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# What Every Engineer Should Know

## About Drive Trains and Linkages

by

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**CREDENTIALS**

The author, a PE, has 50 years of experience designing structures and mechanisms. He holds 13 US Patents ranging from sawmills to spacecraft.

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## ILLUSTRATIONS

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## INTRODUCTION

What do we mean by the term "Mechanism" and what makes it unique? Simply, a mechanism is a collection of moving parts that work together to perform a task. It can be as simple as an electric motor, which has an armature rotating inside a housing, providing torque to drive a load. It can be as complex as the action of a piano, where pressing a key causes a hammer to strike a string in just such a way that a beautiful tone is produced. Mechanisms can be far more complex than that. An old-fashioned adding machine was formed from hundreds of moving parts. These days, however, those really complex machines are being replaced by electronic circuitry, as circuitry gets simpler and more reliable.

Where does one draw the line between "Structure" and "Mechanism"? While all of the parts of a mechanism are structures, structures are not mechanisms. They have no moving parts. And that brings up the chief difference between the two: useable life span. A structure can reasonably be expected to last for centuries (the Eiffel Tower, the Brooklyn Bridge) or even millennia (the pyramids, the Coliseum in Rome). Mechanisms, on the other hand, are only expected to last dozens of years (How many appliances or automobiles have you had?)

What causes that order of magnitude difference in useful life is that if not failing due to loads higher than they are capable of withstanding (earthquakes, hurricanes), structures corrode to death (the I36W bridge in Wisconsin.) Structures that last a long time are generally made of non-corrosive materials such as stone, or are kept protected from the elements by coatings such as paint.

Mechanisms fail by wearing out. A mechanism, by definition, has moving parts. Parts which move against one another, wear. We work hard at minimizing wear, but can't eliminate it completely. So at the end of many years of faithful service, we have a structure which is perfect, but its joints are worn out, or the bearings are worn out, or the seals leak and oil drips on the floor.

The art of mechanism design lies in designing a mechanism that wears very slowly, and confines that wear to easily replaceable components. If those parts are truly easy to replace, then the mechanism can be overhauled, all its wear parts replaced, and it can be returned to many more years of service, **economically.** If the parts are not easy to replace, then the cost of the overhaul will exceed the cost to replace the mechanism, and it will be scrapped.

## TYPES OF MECHANISMS

There are two basic types of mechanisms: Drives and Linkages. A Drive is a mechanism (generally rotary) that takes the input of a rotating shaft and operates on it to perform a different function. As an example, take the pedal sprocket - chain - wheel sprocket of a bicycle. This drive takes the high-torque/low-RPM input from the rider and converts it into a low-torque/high RPM output to drive the rear wheel at a convenient speed.

Prime movers are a subset of drives. A prime mover takes energy from some outside agency (electric current, expanding steam, expanding petroleum combustion products) and through a combination of drives and linkages, converts this energy to rotate a shaft. Probably the simplest prime mover is the electric motor, where electric current flowing through a wire that lies in a magnetic field causes a sideways force on that wire. The wire is attached to an armature, and the force causes the armature to rotate. That motor certainly qualifies as a drive, and generally has two bearings to support the armature on its shaft. Those bearings will eventually wear out, and it would be a shame if you had to scrap all the rest of the motor just because the bearings can't reasonably be replaced.

The second type of mechanism is the Linkage. In general, a linkage will serve one of two functions. Some mechanisms transfer a movement (push or pull) through a distance to perform an operation. Perhaps the ultimate in this type of mechanism was the "engine room telegraph" of old steamships where the Captain had a large dial on the bridge, called the engine room telegraph, which has a handle on the side of it. The Captain moves the handle from, say, "stop" to "half ahead" and an identical instrument located in the engine room will move from "stop" to "half ahead", giving the engineer his orders. These instruments were located hundreds of feet and several decks apart, and had to be extremely reliable. We use many of the linkage concepts that they created back then, today.

Other mechanisms serve as guides to constrain one rigid object to move in a particular path compared to another rigid object. A very good example of this type of linkage is found in the "hinges" of automobile hoods. As you lift a typical hood, the back edge near the windshield needs to come up rapidly to clear windshield wipers, the windshield, etc. What is needed is a hinge line passing approximately through the driver's shoulders, which would be very inconvenient. So the linkage creates a "virtual center" in that location, but since it is "virtual" it doesn't interfere with the driver.

## DRIVE TRAINS

Drive trains are comprised of two or more rotating elements, and their function is usually to provide an output shaft that rotates at a different speed from the input, generally to provide a more suitable torque for the application. For example, the "reduction gear" in a boat accepts the input from an engine that is running at, say, 3000 RPM and 100 foot-pounds of torque, and steps it down to an output of 1000 RPM and 300 foot-pounds of torque to drive a large, efficient propeller. The drive train of an automobile does almost exactly the same thing, stepping the high RPM, low torque output of the motor down to a low RPM with high torque to drive the wheels. For all drive trains, RPM is multiplied or divided exactly by the ratio designed into the gears, or chains or timing belts or V-belts. The torque output of the drive train is also an exact multiple of the ratio, but with efficiency factored in. For many drive trains, efficiency is near 100%, so it can be ignored, but for worm gears and V-belts, efficiency can be quite low, and must be taken into account.

A note about efficiency: Reduction from perfect (100%) efficiency is caused by friction. Friction expresses itself as heat. If you have a 10 horsepower motor driving a load through a drive train that is 90% efficient, one horsepower will be dissipated as heat in the drive train. That is a continuous 746 watts of heat, which will get the drive train hot in a hurry. Conversely, when you hear someone complaining about the inefficiency of their automobile's drive train, ask yourself "Where does the heat go? What element has to be cooled to dissipate the heat?"

There are several types of rotating elements that can provide this RPM/Torque multiplication. The most common of these are: Gears, Chain-and-Sprocket, Timing Belt-and-Pulley, and V-Belt-and-Pulley. I will try to describe each of these to you, with my opinion of the advantages and disadvantages of each.

## GEARS

There are four types of gears in general use: Spur gears, Helical Gears, Angle/Bevel gears, and Worm gears. Spur, Helical, and Angle/Bevel gears transmit torque by a couple of teeth on the drive gear pushing against a couple of teeth of the driven gear. Note that "couple of teeth". Even though each gear may have dozens of teeth, only a few are in contact at any one time. (There is a formula for figuring out how many, and it is complicated). We'll get back to this in a while.

**SPUR GEARS** A spur gear (Figure 1) is a disk much like a wheel, with teeth cut around its periphery. It is designed to mate with a similar gear, which is mounted on a parallel shaft such that the teeth are meshed together. The teeth are cut with a peculiar profile called an "involute". The property of an involute is that when two involutes are rolled together, the contact between the two is almost completely a rolling action, with only a tiny bit of sliding also taking place. The result of this is that with two hardened steel gears running against one another in an oil bath, frictional losses are only around one percent. In other words, a simple, well-designed spur gear box will run at better than 98 percent efficiency. This is one of the great benefits of gears; they're efficient.

Figure 1

Typical Spur Gears

On the other hand, that high force/small contact area is the Achilles' heel of spur gearing. While it can be overcome, still the most important thing to watch for in designing a gear train is wearing of the gear teeth.

The most common way it can be overcome is to make the gears of hardened steel, and enclose them in a dirt-proof enclosure, preferably filled with oil or grease. Sometimes, that requirement is just too hard. I was surprised to find out that the final drive on a typical diesel/electric locomotive is nothing but a huge pinion (the smaller drive gear) on the electric motor shaft, running with a very large gear that is integral with the drive wheel. The assembly is covered with a light, sheet metal cover, mostly to keep rocks and small animals out of the gears.

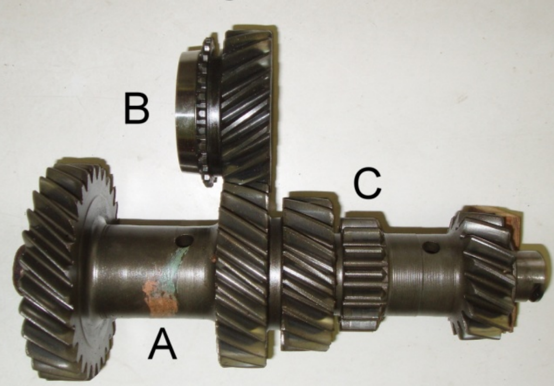
  

**HELICAL GEARS** While spur gears are very useful for many things, they have a flaw that makes them annoying in some applications. As succeeding pairs of gear teeth come into mesh, they contact each other along a line that stretches all the way across the teeth. This contact makes a small but audible noise. As the speed of rotation of the gears gets faster, this noise becomes a very audible whine, which is sometimes offensive. Remember the scenes in old 1930's movies, when our hero takes off in his beautiful Packard roadster with a whine that sounds like a worn-out dirt truck. This was because the low gear in the old transmissions was a simple straight-cut spur gear. In those transmissions, second gear was a helical gear, as were the gears between the input shaft and the cluster gear, so they were quiet. By the 1960's, manufacturers decided to spend a little more money on their transmissions, and cut all gears except reverse as helical gears.

Helical gears are a variation on spur gears, but with the teeth cut at an angle to the primary axis of the gear, usually an angle of from 15 to 30 degrees. When these gears mesh, the teeth first contact only at the outer edges of the gears. As the gears turn, that contact spreads across the faces of the gear teeth until it forms a continuous line across the teeth, and the gears have accepted their full share of the torque. This allows the gears to run almost silently, regardless of RPM. Note that you almost never hear gear whine in modern automobiles. (Automatic transmissions have always used helical gears). All the formulas for spur gears apply to helical gears, but must be applied in the plane of the gear, not at the helical angle of the teeth.

Figure 2 shows a typical automobile cluster gear (A), with most of its gears cut at a helical angle. Meshing with that gear is a second-speed gear (B) that would normally be mounted on the output shaft. (C) is a straight-cut spur gear which would have a straight-cut reverse gear that would slide into mesh with it. This illustrates one of the problems with helical gears. Since the teeth are

Figure 2

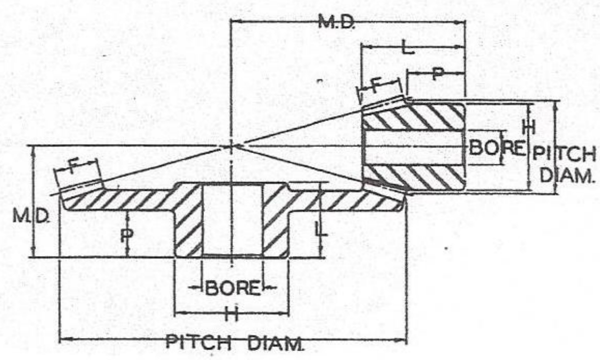



Helical Gears

cut at an angle to the axis of the gear, under load, they develop an axial force, requiring thrust bearings to retain them. In Figure 2, the second speed gear is restrained from moving axially on the output shaft, and is clutched to the shaft if second gear is desired, or unclutched from the shaft if some other gear is selected. The reverse gear is unique, in that it is slid along its shaft into and out of mesh with the gear on the cluster gear. If it was cut on a helix, it would either suck itself into mesh, or spit itself out of mesh, depending on the direction of the torque.

**ANGLE/BEVEL GEARS** If the basic geometry of the spur gears is that of two cylindrical disks mounted on two parallel shafts rolling together, the basic geometry of the Angle/Bevel gears is that of two cones on two shafts mounted at an angle to one another, such that the two cones touch from their apexes to points on their pitch diameters. (Figure 3)

Figure 3




Angle Gear Geometry

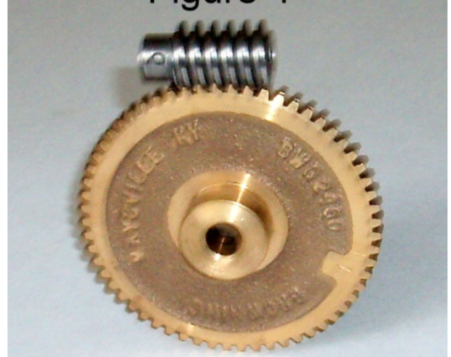
As with spur gears, angle/bevel gears need to have teeth cut in them to drive against one another, and for obvious geometric reasons, those teeth have to taper from a maximum at the pitch diameters to nothing at the apexes of the cones. It is generally found practical to truncate the cones somewhere around one-third of the way from the base to the apex, so, unlike the spur gears which are two disks running together, you have two cones running together, but this time, the disks are mounted on shafts at an angle to one another. From Figure 3, we can see that gears of different sizes (different ratios) running together will have different cone angles.

