Machine DESIGN II

Lecture 4

Clutches

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Introduction

Clutches are required when shafts must be frequently connected and disconnected. The function of a clutch is : first, to provide a gradual increase in the angular velocity of the driven shaft, so that its speed can be brought up to that of the driving shaft without shock; second, when the two shafts are rotating at the same angular velocity, to act as a coupling without slip or loss of speed in the driving shaft.

A clutch is a machine member used to connect a driving shaft to a driven shaft so that the driven shaft may be started or stopped at will, without stopping the driving shaft. The use of a clutch is mostly found in automobiles.

A clutch is a mechanism for transmitting rotation, which can be engaged and disengaged. In a drill for instance, one shaft is driven by a motor, and the other drives a drill chuck. The clutch connects the two shafts so that they can either be locked together and spin at the same speed (engaged), or be decoupled and spin at different speeds (disengaged).

Introduction

A **brake** is a device used to bring a moving system to rest, to slow its speed, or to control its speed to a certain value. The function of the brake is to turn mechanical energy into heat. The design of brakes and clutches is subjected to uncertainties in the value of the coefficient of friction that must necessarily be used.

Introduction

Brakes and clutches are similar, but different from other machinery elements in that they are tribological systems where friction is intended to be high. Therefore, much effort has been directed toward identifying and developing materials that result in simultaneous high coefficients of friction and low wear so that a reasonable combination of performance and service life can be achieved. In previous years, brake and clutch materials were asbestos-fiber-containing composites, but the wear particles associated with these materials resulted in excessive health hazards to maintenance personnel.

Introduction

Modern brakes and clutches use “semimetallic” materials (i.e., metals produced using powder metallurgy techniques) in the tribological interface, even though longer life could be obtained by using the older asbestos-based linings. This substitution is a good example of multidisciplinary design, in that a consideration totally outside of mechanical engineering has eliminated a class of materials from consideration.

Typical brake and clutch design also involves selecting components of sufficient size and capacity to attain reasonable service life.

Introduction

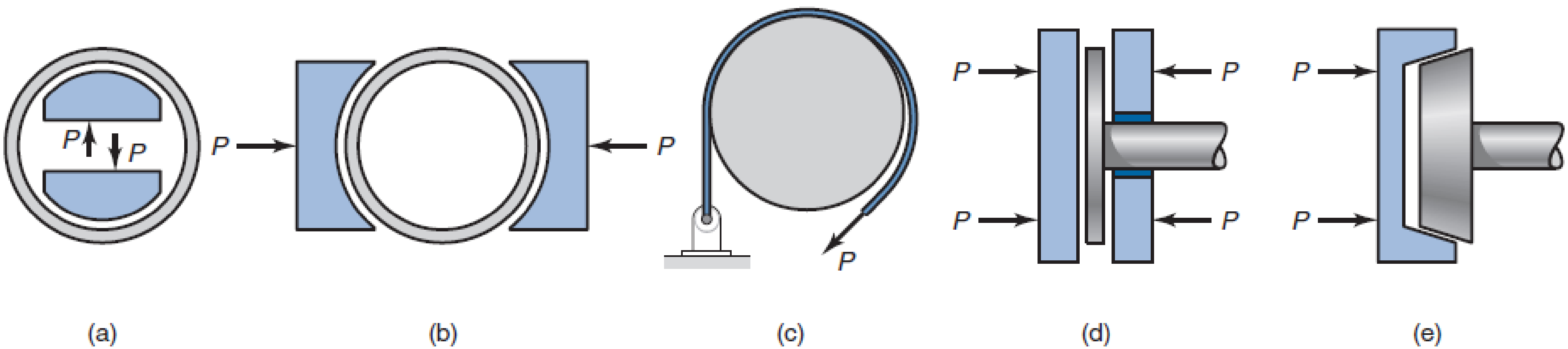
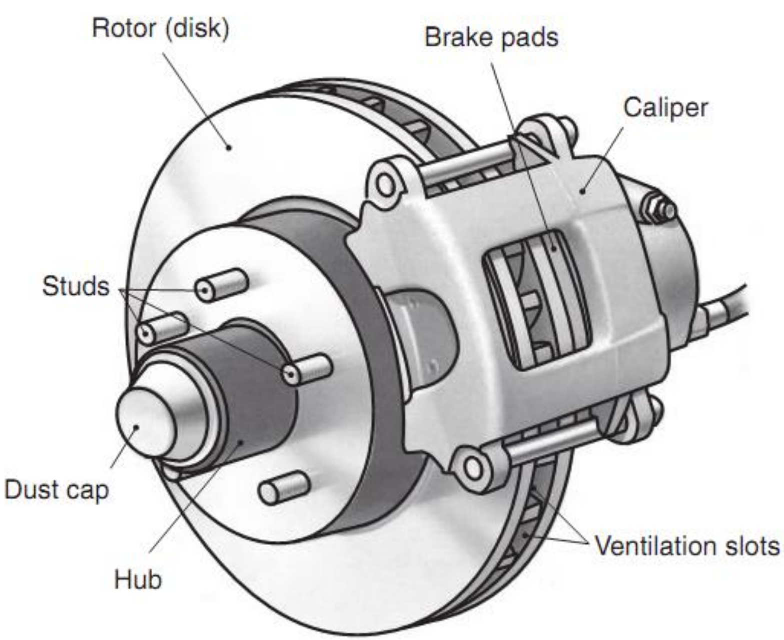


Fig. 1. Types of brake and clutch. (a) Internal, expanding rim type; (b) external contracting rim type; (c) band brake; (d) thrust disk(clutch); (e) cone disk (clutch).

