Chapter 9 Couplings, Clutches and Brakes

Couplings, clutches and brakes are mechanical devices that connect or separate motions. In most cases, they are systems rather than single components. A coupling set makes a permanent connection of two rotating shafts, such as the outlet of an electrical motor, or a spindle, and the input end of a gear box, a generator, etc. A clutch makes a controlled connection of shafts to start a motion; but a brake does the opposite, making a controlled disengagement to stop a motion.

9-1. Couplings

The concept of coupling has been briefly introduced in Chapter 4, Shafts. Here are two of the shaft assemblies again. The far left of each shaft system is a half coupling connected to the shaft end through a key in the radial direction and an end plate by a bolt into the shaft in the axial direction; the other half of the coupling should be on the end of the shaft to be connected. The two halves should be connect by bolts through the bolt holes in the flange of each half.

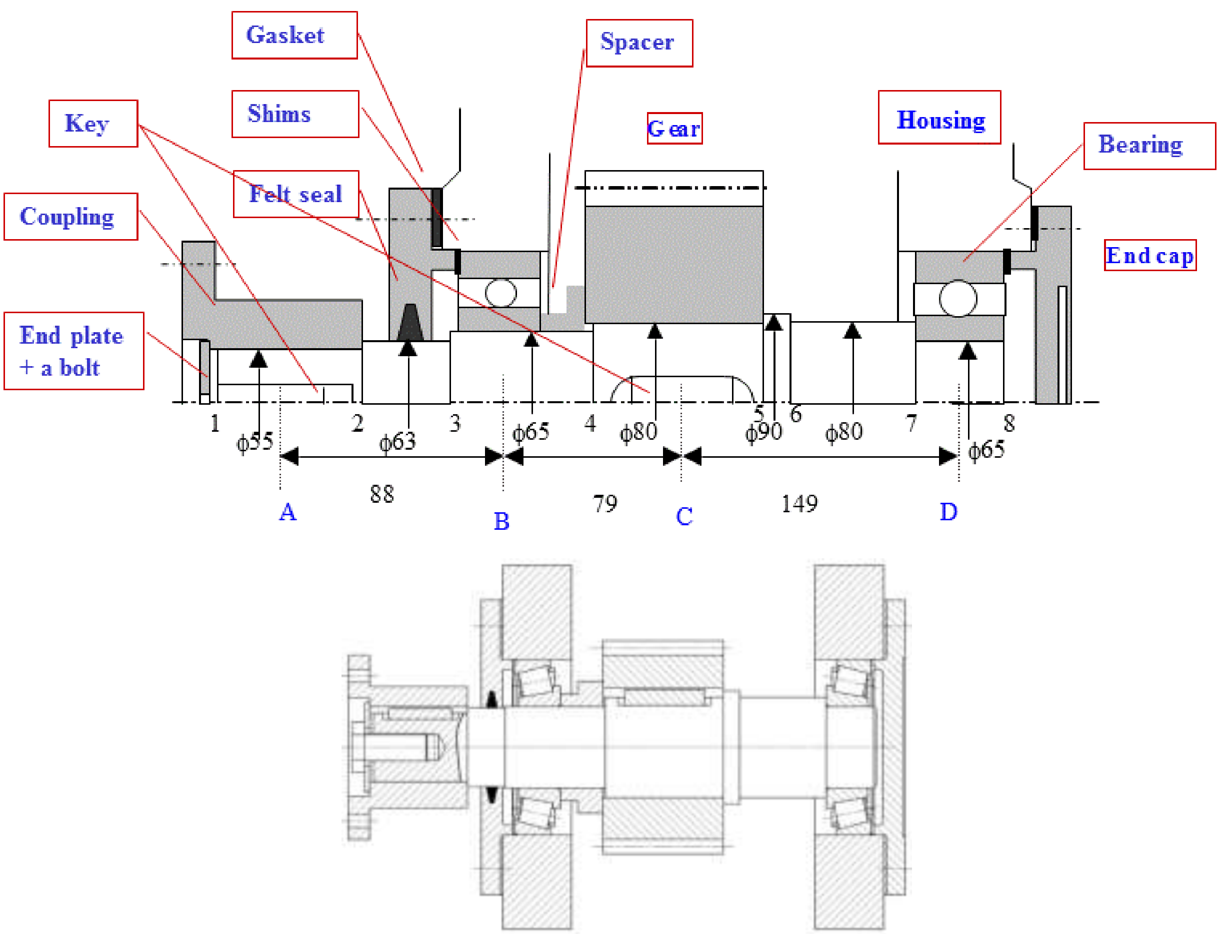
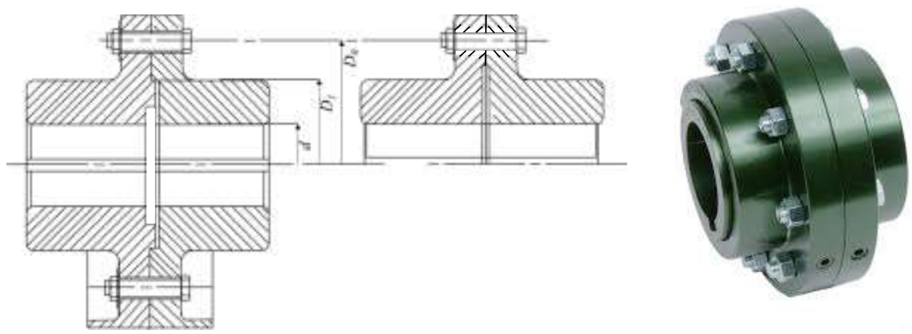


