Part 6 of 6:
POWER TRANSMISSION ELEMENTS

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1 Power Screws

In **Part 2 - Fasteners**, threads were considered as a means of providing a large mechanical advantage (small wedge angle) which was useful in holding and clamping together two or more components. In this section, we focus on another use of threads - for **TRANSMITTING POWER**.

The concept has many applications, from the leadscrew on a lathe, to screw jacks, to mechanical presses, motor car and truck steering mechanisms, etc. Generally, the mechanical arrangement is such that the **POWER SCREW ROTATES** and the **NUT TRANSLATES** (i.e. moves linearly) along the screw, although in applications such as the screw jack the nut rotates and the screw moves linearly to raise the jack.

1.1 Thread forms for power transmission

Several of the thread forms introduced in **Part 2** are used for power screws.

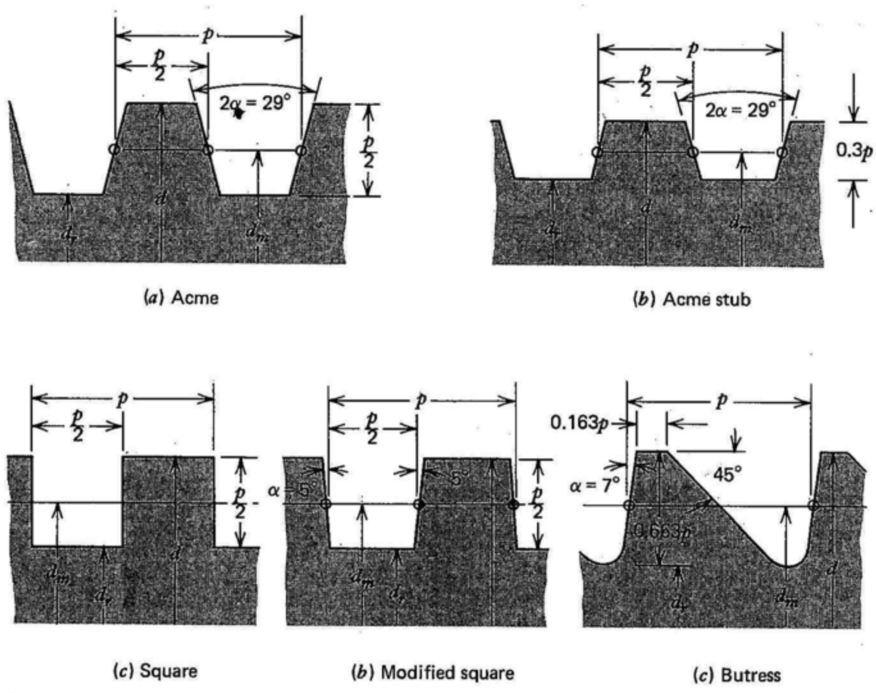


Figure 6-1 Repeated from Part 2 Fig 2-12 showing the thread profiles of a number of threads used for power transmission.

The **ACME** and **BUTTRESS** threads are easier to machine than square threads. The **BUTTRESS** thread can be used only where the applied loading is always in one direction. It is sometimes used in quick-adjust bench vices, in combination with a **SPLIT NUT**. When the two halves of the split nut are moved apart, the gap in the jaws of the vice can be adjusted simply and quickly by sliding the moveable jaw without having to use multiple rotations of the handle.
Juvinall, R C, Fundamentals of Machine Component Design, Wiley 1983, page 279.

1.1.1 Square thread

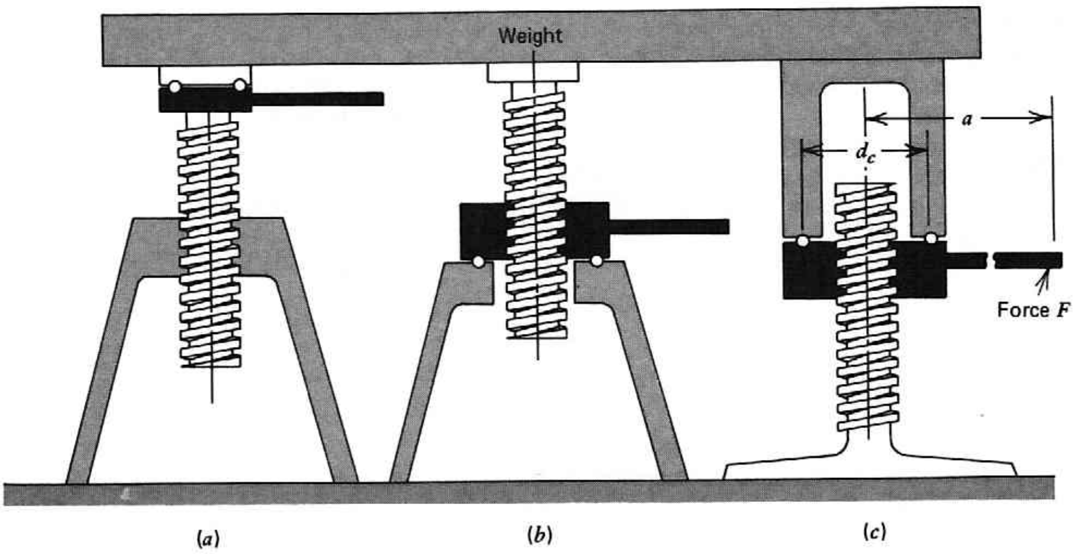


Figure 6-2 Use of a square thread in a lifting jack. In each case, the jack is raised or lowered by exerting a horizontal force F to rotate the black-shaded **LEVER** around the screw. Observe that, in each configuration, friction is decreased by interposing a **BALL THRUST BEARING** (see Project 5) between the rotating **COLLAR** and the stationary frame. **COLLAR FRICTION** is important in determining the efficiency of the jack and efforts are made to keep it as low as possible. In practice, having to rotate the lever through full 360° of movement is often inconvenient and an improved design may use a pair of bevel gears or a worm and wheel turned with a crank-handle. That way, the crank handle can be turned continuously to raise the jack (see Fig 6-3).

Due to its profile, the **SQUARE THREAD** is more difficult to machine than a V thread and is generally only used where strength, low friction and wear resistance make it worthwhile. These threads are used mainly for power transmission. There is no radial force on the nut.

Square threads are used in vices and presses, as well as in screw jacks.

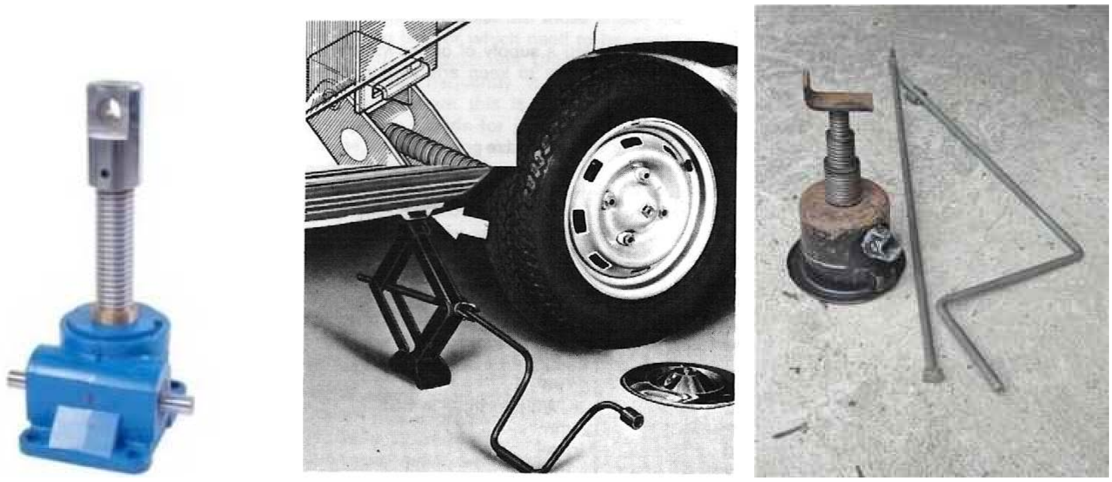


Figure 6-3 As an aside, whilst Fig 6-2 illustrates some engineering principles, this figure shows much more practical jack designs. *Left:* A worm-drive screw jack intended for industrial use. The jack is driven by an electric motor (not shown) driving either end of the horizontal shaft. Inside the housing, the shaft is in the form of a worm gear. The outer periphery of the nut is in the form of a worm wheel. The worm gear meshes with the worm wheel and drives the nut down the square thread to raise the jack. *Centre:* A much more practical jack if you have a flat tyre. The crank handle rotates the threaded rod through a simple bush on the input end and a threaded nut on the far end. Turning the handle shortens the diagonal and raises the jack. The thread may be of square or acme form although it is sometimes some sort of V thread (can be made at lower cost). There are significant advantages in this four-bar scissors-lift design – the mechanism drops to a very low height to slide under a car with a flat tyre yet can lift high enough to enable a wheel to be changed. Also, the crank handle allows continuous rotation to raise the jack in a short time, all of which is much more practical than a screw jack. *Right:* A type of jack sometimes known as a bottle jack. The cranked handle drives a small bevel gear which drives a larger bevel gear to raise the central column. Note the three stage threaded column which allows the maximum height of the jack to be more than three times its lowest height.

http://www.davidbrown.com/screw-jacks.php?
Peugeot 504 Workshop Manual, Mead, J S, Haynes Publishing 1981
Photo by Alex Churches.

1.1.2 Acme

Figure 6-4 There are several thread profiles for Acme threads, two of which are shown in Fig 6-1. Long external Acme threads are often used in powerscrew applications such as the leadscrew on a lathe, where they are usually combined with a **HALF NUT** similar to that shown in this figure. This nut is intended to allow easy and quick engagement or disengagement of the lead screw. With the nut disengaged, the lathe carriage can be moved quickly into the desired position before dropping the nut into engagement with the lead screw. Once engaged, the lead screw drives the carriage at a slow and controlled rate to move a tool along the workpiece to cut material from its periphery. See section on machining in Project 1.
http://www.fdk3co.com/images/halfnut1.png&imgrefurl

The Acme is a strong thread, used frequently for power transmission. One advantage, due to the 14½° taper angle, is that a spring-loaded **HALF NUT** such as that pictured in Fig 6-4 can be used to eliminate clearance despite some wear of the thread, i.e. the half nut is pushed a little deeper into engagement as wear occurs.

1.1.3 Buttress

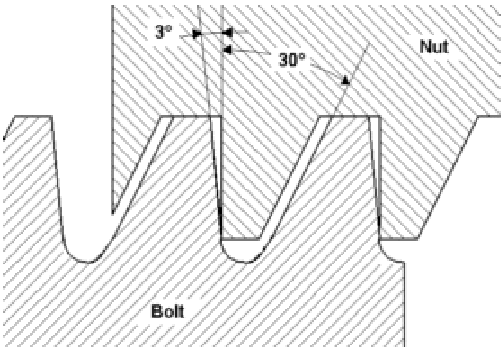


Figure 6-5 Profile of a buttress thread and an example of its use in a woodworker's bench vice. Buttress threads as shown in Fig 6-1 and 6-5 are used when power transmission is always in the one direction. A good example is the bench vice shown. In some vices of this type, a lever is provided to disengage (i.e. lift) a **HALF NUT** (see Fig 6-4) from the buttress thread so that the moveable jaw of the vice can be slid rapidly to the desired opening before final tightening.
http://goods.us.marketgid.com/goods/1780/345738/

1.1.4 Multi-start threads

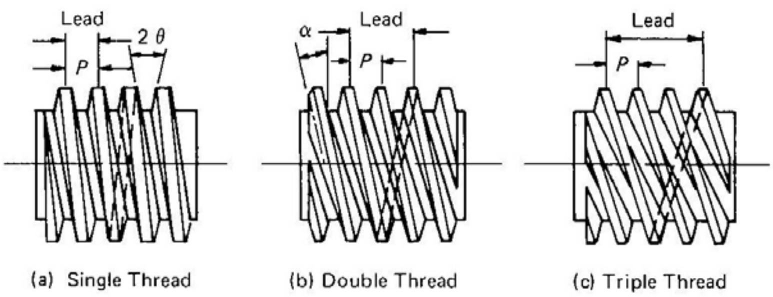


Figure 6-6 Reproduced from Part 2 of these notes (Fig 2-13).
An illustration of single-, double- and triple-start threads with an Acme profile. It may be observed that while the thread **PITCH** remains unchanged, the **LEAD** (or distance a nut would move per turn) has been doubled or tripled in the two-and three-start threads respectively.
Deutschman, A D, Michels, W J and Wilson, C E, Machine Design Theory and Practice, Macmillan, 1975, page 758.

Multi-start threads provide larger axial movement for each turn of the power screw. It also turns out that overall **FRICTION** is **REDUCED** as the wedge angle **INCREASES**. Hence multi-start threads are generally more efficient than single start threads. This is an important factor for power screws.

1.1.5 Ball bearing power screws

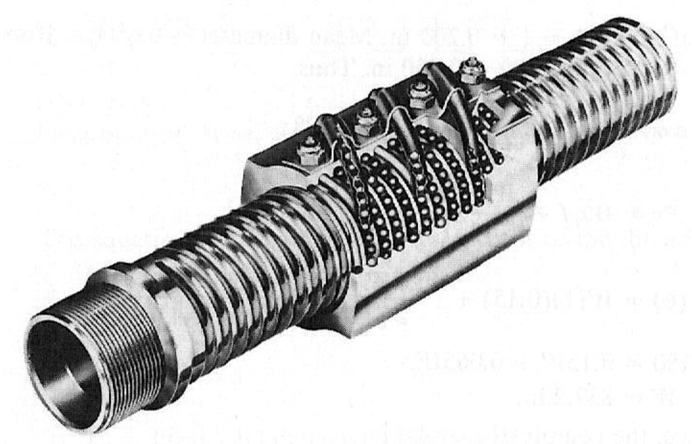


Figure 6-7 An example of a ball-bearing screw, in which contact between the nut and the thread is rolling contact via ball bearings rather than the sliding friction present in a normal nut. The bearings **RECIRCULATE** through return tubes as the nut travels along its thread, so axial movement is limited only by the length of the thread.
Creamer, R H, Machine Design 3E Addison Wesley, 1984, page 395, reproduced courtesy Saginaw Steering Gear Division, General Motors Corp.

In some applications, e.g. the steering system of a car or large truck, it is very important to decrease friction to the lowest possible level. In the case of car and truck steering, this is as much to provide good steering "feel" as to increase transmission efficiency. The sliding friction between the screw and the nut has been replaced by rolling contact of ball bearings between the screw and the nut.

Ball bearing nuts can be made with **PRELOAD** on the balls, so that very precise location can be achieved.

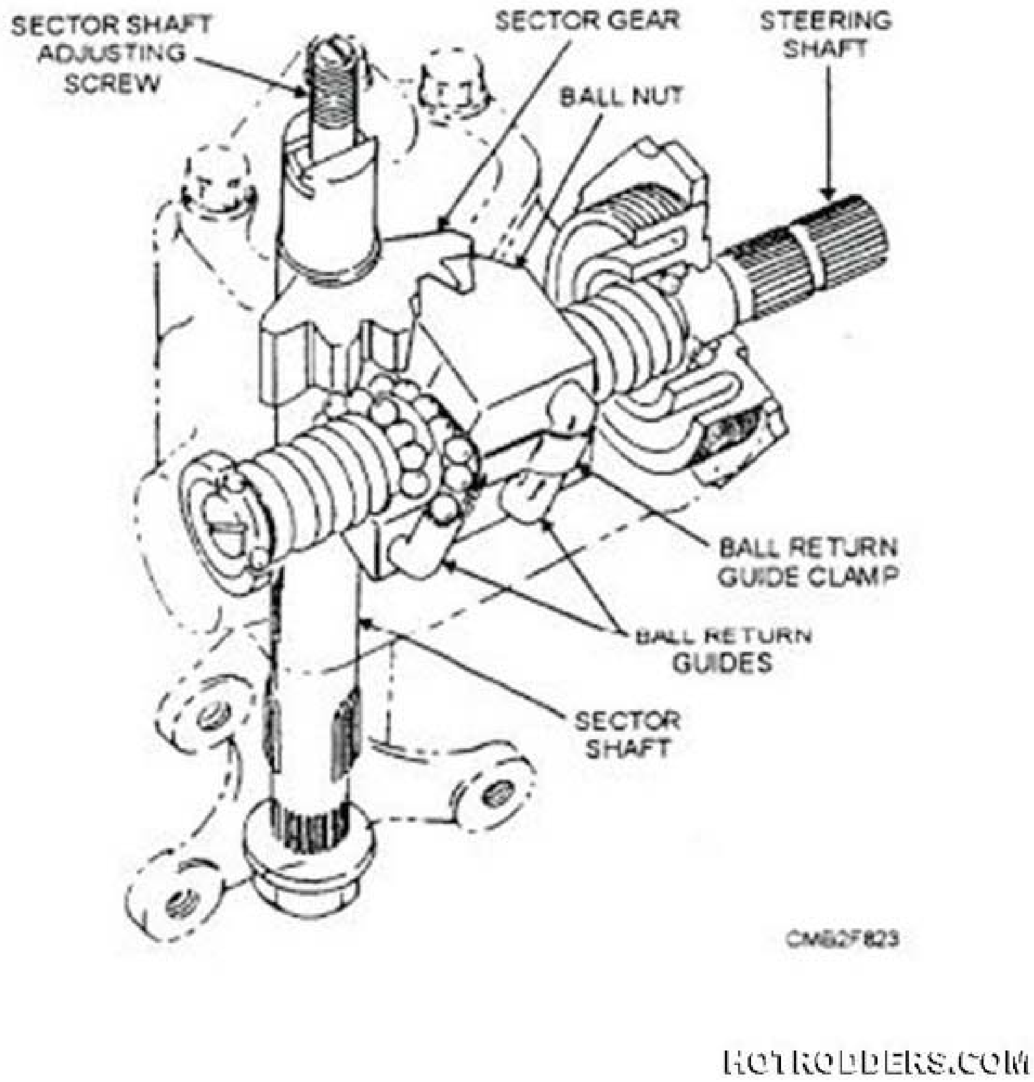


Figure 6-8 An example of an automotive steering box assembly using a **BALL NUT,** often referred to as a **RECIRCULATING BALL** system, to decrease friction and hence reduce the effort needed to steer the vehicle.
http://www.hotrodders.com/gallery/data/500/medium/Steering_gear.jpg