Thrust Pad Clutches and Brakes

A thrust disk has its axis of rotation perpendicular to the contacting surfaces, as shown in Fig. 1.d; Fig. 2 illustrates the components of an automotive disk brake. Basically, a rotor or disk is attached to the vehicle's axle, and a caliper that is mounted on the automobile body contains two brake pads. The pads consist of a friction material supported by a backing plate. Pressurized brake fluid hydraulically actuates the brake cylinder, causing brake pads to bear against the rotor on opposite sides. The pressure applied determines the contacting pressure, friction, and torque, as will be shown below.



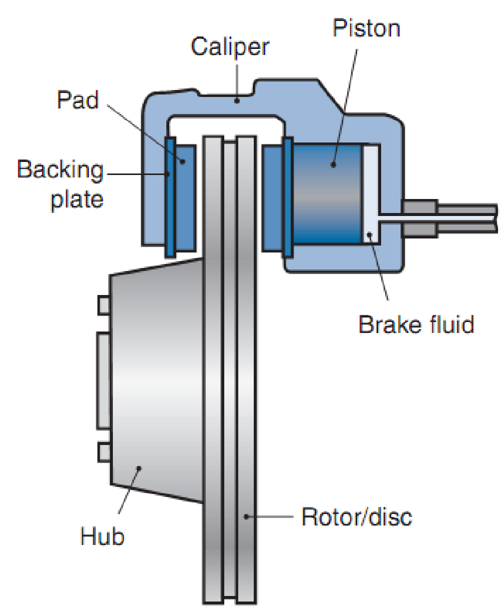


Fig. 2. Thrust Brake

Introduction

Table 1. Typical applications of clutches and brakes.

Type	Application notes
Thrust pad (disc)	Extremely common and versatile arrangement; can be wet or dry; wide variety of materials including carbon-carbon composites for aircraft brakes; preferred for front axles of vehicles because of superior convective cooling; cannot self-lock.
Cone	Higher pressure and torque for the same sized clutch compared to thrust pad due to wedging action of cone; common for lower speed applications with little sliding such as washing machines or extractors, or high-performance applications such as vehicle racing.
Block or short-shoe	Available in a wide variety of configurations and capacities; commonly applied to roller coasters, industrial equipment and positioning devices.
Long-shoe (drum)	Widely applied in vehicles on rear axles; self-locking promotes "parking brake" function; economical and reliable; limited heat dissipation capability.
Pivot-shoe	Used for low-torque applications in architecture, fishing equipement; higher torque applications include hoists and cranes; difficult to properly locate pivot.
Band brakes	Simple, compact, and rugged, widely applied to chain saws, go-karts, motorcycles, and some bicycles; susceptible to chatter or grabbing.
Slip clutches	Used to prevent excessive torque transfer to machinery; available in a wide variety of sizes and capacities; applied to machinery to prevent overload, some garage door operators, cranes as an anti-two blocking device; torque is difficult to control.

Introduction

The material used for lining of friction surfaces of a clutch should have the following characteristics:

- It should have a high and uniform coefficient of friction.
- It should not be affected by moisture and oil.
- It should have the ability to withstand high temperatures caused by slippage.
- It should have high heat conductivity.
- It should have high resistance to wear and scoring.

Introduction

Table 2: Representative properties of contacting materials operating dry, when rubbing against smooth cast iron or steel.

Friction material	Coefficient of friction, μ	Maximum contact pressure,[a] p_{max} kPa	Maximum bulk temperature, $t_{m,\,max}$ °C
Molded	0.25–0.45	1030–2070	204–260
Woven	0.25–0.45	345–690	204–260
Sintered metal	0.15–0.45	1030–2070	204–677
Cork	0.30–0.50	55–95	82
Wood	0.20–0.30	345–620	93
Cast iron; hard steel	0.15–0.25	690–1720	260

[a] Use of lower values will give longer life.

Introduction

Table 3: Coefficient of friction for contacting materials operating in oil when rubbing against steel or cast iron.

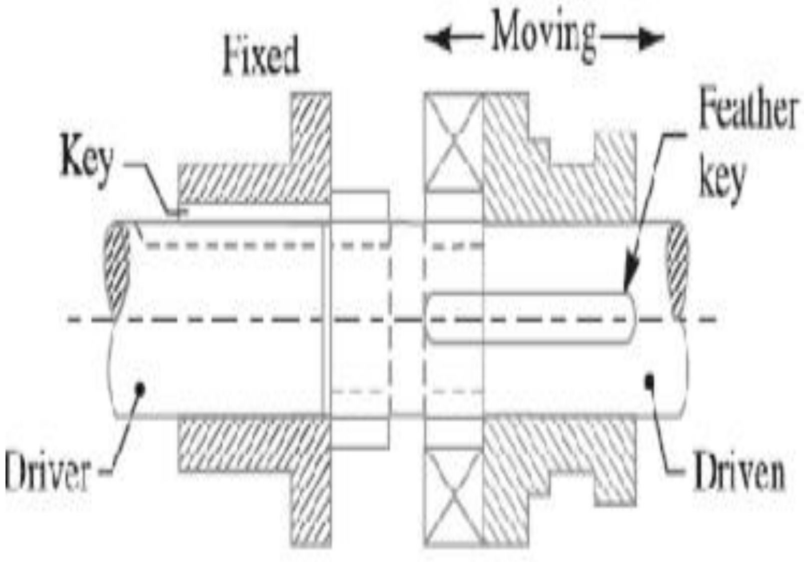
Friction material	Coefficient of friction, μ
Molded	0.06–0.09
Woven	0.08–0.10
Sintered metal	0.05–0.08
Paper	0.10–0.14
Graphitic	0.12 (avg.)
Polymeric	0.11 (avg.)
Cork	0.15–0.25
Wood	0.12–0.16
Cast iron; hard steel	0.03–0.16

Types of Clutches

The two main types of clutches commonly used in engineering practice are:

- Positive clutches.
- Friction clutches.