Figure 9-1 Shaft assemblies; the far left parts are half couplings.

Rigid couplings. The couplings in **Figure 9-1** are rigid couplings. **Figure 9-**2 shows two possible structures of complete coupling designs and connections of the two halves. The key issue for this type of couplings is the alignment of the centerlines of the two shafts. This figure shows two methods, accurate fit of shoulders of the two halves and the use stripper bolts (or shoulder bolts).



(a) (b) (c)

Figure 9-2 Rigid couplings of bolted flanges, alignment by shouldered cylinders (a) and alignment by shouldered bolts (b). (c) shows an assembled set (from internet)

Rigid couplings are simple in structures; however, they do not allow shaft misalignment, or the shafts should be well aligned in assembly.

Shaft misalignments. The following figure shows possible shaft misalignments, which are 1) axial misalignment along the shaft centerlines, 2) parallel misalignment by a shift perpendicular to the shaft centerlines; 3) angular misalignment by rotation of one of the shaft centerline with respect to that of the other; and 4) torsional misalignment, which is the combination of the three mentioned above. The last one is the most commonly seen shaft misalignment. Design of couplings should consider shaft centerline misalignments.

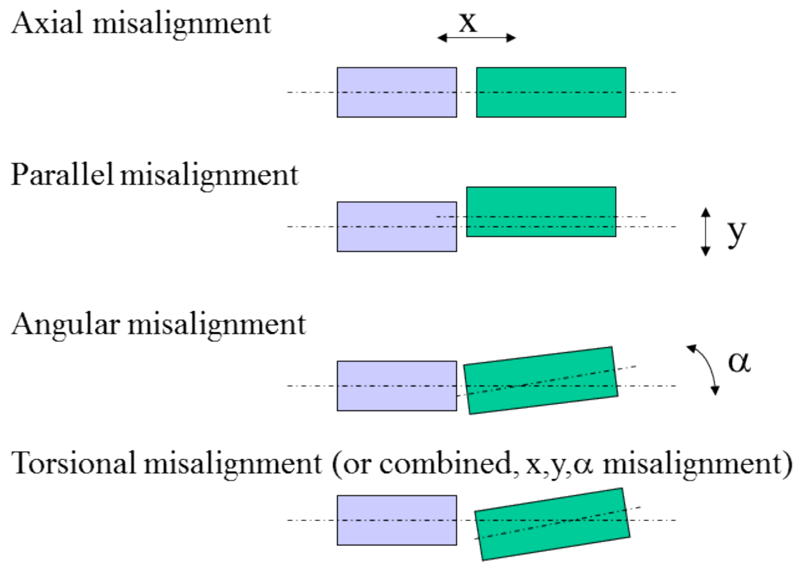


Figure 9-3 Shaft centerline misalignments.