It is common to have the shafts mounted at right angles to one-another. If the gears on each shaft are the same size, they are then called "Bevel Gears" and are a very handy and common way for a rotating shaft to turn a corner.

All of the comments about small number of teeth under high contact pressure with resulting low friction and need of sealed enclosures that were made concerning spur gears are applicable. In addition, the angle of the shafts must be accurate, and the centerlines of the shafts must meet at the same point. If these requirements are not met, the teeth of one gear will only touch the teeth of the mating gear at one point, causing extremely high loads and wear.

**WORM GEARS** A frequent problem with spur or angle gears is that a single pair of gears will only give a maximum gear ratio of around six to one. If one needs, say, sixty to one, it takes three spur gear passes. To get around this limitation, the Worm and Worm Gear were invented. (Figure 4) These are gears that work on an entirely different principle from spur gears. In its simplest form, the worm is exactly a screw thread, with the flanks of the thread bulged a bit to make it an involute in the plane of the worm gear. The worm gear lies in the same plane as the length of the worm. In its simplest form, the worm gear has teeth cut across it at a small angle to match the angle of the spiral of the single tooth of the worm. The worm and worm gear are mounted on shafts that are perpendicular to one another, and offset by a distance equal to 1/2 the sum of the pitch diameters of the worm and worm gear. (The concept of "pitch diameter" will be discussed later in the section about designing with gears.) Each rotation of the worm causes the worm gear to advance one tooth. Thus, a 60-tooth worm gear will give a ratio of 60:1.

Figure 4



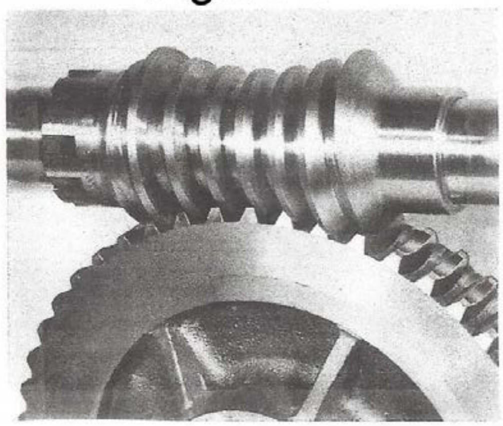
Worm and Worm Gear

To increase the torque capacity, a couple of things are routinely done, but at an increase in cost. First, the teeth of the worm gear are generally expanded from a straight line to a curve which matches the curvature of the worm. If you cut a machine nut in half down the middle you would see teeth curved to fit half-way around the matching screw. In real life, the teeth of the worm gear fit one-third of the way around the worm. This is called a "single enveloping" worm gear. (Figure 5) A similar thing can be done to the worm itself, by cutting the worm from an hour-glass-shaped blank. This way, the teeth entering and leaving the worm gear are larger than the teeth in the center of the mesh, so for a "double enveloping" worm and worm gear, perhaps four teeth share the torque load. (Figure 6)

Figure 5


Single-Enveloping Worms

Figure 6



Double-Enveloping Worm

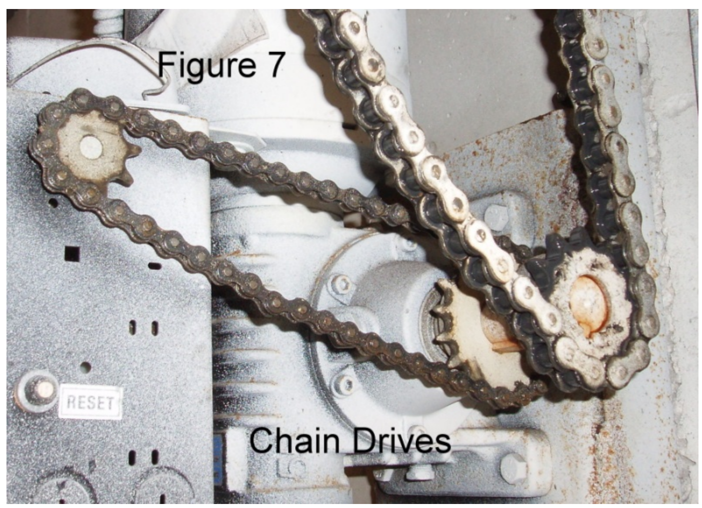
The additional expense for these better patterns of worms and worm gears is not only because the worm and worm gear are more difficult to fabricate, but the shafts have to be exactly perpendicular and mounted at an exact spacing (as in the simplest form). Also, the worm gear has to be kept exactly aligned over the worm (no side-to-side motion) and the worm has to have its waist exactly aligned to the center of the worm gear (no end-to-end motion). This means that both the fabrication and the assembly of the worm gear box are more expensive.

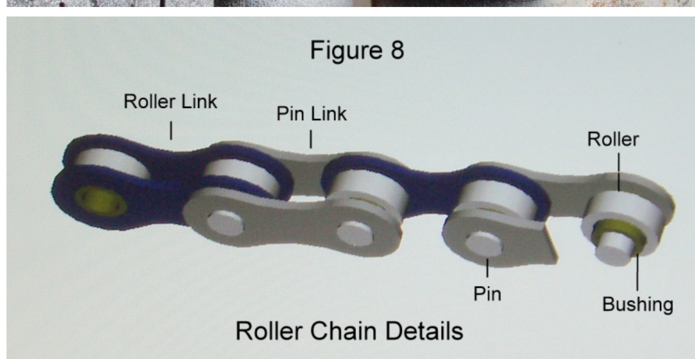
The typical worm has only one tooth, wrapped around the body of the worm like the thread on a machine screw, but there are other configurations less commonly found. A “two lead worm” has two teeth at 180 degrees from one another. In this case, for each revolution of the worm, the worm gear advances two teeth instead of the one for a single lead worm. It is also possible to manufacture worms with three or four teeth, but these are seldom found, since the usual function of a worm and worm gear set is to provide a large gear ratio in a single set of gears. Multiple lead worms cut the ratio by the number of leads.

**EFFICIENCY** A good, hardened-steel spur gear set running in an oil-filled enclosure will have an efficiency greater than 98%. Efficiencies of worms vary greatly, but a typical single-enveloping worm/worm gear set running in a grease-filled enclosure will have an efficiency of around 30%. What this means is that two thirds of the available power is lost as heat and must be dissipated by the surfaces of the enclosure. Only one-third is available to do the work. Nevertheless, for many applications, the cheapness and simplicity of using, for instance, a steel worm running with a plastic worm wheel in a grease-filled housing to match a small, 5000 RPM electric motor to a set of slow-moving windshield wipers will be irresistible.

So, if hardened steel gears in an oil-filled housing is too expensive a solution for the application, what are the alternatives? The first is using plastic gears, or a large plastic gear being driven by a small metal gear. Most plastics are self-lubricating, so as long as the tooth loads are low, a plastic gear train will work fine. They are routinely seen in applications such as computer printers, where they generally outlast the printer. Another alternative is to use another of the drive trains listed below. Let's look at Chain Drives.

**CHAIN DRIVES** (Figure 7)




Many things in our lives utilize chain drives, and there are a few different kinds of chains. The most common chain is the Roller Chain, which is used on most bicycles and motorcycles. Indeed, it was invented in the 1870's as part of the creation of the modern bicycle. The key piece of technology in a roller chain drive is the chain itself. (Figure 8) The chain consists of pairs of metal links, held together by hardened steel pins. Keeping the links separated side to side are rollers which turn freely around the pins. The chain engages sprockets, which are wheels with teeth. These teeth are essentially gear teeth, and their flanks are formed in an involute curve. In use, as each link approaches the sprocket, the roller is contacted by the forward side of a sprocket tooth, and rolls down the flank of that tooth until it is seated in a circular depression between the teeth. The link then rides around the sprocket until it is travelling back toward the other pulley, and then it disengages.

The key to the efficiency of the roller chain is the roller. As long as the roller can turn freely on its pin, it will roll smoothly down the flank of the sprocket tooth, causing a minimum of wear and friction.

The real advantage to a chain drive is that two shafts some distance apart can be coupled together with only two sprockets and one chain. Another advantage to the chain drive is that the gearing ratio between two sprockets is exact, regardless of wear. If you need two shafts to be driven with (for example) a two-to-one ratio, like the camshaft in an engine, then if you use an 18-tooth pulley on the crankshaft and a 36-tooth pulley on the camshaft, the cam will always rotate at exactly half of the crank speed.

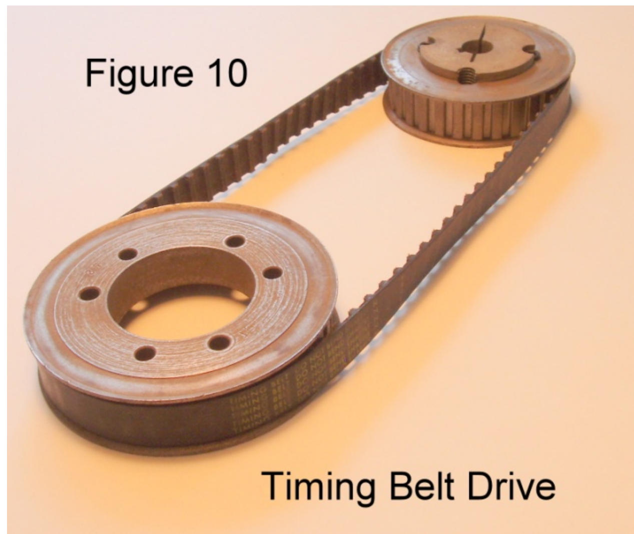
One disadvantage of the chain drive is that there is a substantial load in the chain. The force in the chain can be found by dividing the torque seen at one sprocket by the radius of that sprocket. (Remember to keep units consistent. If the torque is in foot-pounds, then the sprocket radius must be in feet.) This high load in the chain generally means that the chain and sprockets must be made of high-strength steel. This means that the chain must be kept lubricated to minimize wear. That's good, because the lubrication will minimize corrosion in the chain. The good news is that unlike gears, which suck in all sorts of airborne grit and crud into the critical tooth surfaces, with roller chain, most of the wear takes place between the rollers and the pins, and that spot is pretty well protected from grit intrusion. Therefore, most chains are run in the open, and the wear is acceptable.

Figure 9

Silent Chain Drive

A second disadvantage of the roller chain is that in some applications, they are considered noisy. For this reason, chain manufacturers invented "Silent Chain" (Figure 9). In this type of chain, the sprocket teeth are shaped very much like gear teeth, and the chain links have involute gear teeth built into the insides of the chains. These teeth mesh relatively silently with the sprocket, and are a fixture for cam drives for overhead valve engines.

**TIMING BELTS** (Figure 10)




Timing belts, and their associated pulleys are a recent invention, dating from WWII. They are made possible by recent developments in plastics. A timing belt is a flat rubber belt, with stubby, involute teeth on the pulley side of the belt. The belt itself is reinforced by very strong (frequently Aramid) fibers near to the outside of the belt. The rubber teeth are protected from wear by a layer of slippery, wear-resistant plastic cloth bonded to the teeth on the pulley side of the belt. The cloth is frequently made from one of the many varieties of Ultra High Molecular Weight (UHMW) plastic.

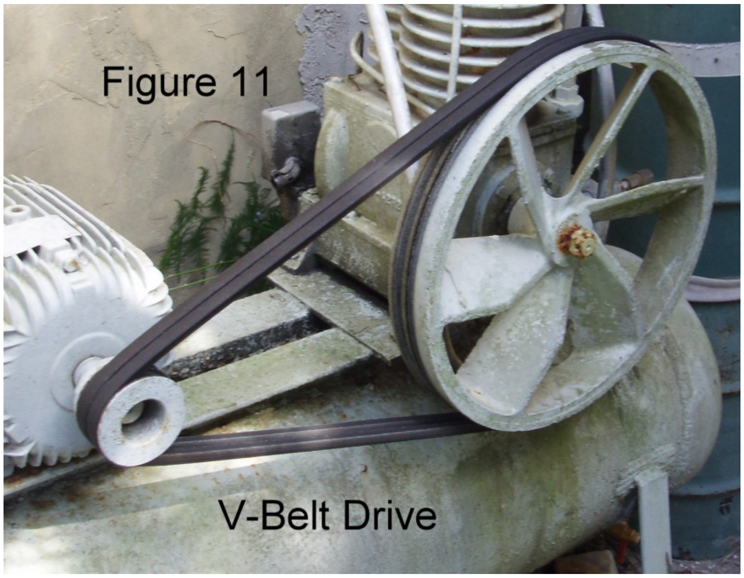
An advantage of the timing belt is that it can be used to allow a driving shaft to drive a driven shaft, even though the shafts are a long way apart. Timing belts share this virtue with chain drives. Another advantage is that thanks to the development of "magic" plastics, timing belts are always run dry, and even without lubrication, they have long service lives. Timing belt/pulley assemblies have become the standard means of driving the cams in an overhead cam engine. On my Honda, the recommended timing belt replacement interval is 100,000 miles, even though the belt is only protected from the environment by a simple plastic housing.