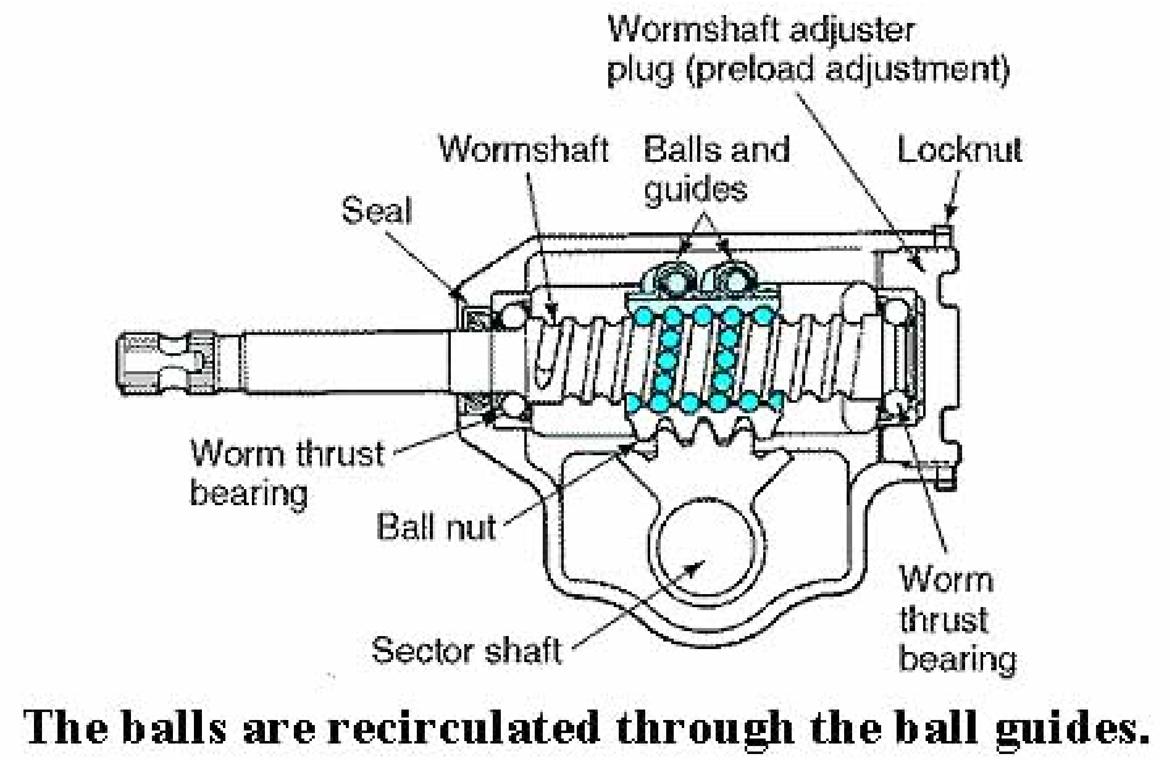


Figure 6-9 A 2D drawing of an automotive steering box assembly, similar in principle to that in Fig 6-8, based on a ball nut.
http://www.imperialclub.com/Repair/Steering/guides.jpg

In Figs 6-8 and 6-9, the steering wheel is connected via the steering column to the splined **STEERING SHAFT** (also called a **WORM SHAFT)** on the right of Fig 6-8 and the left of Fig 6-9. Rotation of the steering wheel causes the steering shaft or worm shaft to rotate and the **BALL NUT** to translate along the shaft. Gear teeth cut on the exterior of the nut mesh with similar teeth on the **SECTOR SHAFT**, causing it to rotate through up to about ± 30°. From Fig 6-8, it may be seen that the lower end of the sector shaft has a **SPLINED** end. A steering component (a lever) called a **PITMAN ARM** is fitted onto this splined end and turns the steering mechanism to steer the vehicle.

Note the use of **ANGULAR CONTACT** or **THRUST** ball bearings (easiest to see in Fig 6-9) to cope with the high axial forces as the steering wheel is turned in either direction. Observe also the use of an **OIL SEAL** on the protruding worm shaft of Fig 6-9. End float or **PRELOAD** of the angular contact bearings is adjusted by means of the large threaded **WORM SHAFT ADJUSTER PLUG** and the adjustment is locked by the **LOCKNUT**. How would you prevent oil from seeping out the threads of the adjuster plug?

2 Shaft Couplings

2.1 Definition

SHAFT COUPLINGS are used to join together or **COUPLE** two shafts belonging to two separate machines or components, each shaft having its own bearings, and the two shafts being more or less co-axial. **Couplings must transmit both angular rotation and torque**.

2.2 Misalignment

Since the two shafts to be coupled are in general each located by their own bearings, **MISALIGNMENT** may occur. The designer should always assume that, when two shafts are coupled, some misalignment will occur.

An example is shown in Fig. 6-10, where a coupling is required to connect an electric motor to the shaft of an air compressor.

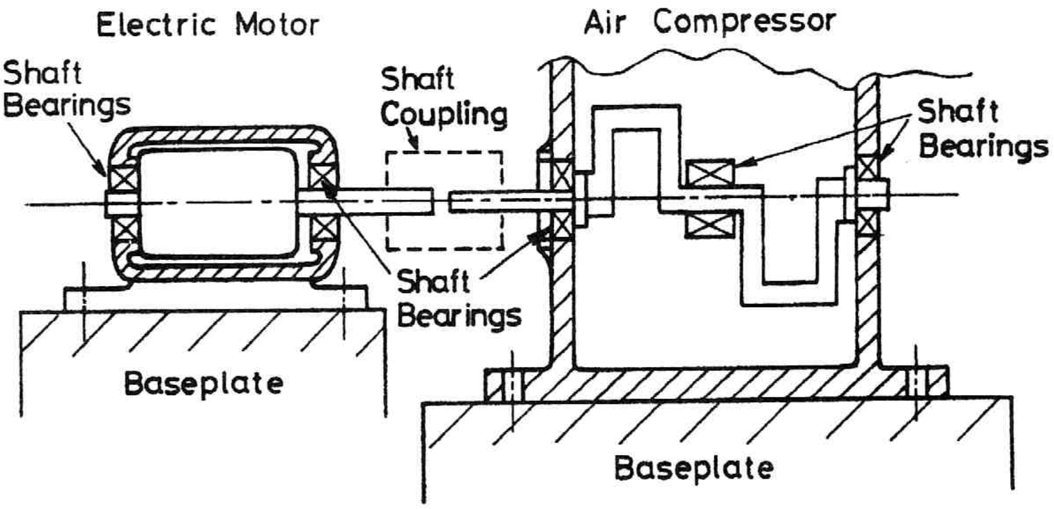


Figure 6-10 A coupling used to connect the shaft of an electric motor to an air compressor. The motor and compressor are mounted on two separate baseplates and it may be very difficult to ensure that the two shafts are accurately aligned under all operating conditions.

2.2.1 Types of misalignment

There are three basic types of misalignment:

- Parallel
- Angular
- Axial

These are illustrated in Fig. 6-11 below.

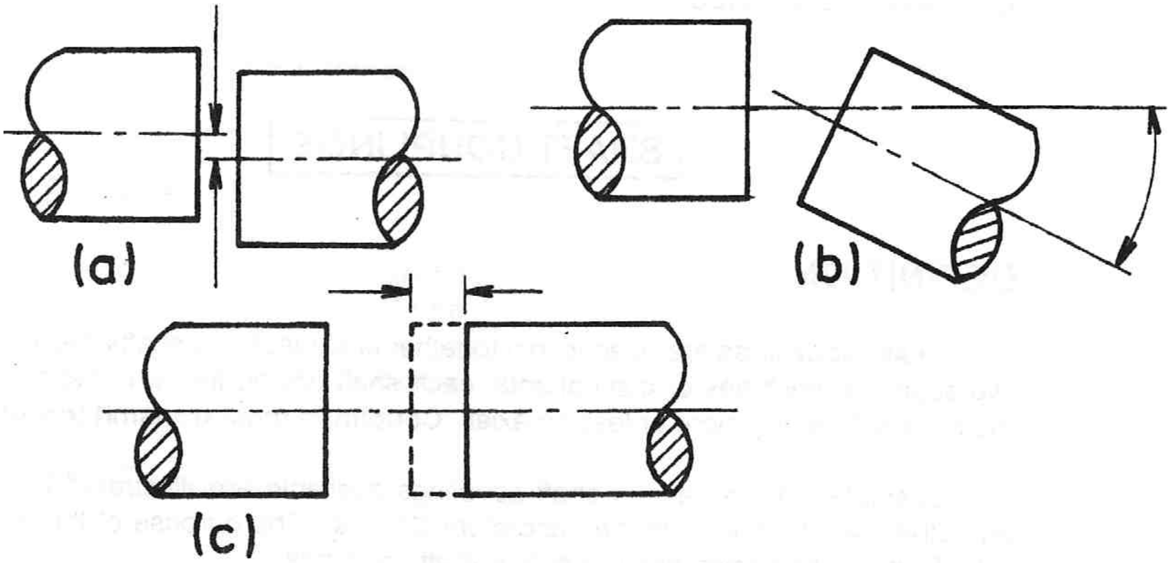


Figure 6-11 Illustrations of the three types of shaft misalignment which can occur. Combinations of the three types can and do occur.

An example of parallel misalignment (6-11(a)) occurs when the motor and compressor of Fig. 6-10 are mounted on their baseplates so that the two shafts are not at the same height. Couplings such as the Oldham Coupling (Fig 6-20 below) will accommodate parallel misalignment.

Angular misalignment (Fig. 6-11(b)) might occur if the base plate of the motor in Fig. 6-10 was horizontal but that of the compressor was not. The rubber-bushed pin-type coupling (Fig 6-15 below) is one coupling which accommodates angular misalignment.

Axial misalignment (Fig. 6-11(c)) occurs when a long shaft expands due to heating or when one shaft is not well located in the axial direction. If the change of axial length is large, a sliding spline joint may be used, otherwise couplings such as the Metaflex coupling (Fig 6-16 below) will accommodate this type of movement.

In the general case, all three types of misalignment may occur together, requiring the use of a coupling such as a Metaflex type with two sets of laminations, as seen in the assembly in Fig 6-16.

2.3 Types of couplings

Figure 6-12 Examples of **RIGID COUPLINGS** which do not allow for any shaft misalignment. *Top:* The coupling on the left uses square keys to transmit torque, the one on the right depends on compressing rubber sleeves and may therefore allow slip to occur if the machine becomes overloaded. *Lower:* Couplings in the lower group are in two halves and are able to be slipped over the two shafts after the machines have been placed in position, whereas those in the top group have to be slid onto their shafts before the machines are positioned.
http://www.hub-4.com/images/news/1078.jpg
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 549
http://www.couplingcorp.com/images/shaft_couplings_ultraflexx2.gif

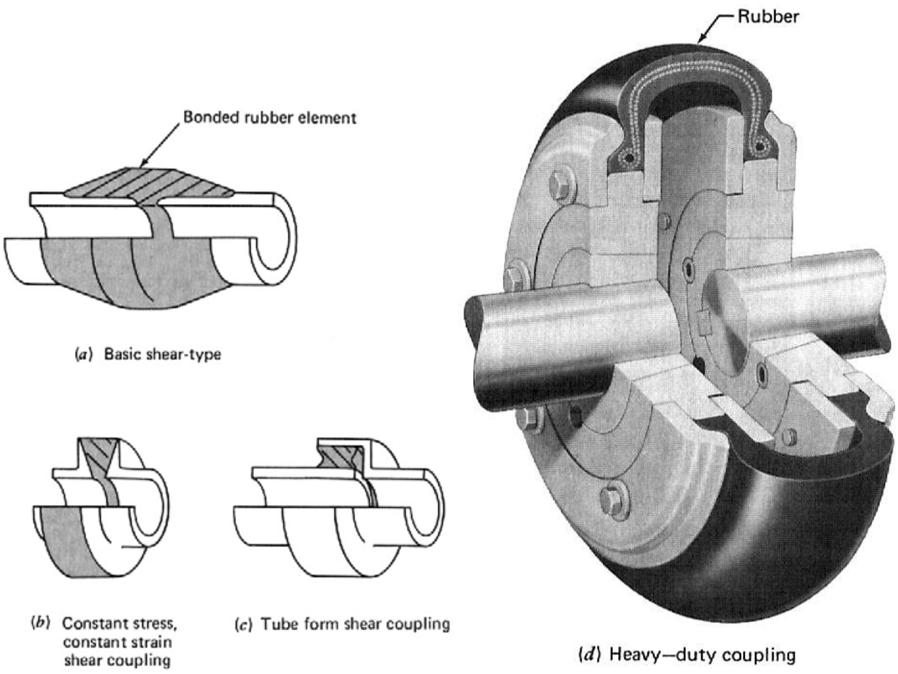


Figure 6-13 Couplings which use rubber in shear to transmit torque while possessing some flexibility and catering mainly for angular misalignment. The heavy duty coupling on the right caters for parallel and axial misalignment as well as angular.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 550

Figure 6-14 A further example of a coupling using rubber or polymer inserts to provide the flexibility needed to cope with angular and axial misalignment.
http://img1.tradeget.com/bestpulleysandcoupling%5CW3TR6NRB1flexible_jaw_couplings.jpg

Figure 6-15 Another variation of coupling using a rubber or polymer ‘barrels’ to cope with axial and angular misalignment.
http://delhi.olx.in/flexible-gear-couplings-gear-shaft-couplings-iid-69416953

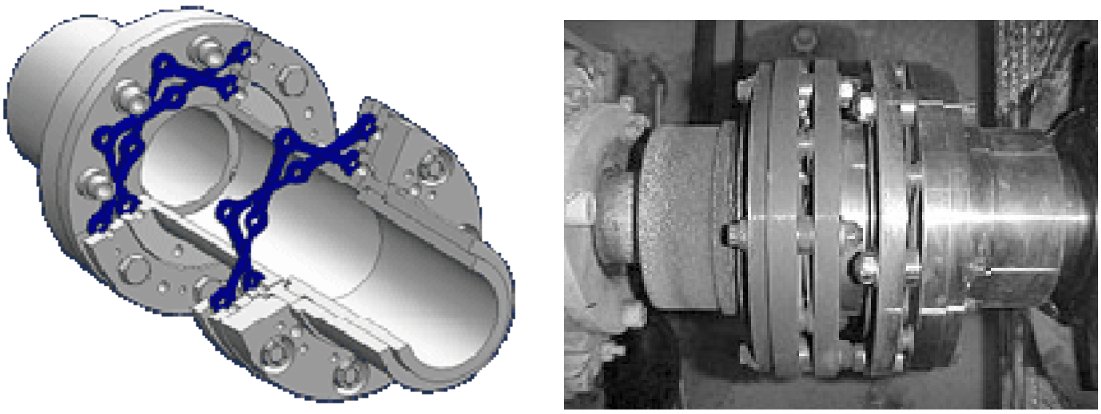
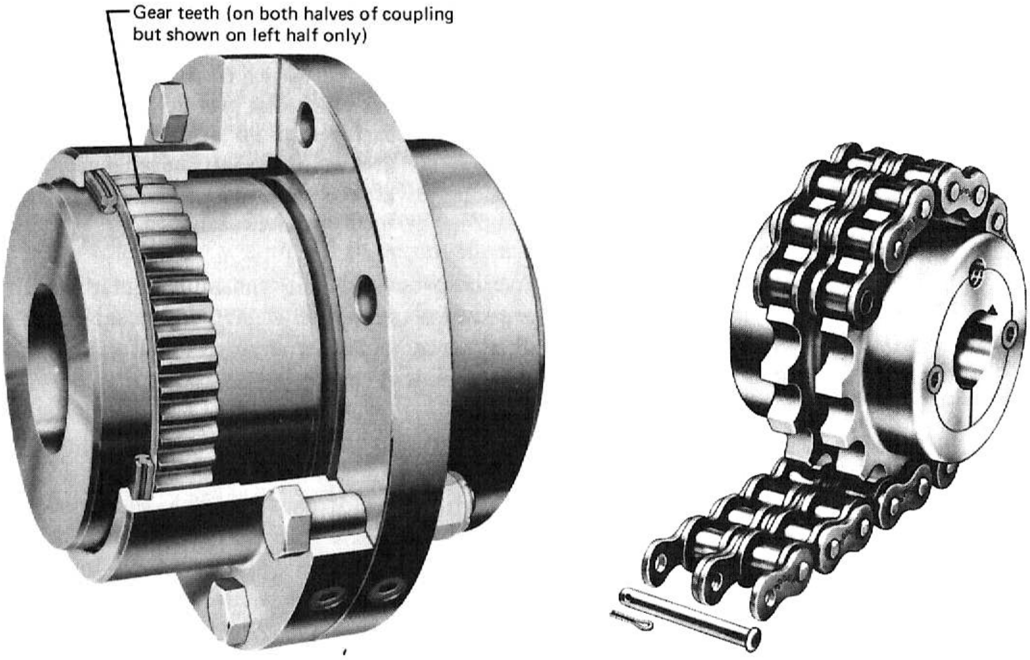


Figure 6-16 Couplings using metal elements for torque transmission. One particular version of this design is known as a Metaflex coupling. The example shown uses the thin blue coloured “springs”, to connect the two halves of the coupling. Each set of “springs” allows angular misalignment and, if two spring elements are combined in series, as in the assembly on the right, some parallel misalignment can be allowed for.
http://www.couplingcorp.com/images/shaft_couplings_ultraflexx2.gif



(b) Gear coupling

(a) Roller chain soupling

Figure 6-17 A gear coupling and a roller chain coupling, each allowing angular and axial misalignment.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 550

Figure 6-18 Used in marine applications, this coupling is a variation of the gear coupling shown in Fig 6-16.
http://www.electrical-res.com/marine-flexible-shaft-coupling

Figure 6-19 Bellows couplings, with the bellows made from spring steel and capable of allowing for axial, parallel and angular misalignment.
http://www.couplingsdirect.com/pdf/Pointers_for_Selecting_Shaft_Couplings.pdf