Flexible coupling. Flexible couplings are designed for shaft connection considering centerline misalignments. **Figure 9-4** shows flexible-pin couplings, in which elastomer pins are used. The flexible-pin design may allow all types of small shaft misalignments; however, wear and damage of the elastomer material limits the application of this coupling to light and smooth loading cases.

Figure 9-4. Flexible-pin couplings using elastomer pins absorb misalignments. Photos are from internet.

Figure 9-5 shows jaw coupling sets consisting of three pieces. The middle one can be made of a different material and has two sets of orthogonal pillars (left) or slots (right), which can slide with respect to the mating pieces to absorb misalignments. This design may allow shaft axial and parallel misalignments. It is for low-speed applications due to the concern of wear of the jaws and the centrifugal force on middle part.

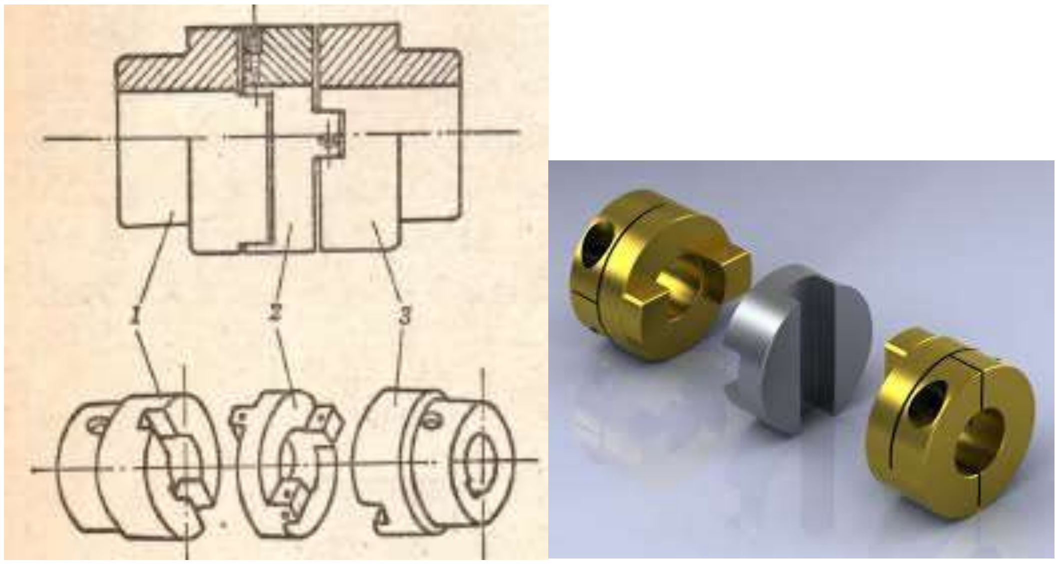


Figure 9-5. Jaw couplings. Left: parts 1 and 3 are identical, with a slot each, while 2 has external pillars. Part 2 may be made of a soft material, such as a soft metal. Right: photo of a jaw coupling, from internet.

9-2 Clutches and Brakes

Clutches and brakes are widely used for controls of system motions, more specifically, for connection and separation of motions. The automotive transmission, for example the one in Chapter 5, which is repeated here in **Figure 9-6** for clarity, has a number of clutches and brakes to accomplish different functions of the two linked planetary gear systems.

A clutch and a brake may share the same structure, which means that a clutch can be made into a brake. Therefore, most of these two devices are discussed together in this section. However, some structures are more proper for the former, some others are more proper for the latter.