The reality of that plastic housing is that it is there to keep tools, neckties, and especially fingers from getting caught between the belt and a pulley on a running engine. If you should be unfortunate enough to get a finger caught in a timing belt, major trauma is going to happen to your finger, since the Aramid fiber is STRONG and has almost no stretch.

**V-BELTS**

All belts operate on friction. Friction from the driving pulley pulls on the belt, and friction from the belt pulls on the driven pulley, causing it to turn. First came flat, leather drive belts, invented sometime in the Middle Ages. Up into well into the 20th century machine tools were driven by leather belts. But leather is not very strong, so belts had to be made from one long rectangle of leather with the ends sewn together. They were not capable of transmitting much torque for their size. Also, leather against dry steel is quite slippery, so to transmit a lot of torque, the belts had to be quite tight against the pulleys. More tightness tears the sewn joint in the belt, and causes the belt to stretch. So somewhere in the latter part of the 19th or early part of the 20th centuries, someone invented the V-Belt. But first, Charles Goodyear had to invent vulcanizing, so that the rubber would last, and people like Harvey Firestone had to learn how to fuse strong cords into rubber. But when that had happened, someone then designed the V-belt and its mating V-pulley. (Figure 11) He gave the V-belt a heavy fabric cord backing to give the strength necessary to resist the high force on the belt when there is a high torque load on the V-belt/V-pulley assembly. He first made the belt flat, and the rubber inner surface transmitted torque a lot better than the old leather belts. But then he had the really bright idea. He made the belt in the shape of a wedge, so that the tension on the belt would try to jam the belt into the pulley groove. With a wedge angle of around 60 degrees, about twice as much force is developed between the belt and the pulley as between a flat belt and its pulley.




Let's go back and look at the whole belt/tension/torque thing. Say we have an electric motor with a two-inch pulley on its shaft, driving an alternator with a two-inch pulley on its shaft. Suddenly, someone puts an electrical load on the alternator, requiring ten inch pounds of torque to drive the alternator at that speed. Since the pulley has a one-inch radius, it will require ten pounds of force at the periphery of the pulley to drive it. If it is a flat belt, the coefficient of friction between the leather belt and the steel pulley will be about .30, so we need thirty pounds of tension on the belt to provide the friction to drive the load. If it is a V-belt, the rubber has a coefficient of friction of about .7, and the wedge of the belt into the pulley groove will double the amount of effective friction available, so we will need only 10/2 ÷ .7 = seven pounds of tension to provide the friction.

The other thing that making the cross-sections of the belt and pulley into a V-shape did was to solve the belt tracking problem. A flat belt, like a leather belt, running on a cylindrical pulley will tend to walk off the pulley when it runs. Our forefathers learned that crowning the pulley (making it barrel shaped) made the belt track stably. With the V-belt, the V of the belt must ride down in the groove of the pulley, and the belt will never come off the pulley. BUT, if the pulleys are not reasonably well aligned, then the belt will rub heavily against one side of the pulley, and will soon wear out.

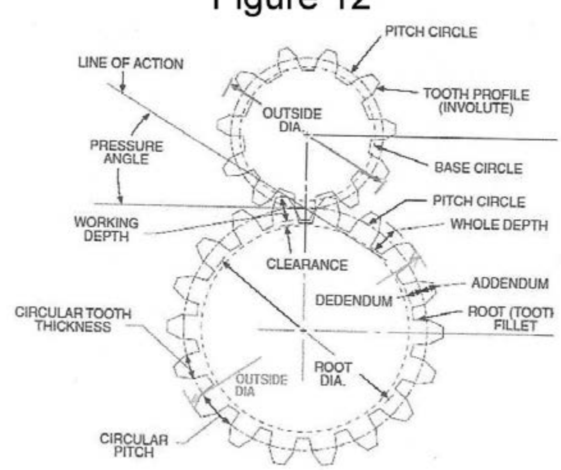
V-belts have certainly stood the test of time. From the beginning, automobiles have used V-belts for everything from fan belts to alternator belts. Today we think of V-belts being obsolete, having been replaced by "serpentine" belts. But if you look closely, a serpentine belt is just a set of six or ten V-belts joined at their flat backs. They run against pulleys that have six or ten grooves in them, instead of just one.

## DESIGNING DRIVE TRAINS

### SPUR GEARS

Gears are sized by their "Pitch Diameter", "Face Width" and tooth size. We know that if you run a "two-inch" diameter gear with a "four-inch" diameter gear, they will turn at a ratio of 2:1. But if you were to measure the diameter of a "two-inch" gear, it would be larger than two inches. The "Pitch Diameter" of a gear is measured at a point about half way out along the teeth, (Figure 12) so the outside diameter is really a function of tooth size.

Figure 12




Gear Terminology

If one were to design a gear train with a 2" pitch diameter gear driving a 4" pitch diameter gear, the centers of the gears would be placed (accurately) 3" apart. (The radius of the 2" diameter gear is 1", the radius of the 4" diameter gear is 2"; 1" + 2" = 3"). Gears can be run on their theoretical centers because the AGMA (American Gear Manufacturers Association) standard dimensions for tooth size, tooth thickness, gear runout, etc., are designed to the maximum metal condition, with all tolerances running small. So the tolerances always make the gaps between teeth on mating gears larger. If properly toleranced, gears should never bind.

Gear teeth are sized using a concept called "Diametral Pitch". (Other pitch standards such as "Circular Pitch" are also used, but diametral pitch is the common one for the US). If a 2" diameter gear has 16 teeth, the tooth size is said to be 8DP. This means that the tooth thickness, spacing, etc., are such that exactly 16 teeth will fit around the pitch diameter of the 2" gear. If the 2" gear had 32 teeth, they would be 16DP, and would be half the size of the 8DP teeth. 8DP teeth are stronger than 16DP teeth, but their tolerances are coarser and so they must run with a larger gap between mating teeth (called "backlash") than 16 DP teeth.

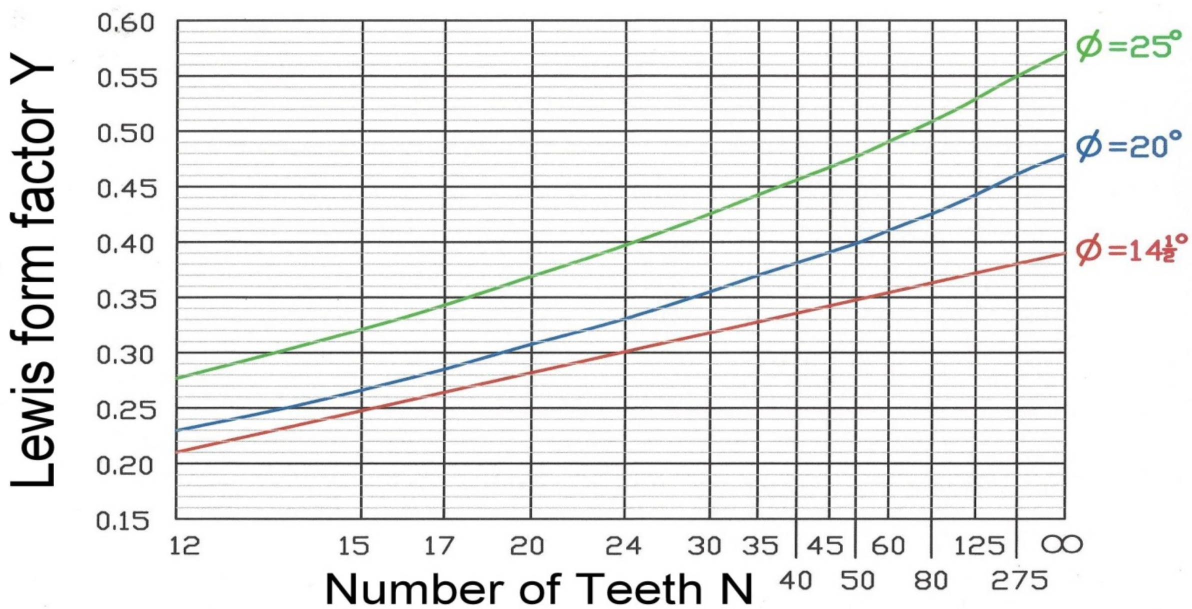
RULE NUMBER 1: Gears that mate with one another must have the same size teeth. An 8DP gear will not run with a 16DP gear, or a 9DP gear, or any other tooth size. Both gears must be 8DP.

Gear strength is a matter of tooth strength. It is the teeth that break, putting the gear out of action. There is an old formula for computing tooth strength, called the "Lewis Formula".

$$\mathrm{S} = \frac{W_t P_d}{\mathrm{FY}}$$

Where S equals the stress in the tooth, $W_t$ is the tangential force on the tooth, $P_d$ is the diametral pitch, F is the face width of the gear, and Y is a constant from a chart that reflects the number of teeth, tooth form, and pressure angle Ø.(Figure 13 below).

Figure 13

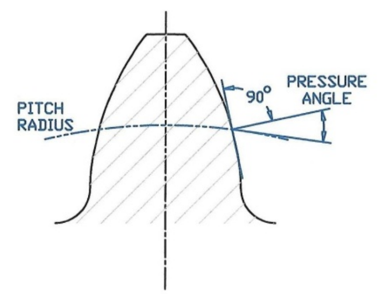



Note that a gear with twice the face width is twice a strong. But from the graph (Figure 13) the smaller the number of teeth, the lower the Y and the higher the stress. This generally means that since an 8DP gear tooth on a 32 tooth gear is stronger than an 8DP gear tooth on a 16 tooth gear, the face width of the 16 tooth gear is generally made wider to increase its strength. That also cures a second problem: When gears are offset, the ends of the teeth bear against the flanks of the mating gear. This causes a stress-raiser, and also provides an escape path for the lubricating oil. Both of these things cause excess wear. It is always good practice to make the smaller gear a little wider than the larger gear.

One final feature of spur and bevel gears that needs to be mentioned is called "Pressure Angle(ø)". The involute curve that generates the shape of the flanks of the gear teeth does not lie along a radial line from the center of the gear. It forms different angles with the pitch radius (because it is a curve), but the angle that the perpendicular to the flank of the gear tooth makes with the radius at the pitch line is called the pressure angle. (Figure 14)

Figure 14




Pressure Angle

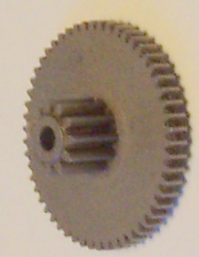
In early years, gears were cut to a pressure angle of 14 1/2 degrees. Almost all modern gears are cut to a pressure angle of 20 degrees. If you think of the pressure angles from both the front and back faces of the gears coming together, they would form a triangle A triangle 40 degrees wide will be thicker at its base than one 29 (2 x 14 1/2) degrees wide. Therefore, 20 degree pressure angle teeth are stronger than 14 1/2 degree pressure angle teeth, but at a cost of slightly more rubbing, so slightly less efficiency. That is why almost all modern gears are cut with a 20 degree pressure angle. BUT, a 20 degree pressure angle gear CANNOT be run with a 14 1/2 degree pressure angle gear.

Let's see how this all works out in practice. Say that we have a 100 HP motor driving a boat propeller through a 2:1 reduction gear. As a first guess, let's try mounting a 2" diameter gear on the motor shaft, driving a 4" diameter gear on the propeller shaft. The 2" gear has 16 teeth (2" x 8 DP), and a 1.25" face width, while the 4" gear has 32 teeth (4" x 8 DP) and a 1" face width. Both gears are 20 degree pressure angle. How did I know all these things? I guessed. Now, let's analyze the gearing and see how close the guess was. 100 ft-lbs is 1200 inch-lbs (units must be consistent). Stress = 1200in-lbs /1 in. radius x 8 (Diametral Pitch) / 1.25" (face width) x .27 (from chart). Thus, the stress at the base of the tooth is 28,400 psi. For the 4" gear, the force is still 1200 lbs x 8 DP / 1" face width x .36 from chart = 26,666 psi. With no safety factor applied, those stress numbers are appropriate for mild steel. Smaller gears would have higher stress, allowing for a lighter housing, but would have to be hardened.

