ACADEMIC YEAR 94-95, SEMESTER ONE

طراحي اجزاء ماشين I

محورها

SHAFTS

مقدمه

محوريا شافت:

عضوي چرخشي يا ثابت، ومعمولا با مقطع دايره اي براي انتقال توان يا حركت از دستگاه محرک، نظير يک موتور الکتريکی، به يک مصرف کننده

بر روی شافت معمولا چرخدنده، پولی و چرخ زنجيرنصب گرديده که انتقال حرکت چرخشی و توان را توسط دنده های درگير، تسمه وزنجير ميسر می سازند

Photo courtesy Emerson Power Transmission Corp.

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اصول طراحی شافت

- Material Selection
- Geometric Layout
- Stress and strength
  - Static strength
  - Fatigue strength
- Deflection and rigidity
  - Bending deflection
  - Torsional deflection
  - Slope at bearings and shaft-supported elements
  - Shear deflection due to transverse loading of short shafts
- Vibration due to natural frequency

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ملاحظاتی برجنس محورها

- Deflection primarily controlled by geometry, not material
- Stress controlled by geometry, not material
- Strength controlled by material property

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جنس محورها

- Shafts are commonly made from low carbon, CD or HR steel, such as ANSI 1020–1050 steels.
- Fatigue properties don't usually benefit much from high alloy content and heat treatment.
- Surface hardening usually only used when the shaft is being used as a bearing surface.

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جنس محورها

- Cold drawn steel typical for $d < 3$ in.
- HR steel common for larger sizes. Should be machined all over.
- Low production quantities
  - Lathe machining is typical
  - Minimum material removal may be design goal
- High production quantities
  - Forming or casting is common
  - Minimum material may be design goal

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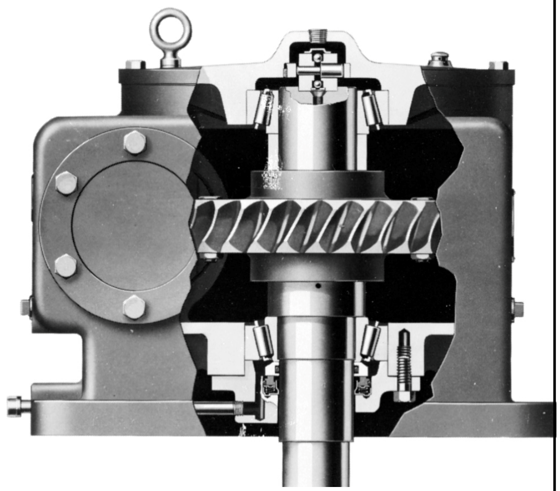
طرح شافت

- Issues to consider for shaft layout
  - Axial layout of components
  - Supporting axial loads
  - Providing for torque transmission
  - Assembly and Disassembly



یک گیربکس حلزونی

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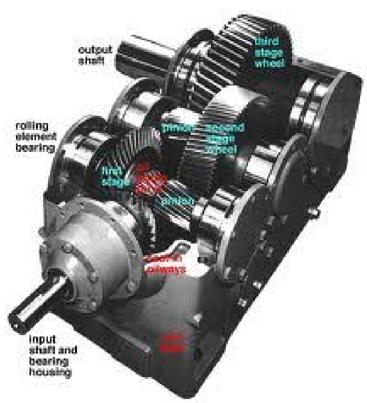
طرح محور در مجموعه های گیربکس