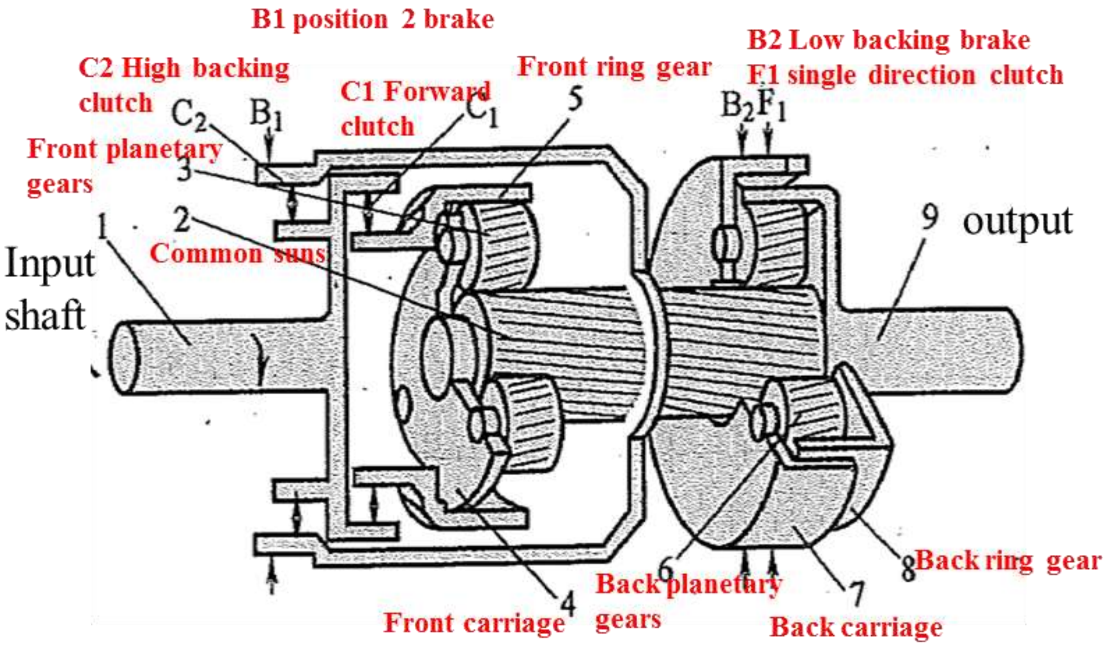


Figure 9-6. Clutches and brakes in the 3-position (3 forward shifts) Simpson planetary gearing system.

Friction clutch (brake). Disk clutch (brake) is a friction clutch (brake), in which friction is the force for the connection. A high-friction material layer (or pat) is usually used to produce a high friction once engaged. **Figure 9-7** illustrates the mechanism of a disk-type of clutches, or brakes.

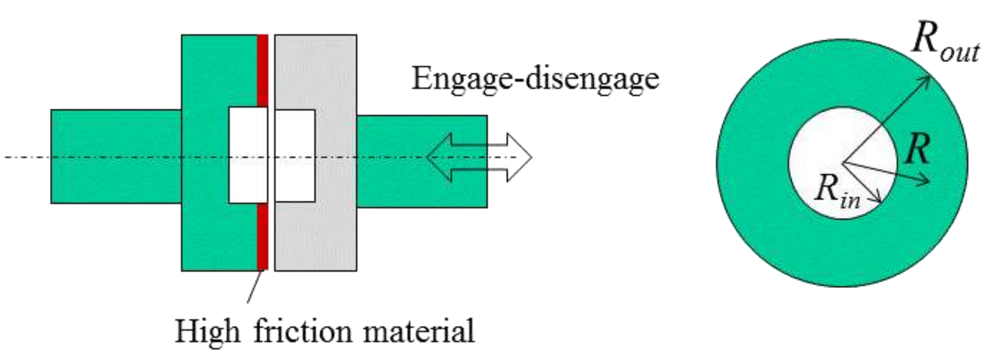


Figure 9-7. Disk clutch (brake) with a high friction material.

For rigid disks, the approaches of the two surfaces should be constant, assuming perfect alignment, which means the wear of the disk surfaces should be uniform in a normal operation.