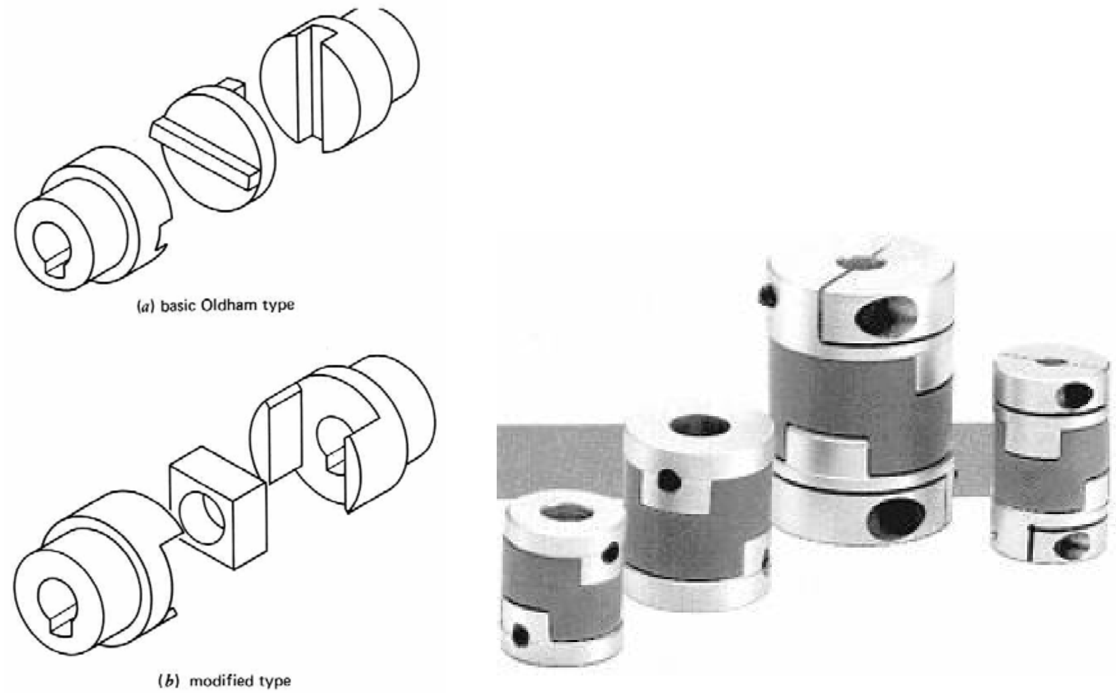


Figure 6-20 The Oldham coupling was one of the pioneers in shaft coupling. The central block is able to move in two mutually perpendicular directions and therefore caters for angular and parallel misalignment.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 552
http://www.couplingsdirect.com/pdf/Pointers_for_Selecting_Shaft_Couplings.pdf

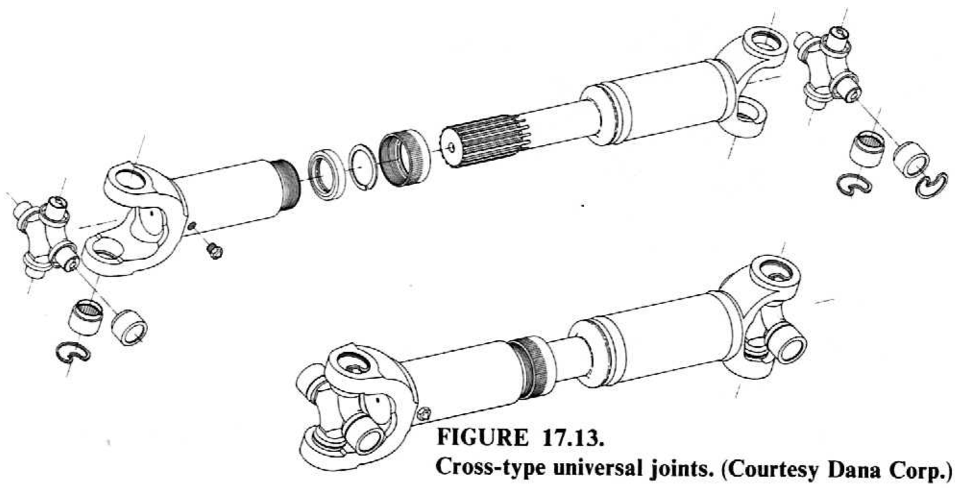


Figure 6-21 For completeness, this driveshaft, using two universal joints, is included with this section. It copes with large parallel, angular and axial misalignments in combination. Its most common application is as the driveshaft between the rear of the gearbox and the differential on a rear wheel drive vehicle.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 552

2.4 Torsional characteristics

2.4.1 Torsional rigidity

In some installations, the two shafts to be coupled may need to retain a given angular relationship at all times. In this case, a coupling possessing **TORSIONAL RIGIDITY** is required, e.g. the Metaflex coupling of Figs 6-16, the gear and chain couplings of Fig 6-17 and the Oldham coupling of Fig 6-20.

2.4.2 Torsional flexibility

A coupling having **TORSIONAL FLEXIBILITY** may be used to absorb energy, thereby reducing shock loading and helping to achieve quiet operation. One example of this type is the heavy duty rubber-tyre-type coupling shown in Fig 6-13. In this case, there may be 10° or more of rotation of one shaft relative to the other, due to the flexibility of the coupling.

2.5 Solid couplings

Despite comments above concerning the need for couplings to accommodate misalignment, **SOLID COUPLINGS** of various types (Fig 6-12) are used in some applications. If such couplings are used, the designer is assuming that misalignment will always be very small and that the shafts themselves are sufficiently flexible to accommodate any misalignment which does occur.

Solid couplings are of course torsionally rigid.

2.6 Coupling selection

Coupling selection needs to be based on the type or the combination of types of misalignment to be catered for, maximum torque to be transmitted (allowing for any shock or impact loading) and any need for torsional rigidity or flexibility. It may not be possible to achieve all requirements with one coupling and other design schemes may be used. For example, a torque limiting clutch might be installed to cope with occasional severe torque overload and additional thrust bearings might be used to control excessive axial misalignment.

2.6.1 Large misalignment

There are occasions when it is necessary to transmit torque from one shaft to another under conditions of very large misalignment or even when the relative positions of the two shafts change during torque transmission.

Consider the drive-line of a typical rear-wheel drive car or truck (Fig. 6-22). The engine and gearbox are mounted at the front and are flexibly mounted to the body or chassis of the vehicle. A driveshaft extends from the rear of the gearbox to drive the rear axle. The rear axle is mounted on springs which allow it to move up and down relative to the body of the vehicle. Torque must be transmitted while the rear axle is moving up and down on its springs. This results in a significant change in the length of the driveshaft, which is accommodated by a sliding spline joint.

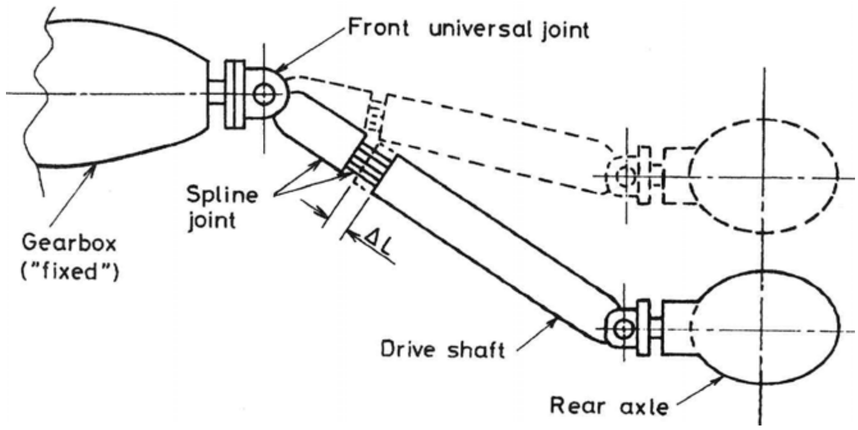


Figure 6-22 The rear axle of a truck or rear-wheel drive car requires the driveshaft to move through relatively large angles and to accommodate significant changes in length, requiring the use of a sliding spline joint.

2.6.2 Universal joints

UNIVERSAL JOINTS allow torque transmission through misalignment angles of up to about 20°. Most universal joints are based on the Hookes Coupling and, as seen in Fig 6-21, the universal joints used in the motor car or truck driveshaft are of this type.

2.6.2.1 Velocity fluctuations

The design of common universal joints such as the Hookes joint is such that, when the joint transmits rotation through an angle, the angular velocity of the driven shaft fluctuates in a cyclic manner relative to the velocity of the driving shaft. If the rotational speed is high or the misalignment angle large, the resulting vibrations may be great enough to be objectionable or to damage the driveshaft components.

2.6.2.2 Overcoming velocity fluctuations

One common design arrangement to overcome the problem of angular velocity fluctuations is to use two universal joints so arranged that the velocity fluctuations introduced by the first joint are cancelled by the second. This is achieved provided:

- Both joints transmit through the same misalignment angle.
- The two joints are in the correct angular relationship (i.e. correctly **PHASED**).

Refer to Fig. 6-21 and 6-22, both of which show a driveshaft with two universal joints. Note that the input and output shafts remain parallel so that the misalignment angles are equal for both universal joints. The input shaft (say the left-hand end) will rotate at constant velocity. The central section of shaft will have fluctuating velocity, alternately faster and slower than the input shaft. (Actually, there are two complete speed fluctuations for each rotation of the shaft.) The second universal joint (at the right-hand end) operates with the same misalignment angle as the first and therefore produces identical velocity fluctuations. If the two joints are assembled in the correct angular relationship or phasing, as for example in Fig. 6-21 and 6-22, the fluctuations are cancelled and the output shaft runs at very close to constant angular velocity.

In the case of the motor car or truck driveshaft (see Fig. 6-22), the rear axle moves up and down and the parallel misalignment changes. However, provided both the input and output shaft (i.e. the gearbox and differential shafts) remain parallel, both misalignment angles remain equal and constant drive velocity is achieved.

2.6.2.3 Axial misalignment

In applications such as the car driveshaft (Fig. 6-22), a significant change in the length of the driveshaft will be required as the rear axle moves up and down on its springs. This is usually achieved by the use of a splined joint, in which the splines are free to slide axially.

2.7 Constant velocity joints

In some applications, it will not be possible to use two universal joints, yet it is required to drive a shaft at constant angular velocity with a large misalignment angle. One application in which this occurs is in the driveshafts used on each front wheel of a front-wheel-drive car. In this case, the inner end of the driveshaft has very little misalignment relative to the transmission housing, while the outer end of the driveshaft is attached to the front wheel and must continue to transmit torque whilst turning through angles up to ± 35°.

To achieve constant angular velocity, special geometry is required. Constant velocity joints frequently use balls running in circular-arc grooves in the inner and outer races, as shown in Fig 6-23.

Figure 6-23 Examples of constant velocity joints which allow large angular movements (up to about 35°) whilst retaining constant angular velocity.
Top Left: The Torvec joint. *Top Right*: The type of joint more commonly used in front wheel drive cars. Note that it is the balls which actually transmit torque from driving to driven member and that the plane of the balls in their cage always halves the angle between the input and output shafts in order to achieve constant velocity.
Lower: A type using double Hookes joints. Constant velocity is transmitted provided the central section of the joint is constrained to run at half the angle between the shafts. http://www.torvec.com/images/CV_Joint.jpg
www.automotive-technology.co.uk/resources.html

For an animation of a CV joint, see
http://commons.wikimedia.org/wiki/File:Simple_CV_Joint_animated.gif

3 Brakes and Clutches

Most brakes and clutches in use today are the products of specialist manufacturers. A wealth of design experience has gone into the development of these components to bring them to their present high standard of performance. However, it is possible to design and manufacture a clutch or brake if the need arises, e.g. for one or two components on a special machine.

Clutches and brakes are frequently considered as a group. This is because both clutches and brakes work on the same engineering principles. It is really only the application which determines whether the particular component will be called a clutch or a brake.

3.1 Definitions

3.1.1 Brakes

In general, brakes are used to apply a **RESISTIVE FORCE** to retard a moving body or to bring it to rest. In most cases, brakes operate on rotational members, so that the resisting force becomes a **RESISTING TORQUE**.

3.1.2 Clutches

In general, clutches allow two adjacent components to be connected or disconnected at will. Often, the required connection is between two shafts, one driven by a prime mover (e.g. an electric motor), the other connected to a machine of some kind. Often (but not always) the function of the clutch is to bring a second shaft up to speed in a gradual manner. In this case, some sort of frictional device may be used.

3.2 Principles of brakes and clutches

In this section the operating principles described apply to both **CLUTCHES AND BRAKES** in their various physical configurations.

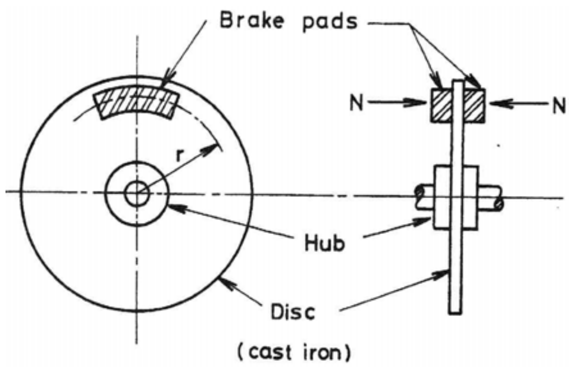


Figure 6-24 Schematic diagram of disc brake.

Fig. 6-24 shows a typical disc-brake arrangement in which the frictional material, the brake pads, occupies only a small arc. If the pads were extended through 360°, it would resemble a typical disc-clutch configuration.

The two (stationary) brake pads in Fig. 6-24 are being pressed into contact with the (rotating) disc. The retarding **FORCE** F due to the frictional contact is

$$F = \underbrace{\mu}_{\text{Coefficient of friction}} \; \underbrace{N}_{\text{Normal force}} \; \underbrace{2}_{\text{2 pads in contact}}$$

The retarding **TORQUE** is therefore

$$T = F \underbrace{r}_{\text{effective radius of pad}}$$

$$= 2 \mu N r$$

From this simple analysis, the conclusions are that **BRAKING OR CLUTCHING TORQUE** is

- Proportional to coefficient of friction between the frictional pairs.
- Proportional to the normal force.
- Proportional to the effective radius of the frictional material.
- Proportional to the number of frictional pairs.

Further, since brakes and clutches of this type work by friction, **HEAT** is generated while ever slipping is occurring. The rate of heat generation is proportional to the **POWER** being dissipated by friction. The total amount of heat generated is proportional to the time during which slipping occurs. The **TEMPERATURE RISE** in a brake or clutch is roughly proportional to the total amount of heat which has been generated.

It is important to avoid overheating of brakes and clutches. From the designer's point of view, the temperature rise may be minimised by

- Minimise power dissipation, e.g. engage clutch at low speed.
- Minimise the duty cycle, e.g. decrease the number of starts and stops.
- Increase the area of the frictional surfaces in contact, e.g. by increasing the number of frictional pairs. This does not decrease the total power dissipation, but spreads the heating effect over a larger area.
- Provide cooling by:
  - Increasing the surface areas.
  - Increasing air flow, e.g. ventilated brake discs.

- Providing liquid cooling, e.g. by internal passages or by running the component in a bath of oil.

3.3 Examples - brakes

3.3.1 Rotating members

Various brake types and designs are in use for different purposes on a wide range of machinery.

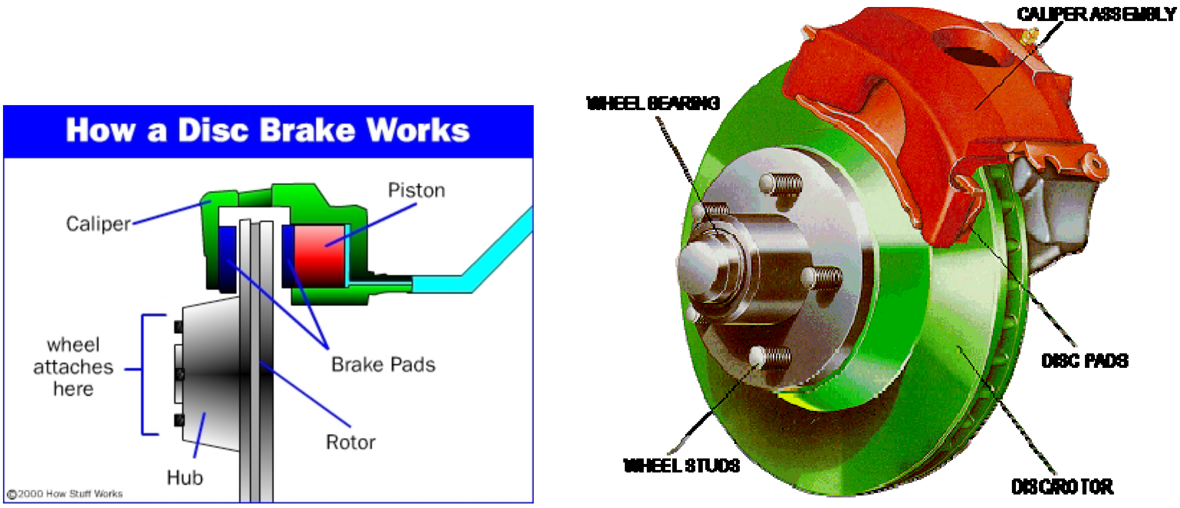


Figure 6-25 Disc brakes are very effective and are used in virtually all recent model cars. The left-hand illustration shows how hydraulic pressure is applied to force two opposing brake pads into contact with the brake disc or rotor to slow the vehicle. The caliper is mounted in a way which allows it to slide (left to right or *vice versa* in the left-hand diagram) to compensate for wear of the brake pads.
The right-hand diagram shows an actual brake assembly in which the disc pads are contained within the caliper assembly.
http://www.jamesglass.org/JGA/2labor/Z_laborIMAGES/00general/0-5_glossary/brake_disc.gif

