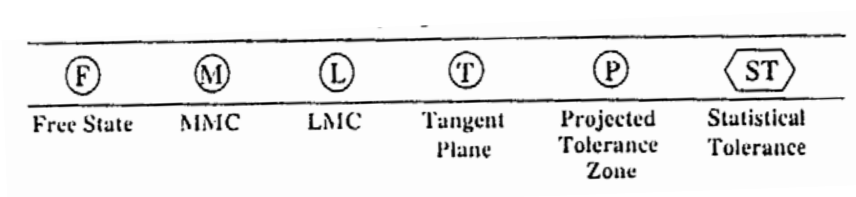


The Position (Location, geometric symbol)

Refer to the previous table on geometric symbols which are also placed in the Feature Control Frame, Total Runout, Straightness, Perpendicularity, Parallelism, Concentricity, etc.

Next reference is to the Tolerance (the total amount of deviation from specified dimension, also known as upper and lower limits), the Tolerance Modifier, then the three Datum References A,B and C(all perpendicular to one another).

Tolerance Modifiers are used with geometric tolerances to further refine the level of control. Below are the Modifier Symbols:



The **M** shown in the Feature Control Frame is the symbol MMC which represents the **Maximum Material Condition**. It is the condition of a part feature wherein it contains the maximum amount of material within the stated limits of size. For an example, as in mating parts (minimum hole size to mate with maximum shaft size).

The symbol **L** is **LMC** which is **Least Material Condition**. This is opposite to MMC. It is the condition of a part feature wherein it contains the minimum (or least) amount of material within the stated limits of size. For an example, the largest hole size and the smallest shaft size.

The symbol **F** is **Free State**. It is the removal of all forces (released state, relaxed state) after a manufacturing process.

The symbol **T** is **Tangent Plane**. The tangent plane is a control feature for the surface of a part, which is generated by high points, which establishes the allowable deviation from the mean dimension.

The symbol **P** is **Projected Tolerance Zone**. This deals with any variations in perpendicularity issues with threaded holes, pins, studs or fasteners which are press–fit. And any interferences with mating parts. The projected zone extends above the functional area of the pin, screw, stud or fastener.

The symbol **ST** is **Statistical Tolerance**. This means that the tolerance is controlled statistically not arithmetically.

This was an overview of the GD&T Standard. This standard conveys the design intent to the finished part or assembly. If followed, the product will function and be interchangeable as designed, because it provides uniformity. Its scope and ability to master takes numerous hours of technical knowledge, use and experience. It is vital to industry because it eliminates questionable fabrication procedures and accelerates the production process. The ASME Y14.5M is the metric version (SI units), and so are the ISO Standards.

Section IV

Production analysis and Quality Control

After the design intent (design criteria) has been addressed both theoretically and in practical application, the actual production (manufacture of the product) must be detailed by a plan which includes a production schedule. There are two areas which we will examine; **first**, production requirements that pertain to the end–user. The goal is to produce a viable product that meets the following criteria:

- Reliability, dependability and resilient
- Cost
- Simplicity in design and operation
- Demand
- Availability: Accessible to purchaser
- Relevance
- Practical
- Functional
- Social and Environmental impact (Green Design)

Having considered the above criteria, the end–user (public, private or corporate), will be the judge and will validate the transaction.

Second, the manufacturer (fabricator) must be able to produce a product that meets the engineer's design intent. Manufacturers must decide on the types of machinery required to produce parts and utilize their quality control departments (Q.C machinery included) in concert with each other to fabricate the end product. Manufacturers are a diverse group. Production and quality control (inspection) equipment varies in degrees of efficiency and precision.

Today, not only can parts be made with great speed and precision, the machinery as a whole system is precise as well. With improved hardware and software technologies the newer CNC (Computer Numerical Control) machine and quality control usage of CMM (Coordinate–Measurement Machine) equipment has revolutionized manufacturing.

Precision machine tools have three important attributes:

1. Accuracy
2. Repeatability

3. Resolution

Accuracy is defined as the maximum spread in measurements made during the machine movement (slides, ways and column movements) during successive cycles at a number of target positions.

Repeatability is defined by how accurate the variations in measurements are in repetitious runs where conditions remain constant.

Resolution is defined as the smallest units of measurement that the system can recognize.

When considering the correct type (quality/precision) of CNC machine, one will pay a premium for unnecessary accuracy, therefore; choose one that meets your tolerance requirements. Other requirements besides tolerance issues are cutting (feed rates) and positioning speeds. Positioning speeds in coordinate–measurement machines used in quality control must be considered. Below are other essential features to consider when purchasing a CNC machine:

- Programming features and compatibilities
- Spindle speeds (RPM)
- Power rating (material removable rates)
- Horsepower
- Setting offsets and set–up ease
- Ergonomics
- Turret speeds and accuracy
- Indexing speeds and accuracy
- Size: Working envelope and capacity
- Rigidity and strength of work system (structural integrity)
- Types and rating of chucks
- Coolant system capacity
- Weight and weight distribution
- Operation safety

Quality Control

Quality control is the system, which insures the desired design intent in a manufactured part, parts or assembly, by proper interpretation of design drawings with various sampling methods. It is a system for achieving and maintaining production integrity and engineering conformity to a quality level in a manufactured product. Quality control maintains the ability to inspect, take sample data and assess what changes are needed to produce a quality part.