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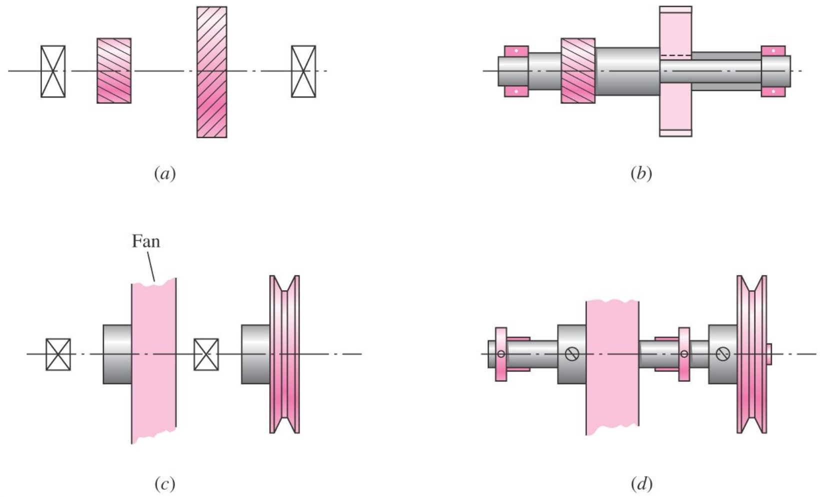
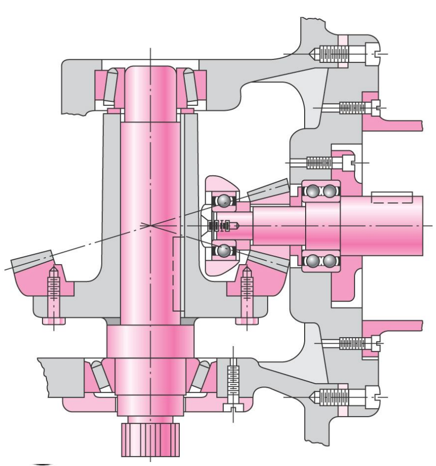
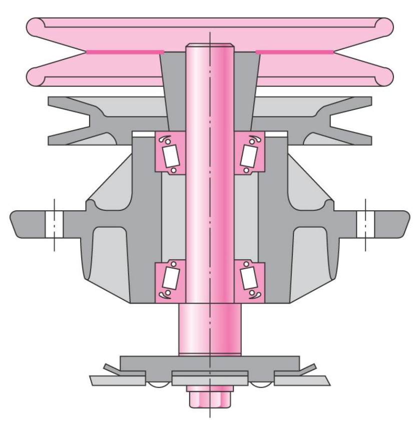
آرایش محوری اجزاء روی شافت



تحمل نیروهای محوری وارد برشافت

- Axial loads must be supported through a bearing to the frame.
- Generally best for only one bearing to carry axial load to shoulder
- Allows greater tolerances and prevents binding





روشهای انتقال گشتاوربه محور

- Common means of transferring torque to shaft
  - Keys
  - Splines
  - Setscrews
  - Pins
  - Press or shrink fits
  - Tapered fits
- Keys are one of the most effective
  - Slip fit of component onto shaft for easy assembly
  - Can design key to be weakest link to fail in case of overload

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روشهای متفاوت اتصال اجزا به شافت