Positive Clutches



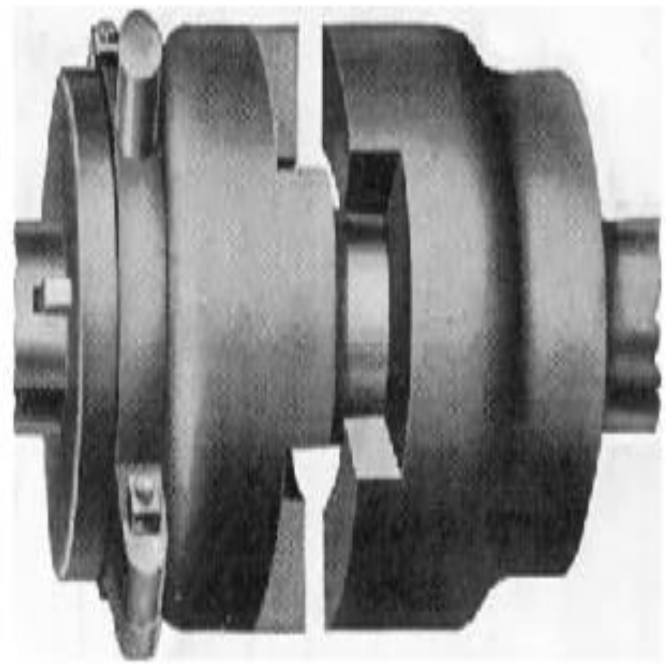


Fig. 3 Square Jaw Clutch

Positive Clutches

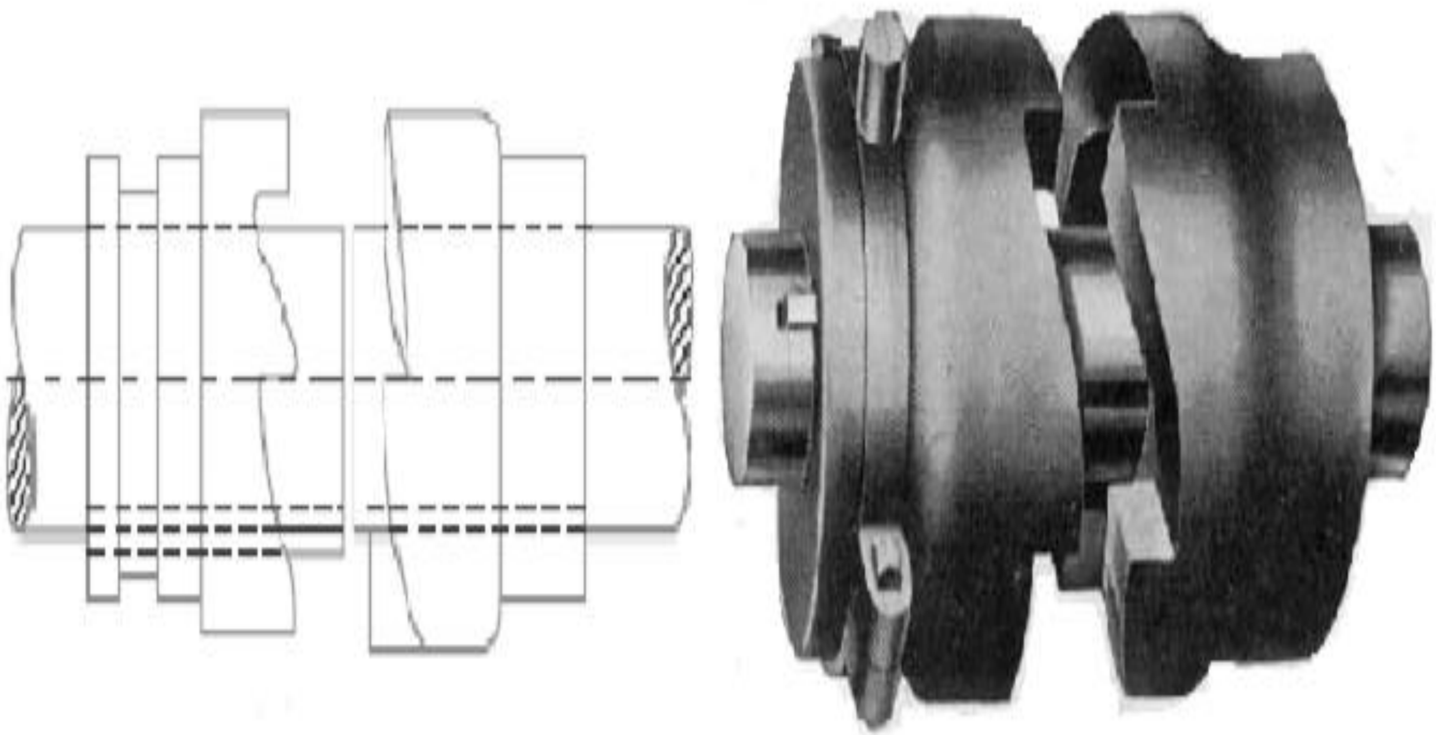


Fig. 4 Spiral Jaw Clutch

Positive Clutches

The positive clutches are used when a positive drive, no slip or power loss due to friction, is required. The simplest type of a positive clutch is a jaw clutch. The jaws of the clutch may be of square type as shown in Fig. 5 or of spiral type as shown in Fig. 6.

The disadvantage of jaw clutch is that it has limited speed and there are shocks at engagement. The use of jaw clutches are frequently applied to sprocket wheels, gears and pulleys. In such a case, the non-sliding part is made integral with the hub.

Friction Clutches

A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. Its application is also found in cases in which power is to be delivered to machines partially or fully loaded. The advantage of friction clutch is that there are no shocks at engagement and can work and high speeds. While, the disadvantage of it is that it has high slipping, power loss, and there is high heat generated.

The most important two types of friction clutch are:

a) Disc or Plate clutch, Fig. 5.

b) Cone clutch, Fig. 6.