Quality control is different from quality assurance. The function of quality control is to take the analytical measurements or samplings and use this information to assess the quality of the analytical data. Quality control's basic area is confined to the manufacturing floor. Where as, quality assurance is on the management level which over sees the entire production regime. This system guarantees the integrity of the data.

Quality control sets the policies and utilizes it's position to provide the necessary procedures to improve and control various processes in manufacturing that will lead to an improved business performance. Quality control, coupled with engineering statistics takes production to an area which uses statistical methods in order to analyze production data and which provides a quality outcome. Variability and it's study is one method; another, is using random sampling. This involves the study of statistics involving average and standard deviation.

Quality control takes corrective action when manufacturing defects are detected or when uncovering system flaws. Quality control is used to eliminate and correct production errors and is to implement and manage the production process with the written design drawings, plans and specifications presented.

Small size companies can also benefit by having a QC program. The plan can be tailored to their production needs and requirements. The cost savings can be beneficial. Rejected and returned parts only disrupt the manufacturing process and increases unwanted costs. The cost of reworking a part or even discarding it is one issue but losing a customer is another. It is a known fact, keeping a satisfied customer is easier than trying to get them back when you lose them. Most companies use data such as delivery schedules and rejection rates to evaluate their suppliers. These quantitative factors can be used to detect whether you meet your customer's production requirements.

Large companies as well as small must be flexible in their production techniques. Large companies depend on consumer driven directives where small companies (sub–contractors) are reliant upon their customer's needs and requirements. No one is excluded from this reality. There is always room for improvement. To analyze properly and constructively can be both beneficial to the production process and to the end–user. There are no shortcuts. Maintain the required standards and manufacture the part right the first time.

Do not over use or abuse the quality control program. Over sampling and unnecessary investigation is time consuming and costly. Use existing programs as a guide to tailor it to your production environment.

Mechanical Systems I Examination

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EXAMINATION

MECHANICAL SYSTEMS I

(There is only one correct answer)

1. As an early inventor, he was credited with inventing the windwheel.
    a. Archimedes
    b. Hero of Alexandria
    c. Leonardo da Vinci
    d. James Watt

2. What joint is associated with linkages?
    a. Revolute
    b. Prismatic
    c. Spherical
    d. all of the above

3. The basic form of cam motion is
    a. uniform
    b. harmonic
    c. uniformly accelerated and retarded
    d. all of the above

4. The most basic and common gear is
    a. spur
    b. helical
    c. bevel
    d. worm

5. This was the year the AFBMA title changed to the ABMA.
    a. 1987
    b. 1993
    c. 1997

d. 2001

6. This type of bearing is not recommended for rotary motion.
   a. single–row radial
   b. double–row
   c. ball bushing
   d. double–row angular

7. Even though it is more expensive, this bearing is used in difficult installations or removals.
   a. plain ball
   b. internal self–aligning
   c. single–row angular contact
   d. split–type

8. A tension test of a test specimen is strictly verified and defined by the
   a. ASME
   b. ASTM
   c. ISO
   d. ANSI

9. Hooke's Law is valid in
   a. the range of elastic behavior
   b. between the yield point and the ultimate strength
   c. between the elastic limit and yield point
   d. none of the above

10. Ashby material selection charts are used because
    a. they analyze the stress–strain curve
    b. they are used in specimen testing
    c. they group materials by a combination of properties
    d. for experimental data

11. A safety factor is used because
    a. of too many unknowns
    b. of limited design constraints
    c. just to be sure
    d. the machine design unit may incur certain unforeseen or unexpected loads.

12. Stress concentration factors are considered
    a. because stress–strain curves may not be reliable
    b. because stress concentration can cause failure
    c. because of a need to refine a design calculation
    d. none of the above

13. Geometric dimensioning and tolerancing is
    a. a comprehensive system of dimensioning
    b. a single way to detail tolerances
    c. not as important as other dimensioning systems
    d. never modified or improved

14. A primary user of GD & T is
    a. quality assurance
    b. design
    c. manufacturing
    d. all of the above

15. These two organizations are preeminent in writing, establishing and standardizing GD & T drawings
    a. ASME & ISO
    b. ASTM & ASME
    c. ISO & SAE
    d. ASME & AIAS

16. In the GD & T system, which geometric control symbol is not an orientation type?
    a. angularity
    b. flatness
    c. perpendicularity
    d. parallelism

17. Geometric control symbols are used
    a. to make design drawings look more complicated
    b. as needed to represent a deficiency
    c. to give significant clarity and directives
    d. none of the above

18. This symbol is not a tolerance modifier
    a. F
    b. L
    c. M
    d. B

19. This must be considered to have a viable product
    a. reliability
    b. demand
    c. cost
    d. all of the above

20. This attribute is not considered in precision machine tools
    a. repeatability
    b. accuracy
    c. resolution
    d. anchoring techniques