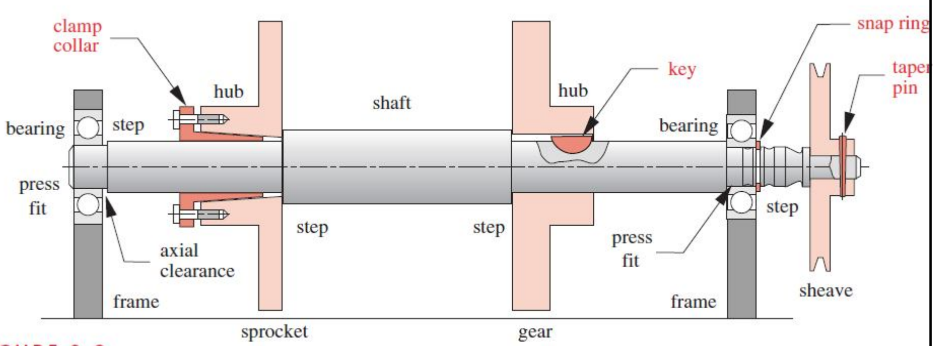

FIGURE 9-2
Various Methods to Attach Elements to Shafts

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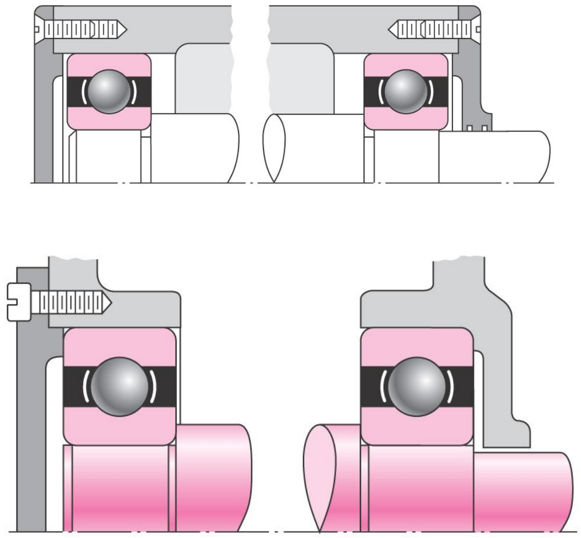
مونتاژ و دمونتاژ اجزای روی محور



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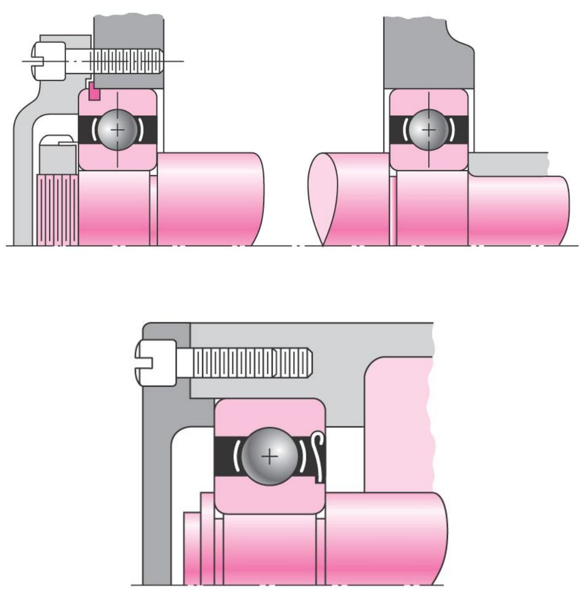
مونتاژ و دمونتاژ اجزای روی محور



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نکات مهم در طراحی محورها

- طراحی کوتاه محور تا حد امکان
- نزدیکی یاتاقانها به محل های اعمال بار(به منظور کاهش ممان خمشی و خیزشافت، افزایش سرعت بحرانی)
- دور نگه داشتن تنش افزاها از محل های با تنش بالا
- ایجاد قوس های ملایم در گوشه پله ها
- افزایش کیفیت سطح محور
- در نظر گرفتن عملیات احتمالی محلی سطح نظیر نورد سرد یا شات پینینگ
- استفاده از شافت توخالی در صورت مهم بودن شاخص وزن

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روند طراحی یک محور با بار ثابت

1. تعیین سرعت چرخشی شافت
2. تعین توان یا تورکی که توسط شافت انتقال می یابد
3. تعیین ابعاد اجزا انتقال قدرت ودیگر اجزایی که روی شافت سوار میشوند
4. تعیین موقعیت یاتاقانهای نگه دارنده شافت
5. ارائه طرح هندسی شافت با در نظر گرفتن اینکه هر جزء درکجا قرار میگیرد و انتقال قدرت چگونه منتقل میشود
6. تعیین اندازه تورک در طول شافت
7. تعیین نیروهای وارده بر شافت
8. ترسیم نمودارهای نیروی برشی و ممان خنثی
9. انتخاب جنس برای شافت و عملیات حرارتی احتمالی

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روند طراحی یک محور با بار ثابت

10. با در نظر گرفتن نوع بارگذاری (ملایم، شوکی، تکرارشونده، معکوس) تعیین تنش طراحی
11. تحلیل و آنالیز همه نقاط بحرانی روی شافت و تعیین حداقل قطرهای قابل قبول به منظور اطمینان از طراحی ایمن
12. تعیین ابعادنهایی شافت

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ملاحظات خیز شافت

- Deflection analysis at a single point of interest requires complete geometry information for the entire shaft.
- For this reason, a common approach is to size critical locations for stress, then fill in reasonable size estimates for other locations, then perform deflection analysis.
- Deflection of the shaft, both linear and angular, should be checked at gears and bearings.