**COMPOUND GEARS** As we have learned earlier, it is hard to get a reduction ratio of more than about 6:1 from a single pair of gears. So, what if you want a ratio of 24:1 or 100:1? The standard way of doing this is to use "compound gears". A compound gear is a large gear and a small gear mounted together on the same shaft. (Figure 15) They are also frequently cut out of the same blank so they are actually a single piece of metal. Typically, you mate a small gear with the large gear of a compound gear, and the small gear of the compound gear with a second large gear. The total ratio will be the product of the ratios of both gear meshes. So, for our 24:1 overall ratio, we would start with a small gear (say 16 teeth) meshing with a large gear (say 96 teeth). This gives us a ratio of 96:16, or 6:1. Then we attach another 16 tooth gear to the hub of the 96 tooth gear, and mesh it with a 64 tooth gear. This gives us a final ratio of 6:1 x 4:1, or 24:1. Note a couple of things: The first set of gears never meshes with the second set of gears. It drives the second set, but does not mesh with it, so they don't have to be the same DP. Also, since torque is multiplied by the reduction ratio, there is six times the torque on the second set as there was on the first set. This indicates that the second set of gears should be a smaller number DP (say an 8DP if the first set was 16DP), or the face width should be wider. Generally both are done, guided by the Lewis formula.

Figure 15



Compound Gear

**GEAR QUALITY** The accuracy of manufacture of gears is guided by standards set up by the **AGMA** (American Gear Manufacturers Association). AGMA standards range from AGMA **Q3** very coarse to AGMA **Q15** exceptionally accurate. Those standards govern everything from tooth form to the runout of the blank that the gear was cut from. They are always toleranced such that there will always be backlash between two mating gears of the same quality, if they are mounted on their theoretical centers.

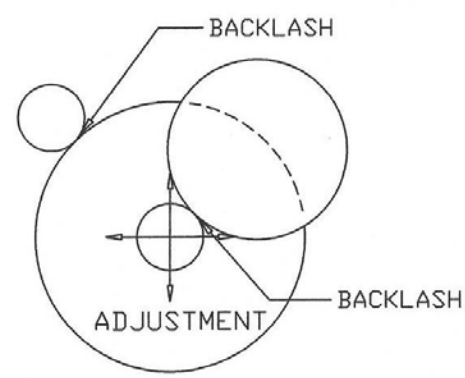
**BACKLASH** Backlash is the term for the gap between meshing gear teeth. Ideally, gears should have a small amount of space between meshing teeth for two reasons: The first is to allow for the tolerances in the manufacture of the gears, such that if the teeth of the gears are a little out of position, or the gears are not quite round, that the teeth do not jam together. The second reason for a little gap is to allow room for lubrication, such that the lubricant is not completely squeezed out from between the teeth.

Gears are toleranced such that if the centers are located precisely where they theoretically should be, backlash will always be positive, and of an acceptable amount. An example of non-adjustable gear centers is found in the transmission of a car, where the transmission case is machined accurately such that the main shaft and the countershaft are in a precise, quiet relationship with one another. An example of adjustable centers is the rear axle (differential) of a rear-wheel-drive car. Both the small pinion gear and the large ring gear are adjustable along their axes of rotation to get the backlash "just right" where the gears run quietly

If there are two or more "gear passes" in the gear train (small gear to large compound gear is one pass, small compound gear to large output gear is another pass), then there are three gear centers (for two passes) involved. The input gear rotates about one center, the compound gear rotates about the second center, and the output gear rotates about the third center. (And so forth for a gear train with more passes.) If a line from the first center to the second center forms an angle with a line from the second center to the third center (Figure 16), the first and third centers can be fixed, and the second center made adjustable to get the correct backlash between both the first and second gear passes. This is frequently desirable, because the input gear should run in alignment with the shaft of the drive motor, while the output gear should run in alignment with the mechanism that is being driven. Therefore, adjusting only the center compound gear minimizes the need for moving large things like the drive motor or the driven mechanism around.

Figure 16




Two-Pass Gear Set

**ANGLE/BEVEL GEARS**

The same rules that apply to aligning and meshing spur gears apply to aligning angle and bevel gears, but angle and bevel gears are more difficult. It is easy to tolerance the locations of spur gears along their shafts such that their faces align properly, but angle/bevel gears must also be aligned accurately along their axes of rotation. Remembering that the gears are the base portions of theoretical cones, we must align the cones such that the tips of the cones coincide exactly (or to a small tolerance). This means that not only do we have to tolerance or adjust the cones along their axes until they touch, but they must also be toleranced or adjusted in the other two dimensions, both left and right and up and down. This can provide quite a challenge for the machinist or the NC machine. This is less of problem today than it used to be, since normal gear location tolerances are on the order of a few thousandths, while NC machines generally hold tolerances to a few tenths of thousandths.

**WORM GEARS**

As was mentioned in the section introducing worm gears, the alignment of their shafts is a significant issue. With a single-enveloping worm gear (the most common kind), the shafts must be at an accurate right angle to one another, and at an accurate spacing from one another such that the teeth mesh properly. Since the worm gear teeth are curved to fit partly around the worm, the worm gear must be located accurately along its axis, so that the worm is located at the center of the curvature of the worm gear teeth. But since the worm is a straight thread like a machine screw, it can be located to a loose tolerance along its axis.

If the worm and worm gear set are the double-enveloping kind, where not only is the worm gear wrapped partly around the circumference of the worm, but the worm is shaped like an hour glass to engage more of the worm gear's teeth (the strongest and most efficient arrangement), now the worm has to be located accurately along its axis. This is to assure that the gaps between the worm gear's teeth that are in mesh with the worm are the same all along the worm. This is made more difficult by the fact that there is a large axial load (along the shaft) on the worm, so its bearings have to be able to take thrust, as well as radial load.

**CHAIN DRIVES**

Compared to gears, chain drives are much easier to design. Basically, you don't "design" a chain drive, you decide on how much horsepower that you must handle over what RPM range, and look in a manufacturer's catalog to find out what size chain is appropriate. Then you select the appropriate sprockets to give you your desired ratio, and then, using the tables in the catalog, determine what length of chain that will give you the desired spacing between the driving and the driven shafts. This is usually called the Center Distance. Remember that a chain is made up of links that are an exact size, so a center distance that requires a chain XX1/3 links long is impossible.

At this point, you are almost home free except . . . compared to gears, chains can wear a lot while still functioning normally. A gear whose teeth have worn a few thousandths will get objectionably noisy, while a chain can wear as much as one percent of its length, and still function smoothly. This is sometimes referred to as the chain "stretching". If the chain is on a motorcycle or an old coaster brake bicycle, you can make the rear axle adjustable, taking up the chain tension that way. But if the chain is driving a camshaft in an engine, changing the center distance is not feasible, so an idler must be used. The idler may be simply adjustable by loosening a fastener, making the adjustment, and tightening the fastener, or it can be spring loaded, putting more load and wear on the chain.

With regard to idlers for both chain and belts, ideally, the idler should be on the slack side of the chain. By this I mean that when the motor drives the load, it does so by pulling on one side of the chain. The other side of the chain is under no load and hangs loosely. This is where the idler should be located. Also, the idler should push the slack side of the chain toward the tight side, thereby increasing the angle that the chain wraps around the smaller of the two sprockets. The more teeth that are engaged by the chain, the less the wear on the chain and the sprockets.

Once the basic geometry has been determined, then the question arises, "Shall we put it in a housing"? On most bicycles, the chain has no housing, to save weight. The additional wear and shorter life due to the chain being exposed to dirt, and having minimal lubrication is considered an acceptable trade-off for the light weight. The timing chain on an automobile engine is covered by an oil-tight housing, and is sprayed with oil. The small amount of weight that this entails is considered an acceptable trade-off for the much longer life afforded by the protection given the chain.

**TIMING BELT DRIVES**

Designing a timing belt drive is very similar to designing a chain drive. You enter the manufacturer's catalog with the desired horsepower and RPM, and take their recommendation as to timing belt size, and thus the associated pulley sizes. You also look up the center distances for the timing belt and timing pulleys that you have selected, and, just like in chains, you can't have a XX1/2 tooth belt. Not only that, but timing belts are only made in certain lengths, so you must try to live with what's available. The good news is that timing belts don't "stretch" like chains do. In fact, the wear on a timing belt during its life is almost inconsequential. The only part of the belt that wears is the slippery fabric that covers the rubber teeth, and that has very little effect on the system geometry. Like chains and V-belts, timing belts fail catastrophically. Generally, the fabric over the teeth wears out, allowing the teeth to be ripped off the belt, allowing the belt to slip on the smaller of the pulleys. Belts should be replaced before they show wear, but that can be a long time. Properly designed automobile engines of today require that the timing belt be changed every 100,000 miles.

Unlike chains, timing belts require no lubrication, and no covers, except for safety reasons. The fibers that form the tensile strength of the belt are so strong that if a finger or other body part gets caught between the belt and a pulley, it will be crushed. Plan accordingly.

Because of the lack of "stretch" in timing belts, and also because of the precision of their manufacture, it is perfectly permissible and normal to run timing belts on fixed centers. No pre-tension of the belt is necessary, but the belt must still be assembled onto the pulleys, or the belt and pulleys assembled onto their shafts. At this point, we need to look at how the belt is kept from tracking off of the pulleys.

Basically, timing belts are kept on their pulleys by flanges on the pulleys. But it only takes two flanges (total) to keep the belt on the pulleys. Two flanges on one pulley, one flange on the left side of one pulley and another flange on the right side of the other pulley, whatever combination makes assembly the easiest. In general, as they come from the catalogs, the smaller pulleys (fewer teeth) have flanges, the larger pulleys (more teeth) don't.

**V-BELTS**

V-belts and their pulleys are easy to design with, long lasting, and adaptable to dirty environments. Their only limitation is that they can't handle much horsepower. When we reach the world of around a dozen horsepower, v-belts start to get bulky and cumbersome. It is time to look at timing belts instead.

V-belts transmit torque by the friction of the sides of the belt against the sides of the pulleys. It takes force to create friction, and that force is manifested as tension in the belt. Provision must be made to provide that tension, either by adjusting the center distance of the pulleys or by adding an idler into the system. In addition, the sides of the belts wear, which reduces the tension, also requiring readjustment. A caution when adjusting the tension on the belt: Each pound of tension on the belt causes two pounds of radial force on the bearings of each of the shafts. Internal combustion engines produce an oscillating torque, which tends to make v-belts "flap". Adjusting the belt tension solely to eliminate the flap can sometimes load the bearings enough to cause their early failure. Belt flap is not really that bad a thing.

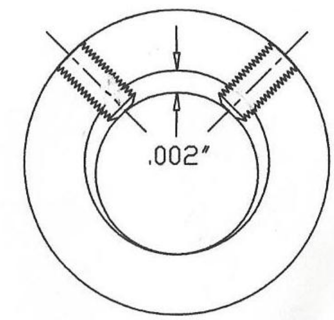
The geometry of a pair of v-belt pulleys is pretty simple, and generally easy to accomplish. The shafts must be parallel, and the pulleys must be in line. As a general rule of thumb, the shafts should be parallel to within a degree, and the pulleys should line up to within 1/16" for pulleys in the range of 6" in diameter, and to within 1/8" for pulleys of a foot or so in diameter. Closer is better.

**ATTACHING GEARS AND PULLEYS TO SHAFTS**

So far, everything that we have talked about involves round things like gears and pulleys that are attached to shafts that rotate in bearings to transmit torque. While the manufacturers of gears and pulleys would like you to think that attaching their product to a shaft was simple and reliable, the reality is complex and difficult.