Introduction

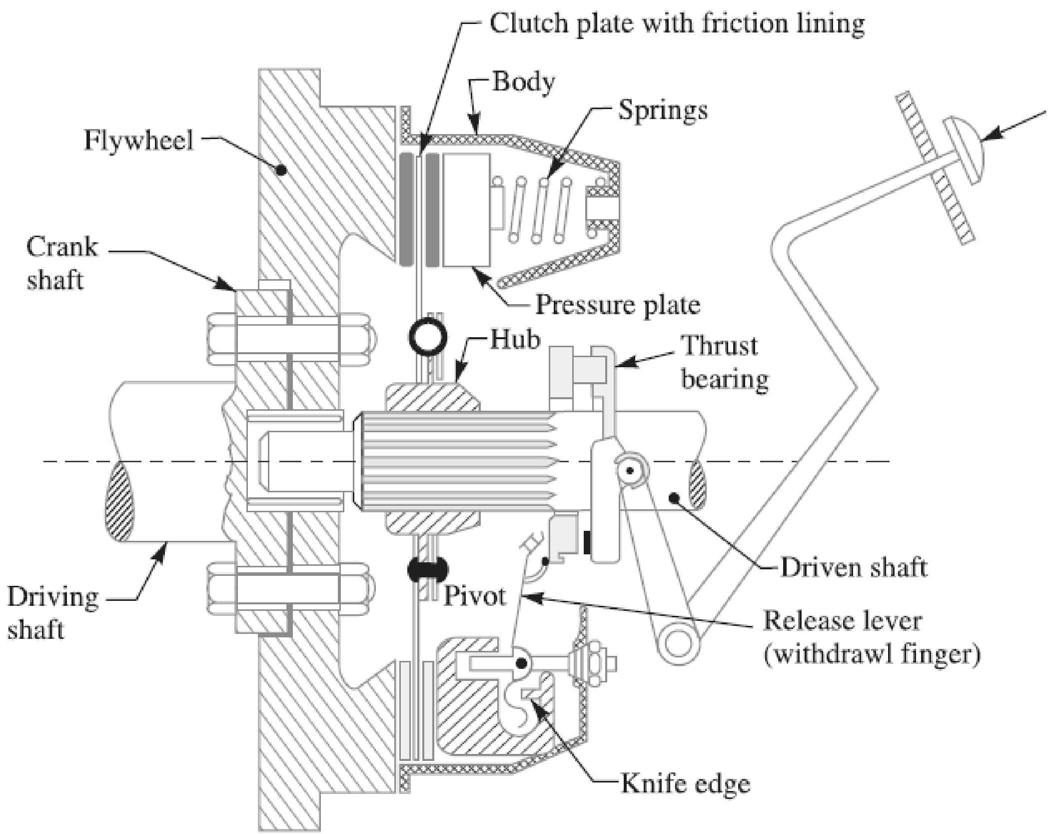


Fig. 5. Disk Clutch

Introduction

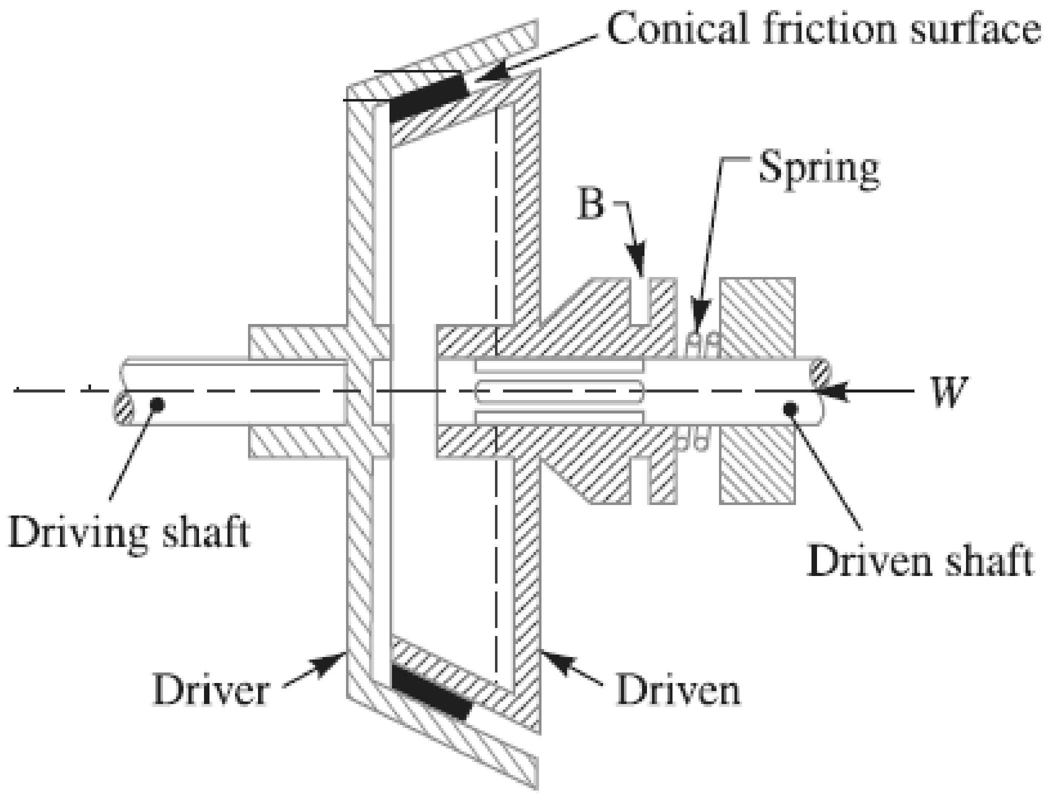


Fig. 6. Cone Clutch.