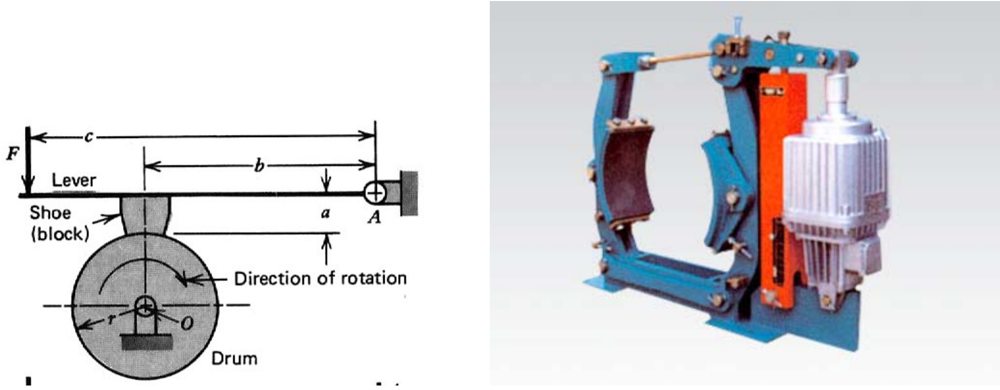


Figure 6-26 Block brakes (left) were initially designed for slow speed, low duty cycle applications such as on the (steel) rim of a wheel of a horse-drawn cart. However, a more modern application is shown on the right. This type can be used as an automatically applied emergency brake, e.g. in the case of a power failure on an elevator.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 552
[http://www.jzzd.cn/eng/client/user/upimage/zdq_product20081209112050473 31.jpg](http://www.jzzd.cn/eng/client/user/upimage/zdq_product2008120911205047331.jpg)

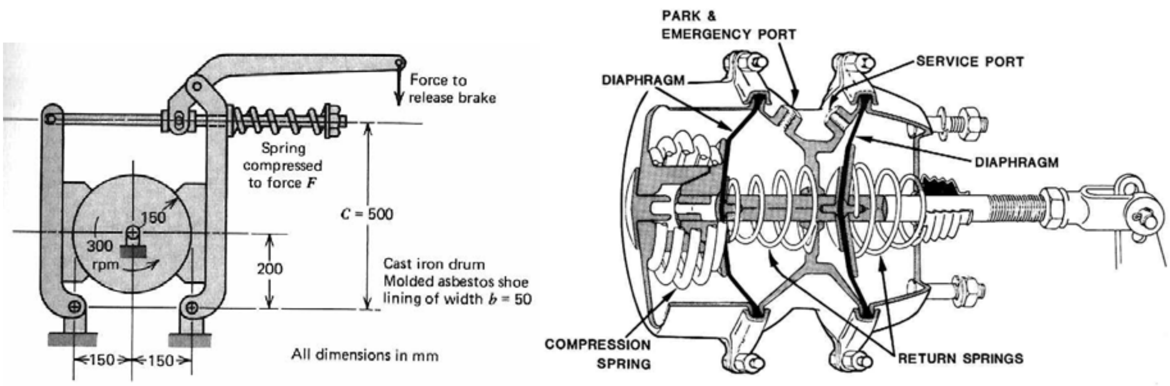


Figure 6-27 Spring brakes are a fail-safe device, since the spring always applies the brake unless there is a force to hold the brake in the off position. In the left-hand diagram, the spring always forces the brake blocks into contact with the rotating drum unless a force is applied to hold the spring in its compressed position. The right-hand diagram is a sectioned view of an air-brake cylinder from a heavy truck or other heavy vehicle. Normal braking is achieved by admitting air under pressure through the SERVICE PORT to the left-hand side of the right-hand chamber. This pushes the brake rod to the right. However, there is a very strong barrel-shaped compression spring in the left-hand chamber which can only be held in its compressed position by air pressure applied to the right-hand side of its chamber through what is marked as the EMERGENCY PORT. The barrel-shaped spring automatically applies the 'emergency' brake if system air pressure fails. It also acts as a parking brake.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 552
Vehicle Inspection Procedure No 24, Roads and Traffic Authority of NSW, April 1991.

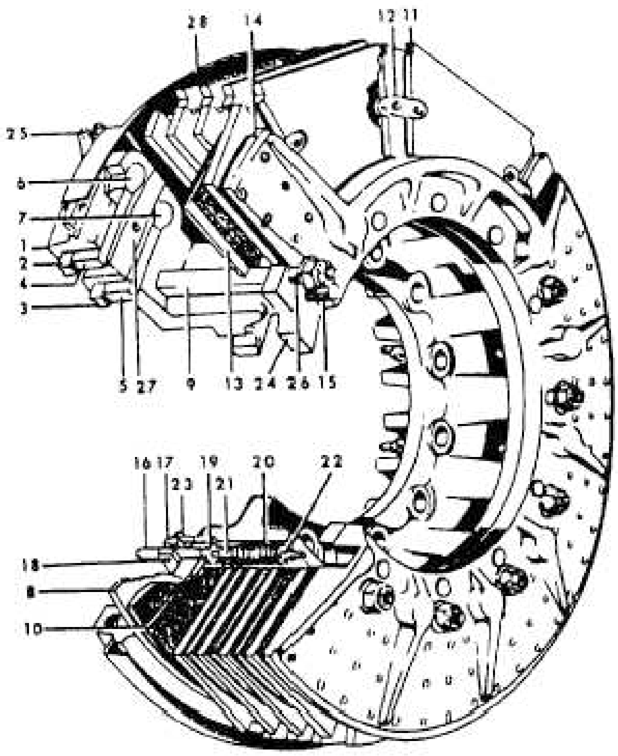


Figure 6-28 Multi-plate disc brakes provide very high braking torques which could not be achieved with a single disc of reasonable diameter. This is achieved by designing in a number of pairs of frictional surfaces, so that each pair contributes to the torque capacity. This diagram is from an aircraft application where the need for very high torque competes with restrictions on size and weight.
http://www.tpub.com/content/aviation/14018/img/14018_479_1.jpg

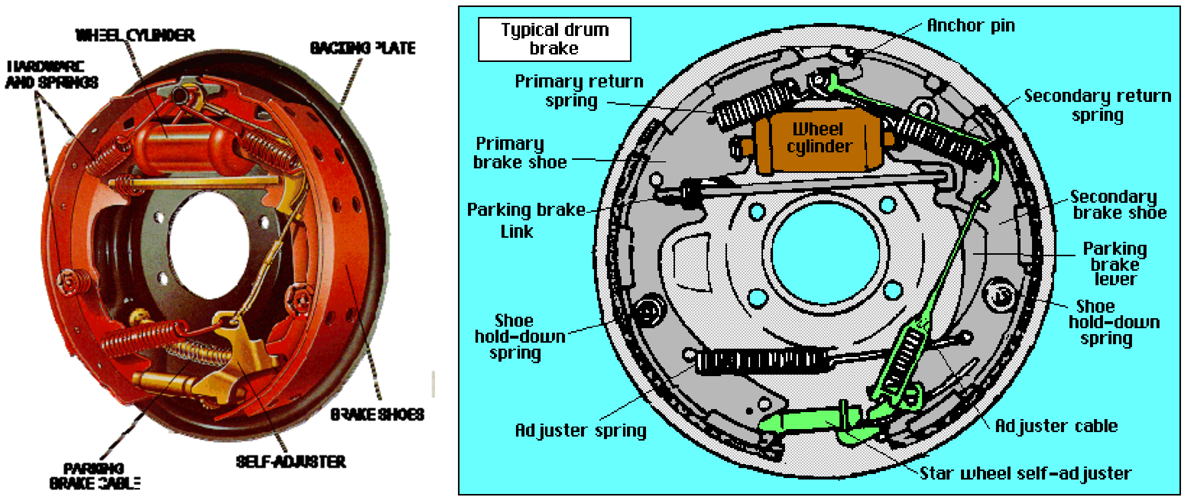


Figure 6-29 Internal expanding shoe brakes (or drum brakes) were used as the main braking system on cars for many years. They are still used on the rear of some cars and are virtually universal on all wheels of heavy trucks, where they are still the most effective system.
http://www.aa1car.com/library/elements/drum_brake.gif

Centrifugal brakes apply themselves automatically as rotational speed increases. See illustration for centrifugal clutches, Fig 6-36.

3.3.2 Linear brakes

Not all brakes work on rotating shafts, discs or drums. Some lifts (elevators), for example, use emergency brakes which act on the long, straight vertical guides located within the lift-well. In the event of power or other mechanical failure, these brakes are automatically applied to prevent the lift car from falling.

3.3.3 Power absorption

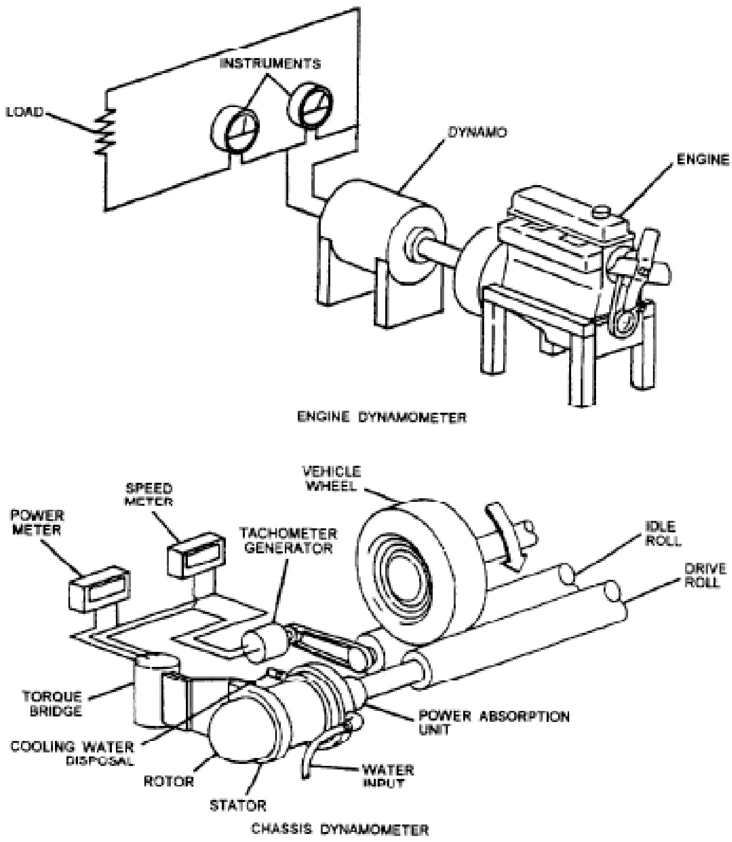


Figure 2-19.—Dynamometers.

Figure 6-30 Some brakes are designed for direct power absorption by electrical, hydraulic or other means. One example (top) is the **ENGINE DYNAMOMETER** which is used to test the power output from an engine. A second type (lower) is known as a **CHASSIS DYNAMOMETER**, in which the whole vehicle is placed on rollers to simulate a road surface and instruments measure speed and torque to calculate power as the vehicle is 'driven' on the rollers. This system obviously takes into account all torque and power losses due to transmission gearing, tyres, etc. Examples of actual engine dynamometers may be seen in the School's L211 (Internal Combustion Engines Lab).
http://www.sweethaven02.com/Automotive01/fig0219.gif

3.4 Examples - clutches

3.4.1 Rotating members

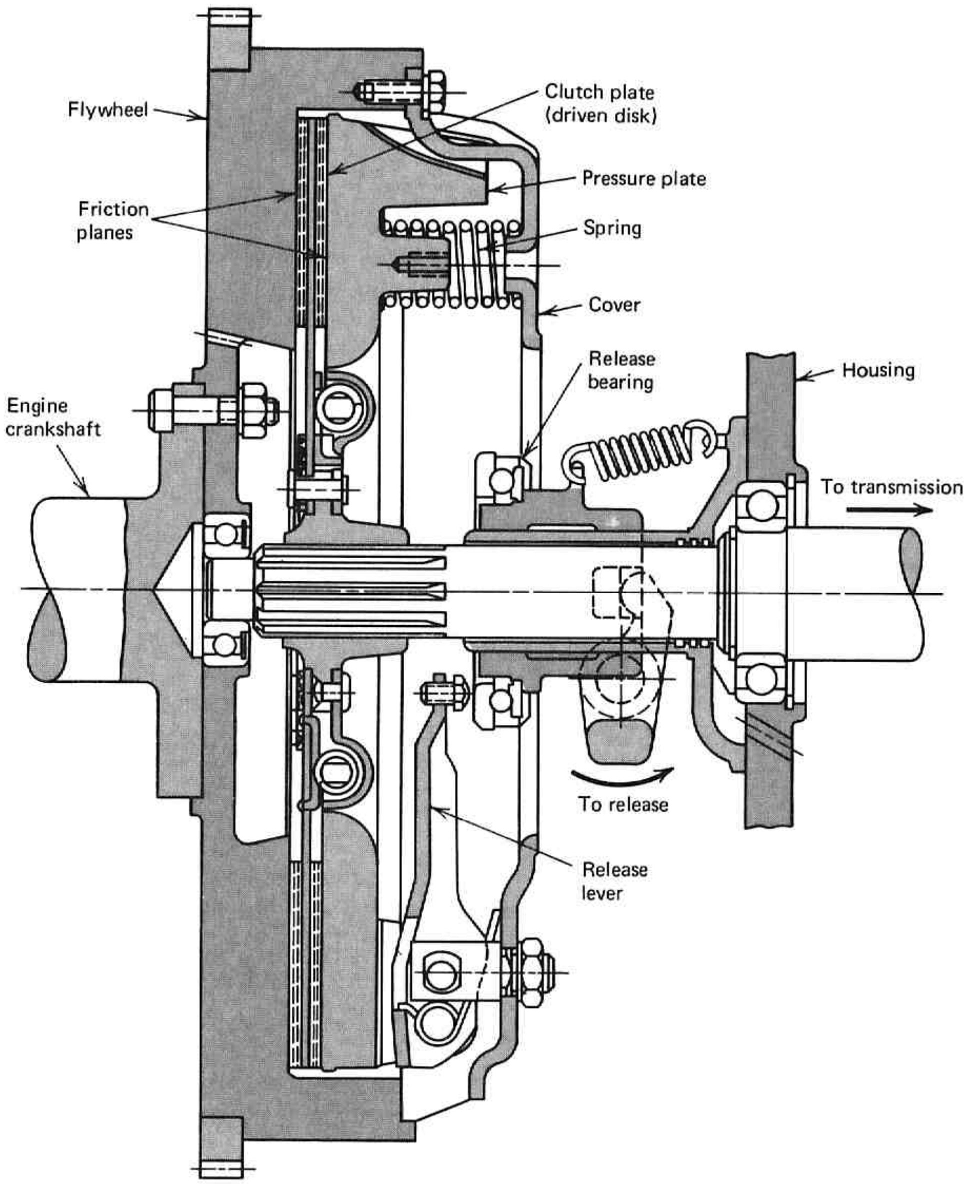


Figure 6-31 An example of an older style of automotive clutch assembly, illustrating many of the principles of clutch design. In this type of clutch, a **CLUTCH PLATE** (driven disc), incorporating two discs of frictional material, is clamped by spring pressure between two metal plates and is driven by those metal plates. In automotive applications, one metal plate is usually the engine **FLYWHEEL** and the other is the clutch **PRESSURE PLATE**. When the clutch release pedal is depressed, the movement is transferred to the **CLUTCH RELEASE LEVERS**, which pivot to separate the two metal plates, allowing the driven plate to come to rest. Gradual release of the clutch pedal allows the clutch plate to be gradually brought up to speed.

Figure 6-32 An example of a modern single plate (disc) clutch commonly used in cars with manual transmission. The photograph shows the **CLUTCH ASSEMBLY** (left) which uses a **DIAPHRAGM SPRING** (refer to Part 4 of these notes) and the double-sided **FRICTION PLATE** (top right). The helical coil compression springs in the clutch in Fig 6-31 have been replaced by a single diaphragm spring (see **Part 4** notes), allowing simpler construction, cost saving and better operating characteristics. The pressed steel component on the lower right in this figure is part of the clutch release mechanism.
http://static.howstuffworks.com/gif/dual-clutch-transmission-11.gif

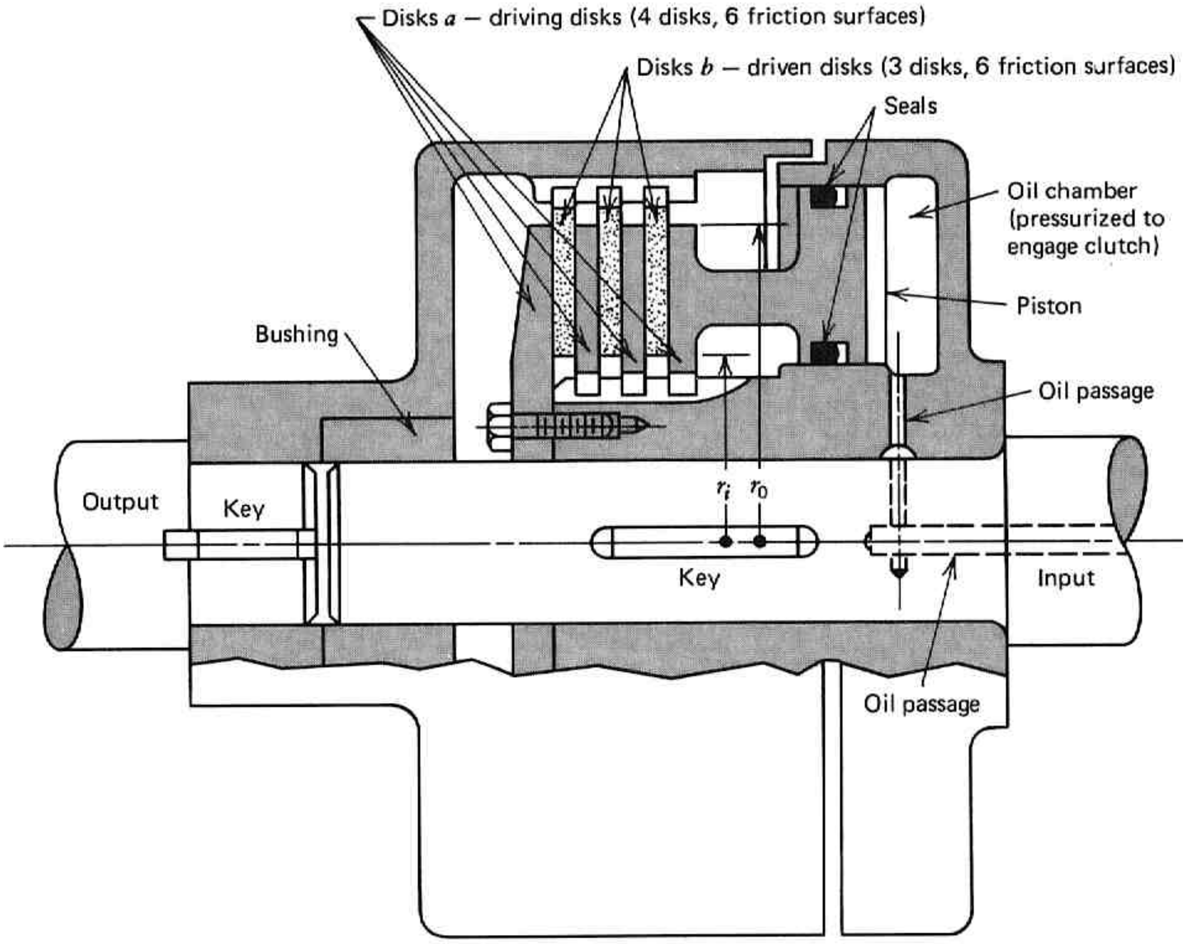
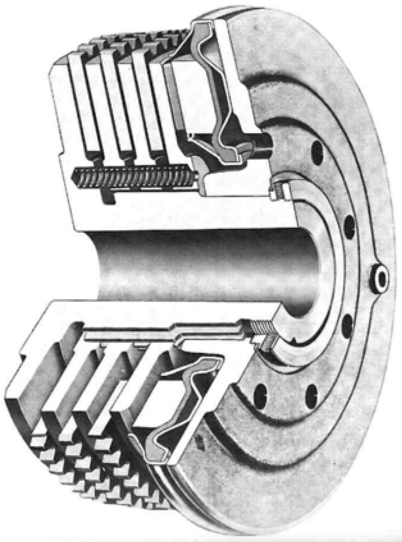


Figure 6-33 Multiple plate clutches are often used in heavy machinery (e.g. earth-moving equipment); they provide very high torque but are often jerky in action. The usual arrangement is to have two types of disc, one type (Discs ***a*** in the diagram) splined to the input member on the left and the second type (Discs ***b***) splined to the output member on the right. When oil under pressure is admitted to the oil chamber, the two sets of plates are clamped together and the clutch transmits torque.