**SETSCREWS** Let's take the simplest and most common attachment, set screws. Suppose we want to attach a gear with a 1/2" bore to a 1/2" shaft. The gear will be supplied with a bore tolerance of .500" to .501". The shaft will be supplied with a diameter tolerance of .500" to .499". This means that in the best case, the gear will have to be pushed firmly onto the shaft, perhaps with a little lubricant, and the worst case, the gear will fit loosely by .002", which will feel quite loose. (Figure 17) The gear is supplied to be attached to the shaft with two cup point setscrews located 90 degrees

Figure 17



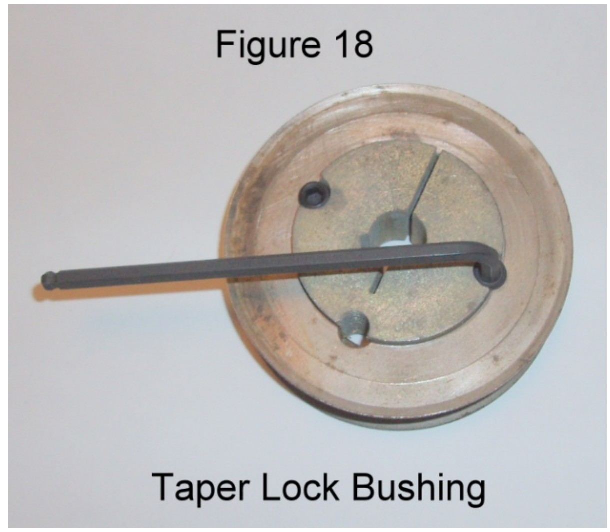

Clearance Between
Pulley and Shaft

apart. When the setscrews are tightened, the gear is forced .001" off center, which for an accurate system, may not be a good thing. Next, imagine a line drawn from half way between the setscrews to the point at which the shaft contacts the bore of the gear. (The only point.) When, in the course of rotation of the gear, the load comes perpendicular to the line, it will try to skid the contact point of the shaft sideways in the bore. Then, 180 degrees of rotation later, when the load is acting the other way, it will skid it back. This "fidgeting" can cause the gear to loosen on the shaft. So you find this out, and decide to take the gear off to fix it. You loosen the setscrews, and the gear slides perhaps 1/16" on the shaft and then locks up, requiring a gear puller to remove it. What has happened? Cup point setscrews, as they are torqued into the shaft, displace small rings of metal that are a couple of thousandths of an inch high. While this burr is still aligned with the setscrew hole, the gear is loose. As soon as the finished bore tries to pass over the burrs, it locks up. Moral: You MUST machine two flats into the shaft to accept the setscrews. While I paint a bleak picture for setscrews, actually, for small, lightly loaded power trains, they work pretty well.

**KEYWAYS** While the torque on a gear, if small, can be taken by the friction of the setscrews, a lot more torque can be taken by a keyway. A keyway can also be used to provide a phase relationship between the gear and the shaft. But while a keyway seems simple in concept, tolerances again make things more difficult. Ideally, the key should be a line-to-line fit in both the keyway in the shaft and the keyway in the gear. But that's the best case. In the worst case, the key is .001" under nominal, and both the keyways are .001" over nominal, so the key is .002" loose in both the shaft keyway and the gear keyway, allowing for +/- .002" of fidgeting between the gear and the shaft. For an electric motor driving a load in one direction, that's fine. For a gasoline engine driving a lawnmower, keys die young.

For a square key, nothing naturally restrains the key from working its way out of the keyway axially, so one of the setscrews should be located right above the key. For a Woodruff key (which is a semi-circular key that fits in a semi-circular keyway in the shaft), it is automatically located by the keyway.




**TAPER-LOCK BUSHINGS** (Figure 18) When gears, pulleys, etc. reach a few inches in diameter, and have to handle a few horsepower or more, many are available with taper-lock bushings, which are a good choice for power trains having to handle significant power, or which have reversing loads (see gasoline engine driving a lawnmower, above). The taper-lock bushing is a simple and effective concept. Where the bore of the pulley would be, there is a large, tapered hole. Fitted into this hole is a bushing that has been bored to fit the shaft size, and then split along the side opposite to the keyway that is on the other side of the bore. The bushing is slipped over the shaft and key, and the pulley slipped over the bushing. Screws (typically three) are tightened, pulling the bushing into the pulley. The tapered bore of the pulley compresses the bushing against the shaft, insuring concentricity and omitting fidgeting. When the screws are properly torqued, the bushing clamps so firmly around the shaft that the entire torque is taken by friction of the bushing against the shaft. To remove the bushing and pulley, the screws are removed from the bushing, and replaced into other holes such that the screws bear against the face of the pulley. As the screws are torqued, the bushing is withdrawn from the pulley, and the whole thing slides off of the shaft.

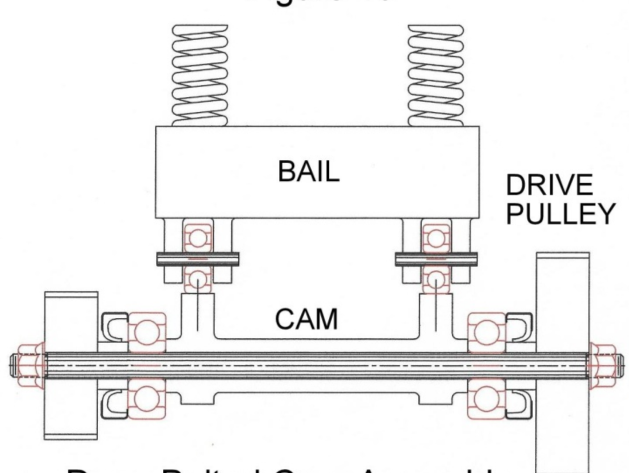
**TWO PULLEYS/GEARS MACHINED FROM THE SAME BLANK** For extreme situations, such as automobile transmissions and some servo systems, there are no reliable ways of attaching gears to a shaft. In this case, the gears or pulleys must be custom-machined from a single piece of stock. In production, this may not be as bad as it sounds, since the hobbing of the gears (machining of the gear teeth) costs the same whether they are made from a single piece or not, and the extra machining of the blank is partially repaid by the lack of setscrews, keyways, etc. (See Figure 2)

A word about servos: The highest grade of servos are ones where the current driving the motor is supplied by a linear amplifier, and the amplifier is wired to work in all "four quadrants" of servo performance. The quadrant that poses a challenge in keeping the gears/pulleys from slipping on their shafts is the one where the load is being driven at high speed in one direction (say, clockwise), and the motor is called upon to reverse its direction instantaneously. In normal operation, the rotation of the motor causes the motor windings to generate a voltage that opposes the voltage input to the motor. This voltage is called "Back EMF" The difference between the input voltage and the back EMF is what actually drives the load, and that's what the motor is rated for. If the motor and the load are not moving, no back EMF is generated. If power is then applied to the motor, the full voltage is applied across the motor windings, and the motor produces a torque about four times its rated torque. Now if the motor and load are rotating at full speed, producing a lot of back EMF, and the motor drive voltage is reversed, it adds to the back EMF, and the motor will produce 8 - 10 times its rated torque. This is what the gear sees, and if the drive train has only been designed to handle the motor's rated torque, bad things will happen.

**DRAW BOLTING** This is a little-known but extremely effective technique for attaching gears/pulleys to their driving shafts. If there is one thing that I would like you to take away from this course, it is this technique.

As a very good (and amazing) example of how it can be used, let me describe the technique the first time that I saw it. Once upon a time, I worked for a company that made machines that punched IBM cards. (That was a long time ago.) The punch had a drive shaft that drove a three-lobed roller cam at 2200 RPM. The cam drove a structure called a "bail" about 1/4" up and down at 6600 cycles per minute. The bail was held against the cam by two springs of about 100 pounds compression each. (Figure 19) (If you think that this whole thing looks like an automotive valve train, you are right.) The problem was that the springs were so strong that it took the full stall torque of a one-horsepower motor to get the cam follower over the nose of the cam to get the punch running, but as soon as the follower got over the nose of the cam, the springs pushed the follower rapidly down the cam,

Figure 19




Draw-Bolted Cam Assembly

almost completely reversing the torque on the drive shaft.

The drive shaft was only 1/2" in diameter. It had a 1/16" square keyway down the length of the shaft, not to resist the torque, but to assure that the cam was phased properly with the output pulley. The whole thing went together like a shish-kebab. The cam was a smooth slide fit over the shaft and keyway, with a bearing on either end of it. Outside the bearings were two spacers that had oil seals running against them. Outside the spacers were a driving pulley on one end and a pulley to drive a companion mechanism on the other. Bearing against each pulley was a 7/16", fine-thread nut. These nuts were torqued to 20 pounds feet. This punch was in production for years, and there was never an instance of a pulley slipping on the shaft, or the cam slipping on the shaft, or wear on the key.

Think about it for a minute. We torqued the nuts to 20 pounds-feet, so there was a frictional force of 20 pounds-feet on one face of the pulley. But the nut was pressing the pulley back against the face of a spacer, so there was another 20 pounds-feet of friction on the pulley from that source. So we have perhaps 40 pounds-feet of friction holding a pulley from rotating against a load of 9 pound-feet. A huge safety factor. And the real payoff comes during assembly and disassembly. When the nuts are loosened and removed, the shaft is pulled smoothly out of the pulleys, cams, bearings, etc., freeing everything.

I have used this technique many times since, but never in as brutal an application as the one above. It has always been an easy and cheap method of attachment, and has never failed me.

## BEARINGS

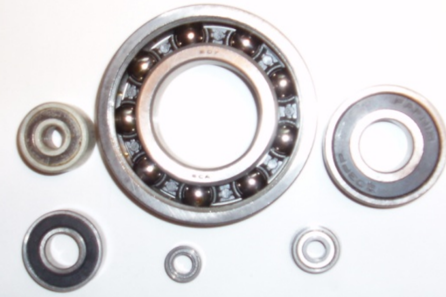
The place where drive trains and linkages intersect is bearings. If it moves, it must have bearings, even if the bearing is a simple as a hole in a steel link pivoting around a steel pin.

Bearings come in two types, rolling-element bearings and plain bearings, and it is not always obvious which type is better for the application.

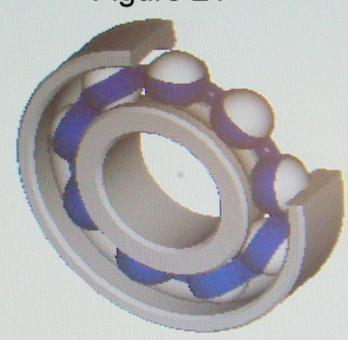
**BALL BEARINGS** Ball bearings come in two different types, radial and angular contact. A radial bearing (Figure 20) is designed to take a radial load, that is, a load in the plane of the bearing. But because of its construction, balls rolling in grooves in the races, it can take a modest end load before the balls try to run out of their races. This end load is usually about one third of the radial load. In contrast, angular contact bearings (Figure 21) are designed to take a thrust load. They do this by having asymmetrical grooves. If an angular contact bearing was lying on the table before you, the outer race might have a groove with a high lip on the bottom side and a very low lip on the upper side. Meanwhile, the inner race groove will have a high lip on the upper side and a very low lip on the bottom side. When you push down on the inner race, the high lip on the inner race presses against the balls, which press against the high lip on the outer race. This allows the bearing to carry high thrust loads. But if you should push up on the inner race, the bearing will come apart at a very low load, as the balls just climb up over the very low lips. (That's how the bearings are assembled).

Figure 20



Radial Ball Bearings

Figure 21



Angular Contact Bearing

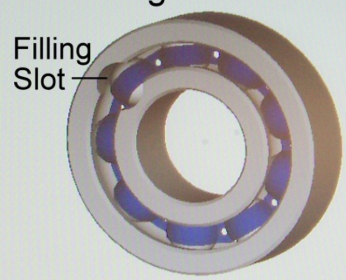
Angular contact bearings are frequently used in pairs, one at each end of a shaft. Generally, some means is provided to preload the bearings one against the other to eliminate free play. It is not unusual to have a spacer of a very precise length between the inner races, and the outer races located in recesses in the housing, with the distance between the lands of the recesses controlled very precisely. The shaft can then have nuts on each end, torqued against the inner races and the spacer. This forms a very stiff, reliable assembly, but at significant cost.

Figure 22

Conrad Bearing

Assembly divides the radial bearings into two classes, Conrad and full-complement. Most bearings are of the Conrad type, (Figure 22) which are assembled by pushing the inner race down against the outer race, leaving a large gap at the top. There is then room for the balls to be inserted into the gap, and then slid down the grooves, until no more balls can be inserted. The races are then centered, and the balls distributed evenly around the bearing. One will then note that only about half of the balls that could be there actually are. Then two halves of a "retainer" are fitted in, one from each side, holding the balls in their places. The two halves of the retainer are then riveted together. Retainers hold the balls in their places, while being made of soft enough material to not scratch the balls.

Figure 23




Filling Slot Bearing

But obviously, if we could put more balls into the bearing, it would carry more load. Thus we have the "full complement" bearing. (Figure 23) This is made by having a slot cut into each race from the outside to the center of the race. When the races are aligned, the slots allow just enough room to pass a ball through the slot into the race. The bearing can then be filled with as many balls as will fit around the races. A retainer is then added both to keep the balls from scratching one another, and to keep balls from falling out through the filling slot. The one drawback to this type of bearing is that it can't take any axial load, because of the filling slot.