DESIGN OF A DISC CLUTCH

The objective is to obtain the axial force F necessary to produce a friction force to transmit a torque T. Fig. 7 shows a friction disk having an outside radius r_o and an inside radius r_i.

Consider an elementary ring of radius r and thickness dr as shown in Fig. 7.

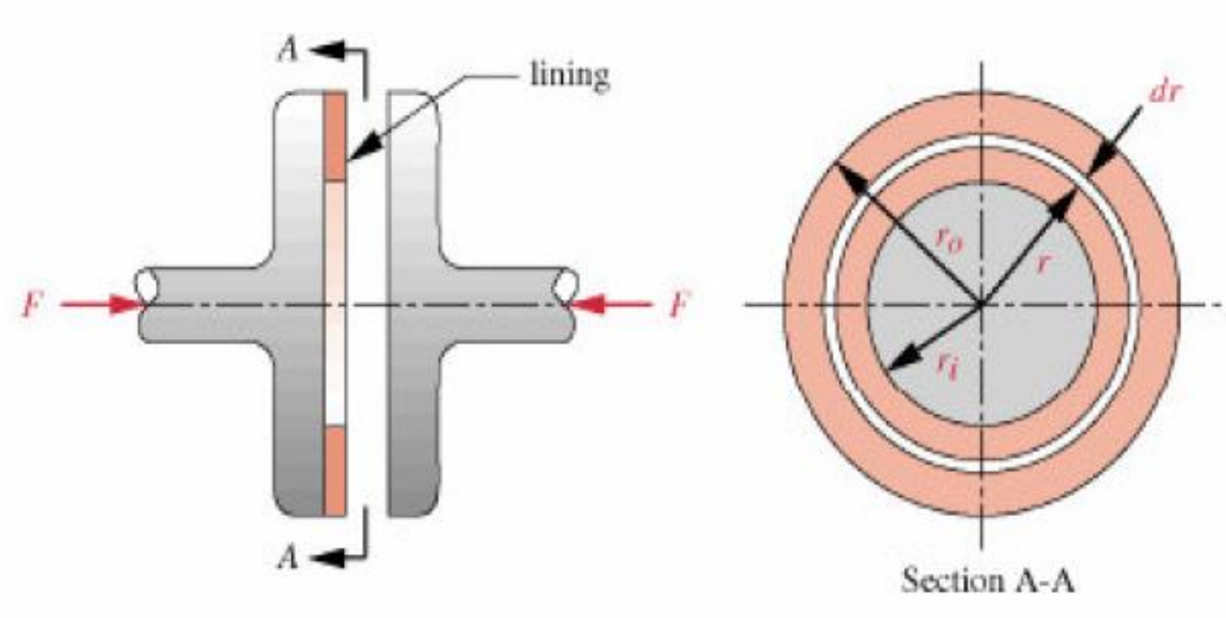


Fig. 7. Single Disk Clutch

DESIGN OF A DISC CLUTCH

The area of the contact surface or friction surface is

$$\mathrm{dA} = 2\pi \mathrm{r}\,\mathrm{dr} \qquad (1)$$

$\therefore$ Normal or axial force on the ring, with a contact pressure P, is

$$\mathrm{dF} = \mathrm{P} * \mathrm{dA} = \mathrm{P} * 2\pi \mathrm{r}\,\mathrm{dr} \qquad (2)$$

$\therefore$ Total force acing on the friction surface,

$$\mathrm{F} = \int_{\mathrm{r_i}}^{\mathrm{r_o}} \mathrm{P} * 2\pi \mathrm{r}\,\mathrm{dr} \qquad (3)$$

and the frictional force on the ring acting tangentially at radius r, with a coefficient of friction f,

$$\mathrm{fdF} = \mathrm{P} * 2\pi \mathrm{fr}\,\mathrm{dr} \qquad (4)$$

$\therefore$ Frictional torque acting on the ring is

$$\mathrm{dT} = \mathrm{fdF} * \mathrm{r} = \mathrm{P} * 2\pi \mathrm{fr}^2\,\mathrm{dr} \qquad (5)$$

$\therefore$ Total Frictional torque acing on the friction surface,

$$\mathrm{T} = \int_{\mathrm{r_i}}^{\mathrm{r_o}} \mathrm{P} * 2\pi \mathrm{fr}^2\,\mathrm{dr} \qquad (6)$$

DESIGN OF A DISC CLUTCH

The normal force can be supplied mechanically, hydraulically, pneumatically or electromagnetically. The pressure between the contacting surfaces may approach a uniform distribution over the surface if the disks are flexible enough. In such cases, the Wear will be greater at larger diameters because wear is proportional to the pressure times the peripheral velocity (PV) where the velocity increases linearly with radius. As the wear increases towards the outer diameters of the disks, the loss of material will change the pressure distribution to a non uniform one, and the disks will approach a uniform wear condition where PV = constant. A new clutch may be close to the uniform-pressure condition and approaches the uniform wear conditions with use.

For Uniform Pressure

When the pressure is uniformly distributed over the entire area of the friction face, i.e. P=Const., the axial force F, from Eq. 3, will be

$$\mathbf{F = 2\pi P \int_{r_i}^{r_o} r\, dr = \pi P\left(r_o^2 - r_i^2\right)} \qquad (7)$$

and the frictional torque T, from Eq. 6, is

$$\mathbf{T = 2\pi Pf \int_{r_i}^{r_o} r^2\, dr = \frac{2\pi Pf}{3}\left(r_o^3 - r_i^3\right)} \qquad (8)$$

For Uniform Wear

Since,

wear $\propto \mathrm{PV} \propto \mathrm{P\omega r}$ and a uniform wear is assumed

Then, $\mathrm{PV} = \mathrm{P\omega r}$ = Const. & ω is constant for the plate

$\therefore$ Pr = Const.