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ملاحظات خیز شافت

- Allowable deflections at components will depend on the component manufacturer's specifications.

Table 7–2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	< 0.0005 rad
Transverse Deflections	
Spur gears with $P < 10$ teeth/in	0.010 in
Spur gears with $11 < P < 19$	0.005 in
Spur gears with $20 < P < 50$	0.003 in

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ملاحظات خیز شافت

- Deflection analysis is straightforward, but lengthy and tedious to carry out manually.
- Each point of interest requires entirely new deflection analysis.
- Consequently, shaft deflection analysis is almost always done with the assistance of software.
- Options include specialized shaft software, general beam deflection software, and finite element analysis software.

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Example 7-3

This example problem is part of a larger case study. See Chap. 18 for the full context.

In Ex. 7–2, a preliminary shaft geometry was obtained on the basis of design for stress. The resulting shaft is shown in Fig. 7–10, with proposed diameters of

$$D_1 = D_7 = 1 \text{ in}$$
$$D_2 = D_6 = 1.4 \text{ in}$$
$$D_3 = D_5 = 1.625 \text{ in}$$
$$D_4 = 2.0 \text{ in}$$

Check that the deflections and slopes at the gears and bearings are acceptable. If necessary, propose changes in the geometry to resolve any problems.

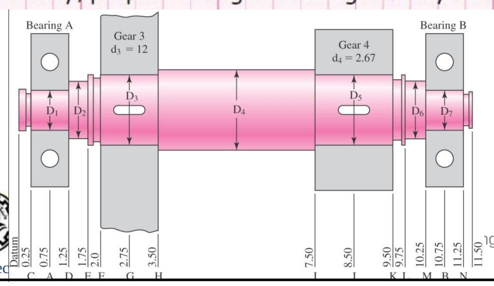

Fig. 7-10

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Example 7-3

Solution

A simple planar beam analysis program will be used. By modeling the shaft twice, with loads in two orthogonal planes, and combining the results, the shaft deflections can readily be obtained. For both planes, the material is selected (steel with $E = 30$ Mpsi), the shaft lengths and diameters are entered, and the bearing locations are specified. Local details like grooves and keyways are ignored, as they will have insignificant effect on the deflections. Then the tangential gear forces are entered in the horizontal *xz* plane model, and the radial gear forces are entered in the vertical *xy* plane model. The software can calculate the bearing reaction forces, and numerically integrate to generate plots for shear, moment, slope, and deflection, as shown in Fig. 7–11.

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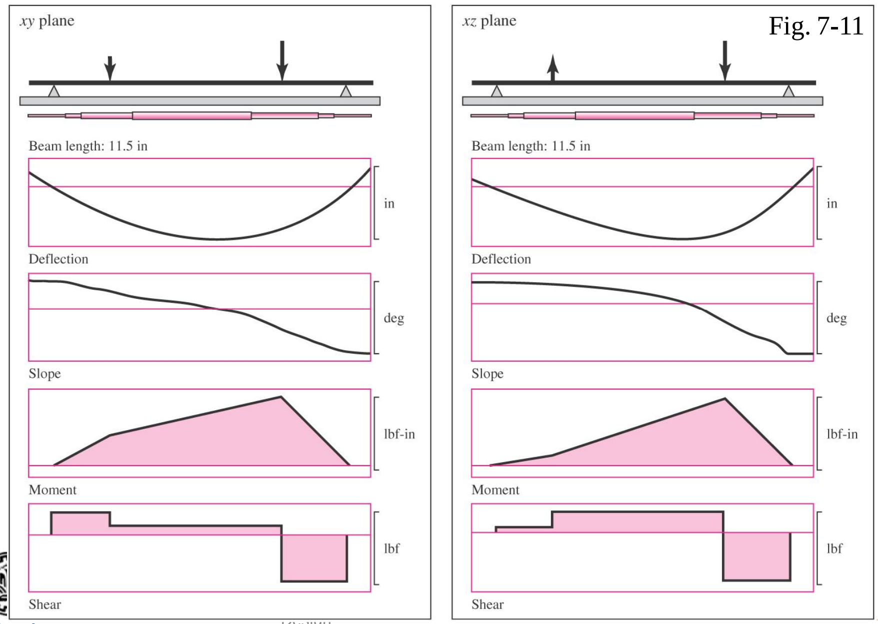
Example 7-3