The basic ball bearing consists of two races, a complement of balls, and a retainer to keep the balls properly located. A bearing like this needs to be in a sealed, oil-filled housing in order to last. Bearings are also readily available with either seals or shields protecting the bearing from contamination. (Figure 24) These seals and shields can be furnished on either one or both sides of the bearing. Many bearings are used with two seals which are used to retain grease within the bearing, and to keep

Figure 24

Sealed Bearing Shielded Bearing

dust and dirt out of the bearing. This is a very effective setup, and is commonly found in automotive alternators, lawnmower blade spindles, and the like. However, seals have a small amount of drag, and they also wear out eventually. Instrument bearings are frequently furnished with shields, which are thin metal plates fixed to the outer race and running with a very small clearance to the inner race. The clearance is small enough to keep most contamination out, and will keep a charge of oil in the bearing. They work well in a lab or office environment, but should not be used outdoors.

**TAPERED ROLLER BEARINGS** Up until the early part of the 20th century, railroad freight cars rolled on steel axles turning in babbit (a soft metal) bearings lubricated by oil-soaked cotton waste. These were constantly causing trouble, and in addition, had a lot of friction as the cars started to move, before the oil had formed a lubricating film. Ball bearings, which were new at the time could not be used, because a ball bearing large enough to handle the load would be too big to fit in the space formerly occupied by the babbit block and the cotton waste box. (Ball bearings are quite large radially for their load capacity.) Replacing the entire truck and its four wheels was unacceptably costly to the railroads. A Connecticut company called Timken invented a bearing that has become an industry staple, not for just the railroad industry, but for automobiles, power transmissions, and many other applications. This is called the Tapered Roller Bearing. (Figure 25)

Figure 25

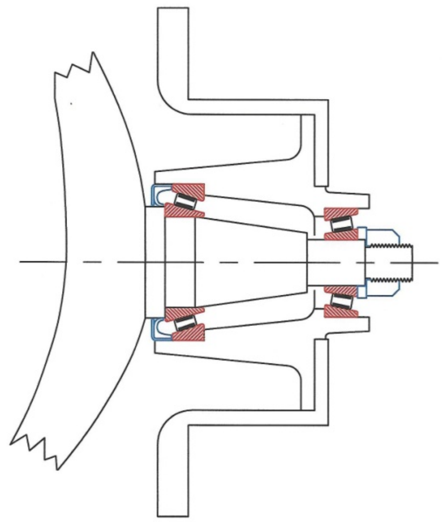
Tapered Roller Bearing

The tapered roller bearing has an inner race with a cylindrical bore, so that it can be slid on a shaft. The bearing surface is conical, with a taper of around 10 to 15 degrees. Rolling on this taper are a set of rollers which are also tapered, a taper that comes to the same point as the taper of the inner race. These rollers are spaced apart by a retainer like the balls in a ball bearing, and are frequently prevented from coming off the inner race by a shoulder on the tapered surface of the inner race. This entire assembly of race, rollers, and retainer fits into an outer race (called in the trade a "cup".) which has a taper that also comes to the same point as the inner race and the rollers. The outer surface of the outer race is generally cylindrical, so that it will fit into a bore in the housing. Because the rollers contact the races along a line against each race, instead of a point on each race as does a ball bearing, the rollers can take much more load for their size than can the balls of a ball bearing.

This type of bearing is an angular-contact bearing, and has similar properties. It can take large radial loads, but reacts to radial loads by requiring thrust loads just to keep the bearing together. But because of the cone-into-a-cone geometry, tapered roller bearings can take large thrust loads as well. A typical use of tapered roller bearings is in the front wheels of automobiles. (Figure 26) In this application, an inner race and roller assembly is slid onto the front axle until it bottoms out against a shoulder, with the small end of the cone facing outward. The wheel hub, with a "cup" pressed into each end of its bore is slid over the axle until the inner cup bears against the inner race assembly. A second inner race and roller assembly is slid over the axle until it bears against the outer cup, and a nut is tightened against the inner race until a small preload is developed between the two bearing assemblies. This two-bearing assembly comfortably handles the weight of the car as a radial load, and the lateral forces of cornering as a thrust load on one of the two bearings. The assembly is lubricated by grease held in by a seal at the inner end and a cap at the outer end, pressed in over the nut.

Figure 26



Front Wheel Hub

**NEEDLE BEARINGS** A third type of bearing in common use today is the needle bearing, (Figure 27) frequently found in outboard motors and automotive transmissions. Needle bearings have a very high load capacity in a very small radial space. They usually consist of a thin outer shell made of very hard steel, containing as many long, thin rollers as can be packed into the housing. Generally, there is no inner race, that function being provided by a hardened shaft. They are usually run in an oil-filled housing, and the part of the housing that contains the bearing must be strong and rigid, as the load from the needles is transmitted right through the thin shell and

Figure 27

Needle Bearing

supported by the housing. There is no retainer. The function of the retainer is provided by the oil film, which keeps the needles microscopically apart.

The theory behind the needle bearing is simple, but not obvious. In a ball bearing, a few of the balls support the load. The load is transmitted through the inner race through a very small round contact patch into the ball, and from there through a somewhat larger contact patch to the outer race. The load carrying capacity of the bearing is a function of the stress at the contact patch and the strength of the material of the bearing. In a needle bearing, first, there are many more needles sharing the load. Then the contact patch is a line along the length of the needle between the inner race and the needle, and between the needle and the outer race, as with the ball bearing. These contact lines have much more area than the small round patches of ball bearings, so much more load can be carried.

In designing a mechanism with needle bearings, first, the oil film <u>must</u> be maintained, and second, the shaft must be carefully aligned with the axis of the bearing. If the shaft is skewed in the bearing, only one end of the needles carry the load, reducing the size of the contact patch and hence the load carrying capacity. Also, skewing the shaft may cause the needles to slew crosswise in the shell, jamming them.

**OTHER TYPES OF BEARINGS** There are many other types of rolling element bearings available, but they are not very common. Some of the types are the following: Ball bearings can be manufactured with two ball grooves in the same race, called "double row bearings". Angular contact bearings can be manufactured with very precise control over the widths of the races, such that the bearings can be mounted clamped against one another. If they are mounted face-to-face, the pair can take large thrust loads in both directions. If they are mounted back-to-face, they can share the load, hence carrying twice the thrust load, but in only one direction. Roller bearings look like ball bearings, but instead of balls, they have short, thick rollers. Because of their longer contact patches, they can carry very high radial loads, but no thrust loads. Spherical roller bearings are like roller bearings, but the rollers are barrel shaped, and run in races where the ball groove is gently curved crosswise. This actually means that the ball groove in the outer race forms a section of a sphere. This allows the bearings to be assembled by pushing the inner race with its spherical rollers and retainer crosswise into the outer race, and then rotating the inner race assembly 90 degrees to align the two races and seat the rollers in the ball groove in the outer race. It also means the bearing has some self-aligning capability, that is, that the shaft and inner race can be run at a small (few degree) angle with the outer race and housing.

**BEARING CAPACITY AND HARDNESS** Rolling element bearings are commonly made from AISI 52100 steel. This steel has a very high carbon content which allows it to be hardened to Rockwell C hardness of 62, which is about as hard as a file. Both capacity and life drop dramatically with decreasing hardness. A bearing made from steel with a hardness of only Rockwell C 40 (about as hard as a screwdriver) has only about 1/2 the capacity and life of one at Rc 62. This means that there is no such thing as a "stainless steel" bearing, although many are advertised. The best "stainless" material is AISI 430, which can be hardened to Rc 60, but which is not very corrosion resistant. Bearings made of AISI 300 series stainless are so soft as to be virtually useless.

This hardness issue is also a problem for bearings where the shaft acts as the inner race, such as some needle bearings. In order to develop adequate load and life, the shaft must be hardened to Rc 62, which usually means case-hardening the area of the shaft that serves as the inner race.

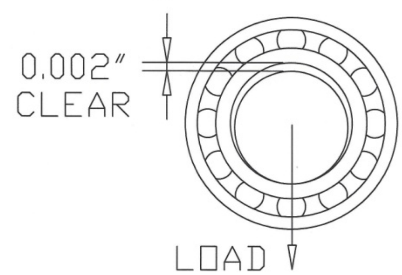
**REPLACING BEARINGS** One thing that all bearings have in common is that they are wear parts, that is, they are designed to have a limited (but hopefully long) life, and be replaced when they wear out. Many of us have had the experience of having an expensive appliance or power tool or similar that had to be scrapped because one of its "bearings" was a hole in the plastic housing, and it finally wore out. Yet even when the bearings are dedicated, unique entities that should be replaceable, it is not uncommon to have a situation where there is no practical way to get the worn-out bearing out, so that it can be replaced. Let's look at some of the reasons for that, and some possible solutions.

You take the armature out of an electric motor, and it has a ball bearing on either end. You can feel that the bearings are worn, and you would like to slip them off to replace them, but you can't because they are pressed on, and must be removed with a gear-puller. Is this done just for the manufacturer's convenience? No, there is a very good and important reason for the bearings to be a press fit.

In order to slide a bearing onto a shaft, there must be clearance between the bearing and the shaft. For small shafts (less than an inch), anything less than 0.001" clearance will bind. In addition, both the bearing and the shaft have manufacturing tolerances, even though they are small. So, worst case, you wind up with the shaft being 0.002" smaller in diameter than the inner race of the bearing. Which means that the circumference of the shaft is 0.00628" smaller than the circumference of the bearing. Now, normally, the motor is used such that the shaft rotates and the load is "stationary", meaning that the load always comes from the same direction. Think of the motor driving a V-belt, for example. So the load pulls the shaft to the bottom of the bearing, and the shaft rolls along the bottom of the bearing as the bearing turns beneath it. (Figure 28) So like a hamster running in its cage, we have the soft steel shaft running in the hard steel bearing inner race, with no lubrication. In time, the shaft will wear smaller, and the difference in speed between the shaft and the bearing will be more pronounced. The shaft will wear out, not the replaceable bearing.

Figure 28




Shaft Loose in Bearing

Now, take an automotive alternator apart. You take the case apart, and the rear sleeve (plain) bearing slips right off the shaft. A plain bearing (generally oilite) can be used there because most of the load is on the front bearing next to the drive pulley. Then you remove the nut holding the pulley to the shaft, and not only does the pulley come off, but the armature drops out of the front bearing. Why is the front bearing not pressed on? It sees the same load situation as the front bearing on the motor armature, so what is the difference? Remember the drawbolting technique explained earlier. The nut clamps both the pulley and the inner race of the bearing against a shoulder on the shaft. There is clearance between the bearing and the shaft, but the shaft can't walk around the inside of the bearing because the nut clamps it firmly, even against the pull of the belt.

It is hard to overemphasize how important it is to totally prevent the shaft from "walking" inside the inner race. I have seen a good 3/8" shaft made from 416 stainless steel that was running at 3600 RPM wear itself into junk in a dozen hours of test time. For a machine that should last 20 years, that is a problem. There are many "patent" solutions for this problem, but they are generally expensive, and frequently don't work in the application. One that does work is the use of "Loctite" anaerobic adhesive. In practice, slide the bearing over the shaft, apply a drop of the proper grade of Loctite between the shaft and the inner race, and the bearing is bonded to the shaft. However, in order for it to work, both the shaft and the inner race must be spotlessly clean. One salami-stained fingerprint and the bond will fail. Finally, when you need to replace the bearing, it is glued to the shaft, and must be removed somehow.

Let's next look at the "shaft stationary, housing rotating" case. This is typical of front wheel bearings. The shaft is an integral part of the front spindle, and remains stationary, while the wheel rotates around it. In this case, since the load is always vertical and the shaft is stationary, the inner race of the bearing does not walk around the shaft, because neither of them turn. But the housing (wheel hub) turns, and if the outer race of the bearing was loose in the wheel hub, it would walk around the inside of the hub, wearing it out. So we find that the outer races ("cups") of the tapered roller bearings are pressed into the hub. Fortunately, tapered roller bearings slide apart easily, so it is easy to remove the front wheel for service.

Look back at Figure 26. The upward force on the wheel from the weight of the automobile will work on the tapered roller bearings to force them apart. The inner race near the spindle will be forced against the shoulder of the spindle, and the inner race near the nut will be forced against the nut, so that is solid. Meanwhile, from Newton, the cup nearest the hub will be forced outward against the shoulder in the hub, and the cup nearest the nut will be forced inward against a shoulder in the hub, so even without pressed fits, both cups are retained. But still, they are pressed in, requiring that they be driven out for replacement, because if they were not pressed in, they would walk around the inside of the hub, wearing it out.