Therefore, $\mathrm{Pr} = \mathrm{P_{max.}r_i} = \mathrm{P_{min.}r_o}$ (9)

and from Eq. 3, the normal force on the ring is

$$\mathbf{F = 2\pi P_{max.} r_i \int_{r_i}^{r_o} dr = 2\pi P_{max.} r_i\,(r_o - r_i)} \qquad (10)$$

The frictional torque T, from Eq. 6, is

$$\mathbf{T = 2\pi\ f\ P_{max.} r_i \int_{r_i}^{r_o} r dr = \pi\ f\ P_{max.} r_i \left(r_o^2 - r_i^2\right)} \qquad (11)$$

Uniform presuure

$$\mathbf{T = \frac{2\pi Pf}{3}\left(r_o^3 - r_i^3\right)}$$

Notes

- For a single disc or plate clutch, normally both sides of the disc are effective. Therefore a single disc clutch has two pairs of surfaces in contact, i.e the total torque being calculated from Eq. 8 or Eq. 11, must be doubled.
- The uniform pressure approach gives a higher friction torque than the uniform wear approach. Therefore in case of friction clutches, uniform wear should be considered, unless otherwise stated.

Notes

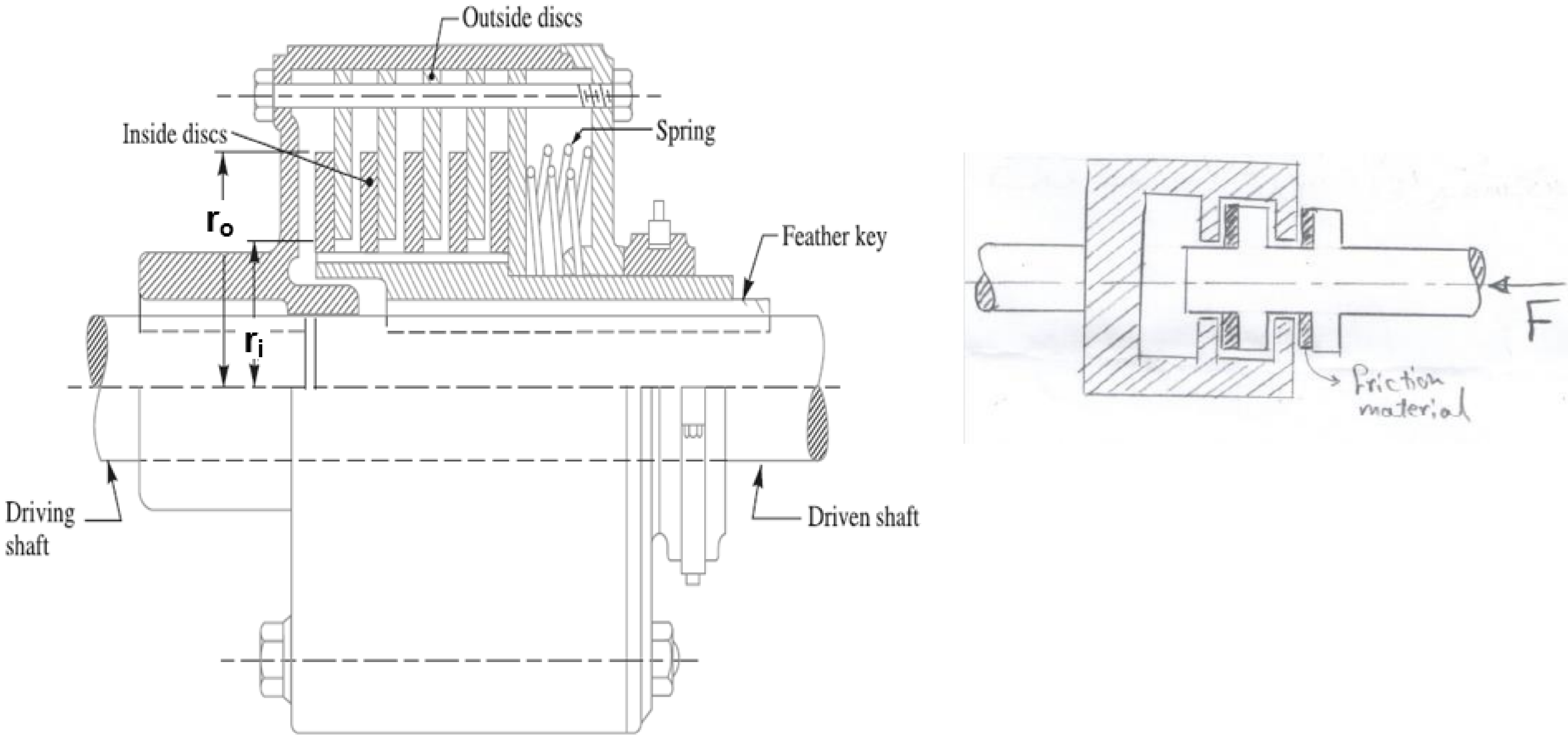


Fig. 8. Multiple Disk Clutch.

Notes

A multiple disc clutch, as shown in Fig. 8, may be used when a large torque is to be transmitted. It is extensively used in motor cars and machine tools. The total frictional torque acting on the friction surfaces may be obtained from,

$$\mathbf{T_{multiple}} = \mathbf{n} * \mathbf{T_{single}} \qquad (12)$$

Let n_1 = Number of discs on the driving shaft

n_2 = Number of discs on the driven shaft.

$\therefore$ Number of pairs of contact surfaces n,

$$n = n_1 + n_2 - 1 \qquad (13)$$

Clutch Example 1:

Plate clutch having a single driving plate with two contact surfaces on each side is required to transmit 110 kW at 1250 r.p.m. The outer diameter of the contact surfaces is to be 300 mm. The coefficient of friction is 0.4.

a. Assuming a uniform pressure of 0.17 N/mm^2; determine the inner diameter of the friction surfaces.
b. Assuming the same dimensions and the same total axial thrust, determine the maximum torque that can be transmitted and the maximum intensity of pressure when uniform wear conditions have been reached.