Juvinall, R C, Fundamentals of Machine Component Design, Wiley, 1983, page 556.



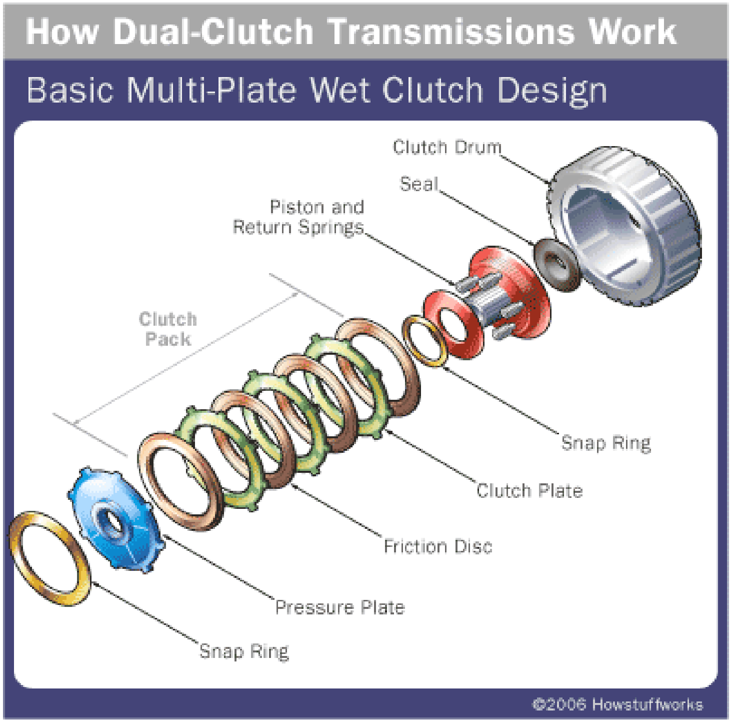


Figure 6-34 Examples of multiple plate disc clutches. *Left:* A "dry" clutch, intended to run with the frictional material dry, i.e. free from oil or other liquid. *Right:* An exploded view of the components of a "wet" clutch, intended to run in a bath of oil or special lubricant. Whilst this will reduce the coefficient of friction of the plates, adequate torque transmission is obtained by the multiple discs and oil flow contributes to cooling of the whole assembly.
http://static.howstuffworks.com/gif/dual-clutch-transmission-11.gif

Each sprag is essentially a strut placed between the races in such a way that it transmits power from one race to the other by a wedging action when either race is rotated in the driving direction. Rotation in the other direction frees the sprags and the clutch is disengaged, or overruns. Either race may be the driven member.

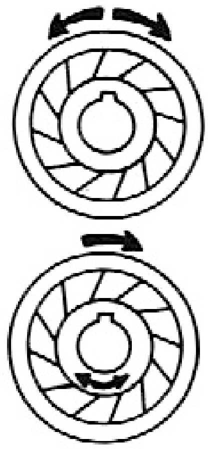


Figure 6-35 Sprag and free-wheeling clutches drive the output shaft in only one direction and are often used where the input shaft may change its direction of rotation but it is not desirable to reverse the direction of rotation of the output shaft. In operation, the series of small sprags tilt whenever the direction of shaft rotation is reversed, either locking the inner and outer races together or allowing slip to occur.
Deutschman A D, Michels W J, Wilson C E, Machine Design Theory and Practice, Macmillan, 1975, page 694.

Figure 6-36 An example of a sprag clutch. The small white elements move to engage or disengage the drive, depending on the direction of rotation.
http://gottransmissions.com/blog/wp-content/uploads/2009/03/onewaybearingspraguedetail_700.jpg

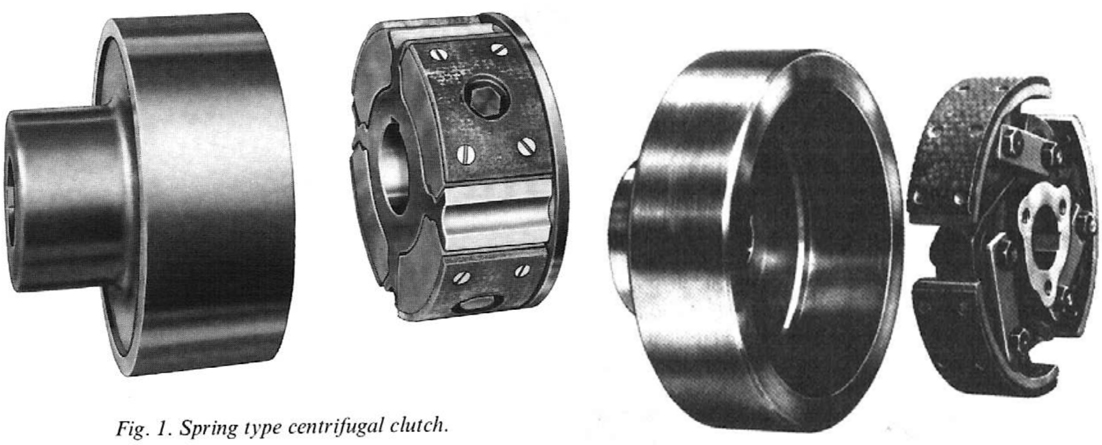


Figure 6-37 A typical example of the use of a centrifugal clutch of the type illustrated is on go-carts. The engine may be started and continue to run at low speed without driving the cart - it is only when the accelerator is depressed, engine speed increases, and the blocks of frictional material are thrown outwards by centripetal force that the cart moves. Many chain saws use a similar principle to stop movement of the cutting blade when the engine is idling.
Bell, Peter C (Ed) Mechanical Power Transmission, Macmillan 1971, page 18
Shigley, J E, Mischke, C R, Mechanical Engineering Design, 5E, McGraw Hill, 1989, page 630.

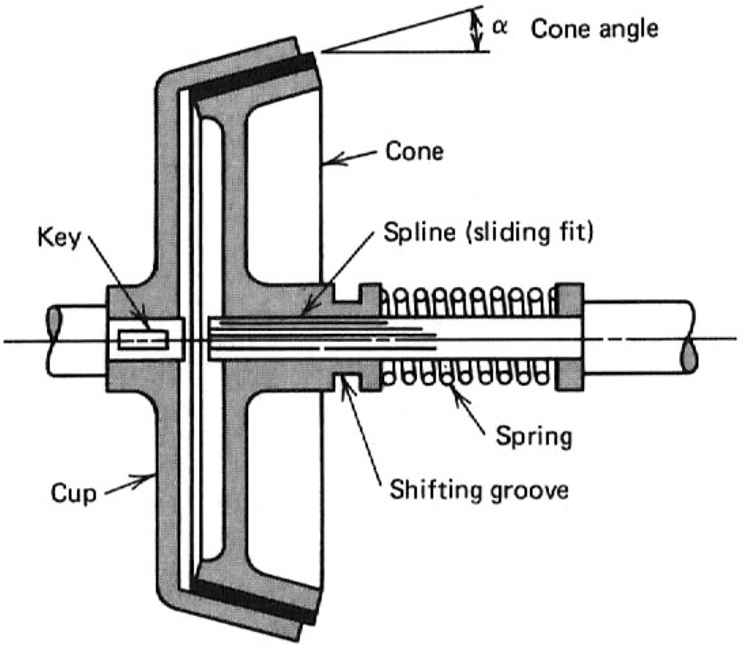


Figure 6-38 A schematic of a cone type friction clutch. Cone clutches transmit more torque than disc clutches of the same diameter because of the wedging action of the friction member into the cone. They can be very jerky in their engagement and will overheat if used for frequent stops and starts.
Juvinall R C, Fundamentals of Machine Component Design, Wiley, 1983, page 552

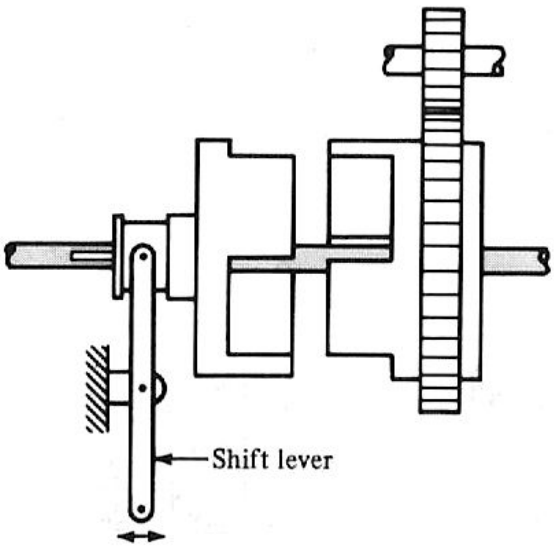


Figure 6-39 **DOG CLUTCHES** are used where the drive is required to be positive (i.e. no slip at all) yet able to be disconnected at will. The dog (or **CLAW**) clutch can only be engaged while the shafts are stationary or rotating at very low speed.
Shigley, J E, Mischke, C R, Mechanical Engineering Design, 5E, McGraw Hill, 1989, page 655.

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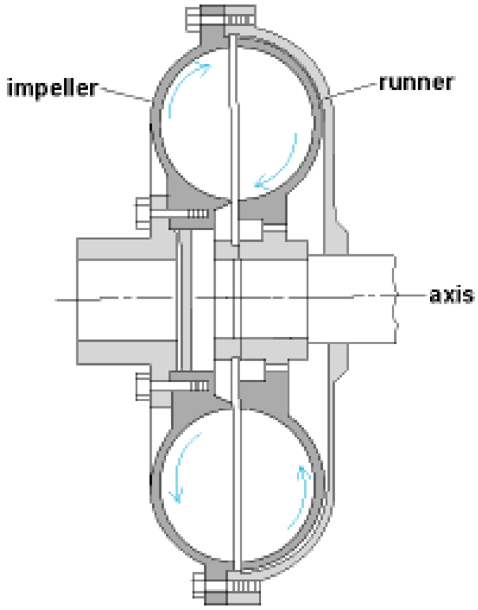


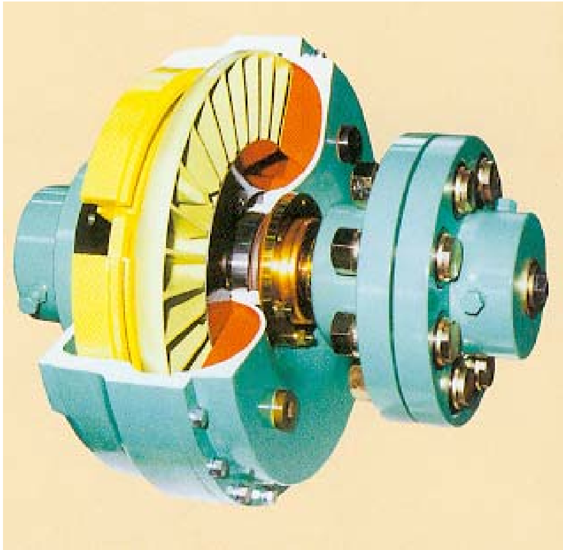


Figure 6-40 Fluid couplings transmits torque hydraulically from one shaft to another without the friction and consequent wear associated with the clutches previously described.
http://www.accessscience.com/popup.aspx
http://img.diytrade.com/cdimg/229408/1050143/0/1094632271/Shinko_fluid_coupling.jpg

3.4.2 Other clutches

3.4.2.1 Rotary clutches

ELECTROMAGNETIC friction clutches are similar to the friction clutches shown above except that the force on the friction member is provided electromagnetically instead of by mechanical springs.

Another type of clutch is the **DRY POWDER CLUTCH** in which the space between the driving and driven members is filled with a dry powder having ferro-magnetic properties. When an electro-magnetic field is applied, the clutch engages and transmits torque. The benefit of these clutches is their ability to be engaged and disengaged very rapidly. They are generally used for low power applications.

3.4.2.2 Linear clutches

One example of this type of clutch is seen in many cable-car systems, which use a clutch to attach the cars to the continuously moving cable. This allows the cars to be detached so that passengers have time to get in or out. Sydney's Taronga Park

Zoo uses this method as do many chair lifts in the ski fields and San Francisco's famous cable trams operate on a similar principle.

3.5 Effects of overheating - brakes and clutches

In automotive parlance, we often hear or read of **BRAKE FADE**. This refers to the fact that, as brakes become overheated, they become less effective. The driver of a car or truck using the brakes to descend a long hill thus finds that more and more force must be applied to the brake pedal to produce the desired retardation. Eventually, a stage may be reached where the driver is unable to apply sufficient force and the vehicle speed increases uncontrollably.

The effects of overheating are

- The coefficient of friction of the frictional material may change markedly, often decreasing with increased temperature.
- With drum brakes, the actual diameter of the drum increases with temperature. As the drum continues to expand, it may not be possible for the brake shoes to be forced into contact with the drum. Hence the normal force between the frictional material decreases and braking is lost.
- Excessive temperature damages the brake parts. Rubber seals may harden and crack. Cast iron brake drums or discs glaze, "burn", distort and crack.
- With hydraulic systems, the hydraulic fluid may vaporise, causing total loss of braking.

Similar problems can occur with clutches which are used for frequent stops and starts, particularly where high torques and high rotational speeds are involved.

3.6 Brake actuating systems

This section deals specifically with the actuation of **BRAKING SYSTEMS**. However, the methods described are to be seen as general engineering systems which may also find application to the control of clutches or other components.

3.6.1 Mechanical

There is a mechanical component in the actuation of all friction-brake systems, e.g. moving the frictional component into contact with the rotating drum or disc. However, some systems are entirely mechanically actuated, e.g. the block brake in Fig 6-26. Such systems are normally actuated by direct human effort via hand lever or foot pedal.

3.6.2 Hydraulic

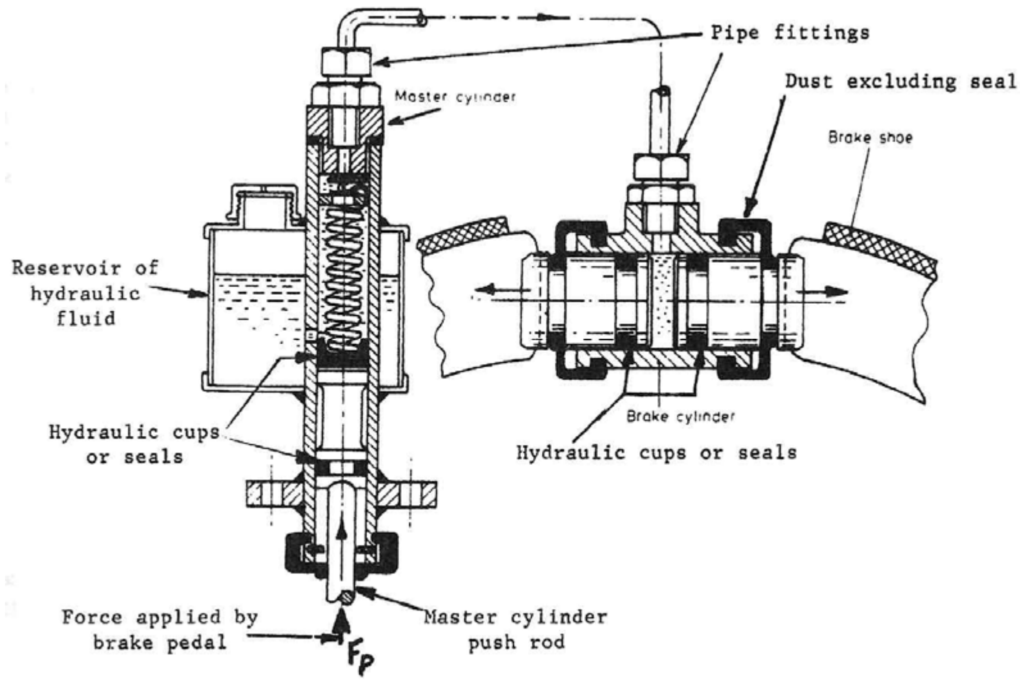


Figure 6-41 A schematic of a simple hydraulic brake system using principles typical of those used on early model cars with drum brakes. Later developments use more complex master cylinders having two pistons in order to create two separate hydraulic circuits as a precaution against brake failure. However, the simpler system shown here allows attention to be focussed on the operating principles.
Reproduced from J. Carvill, *The Student Engineer's Companion*, Butterworths, 1980.