A problem with the use of ball bearings is that the manufacturer's tolerances are very precise, and beyond the capabilities of many shops. As an example, let us look at the tolerances that Fafnir (division of Torrington) requires for the Housing Stationary, Load Rotating case for the very common #203 bearing: Housing: 1.5748 - 1.5754 (mean fit .00055 loose: Shaft: .6695 - .6692 (mean fit tight .00020). I repeat, these are the tolerances that <u>you</u> must provide for the bearing to slip easily into your housing, and press gently onto your shaft. Yes, I know that bearings will still work for a while with looser tolerances, but for the full rated life of the bearing, those tolerances should be held. Which gets us back to drawbolting, which allows us to use looser tolerances on the shaft, since the clamping force on the inner race of the bearing prevents the shaft from "walking" in the bearing.

**PLAIN BEARINGS** Plain bearings are the place where drive trains and linkages intersect. They both require bearings, even if it be as simple as a pin through a hole in a link. (Think of a pair of scissors). A plain bearing, basically, is a simple sleeve of soft material that a shaft of harder material can run in with little wear. In earlier times, these bearings were made of wood (lignum vitae), babbit (a soft metal related to solder), and soft brass or bronze. In any case, the bearing/shaft combination worked better and lasted longer if it was lubricated. The babbit or lignum-vitae bearings of the covered wagons were lubricated by grease or lard. The lignum-vitae propeller shaft bearings on steamships were lubricated by grease or water. The babbit bearings on freight cars were lubricated by oil retained in containers of cotton waste. By the post-WWII era, Oilite had been invented by the Chrysler Corporation. Oilite is a product that is formed from tiny bronze balls sintered together under heat and pressure, then the spaces between the balls are filled with oil. Visually they look like solid bronze, but in reality, they are over half oil by volume. A tip: never buy Oilite bearings that have been stored in cardboard boxes. The cardboard will absorb the oil right out of the bearing.

In more recent times, plain bearings made of several kinds of plastic have been developed. These have the wonderful feature of not requiring lubricant, which means first, that they don't have to be periodically relubricated, and second, that dirt drawn into and held by the lubricant doesn't form a messy sludge.

While many manufacturers offer proprietary formulations for bearing material, the three common bearing materials are nylon, Teflon, and polyethylene, generally in its high-molecular-weight form (UHMWPE). They all have their advantages and their disadvantages.

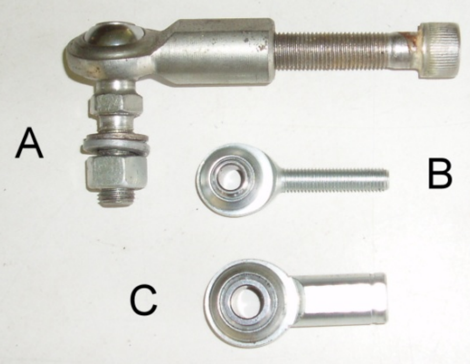
It is not automatic that a rolling-element is ideal and a plain bearing is the cheap, fall-back position. For example, one of the largest electric motor manufacturers uses oil-lubricated bronze bearings in preference to ball bearings, and claims better life with the bronze.

**ROD END BEARINGS** Rod End Bearings form a wonderful segue to the world of linkages, since they were originally designed to form the pivot points in linkages. They come in many forms, some as simple bearings looking like ball bearings, some with shafts built into their housings, with either male or female threads, and some with shafts built into their center balls, with either male or female threads. What is distinctive about rod end bearings is that their inner race is a ball with a hole in it. This hole (or, in some cases a shaft) allows the rod end bearing to be attached to whatever part of the linkage is appropriate. Between the ball and the housing is generally a bearing surface with a spherical inner surface fitting the ball. This surface can be a plane bearing, such as brass or plastic, or in some cases, even a race of balls, like a ball bearing. In any case, it forms a nice, long-lasting bearing surface for the linkage to pivot on. (Figure 29)

The neat feature of this ball within a spherical socket arrangement is that the axis of the hole in the ball can be swung sideways through an appreciable angle, usually around 15 degrees, to allow for either accidental or intentional misalignment of the rods coming into and out of the linkage.

Figure 29




Rod End Bearings

In Figure 29, "A" shows a rod end bearing with an integral attaching stud, and a female thread, as illustrated by the inserted bolt. "B" shows a rod end bearing with a male thread, while "C" shows one with a female thread. Both male and female threads are available in either right- or left-hand threads, so it is easy to make up an adjustable-length push rod.

I believe that rod end bearings originally came from the aircraft industry, because airframes are so flexible that a linkage that was designed to operate in a plane would bind up as the linkage was flexed out of plane. Thus the spherical center, and once that was invented, many other applications were found for it. One non-aircraft application is in carburetor linkages on automobiles, and another is for suspension parts in high-performance automobiles, where the compliance of rubber bushings is intolerable, and the expense of rod end bearings is acceptable.

## LINKAGES

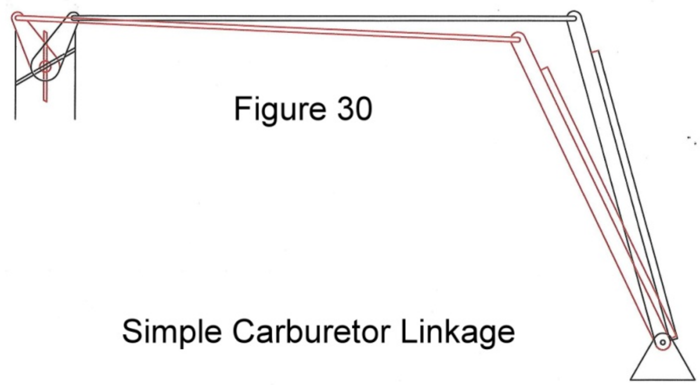
Linkages are generally used either to transmit motion through a distance, such as the carburetor linkage or the clutch release linkage used on older cars, or to constrain a motion to a particular path, such as the linkage controlling the motion of the hood on most cars.

**LINEAR MOTION TRANSMISSION** We start by looking at the "transmit motion through a distance" kind of mechanism. The most basic elements of such a linkage are the lever, the bellcrank, and the pushrod.

Let's look at how one would design such a mechanism. To start with, you usually have a known input and a desired output. For example, we have a pedal on the floor of our car that is moved by our foot about three inches, and we need to rotate the butterfly which controls the airflow into our carburetor or fuel injector through an arc of about 60 degrees. Secondary considerations are that the butterfly should have a strong enough spring on it to guarantee that it will return to idle when you take your foot off the pedal, but not so much force that your foot gets tired on a long drive. A further consideration is that you want gentle, exact control of the butterfly when you are driving in traffic, but you can tolerate much less control when you "floor it". A final consideration is that we don't have all the space in the world to put this linkage, since the engine itself is a big lump, and blocks many of the places that we might want to pass our linkage through.
If what you are trying to control is located in a straight line from where you are, you are almost home free. All you have to do is to design a pair of levers, and a rod to connect them. So we first have to look at the angles that the things that we are connecting have to work through. For example, a carburetor has a throttle butterfly that at idle is about 30 degrees to the horizontal. At full throttle, the butterfly needs to be aligned with the bore of the carburetor, or about 90 degrees to the horizontal. So the throttle butterfly needs to rotate through an angle of 60 degrees. But the accelerator pedal would not be comfortable at all if it rotated through 60 degrees. Perhaps around 10 degrees would be a better angle. This suggests that the lever formed by the foot pedal be about six times as long as the lever on the carburetor. Since the foot pedal is about 9 inches long, a 1 ½" link on the throttle butterfly would give us a lever ratio of 6:1. The point of attachment of the connecting rod on the accelerator pedal will move about the same distance through the air as the point of attachment of the connecting rod to the top of

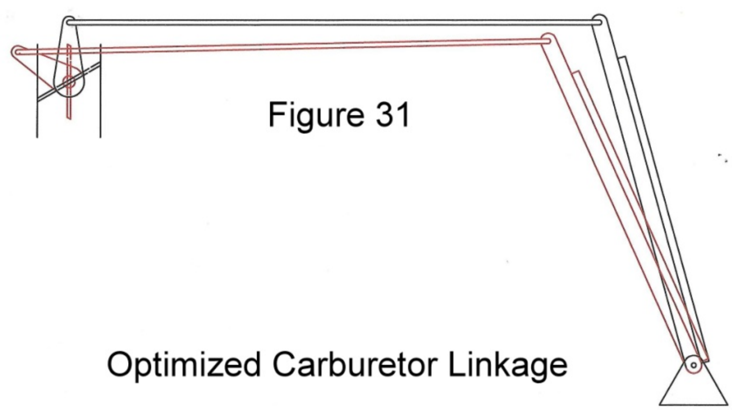
the carburetor link. (Figure 30) (In the following illustrations, the black lines show the linkage at rest, while the red lines show the linkage actuated).

If we want to, we can optimize this linkage some. First, we can set the linkage up to maximize the force that gets to the carburetor from a given load on the accelerator pedal. To do this, we tune the length of the connecting rod so that when the accelerator pedal is half-way through its travel, the carburetor link is half-way through its travel.




That will work fine, but is minimum force on the accelerator pedal the thing to optimize for? We have plenty of strength in our foot, but we need careful, precise control of the power of the motor when driving in traffic at part throttle. It would be nice if the first few degrees of the pedal travel would only move the carburetor butterfly a small amount, while the last few degrees of the pedal travel would open the carburetor butterfly a lot. How do we arrange that?

Figure 31 shows one possible arrangement. Since the pedal only moves through ten degrees, there is not much room for improvement, but since the carburetor butterfly moves through sixty degrees, we can do some optimizing there. If we start our travel with the butterfly link at ninety degrees to the actuator rod, one degree of movement of the pedal will move the butterfly six degrees, but at the end of the travel, one degree of the pedal will move the butterfly 9 1/2 degrees. So we have the best sensitivity that we can get at the beginning, and quick opening of the throttle at the end.



The linkage that we have designed might be all right for a Model A, where the updraft carburetor is mounted low on the engine, nearly in line with the foot pedal. But now Mr. Ford has changed over to a V-8 engine with a downdraft carburetor mounted up high on the engine, so our pushrod would go right through parts of the engine. As one solution to this problem, we need to introduce the third element of this kind of linkage, the bellcrank. Philosophically, a bellcrank is formed from two levers joined at their pivot ends. One lever can be designed to be at the best angle for the input pushrod, and the other can be designed to be at the best angle for the output pushrod. The bellcrank is thus able to carry the action of the pushrods through an angle, such as around a corner. But while a bellcrank that looks like rabbit ears, or Churchill's "V for victory" fingers is not an uncommon shape, it is not always the best structural shape. More about that later.

Figure 32 shows the same "optimized" linkage as Figure 31, but with the carburetor raised up an indeterminate amount to its new position high on the engine. This time, instead of the linkage going straight to the carburetor, it goes to a bellcrank mounted on a bracket on the front side of the firewall. That bellcrank is designed such that its vertical leg will be 90 degrees to the incoming pushrod half-way though its angular travel, and the horizontal leg will be at 90 degrees to the outgoing pushrod at half-way through its travel. Both legs are the same length, so the outgoing pushrod will move exactly the same distance as the incoming pushrod. At the upper end of the outgoing pushrod is another bellcrank, also mounted to the forward side of the firewall. Again, the lower leg is designed to be at 90 degrees to the vertical

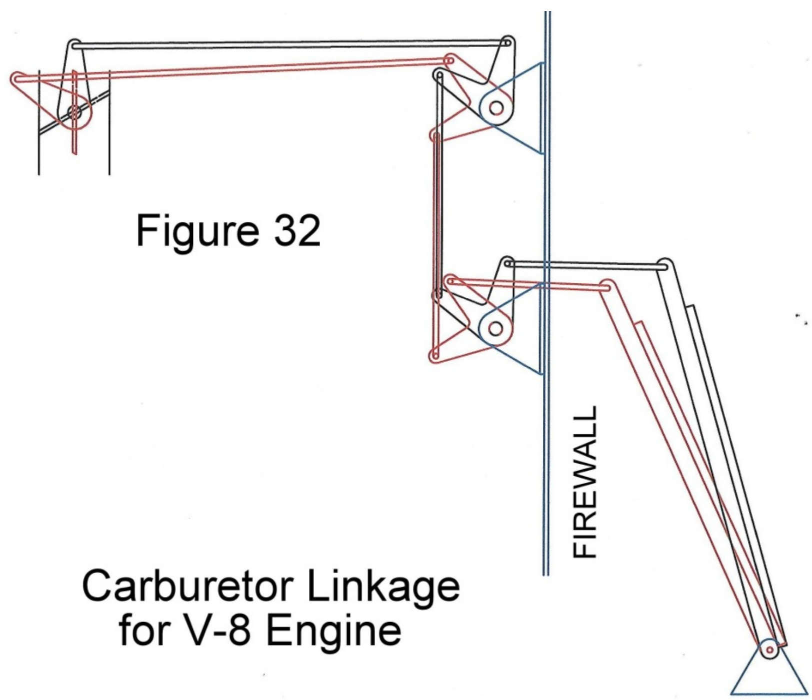


Figure 32
Carburetor Linkage for V-8 Engine

pushrod at half-way through its travel, and the upper leg is designed to be at 90 degrees to the horizontal pushrod to the carburetor. Thus we get the same effect as the "Optimized Carburetor Linkage" in Figure 31.