If so, the pressure on the surfaces will not be constant, and the maximum pressure, p_{max}, should be at the inner radius, R_{in}, and

$$\frac{p}{R_{in}} = \frac{p_{\max}}{R} \qquad (9\text{-}1)$$

The total normal force on the disk, which is also the actuating force, is the integration of the pressure over the disk surface,

$$F = \int_{R_{in}}^{R_{out}} \left(\frac{p_{\max}}{R} R_{in} \right) 2\pi R dR = 2\pi p_{\max} R_{in} \left(R_{out} - R_{in} \right) \qquad (9\text{-}2)$$

The torque is the integration of the frictional torque, assuming friction coefficient μ of the interface.

$$T = \int_{R_{in}}^{R_{out}} \mu \left(\frac{p_{\max}}{R} R_{in} \right) (R) 2\pi R dR = \pi p_{\max} R_{in} \left(R_{out}^2 - R_{in}^2 \right) = \pi p_{\max} R_{in} \left(R_{out} - R_{in} \right) \left(R_{out} + R_{in} \right)$$
$$= \frac{\mu F}{2} \left(R_{out} + R_{in} \right) = \mu F R_{av}$$
(9-3)

The torque can be expressed by the product of friction coefficient μ, normal force F, average radius R_{av}. Theoretically, pressure is infinite at R = 0, therefore, the disks should have annular shapes to avoid R = 0.

If the friction material is flexible (soft), constant pressure p can be expected. Correspondingly, wear should be non-uniform. More wear should be expected at the outer radius of higher speed; however, R=0 should also be avoid due to drastic velocity direction change there. The normal force and frictional toque are as follows.

$$F = \int_{R_{in}}^{R_{out}} p 2\pi R dR = \pi p \left(R_{out}^2 - R_{in}^2 \right) \qquad (9\text{-}4)$$

$$T = \int_{R_{in}}^{R_{out}} (\mu p R) 2\pi R dR = \frac{2}{3} \pi p \mu \left(R_{out}^3 - R_{in}^3 \right) = \frac{2}{3} \mu \frac{\left(R_{out}^3 - R_{in}^3 \right)}{\left(R_{out}^2 - R_{in}^2 \right)} F \qquad (9\text{-}5)$$

Figure 9-8 shows two other shapes of frictional clutches (brakes); one uses more disks to enlarge the friction effect, and the other has a conic friction interface for convenient engaging, where the actuating force is enlarged by the wedge, which is the normal force at the frictional interface for higher friction. Figure 9-9 shows details of a multiple-disk brake, where two sets of disks are used, one mounted on the shaft while the other in the drum.

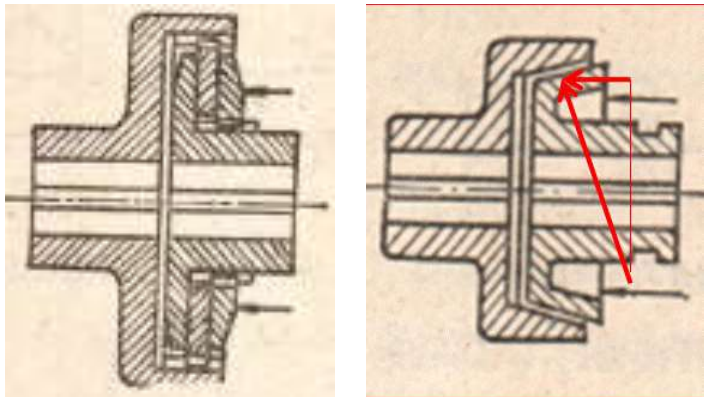


Figure 9-8. More friction clutches (brakes); left, more disks to enlarge the friction effect, right, conic friction interface for convenient engaging and for enlarging the normal force at the friction interface.