Solution:

Given:

Power = 110 kW ; N = 1250 r.p.m ; $r_o = \frac{300}{2} = 150\ mm$; f=0.4 ;

$P_{max.} = 0.17\ MPa$; n =2

(a) Since,

$$power = T * \omega = T * 2\pi N$$

$$\therefore T = \frac{power}{2\pi N} = \frac{60*110*10^3}{2\pi*1250} = 840\ N.m$$

From Eq. 8, the transmitted torque for uniform pressure clutch is

$$T = n * \frac{2\pi Pf}{3}\left(r_o^3 - r_i^3\right)$$

$$840 * 10^3 = 2 * \frac{2\pi * 0.17 * 0.4}{3}\left(150^3 - r_i^3\right)$$

$$\therefore\ r_i = 75\ mm \qquad \text{or, } d_i = 150\ mm$$

The total axial thrust for uniform pressure, from Eq. 7, is

$$F = \pi P\left(r_o^2 - r_i^2\right) = \pi * 0.17(150^2 - 75^2)$$

$$= 9011\ N$$

Solution:

(b) for uniform wear, the axial thrust , Eq. 10, is

$$F = 2\pi P_{max.} r_i \left(r_o - r_i\right)$$

$$\therefore 9011 = 2\pi P_{max.} * 75\left(150 - 75\right)$$

and, $P_{max.} = 0.255\ \mathrm{MPa}$

The Maximum torque for uniform wear, from Eq. 11, is

$$T = n * 2\pi\, f\, P_{max.} r_i\left(r_o^2 - r_i^2\right)$$

$$T\ = 2 * 2 * \pi * 0.4 *\ 0.255 * 75 * \left(150^2 - 75^2\right)$$

$$T = 811\ \mathrm{N.m}$$

Clutch Example 2:

A single plate clutch, effective on both sides, is required to transmit 25 kW at 3000 r.p.m. Determine the outer and inner diameters of frictional surface if the coefficient of friction is 0.255, ratio of diameters is 1.25 and the maximum pressure is not to exceed 0.1 N/mm^2. Also, determine the axial thrust to be provided by springs. Assume the theory of uniform wear.

Solution:

Given:

Power = 25 kW ; N = 3000 r.p.m ; $\frac{r_o}{r_i} = 1.25$; f=0.255 ; $P_{max.} = 0.1\ \mathrm{MPa}$; n =2

$$T = \frac{\mathrm{power}}{2\pi N} = \frac{60 * 25 * 10^3}{2\pi * 3000} = 79.6\ \mathrm{N.m}$$

$$T = n\pi\, f\, P_{max.} r_i \left(r_o^2 - r_i^2\right)$$

$$79600 = 2 * \pi * 0.255 * 0.1\ r_i \left((1.25 r_i)^2 - r_i^2\right)$$

$\therefore\ r_i = 96\ \mathrm{mm} \quad \& \quad r_o = 120\ \mathrm{mm}$

$$F = 2\pi P_{max.} r_i\ (r_o - r_i) = 2 * \pi * 0.1 * 96(120 - 96)$$

$$F = 1447\ \mathrm{N}$$

Clutch Example 3:

A multi-disc clutch has three discs on the driving shaft and two on the driven shaft. The inside diameter of the contact surface is 120 mm. The maximum pressure between the surface is limited to 0.1 N/mm^2. Design the clutch for transmitting 25 kW at 1575 r.p.m. Assume uniform wear condition and coefficient of friction as 0.3.

Solution:

Given:

Power = 25 kW ; N = 1575 r.p.m ; $r_i = 60 \text{ mm}$; f=0.3 ; $P_{max.} = 0.1 \text{ MPa}$; $n_1 = 3$; $n_2 = 2$

$$T = \frac{\text{power}}{2\pi N} = \frac{60 * 25 * 10^3}{2\pi * 1575} = 151.6 \text{ N.m}$$

The number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1 = 3 + 2 - 1 = 4$$

since,

$$T = n\pi \, f \, P_{max.} r_i \left(r_o^2 - r_i^2\right)$$

$$\therefore 151600 = 4 * \pi * 0.3 * 0.1 * 60(r_2^2 - 60^2)$$

$$\therefore \; r_i = 102 \text{ mm}$$

DESIGN OF A CONE CLUTCH

Consider a pair of friction surfaces of a cone clutch as shown in Fig. 9.

Let,

2α = Cone angle.

b = Width of the friction surfaces (Lining width or face width).

r_i = Inner radius of friction surface.

r_o = Outer radius of friction surface.

f = Coefficient of friction between the contact surfaces.

P = Pressure between the contact surfaces.