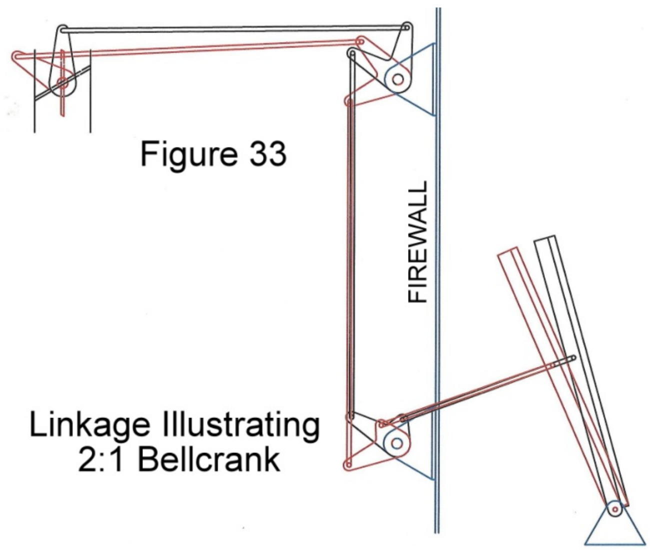



Figure 33

Linkage Illustrating
2:1 Bellcrank

Figure 33 shows a second way to design a linkage for a “low pedal/high carburetor” situation. This is to illustrate another design freedom available by using a bellcrank This time, instead of taking the pushrod from the "toe" of the pedal, we attach it to about the middle of the pedal. This means that we only have half as much stroke to move our carburetor butterfly. So we make that back up by designing a bellcrank with one lever only half as long as the other. This way, the shorter stroke from the new pushrod can be magnified up to what it was before, so the rest of the linkage operates normally.

Before we leave the bellcrank, let’s look at making it stronger. Our basic design is a simple “L” shape, with both legs in bending. If we convert the “L” to a triangle, we have a shape where the bending is converted to tension and compression in the legs of the triangle, so it is stronger and stiffer. While many times the "L" -shaped design of the previous examples will work just fine and is cheap to make, sometimes the structure of the bellcrank is important. A while ago, I had to design a linkage for a spacecraft application. The lengths of the legs of the bellcrank were about 3 inches, but the load in the pushrod was 1200 pounds. Since weight is extremely important in a spacecraft, the final design came out something like Figure 34.

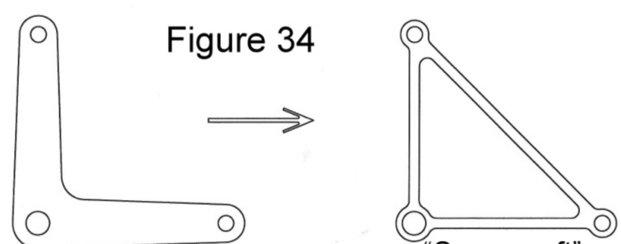


Figure 34

“Typical” Bellcrank

“Spacecraft”
Bellcrank

**DESIGN OPTIMIZATION**
Things never come out perfectly, so let's see what's important. In this case, it is important that the butterfly in the carburetor go from idle to wide open, or the engine will not develop its full horsepower. Is it important that the accelerator pedal travel through exactly 10 degrees? I don't think so, but I am not a human factors expert. In the event, by the time I had gotten the linkage in Figure 31 designed, the pedal travel was down to 8.8 degrees. I don't think that the driver will notice, but if it was important, it could have been tuned by moving the pushrod up or down the pedal, or changing the length of one of the bellcrank arms.

**PUSH-PULL CABLES** All of these sample linkages from the '30's have the same problem. The motor is mounted on rubber mounts, while the foot pedal pivots about a point firmly mounted to the body of the car. If under acceleration, for example, the motor lurches a fraction of an inch backward, that pushes the butterfly back against the pushrod, opening the throttle farther, causing more acceleration. This "feedback" can make it difficult to drive the car smoothly. Many interesting linkages were designed to fix this problem, but it wasn't until the '60's that the problem was finally solved. The invention of high-molecular-weight polyethylene allowed a cable housing to be lined with the polyethylene that was highly wear-resistant and nearly friction-free. This permitted the design of a "push-pull" cable that would last the life of the car.

A "push-pull" cable consists of a (usually) stranded steel core sliding through a polyethylene-lined flexible housing. Push-pull cables are usually used in the "pull" mode, since the core is usually much stronger when pulled than when pushed. In either case, the housing can be routed through a tortuous path, but if the ends of the housing are securely fastened, a pull that moves one end of the core 1" will cause the other end of the core to also move 1". Figure 35 shows a push-pull cable used to replace the throttle linkage shown in Figure 31.

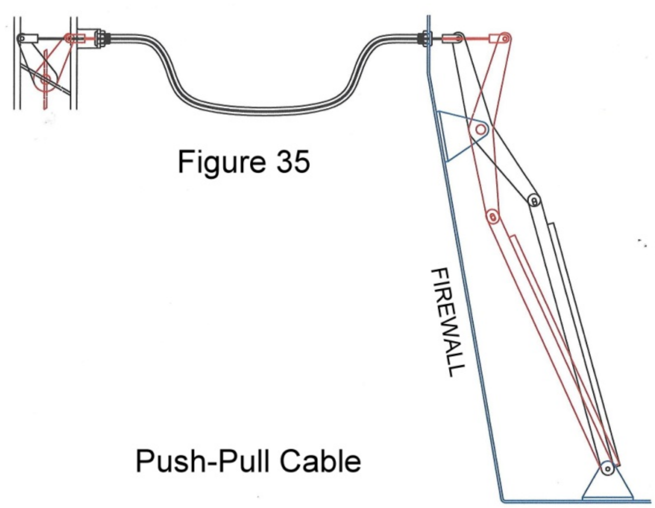


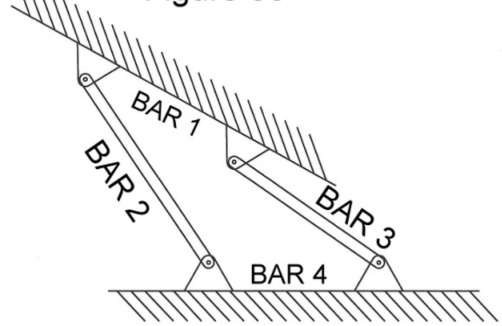
Figure 35
Push-Pull Cable

The pedal end of the housing is firmly attached to the body so movement of the engine does not change the relationship between the pedal end of the housing and the pedal end of the core. The butterfly end of the housing is firmly attached to the engine, so movement of the engine does not change the relationship between the butterfly end of the housing and the butterfly end of the core. Feedback is eliminated.

In this design, we have another variation on the bellcrank, where it has been reduced to a simple lever, transmitting the force from the pedal up the firewall to the end of the push-pull cable. Note, however, that since the upper end of the pedal travels in one arc, and the bottom end of the bellcrank travels in a different arc, the upper end of the pedal is slotted to compensate.

Outboard motors are frequently steered by a push-pull cable used in both the push and pull modes. Their cores are quite large.
A unique use of a push-pull cable was in the old Saab automobile. One end of the flexible housing was attached to the firewall, and that end of the core was attached to the clutch release pedal such that depressing the pedal pulled back on the core. The cable was then bent in a smooth 180-degree arc back so that the other end of the cable was near the end of the clutch-release lever. Normally, the end of the flexible housing would have been firmly attached to the clutch housing near the clutch-release lever, and the core would have pulled in a forward direction. But the clutch-release lever needed to be pushed backward to function properly. So the end of the core was fixed firmly to a projection on the clutch housing, and the end of the flexible housing was fixed to the clutch-release lever. At first glance, it appeared that the driver was pushing against fixity, but the tension in the core caused the housing to move backward, actuating the clutch-release lever. Weird, but it worked fine.

Figure 36



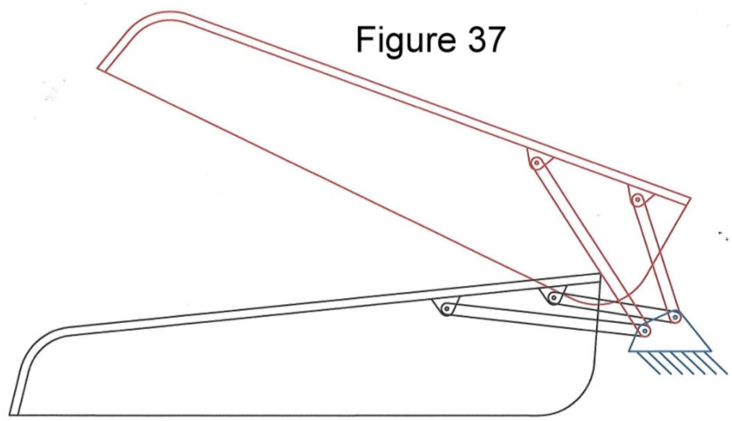

Four-Bar Linkage

**THE FOUR-BAR LINKAGE** One of the most common and useful linkages is the “four-bar” linkage, so-called because it consists of four links. In real life, two of these bars are usually large objects, and the other two are bars that connect them. (Figure 36)
A classic example of a four-bar linkage is the linkage connecting the hood to the body of most automobiles. On simple automobiles, like the

WWII Jeep, the hood is connected to the body with a simple hinge, but as styling got more elaborate, a simple hinge would cause parts of the hood to interfere with the body as the hood was raised. To correct this, stylists started attaching the hood with a four-bar linkage that caused the back part of the hood to raise up and move forward as the front part of the hood is raised. (Figure 37) The same thing could have been achieved if a hinge line could have been located about at the driver's shoulders, but that would have been awkward. Thus we have one of the most-used features of the four-bar linkage, that of creating a "virtual hinge line". Locating the virtual hinge line is easily done graphically. The centerlines of each of the two moving links (from hinge pin to hinge pin) are extended to infinity. The point at which they cross is the virtual hinge line, that is, the point about which the moving piece (hood in this case) is pivoting about the fixed piece (the body in this case) <u>at that instant in time</u>. As the linkage moves, the point at which the centerlines of the links intersect moves.

Figure 37



Hood Four-Bar Linkage

Another common use of the four-bar linkage is in the front suspension of automobiles. In the '30's, it was found that cars rode better and handled better if the wheels moved independently of one another. This can be accomplished in many ways, but the most common, for many years, was to attach the wheel to a spindle, which was connected at top and bottom to a link to the chassis. If the links were the same length, the wheel would move up and down, but would also move in an arc that was the length of each of the links. This arc would cause the tire to scrub across the pavement, causing tire wear and also causing the tire to lose traction. So, by experimentation, they came up with a linkage geometry where the upper link was shorter than the lower link. With this geometry, the tire both moves up and down, but also rocks from side-to-side. This rocking permits the point of the tire that contacts the road to move substantially up and down, minimizing scrubbing. Today, many cars have all four wheels suspended this way, but as I said before, there are many other ways to accomplish the same things.

## CONCLUSION

We have discussed many types of bearings, but have left many types unmentioned. We have discussed several types of drive trains, but there are many more that have niches. We have discussed a few different linkages, but there are many more. In short, in a four-hour course, we have covered very lightly, what would take a lifetime to learn. My principal focus has been to look at things from the practical side, from the technician's point of view, rather than the professor's. It is the responsibility of the design engineer to combine the professor's theory with the practical considerations that permit easy manufacturing and service. I hope that this helps.