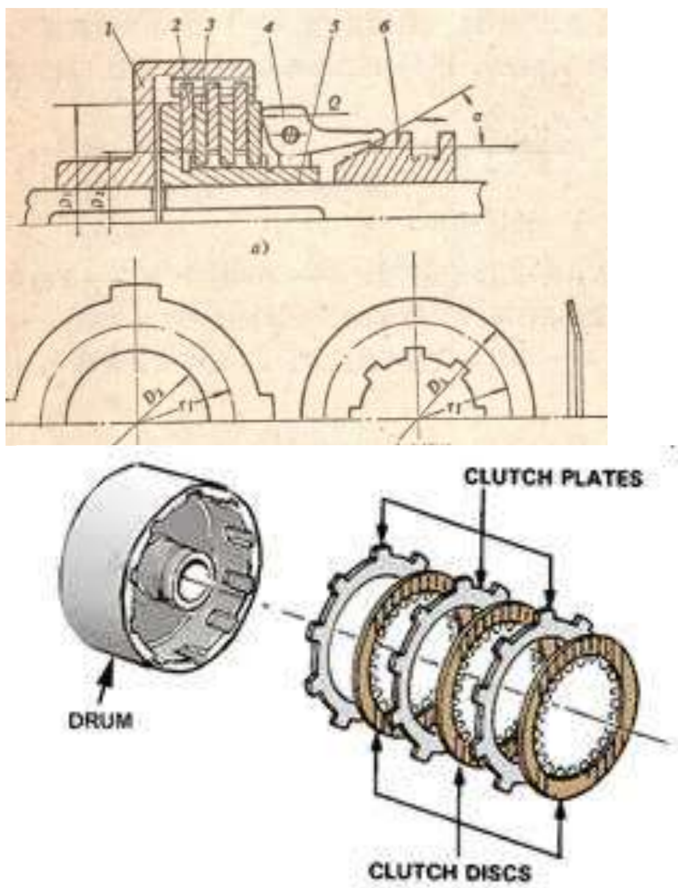


Figure 9-9. Multiple-disk brake; Top, 1: drum mounted on one shaft, 2&3: friction disks, one set is mounted on the right shaft while the other in the drum linked to the left shaft, 4: pressure block, 5) another shaft, and 6) actuator. Bottom, a photo from internet, showing the drum and the two sets of disks.

Figure 9-10 shows one more type of friction brake, which uses a level to press the shoe onto the drum to stop the rotation of shaft of the drum. The shoe is made of a frictional material. Forces can be easily calculated.

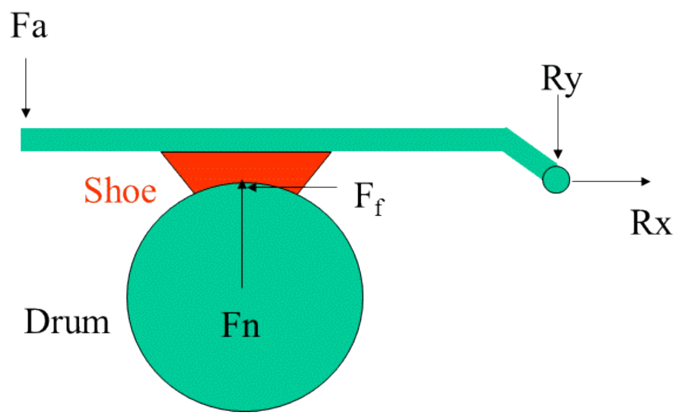


Figure 9-10. Frictional brake containing a shoe and drum.

Toothed clutches. Toothed clutch connects shafts through tooth engagement. This type of the structures is not used for brakes because it cannot dissipate large amount of heat due to their relatively smaller areas of tooth contact. **Figure 9-11** shows two simple toothed clutches; one uses surface teeth, and the other looks like an internal gear set. Toothed clutches use 3~60 teeth, smaller numbers are used for larger torque transmissions.

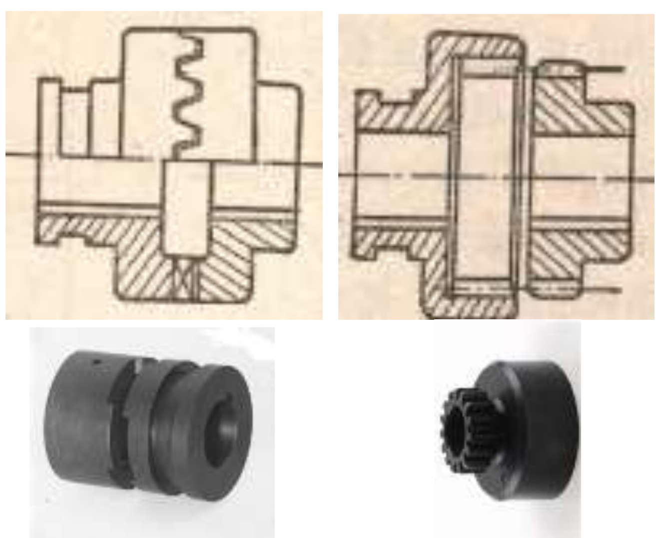


Figure 9-11. Simple toothed clutches. Top left: surface tooth meshing, top right: internal tooth meshing. The bottom photos are examples from internet.