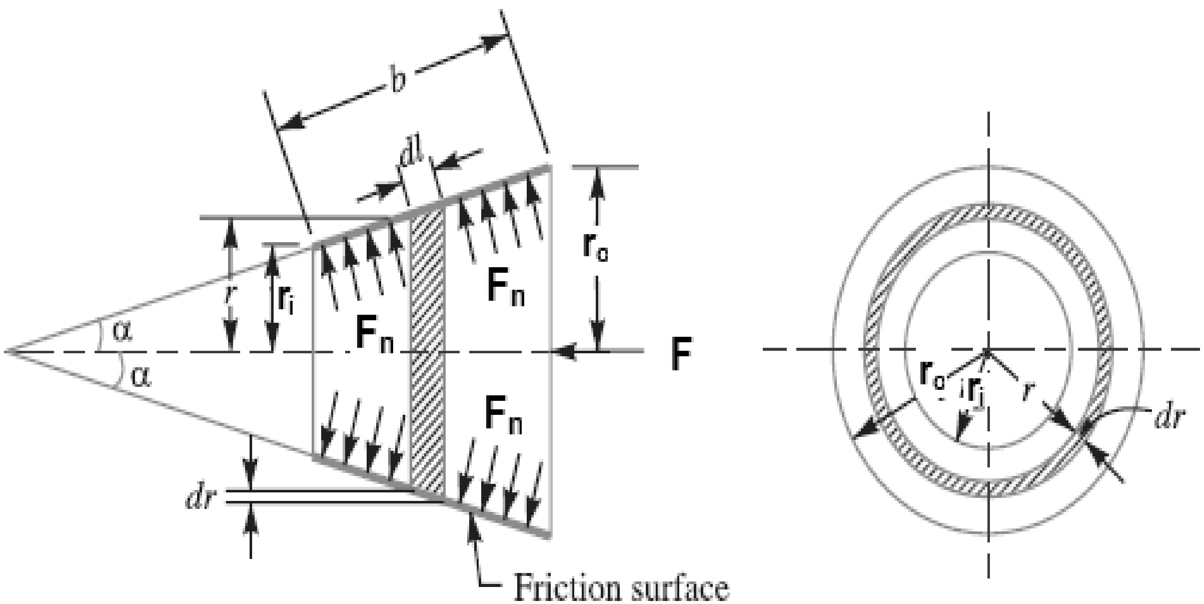


Fig. 9. Friction surfaces of cone clutch

DESIGN OF A DISC CLUTCH

Consider a small ring of radius r and thickness dr as shown in Fig. 9. Let dl is the length of ring of the friction surface, such that,

$$\mathrm{dl} = \frac{\mathrm{dr}}{\sin\alpha}$$

$$\therefore\ \mathrm{dA} = \frac{2\pi r \mathrm{dr}}{\sin\alpha} \qquad (14)$$

We know that the normal force acting on the ring,

$$\mathrm{dF_n} = \mathrm{P} * \mathrm{dA} = \mathrm{P} * \frac{2\pi r\mathrm{dr}}{\sin\alpha} \qquad (15)$$

Therefore, the axial force on the clutch is

$$\mathrm{dF} = \mathrm{dF_n}\ \sin\alpha = 2\pi \mathrm{Pr\, dr} \qquad (16)$$

This equation is similar to that for disc clutch. So, the axial force for cone clutch, for both uniform-pressure and uniform- wear approaches; can be obtained from Eq. 7 and Eq. 10, respectively.

$$\mathbf{F = 2\pi P \int_{r_i}^{r_o} r\, dr = \pi P\left(r_o^2 - r_i^2\right)} \qquad \text{for uniform-pressure}$$

$$\mathbf{F = 2\pi P_{max.} r_i \int_{r_i}^{r_o} dr = 2\pi P_{max.} r_i\, (r_o - r_i)} \qquad \text{for uniform-wear}$$

DESIGN OF A DISC CLUTCH

We know that frictional force on the ring acting tangentially at radius r,

$$\mathrm{fdF_n} = \mathrm{f\,P} * \frac{2\pi \mathrm{rdr}}{\sin\alpha} \tag{17}$$

$\therefore$ Frictional torque acting on the ring is

$$\mathrm{dT} = \mathrm{fdF} * \mathrm{r} = \mathrm{P} * \frac{2\pi \mathrm{fr}^2}{\sin\alpha}\,\mathrm{dr} \tag{18}$$

Comparing Eq. 18 with Eq. 5 for disc clutch,

$\therefore$ The total frictional torque T is

For uniform-pressure,

$$\mathbf{T} = \frac{\mathbf{2\pi Pf}}{\sin\alpha}\int_{\mathbf{r_i}}^{\mathbf{r_o}} \mathbf{r^2\,dr} = \frac{\mathbf{2\pi Pf}}{\mathbf{3}\sin\alpha}\left(\mathbf{r_o^3} - \mathbf{r_i^3}\right) \tag{19}$$

For uniform-wear,

$$\mathbf{T} = \frac{\mathbf{2\pi\ f\ P_{max.}r_i}}{\sin\alpha}\int_{\mathbf{r_i}}^{\mathbf{r_o}} \mathbf{rdr} = \frac{\mathbf{\pi\ f\ P_{max.}r_i}}{\sin\alpha}\left(\mathbf{r_o^2} - \mathbf{r_i^2}\right) \tag{20}$$

Clutch Example 4:

An engine developing 45 kW at 1000 r.p.m. is fitted with a cone clutch built inside the flywheel. The cone has an angle of 25° and an outside diameter of 400 mm. The coefficient of friction is 0.2. The normal pressure on the clutch face is not to exceed 0.1 N/mm^2. Determine:

The face width required

The axial spring force necessary to engage the clutch.

Solution:

Given:

Power = 45 kW ; N = 1000 r.p.m ; $r_i = 200$ mm ; f=0.2 ;

$P_{max.} = 0.1$ MPa ; 2α = 25°.

$$T = \frac{power}{2\pi N} = \frac{60 * 45 * 10^3}{2\pi * 1000} = 430 \text{ N.m}$$

Assume uniform-wear,

From Eq. 20

$$T = \frac{\pi \, f \, P_{max.} r_i}{\sin \alpha}\left(r_o^2 - r_i^2\right)$$

$$\therefore \; 430000 = \frac{\pi * .2 * 0.1 * 200}{\sin 12.5}\left(r_o^2 - 200^2\right)$$

$$\therefore \; r_o = 220 \text{ mm}$$

Solution:

Since,

$$\sin\alpha = \frac{r_o - r_i}{b}$$

$$\therefore \; \sin 12.5 = \frac{220 - 200}{b}$$

$$\therefore \; b = \; 90 \text{ mm}$$

$$F = 2\pi P_{max.} r_i \,(r_o - r_i) = \; 2\pi * 0.1 * 200(220 - 200)$$

$$\therefore \; F = \; 2515 \text{ N}$$