In a typical hydraulic braking system, the operator applies force to the system via the **MASTER CYLINDER PUSH ROD** such as that seen at the bottom of Fig 6-41. This push rod presses onto the master cylinder **PISTON** and the force on the piston produces a **PRESSURE** in the **HYDRAULIC FLUID**. By Pascal's Principle, this pressure is transmitted throughout the hydraulic system and, through connecting pipes, to the **BRAKE CYLINDER**, sometimes called the **SLAVE CYLINDER** or **WHEEL CYLINDER**. Often, the brake cylinder has a larger diameter than the master cylinder and, since the same pressure acts throughout the system, the brake cylinder may exert a much larger force than the input force at the master cylinder. In the system of Fig. 6-41, the brake cylinder pistons push directly onto the **BRAKE SHOES** which have the frictional material attached to them. Brake cylinders are often "double -ended" as in Fig. 6-41, so that they can push simultaneously on two brake shoes.

There may be more than one brake cylinder in one hydraulic system. In a typical car, there are four brake cylinders, one for each wheel. The brake cylinders may take the form of disc brake calipers for the front wheels (see Fig 6-24) with cylinders such as those in Fig. 6-41 or 6-29 at the rear wheels. In that case, the designer must carefully "balance" the braking effort due to the different actuating cylinders in order to obtain good overall braking performance.

Note that in this system the force to actuate the brake comes entirely from the operator, although the mechanical advantage of the system is very high because the travel of the hydraulic cylinders is very small.

3.6.3 Pneumatic

In **PNEUMATIC SYSTEMS**, energy is stored in compressed air and this air is used to actuate the brakes when required. Note that human effort in this case is confined to opening one or more **CONTROL VALVES** which direct the flow of air. In other words, this is a full power system. Pneumatic systems are used almost universally to actuate the drum brakes on heavy trucks and coaches.

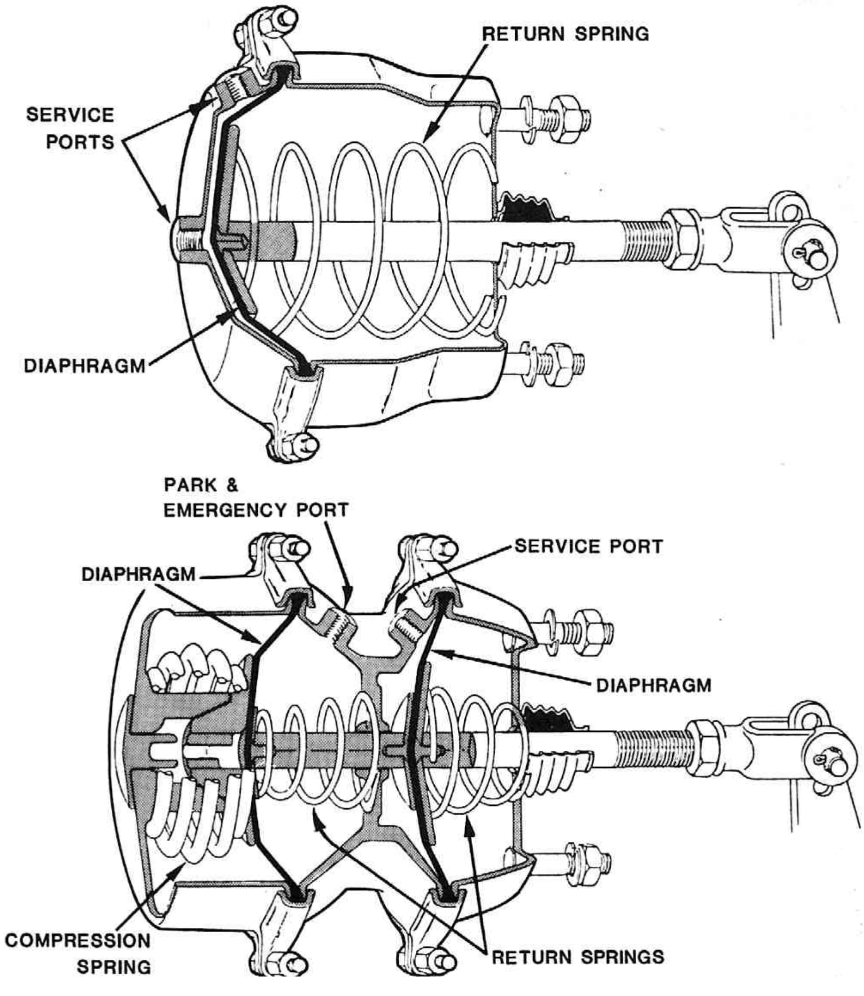


Figure 6-42 Examples of the pneumatic brake operating cylinders used on heavy vehicles. It is pneumatic pressure applied by the vehicle's compressed-air system which pushes the brake rod to move the lever on the right of the diagram to move the brake shoes into contact with the brake drum. The lower cylinder incorporates a spring brake used as an emergency brake as well as a standard park brake. Compare with the explanation in Fig 6-26.
RTA Vehicle Inspection Procedure No 24, April 1991, page 14.

3.6.4 Electromagnetic

In this type of brake, the energy to move the brake shoes into contact with the brake drum is supplied by electricity. One method is to use a solenoid coil. The operator exerts a control function only. Obviously, a source of high amperage electric current is essential if the brakes are to operate reliably.

This type of brake is sometimes used on medium-size trailers towed by cars which do not have high-pressure air available. In that case, for safety reasons, the trailer is required to have a high-current battery located on the trailer. This allows emergency braking to be actuated if the trailer breaks away from the towing vehicle.

3.6.5 Vacuum assisted

Most modern cars continue to use a hydraulic brake system, although with many safety related features such as two separate hydraulic circuits, as may be seen from Fig 6-43. To decrease the force which the operator must apply to the brake pedal, a vacuum cylinder is used to assist (i.e. add to) the effort exerted by the operator.

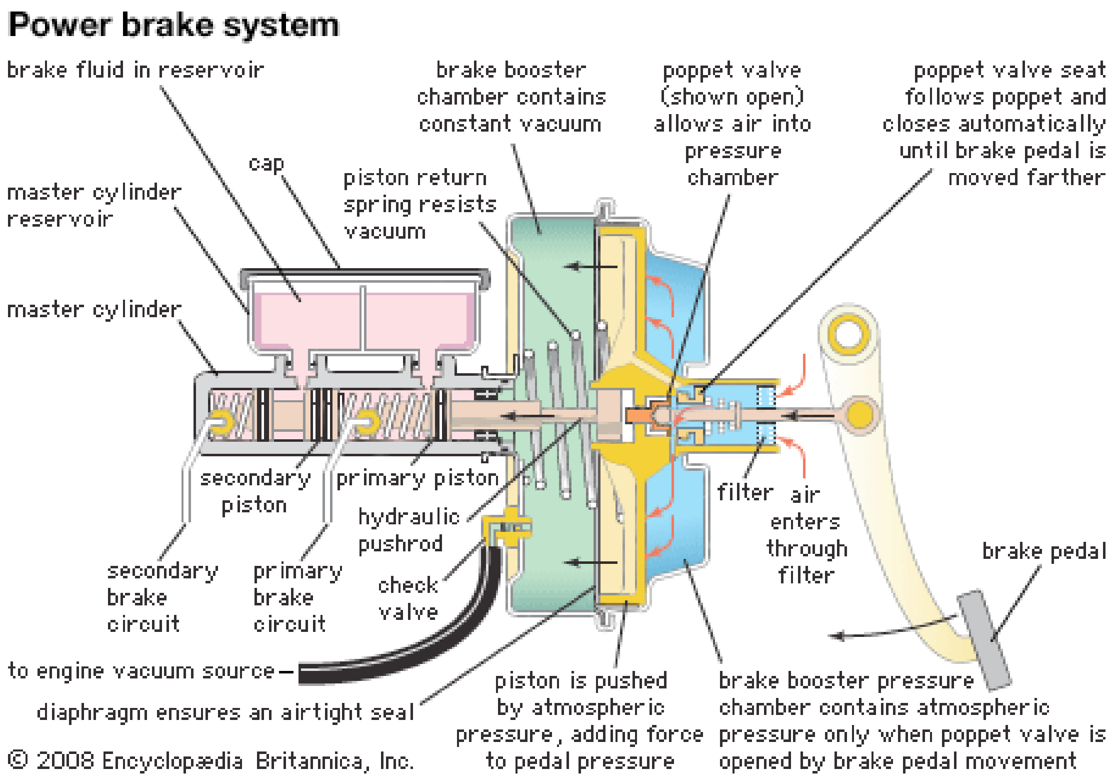


Figure 6-43 A schematic of a vacuum-assisted hydraulic braking system. In this system, the effort required by the driver to apply the brakes is reduced by utilising the partial vacuum created in the engine's air intake system.
http://www.britannica.com/EBchecked/topic-art/77441/47836/Vacuum-assisted-power-brake-for-an-automobile

Vacuum for the assisting or "boosting" cylinder is obtained by connecting it to the engine's intake manifold. Note that the brakes can still be used (although significantly more effort will be required) if the engine is not running and there is no vacuum. The vacuum cylinder is normally constructed so that, even with the engine stopped, vacuum assistance is available for two or three brake applications before the vacuum is depleted.

3.6.6 Spring brake system

See Figs 6-27 and 6-42. This is a fail-safe design in which the spring holds the brake on unless another force overcomes the spring and pulls the brake off. Spring brakes are always used on vehicles with full air brakes. Loss of air pressure causes the spring brakes to apply automatically as an emergency brake. Spring brakes are also used as the parking brake on heavy vehicles using air brakes, since all air pressure will be lost if the vehicle stands for a long period.

Spring brakes are also used in lifts, hoists and similar appliances. They apply automatically if electric power is lost.

4 Belt Drives

A **BELT DRIVE** is used to transmit power from one shaft to another. The drive is transmitted by a continuous flexible belt which runs on pulleys mounted on the two shafts. Belt drives have a number of advantages in some circumstances, including the ability to transmit power between shafts whose centres are some distance apart. Speed changes are also readily achieved, installation and maintenance costs are relatively low, and no lubrication is needed.

There are several different types of belts, as described in the notes below.

4.1 Flat belts

Historically, flat leather belts were the first belts to be used in industrial applications. They were made of strips of leather, cut from ox-hides with the ends joined together by means of leather laces. Later developments include rubber/canvas or other synthetic polymer and cord combinations manufactured as one continuous loop.

As shown in Fig. 6-44, flat-belt **PULLEYS** or **SHEAVES** are usually cambered or **CROWNED** to prevent the belt slipping off during use. Do you think this design feature will be successful? If so, why? An alternative method of preventing the belt from slipping off is to provide flanges on the pulley. Under what circumstances do you think flanges would be used?

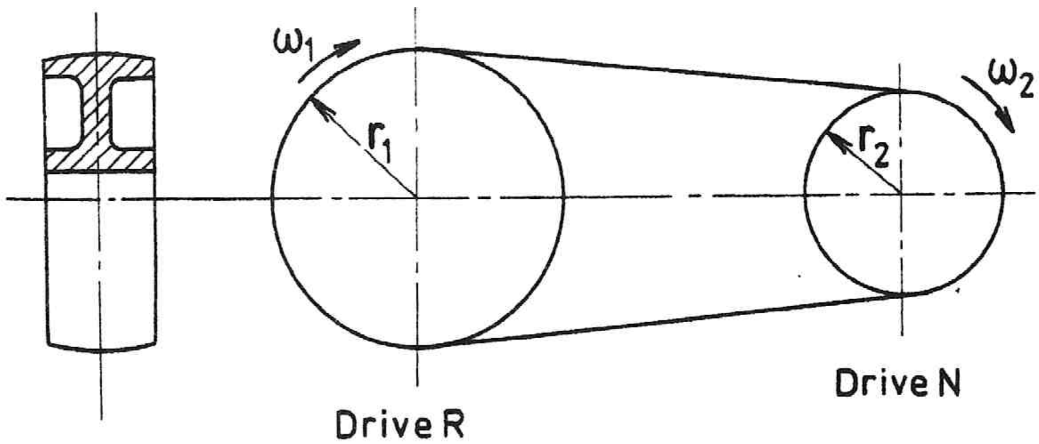


Figure 6-44 Schematic diagram of a flat-belt drive. Note the distinction between drive**R** and drive**N** pulleys.

Figure 6-45 An example of a flat-belt drive using a **STEPPED PULLEY** or **SHEAVE,** to provide drives to two machines which need to be driven at different speeds.
http://www.diracdelta.co.uk/science/source/b/e/belt%20drive/flatbeltdrive001.jpg

4.1.1 Speed and torque ratios

From Fig. 6-44, it can be seen that a flat belt runs directly on the outer surface or periphery of the pulley. It follows that the underside of the belt and the periphery of the pulley must be running at the same linear velocity, $v = r\omega$. (This statement neglects the small amounts of slip and creep which may occur when the belt is transmitting torque.) Therefore, by definition, the outer surface of the pulley is the **PITCH DIAMETER** for the pulley (see notes regarding pitch diameters for Part 4 - Gears). Calculations of **BELT SPEED** and shaft speed ratios for **flat belts** will therefore be based on the **OUTSIDE DIAMETER** (OD) of the two pulleys.

Consider the belt drive shown in Fig. 6-44, where the left-hand pulley of radius r_1 (diameter D_1) is the driveR, running at angular velocity ω_1 rad/sec (N_1 revolutions per minute (rpm)). The right-hand pulley is the driveN pulley with parameters r_2, D_2, ω_2, N_2. Since the two pulleys are linked by the relationship $v = r_1\omega_1 = r_2\omega_2$, the **DRIVE SPEED RATIO** is given by

$$\frac{N_1}{N_2} = \frac{\omega_1}{\omega_2} = \frac{r_2}{r_1} = \frac{D_2}{D_1}$$

This drive therefore serves to **increase** the speed of the driveN shaft. Furthermore, Power $P = T_1\omega_1 = T_2\omega_2$, so that $\frac{T_1}{T_2} = \frac{\omega_2}{\omega_1}$. Hence, as was the case for gears, the small pulley has the faster rotational speed and the lower torque. Note, however, that both pulleys rotate in the **same direction**.

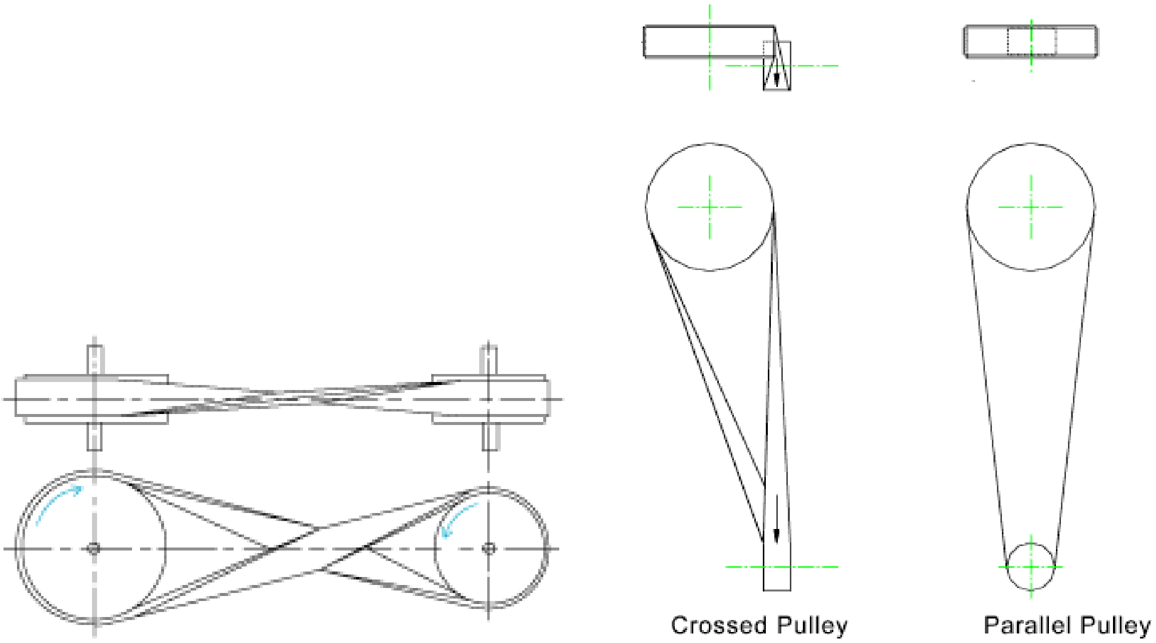


Figure 6-46 *Left:* A schematic of a **CROSSED-BELT** drive, which reverses the direction of rotation of the driven shaft. There is some friction where the belts cross. *Right:* It is also possible to set up a flat belt to drive shafts which are mutually perpendicular in what is here referred to as a **CROSSED PULLEY** drive, often called a **RIGHT-ANGLE BELT DRIVE.**
http://content.answers.com/main/content/img/McGrawHill/Encyclopedia/images/CE078100FG0020.gif

4.2 V belts

As machines became more powerful and faster, flat leather belts were found to have some shortcomings, particularly related to the strength of the material and the difficulty of joining the ends. Also, wide belts were required to transmit higher torque and the resulting pulleys became very bulky and heavy.

V belts are manufactured as continuous loops, using long cords or fibres which are wound round and round the belt before being impregnated with rubber. The cords are often strong plastic such as nylon or polypropylene. Steel cords are also used for heavy-duty belts.

V belts are made to run in the corresponding V-shaped grooves in the pulleys (Fig. 6-47). The wedging action of the belt being pulled into its groove by belt tension greatly increases the normal force and therefore increases the torque capacity of the belt. V belts, for a given torque transmission, are much more compact than a flat belt drive.