Figures 9-12 shows detailed structure of a toothed clutch. Three different tooth shapes can be designed; square, trapezoidal, and triangular. Square teeth are good for two directions of motions (forward and backward), but not convenient for clutch engagement; trapezoidal teeth are good for wear compensation and large torque transmission; triangular teeth have the highest tooth strength, but they can only transmit one-direction torque.

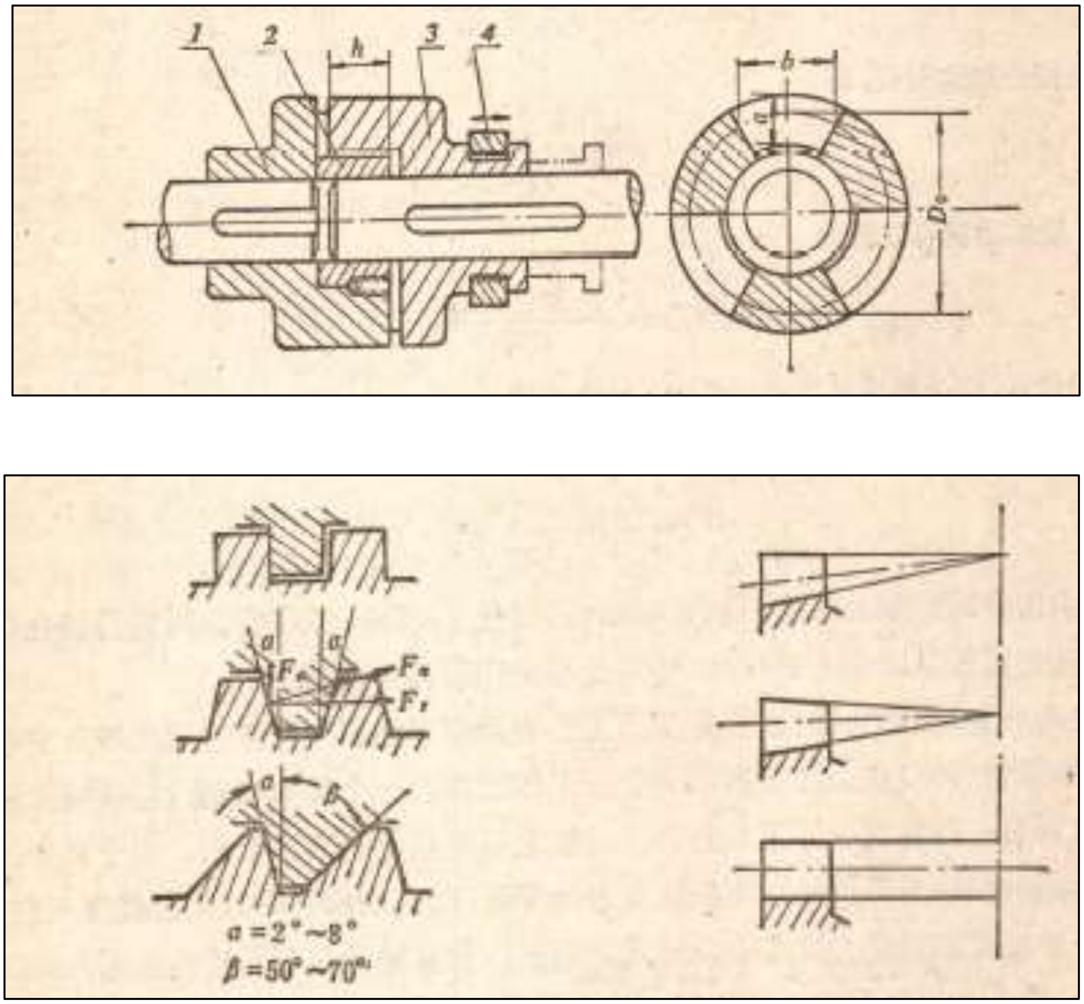


Figure 9-12. Design of toothed clutches. Top, 1&3: toothed drums, 2: shaft sleeve, 3: actuator. Bottom, three types of tooth shape designs and three alignment designs.

Over-running clutch. Figure 9-13 shows an over-running clutch; it uses the competition between the spring force and the centrifugal force to determine system engagement and separation. In this system, the shoes are rotating with one shaft while the drum is mounted on the other shaft. The shoes do not touch the inner surface of the outer cylinder unless the centrifugal force is larger than the spring force. If so, the outer cylinder and the shoe shaft will rotate together.

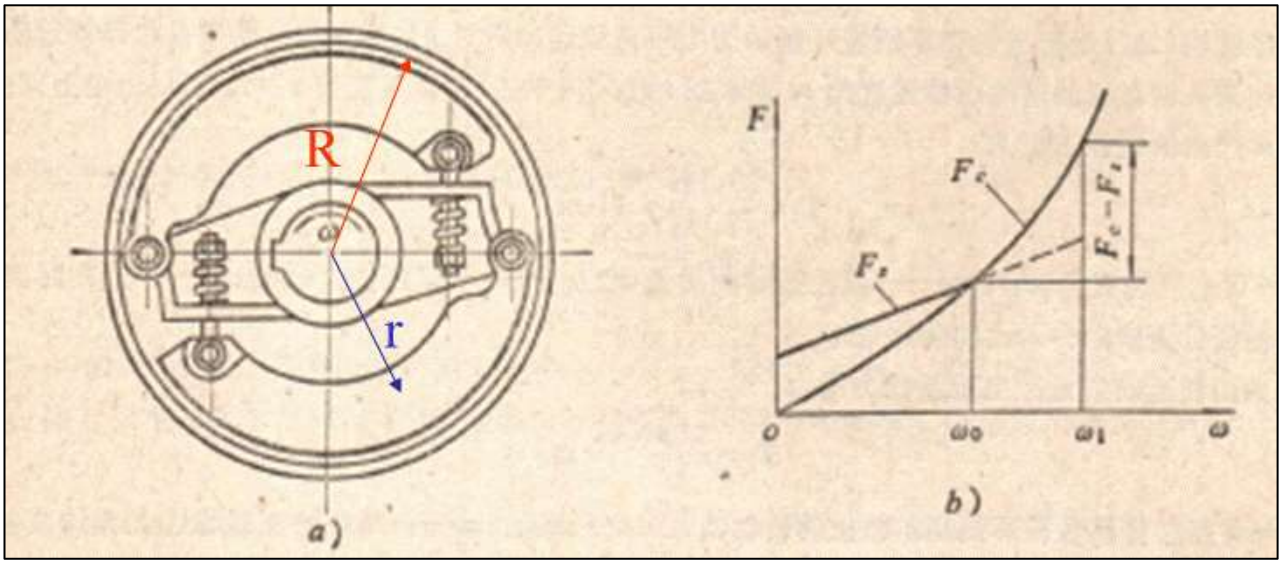


Figure 9-13. Over-running clutch formed by rotating shoes, each mounted on a hinge and a spring (left) and subjected to both the centrifugal and spring force (right).

The spring force is $F_s = kx$, the centrifugal force is $F_c = m\omega^2 r$, where m is the rotation mass of the shoe set and r the equivalent radius of the rotation mass of the shoe. If N is the number of pads and μ the friction coefficient, the friction torque = $(F_c\text{-}F_s)\ \mu RN$. Note that the maximum spring force is achieved at the contact of the shoe and the drum inner surface, while the centrifugal force is a function of the rotation speed. The higher the rotation speed is, the stronger the engagement force. Therefore, this over-running clutch has a lower limit of engagement; besides, it works for one rotation direction only to avoid impact during the engaging process.

Chapter Summary. This chapter introduces the concept of couplings, clutches, and brakes. Several structures of each are discussed, which are the rigid and flexible couplings and their structures, friction clutches and brakes, toothed clutches, and an over-running clutch.

References

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Media: Several U-Tube movies are on line. 315 students should watch the short U-Tube movies before the class.