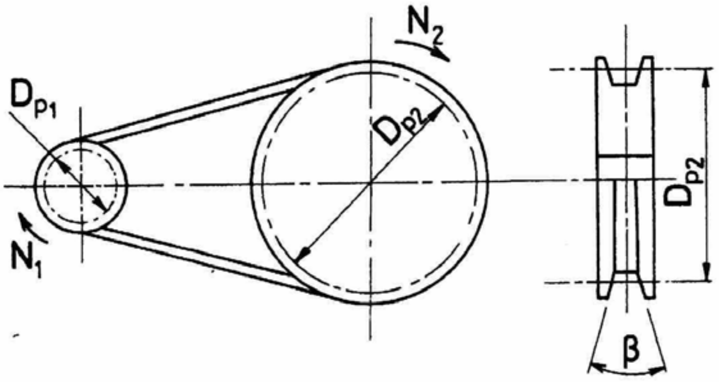
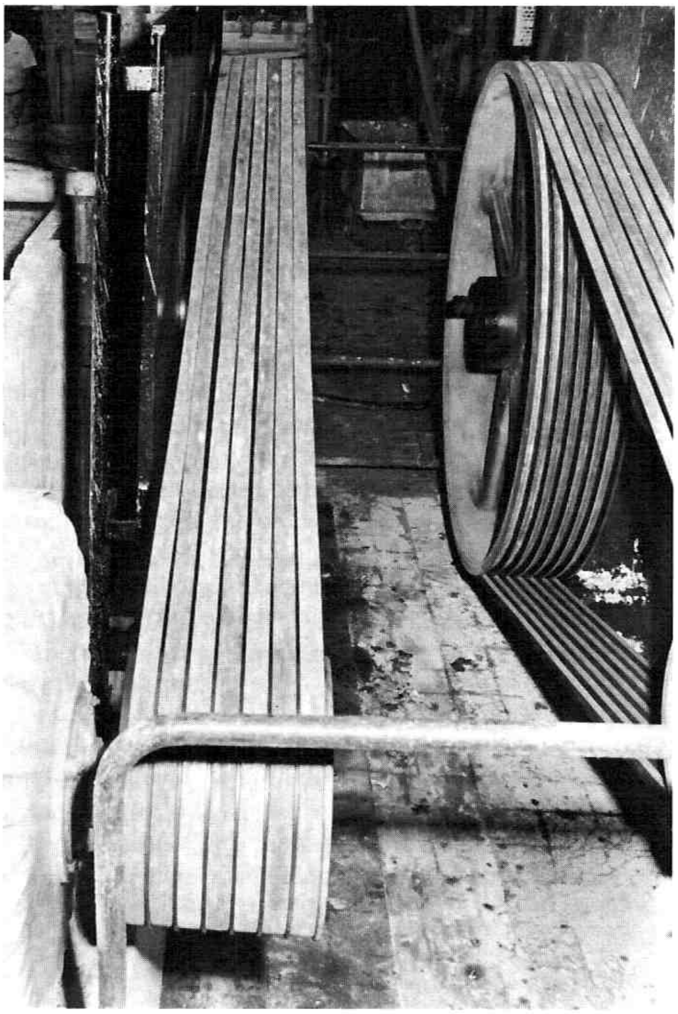


Figure 6-47 Schematic diagram of a V-belt drive showing pitch diameters D_{p1} and D_{p2} and groove angle β.



Matched teams of V-belts transmit machine drive at a paper mill (in the Netherlands) (courtesy The Goodyear Tyre & Rubber Co Ltd).

Fig 6-48 An example of a real-world V-belt drive, showing the use of multiple V belts to transmit the power required for a particular application.
Bell, Peter C (Ed) Mechanical Power Transmission, Macmillan, 1971, page 106

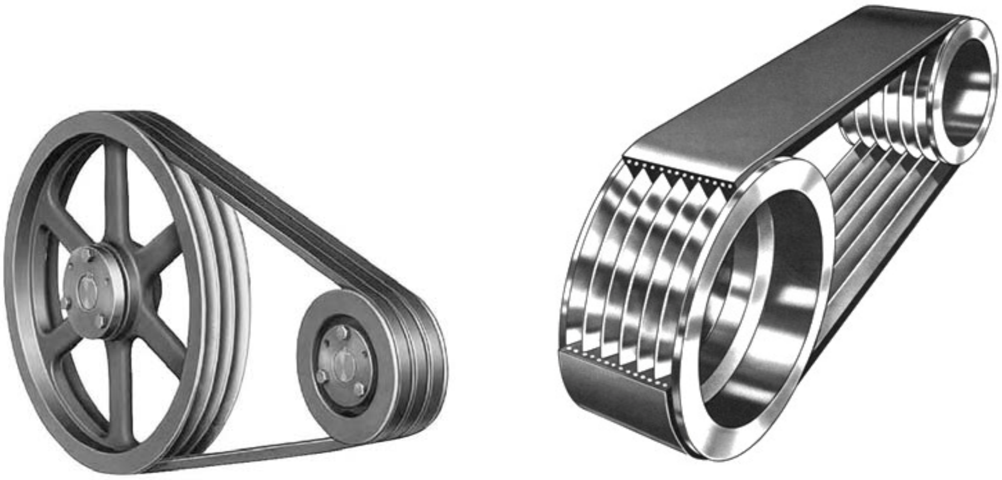


Figure 6-49a An example of a triple v-belt drive (left) and a 6-groove **POLY-V BELT DRIVE** (right). More belts or more grooves in the poly-v drive increase the power which can be transmitted.
http://www.google.com.au/search?client=safari&rls=en&q=v+belt+drives+pictures
http://www.monarchbearing.com/images/poly-v-belt-drives.jpg

Figure 6-49b A further example of a "real world" belt drive. The blue electric motor drives through a **SHAFT COUPLING** to a **JACK SHAFT** mounted on **PLUMMER BLOCKS,** then to a **SPEED-REDUCTION BELT DRIVE** (which appears to be a poly-V belt) and finally to an unidentified piece of equipment (probably with a vertical output shaft). As can be seen by the men in the background, this is quite a large piece of equipment. http://www.naismith.com.au/pdf/timingpb.pdf

All V belts are made to a standard wedge angle β. However, it is found that the required angle of the pulley groove depends on the radius of the pulley. Small pulleys may have an angle β=34° (see Fig. 6-47) and large ones β=38°. This occurs because the shape of the belt cross-section changes as it is bent to the radius of the pulley.

Note that V belts **must not** touch the bottom of the V groove, since the wedging action would then be lost.

4.2.1 Speed and torque ratios

As shown in Fig. 6-47, V belts run in V-shaped grooves cut into the pulleys (or sheaves). In this case, the effective pitch diameter is smaller than the OD of the pulley. It is roughly at the mid-point of the belt cross-section, and is indicated as Dp1 and Dp2 in Fig. 6-47. Pulley manufacturers specify their pulleys by pitch diameter, **not** their OD.

Using the appropriate pitch diameters, the speed ratio for V belts is similar to that for flat belts:

$$\frac{N_1}{N_2} = \frac{\omega_1}{\omega_2} = \frac{D_{p_2}}{D_{p_1}}$$

Fig. 6-50 shows the relevant forces and torques in the belt drive. We imagine that the belt has been cut away between the two pulleys and replaced by the forces F_T and F_S on the tight and slack sides respectively.

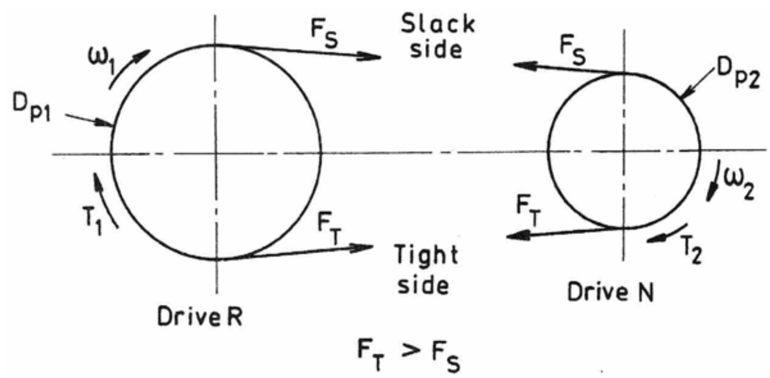


Figure 6-50 Forces and torques in a belt drive.

The left-hand pulley is the driveR (i.e. attached to the motor) and the right-hand the driveN (i.e. attached to the machine). In this case, the lower portion of the belt is the "tight" side and the upper portion is the "slack" side, giving belt forces F_T and F_S respectively ($F_T > F_S$). Pulley speeds, torques and pitch diameters are as defined in Fig. 6-50.

Now, for the driveR and driveN pulleys respectively

$$T_1 = (F_T - F_S)\frac{D_{p_1}}{2}$$

$$T_2 = (F_T - F_S)\frac{D_{p_2}}{2}$$

$$\therefore \qquad \frac{F_T - F_S}{2} = \frac{T_1}{D_{p_1}} = \frac{T_2}{D_{p_2}}$$

and Torque Ratio is $\frac{T_1}{T_2} = \frac{D_{p_1}}{D_{p_2}}$.

Since Speed Ratio is $\frac{\omega_1}{\omega_2} = \frac{D_{p_2}}{D_{p_1}}$,

it follows that $T_1\omega_1 = T_2\omega_2$,

i.e. the same power is transmitted by each pulley, as must be the case. This simple analysis neglects the small power losses which occur in any belt drive.

4.2.2 Power transmission

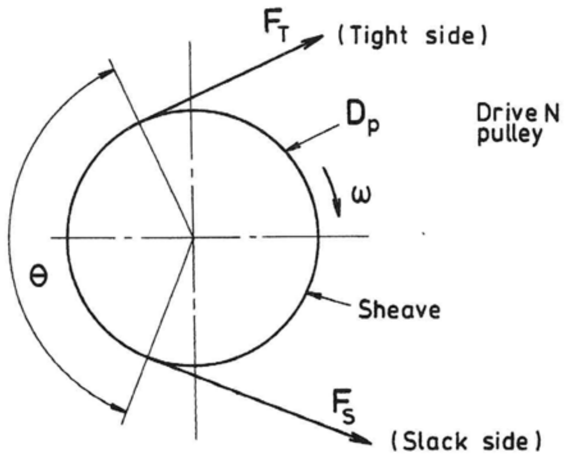


Figure 6-51 Angle of contact and belt forces acting on a driveN pulley.

A belt drive reaches its maximum torque transmission just before the belt begins to slip over the surface of the pulley, i.e.

$$T_{max} = (F_T - F_S)_{max}\frac{D_p}{2}$$

Now, $(F_T - F_S)_{max}$ is a function of the coefficient of friction (μ) between belt and pulley, the angle of wrap (θ) of the belt around the pulley and the normal force between the belt and the pulley. The normal force comes from the belt being stretched tightly between the pulleys. This is referred to as belt tension.

[**For background**. Designers of belt drives often make use of the following approximate theoretical relationships:

$$\frac{F_T}{F_S} = e^{\mu\theta} \qquad \text{for flat belts;}$$

$$\frac{F_T}{F_S} = e^{\frac{\mu\theta}{\sin(\beta/2)}} \qquad \text{for V belts;}$$

where e is the base of natural logarithms, β is the groove angle for V belts. The term sin(β/2) accounts for the increase in normal force on the belt due to its wedging in the groove.]

Hence the **torque transmission** for a belt drive can be increased by:

- Increasing the wrap angle, θ.
- Increasing the coefficient of friction, μ.
- Increasing belt tension, thereby increasing the normal force.
- Increasing the pulley diameter, Dp.
- Using multiple belts.

Furthermore, since P = Tω, **power transmission** can be increased by increasing the speed of both pulleys.

From Fig. 6-52, it can be seen that:

- The small pulley will always be the critical one, due to its small diameter and its smaller wrap angle (always less than 180□).
- Increasing the shaft centre distance increases the wrap angle on the small pulley and increases the torque capacity of the drive.

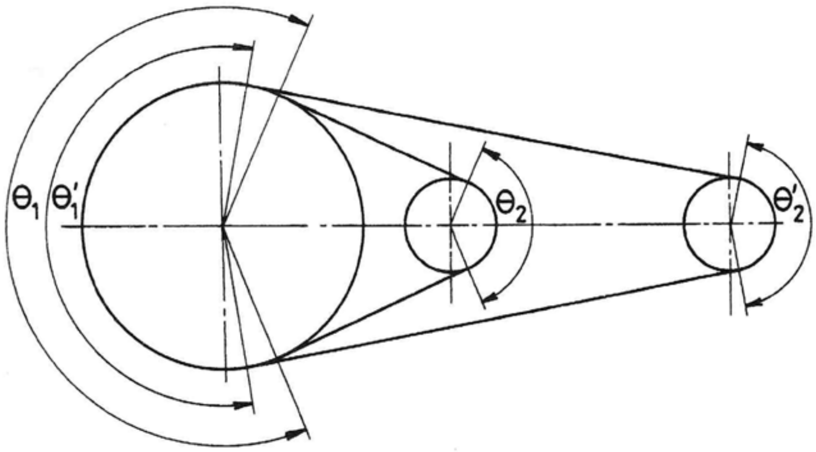


Figure 6-52 Effect of centre distance on the wrap angle of a belt drive.

4.3 Belt tension adjustment

The following methods are generally applicable to all types of belt drives.

4.3.1 Adjust centre distance

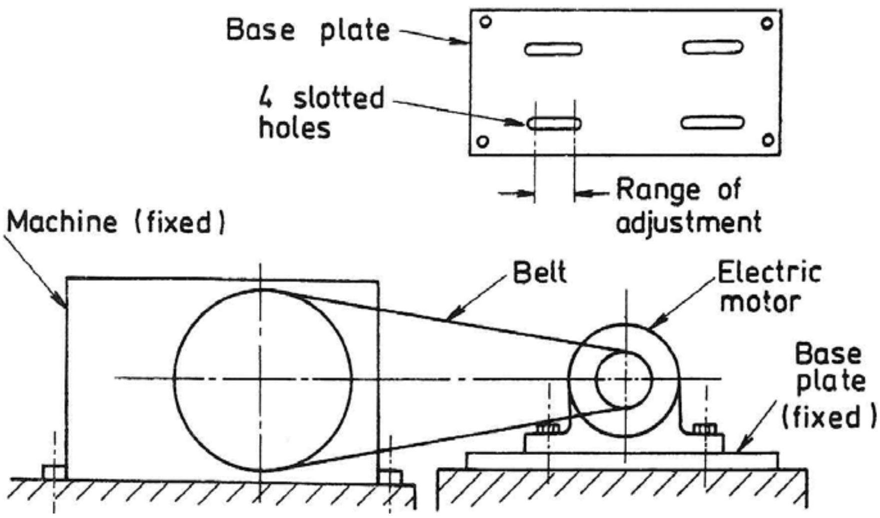


Figure 6-53 Belt tension adjustment by moving the electric motor.

In Fig. 6-53, the driven machine is fixed in position, but the belt can be tensioned by slackening the motor mounting bolts, moving the motor along the slotted holes in its base plate and then re-tightening the bolts. In larger drives, where belt tensions are very high, some form of threaded adjuster is usually provided.

4.3.2 Use of belt-tensioning pulleys

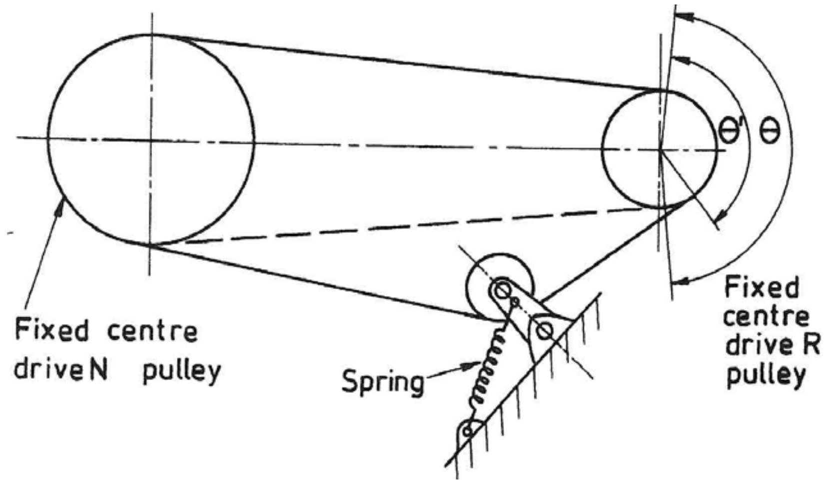


Figure 6-54 Belt tension adjustment by means of a jockey pulley.

Where the shaft centres are fixed, a belt may be tensioned by means of an **IDLER PULLEY** or **JOCKEY PULLEY** (Fig. 6-54). Such pulleys are often spring loaded so that they maintain belt tension automatically as the belt wears in service.

Note that the arrangement of Fig. 6-54 **decreases** the **WRAP ANGLE** θ on the small pulley and therefore decreases the torque which can be transmitted.

By contrast, the idler pulley in Fig. 6-55 increases the wrap angle. However, it causes some **reverse bending** of the belt and this may shorten the belt life to some extent.

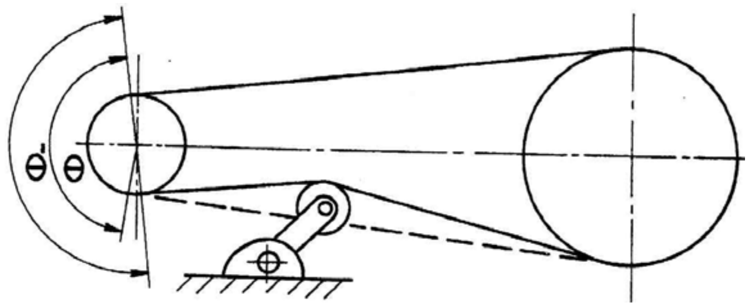


Figure 6-55 An alternative method of adjusting belt tension by means of a jockey pulley, which would normally be spring loaded.

4.4 Speed change

In some circumstances, it is of advantage to have several different speeds available. For example, a bench drill requires a high rotational speed when using very small drills, in order to increase the cutting speed. When drilling large holes, low speed (and high torque) are required.

4.4.1 Stepped pulleys

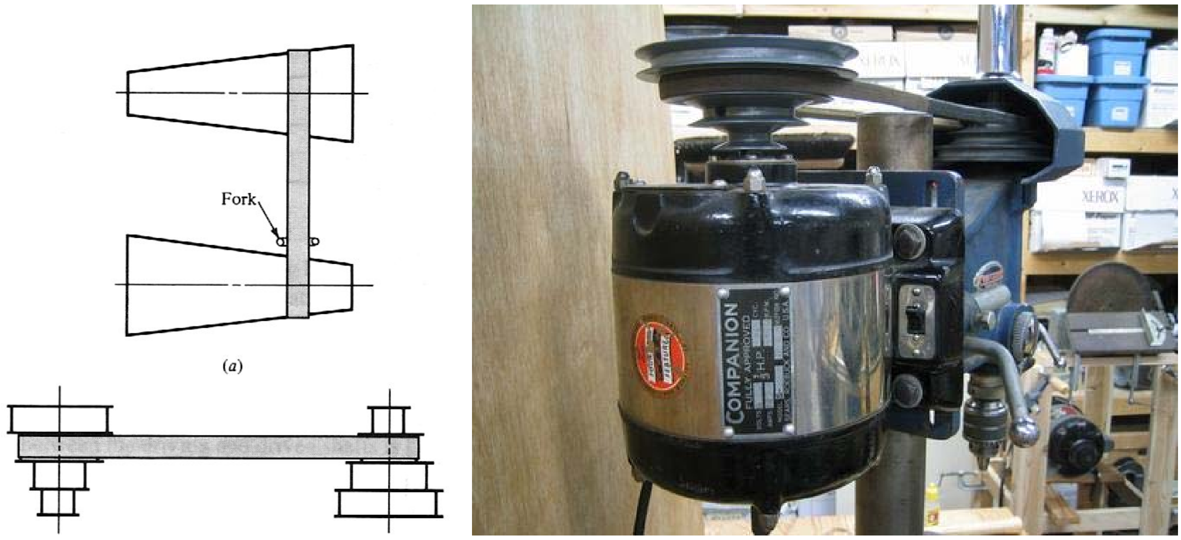


Figure 6-56 Examples of stepped pulleys to provide a range of speeds on a flat belt drive (lower left) and on a V belt drive on a bench drill (right). The schematic of the cone pulleys (upper left) illustrates the principle of an infinitely variable speed drive in which the **FORK** is used to guide and hold the belt in the required position.
Shigley, J E, Mischke, C R, Mechanical Engineering Design, 5E McGraw Hill, 1989, page 668
http://www.owwm.com/PhotoIndex/detail.aspx?id=1530

Fig 6-56 shows how stepped pulleys are used to achieve speed change on a flat belt drive and a V belt drive. The four belt positions in each case can be seen to provide either a speed reduction or a speed increase, as required. The pulley sizes are chosen so that the belt length and belt tension remain unchanged as the belt is moved to the different positions, so no change in centre distance is needed.

The concept of the infinitely variable drive (top left in Fig 6-56) is to move the belt by means of a fork or guide along the two conical pulleys, equivalent to an "infinite" number of stepped pulleys.

4.4.2 Variable speed belt drives

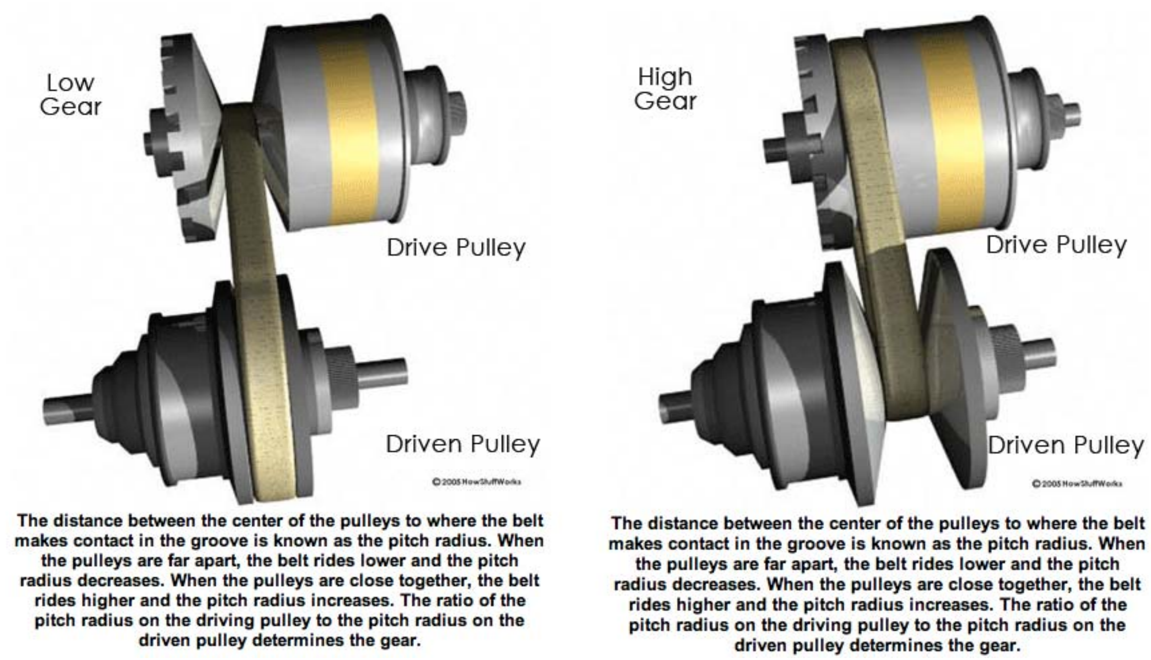


Figure 6-57 An illustration of an infinitely variable belt drive in which the two sections of the V pulleys are able to move axially to change their effective pitch diameter. http://auto.howstuffworks.com/cvt2.htm

Several different types of variable speed belt drive units have been used over past decades and there has been significant development over recent years, particularly in automotive drive systems.

The operating principle is generally to change the effective pitch diameter of the two pulleys, as seen in Fig 6-56 and 6-57, one increasing and the other decreasing, so that belt length and belt tension remain constant. The overall speed ratio may be made to vary to a factor of 4:1 or 5:1.

Variable speed drives have generally been regarded as light duty drives but recent developments are increasing their capability.

4.5 Timing belts

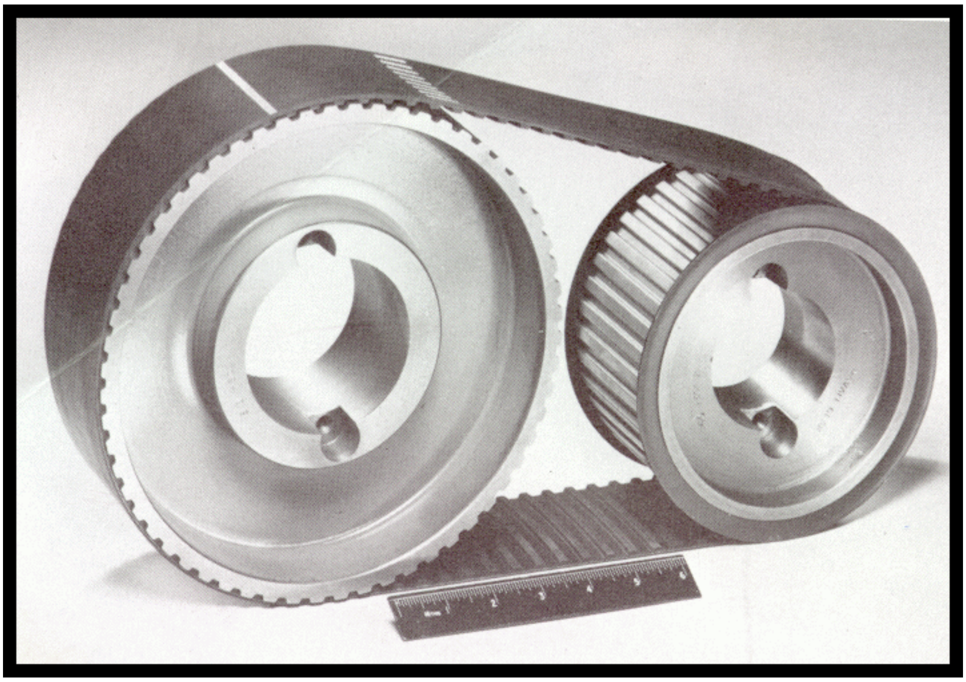


Figure 6-58 A photograph of two pulleys driven by a timing belt. Note the guide flanges on the right-hand pulley to ensure the belt does not drift off the pulleys.
http://www.google.com.au/imgres?imgurl

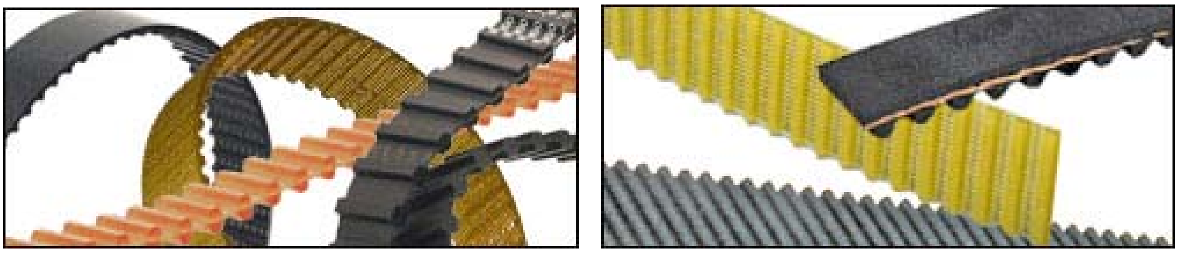


Figure 6-59 Examples of some commonly available timing belts. Note that on some belts teeth are formed on both sides of the belt and in a complex drive both sides of the belt will be used to drive different components.
http://www.sdp-si.com/eStore/CoverPg/belts.htm

Fig 6-58 illustrates the principle of a simple timing belt drive. Its principal advantage is that the teeth or lugs formed on the belt match the grooves on the pulleys and, acting almost like a chain running on sprockets, form a positive drive with fixed angular relationship between driving and driven pulleys. Timing belts are now used almost universally on car engines to drive the camshaft which operates the valves and must therefore have an accurate phase relationship relative to the crankshaft. Such a drive has a long life, is quiet in operation and is economical.

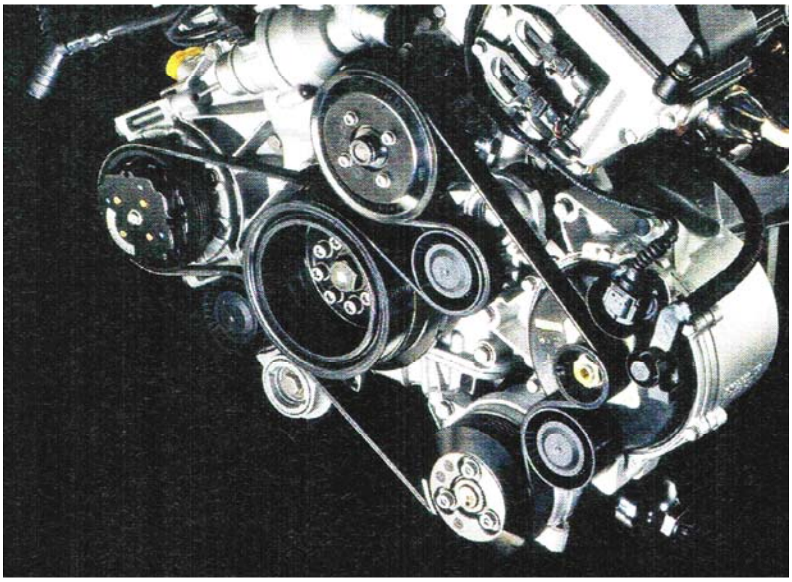


Figure 6-60 An example of a more complex toothed belt drive often referred to as a **SERPENTINE DRIVE.**
Automotive Engineering International, October 2009, page 7.

4.6 Characteristics of belt drives

- Belt drives are suitable for medium to long centre distances. Compare with gears, which are suitable only for short centre distances.
- Belt drives have some slip and creep (due to the belt extending slightly under load) and therefore do not have an exact drive ratio.
- Belts provide a smooth drive with considerable ability to absorb shock loading.
- Belt drives are relatively cheap to install and to maintain. A well-designed belt drive has a long service life.
- No lubrication is required. In fact, oil must be kept off the belt.
- Belts can wear rapidly if operating in abrasive (dusty) conditions.

5 Chain Drives

There are similarities between chain drives and belt drives, and many of the operating principles apply to both. The analysis of speed and torque relationships developed above for belts apply equally to chain drives, always working with pitch line velocities or their equivalent.

5.1 Examples of chain drives

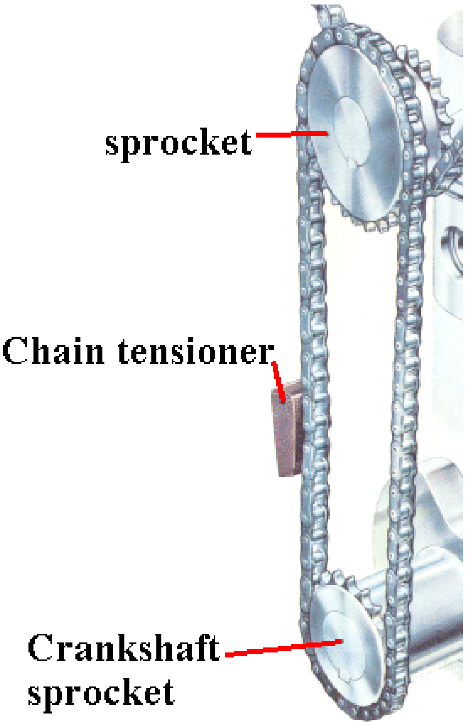


Figure 6-61 A simple roller chain drive used in an automotive application. The chain and sprockets would be encased within a cover or housing as part of the engine to maintain cleanliness and to provide lubrication. In this layout, shaft centres are fixed. Note the use of a **CHAIN TENSIONER** to prevent excessive deflection or "whipping" of the slack side of the chain.
http://www.motorera.com/dictionary/ch.htm

Figure 6-62 An example of a roller chain drive used in industry. The application appears to be a series of powered conveyor rollers, all driven from one motor. There does not appear to be provision for lubrication in this drive system.
http://www.nleco.com/CatalogPDFs/ChainDrivenConveyors.pdf

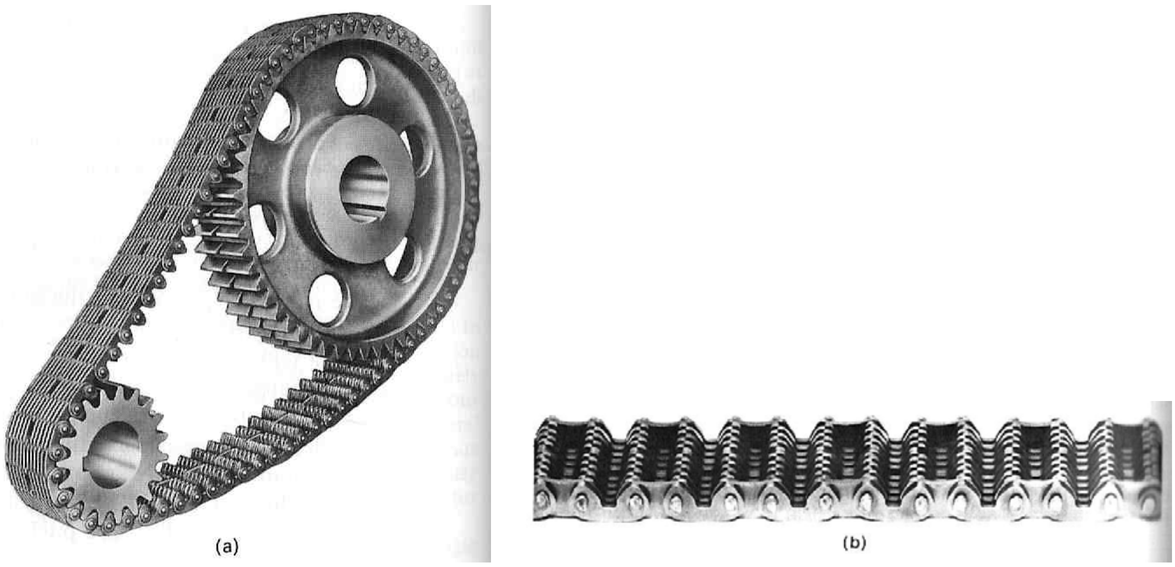


Figure 6-63 (a) A chain drive of the type described as a **SILENT CHAIN,** which does run more quietly than a roller chain. (b) A view of the links of a silent chain. The plates on each side of the links align the chain on its sprockets.
Deutschman A D, Michels W J, and Wilson C E, Machine Design Theory and Practice, Macmillan, 1975, page 672.

5.2 Characteristics of chain drives

Some points to note are:

- Chains provide a positive drive suitable for use where timing/phasing is required. Before the development of timing belts, chains were frequently used for driving the camshafts of motor car engines.
- Chains transmit shock loading, whereas belts tend to absorb any shock loading which may occur.
- The speed ratio is determined by the number of teeth on the two chain wheels or sprockets, although calculations based on sprocket pitch diameters and pitch line velocities are equally valid.
- A chain drive is more costly to set up than a belt drive, but has a long life.
- Very high torques may be transmitted by chains, beyond the capacity of belt drives. The drive does not depend on friction.
- Chains generally require lubrication and a heavy duty chain drive may require a sealed housing incorporating either bath or jet lubrication, thereby increasing cost.
- Abrasive material rapidly destroys a chain drive.
- It is best not to use chain drives on very long centre distances because the long lengths of chain tend to "whip".

6 Concluding Remarks

This set of six resource documents is intended for use as an introduction to engineering hardware for students in the early years of a course in Mechanical Engineering. I have attempted to steer a middle path between lots of detail and a simplified presentation, focussing on the principles of the mechanical components being considered. The documents, even when supplemented by lectures, are not intended to be anything more than an introduction to some basic engineering hardware. To use an analogy, they represent no more than the tip of a very large iceberg, with the real detailed understanding still buried beneath the surface. It is my hope that students will be able to use a basic understanding of hardware components gleaned from these notes as the foundation for further study and understanding.