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Example 7-3

The deflections and slopes at points of interest are obtained from the plots, and combined with orthogonal vector addition, that is, $\delta = \sqrt{\delta_{xz}^2 + \delta_{xy}^2}$. Results are shown in Table 7–3.

Point of Interest	*xz* Plane	*xy* Plane	Total
Left bearing slope	0.02263 deg	0.01770 deg	0.02872 deg 0.000501 rad
Right bearing slope	0.05711 deg	0.02599 deg	0.06274 deg 0.001095 rad
Left gear slope	0.02067 deg	0.01162 deg	0.02371 deg 0.000414 rad
Right gear slope	0.02155 deg	0.01149 deg	0.02442 deg 0.000426 rad
Left gear deflection	0.0007568 in	0.0005153 in	0.0009155 in
Right gear deflection	0.0015870 in	0.0007535 in	0.0017567 in

Table 7–3

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Example 7-3

Whether these values are acceptable will depend on the specific bearings and gears selected, as well as the level of performance expected. According to the guidelines in Table 7-2, all of the bearing slopes are well below typical limits for ball bearings. The right bearing slope is within the typical range for cylindrical bearings. Since the load on the right bearing is relatively high, a cylindrical bearing might be used. This constraint should be checked against the specific bearing specifications once the bearing is selected.

The gear slopes and deflections more than satisfy the limits recommended in Table 7-2. It is recommended to proceed with the design, with an awareness that changes that reduce rigidity should warrant another deflection check.

سرعت بحرانی محور

- A shaft with mass has a critical speed at which its deflections become unstable.
- Components attached to the shaft have an even lower critical speed than the shaft.
- Designers should ensure that the lowest critical speed is at least twice the operating speed.

سرعت بحرانی محور

- For a simply supported shaft of uniform diameter, the first critical speed is

$$\omega_1 = \left(\frac{\pi}{l}\right)^2 \sqrt{\frac{EI}{m}} = \left(\frac{\pi}{l}\right)^2 \sqrt{\frac{gEI}{A\gamma}} \qquad (7\text{–}22)$$

- For an ensemble of attachments, Rayleigh's method for lumped masses gives

$$\omega_1 = \sqrt{\frac{g \sum w_i y_i}{\sum w_i y_i^2}} \qquad (7\text{–}23)$$

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اجزاء سوارشونده روی محور- چرخدندها

Figure 13–1

Spur gears are used to transmit rotary motion between parallel shafts.

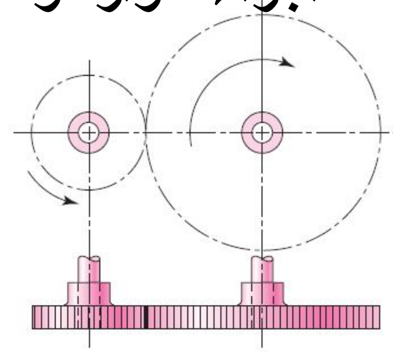
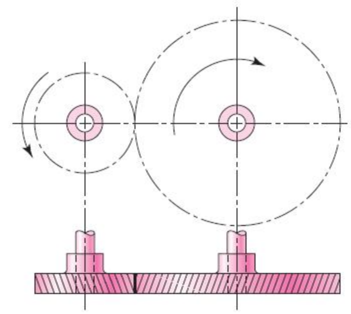


Figure 13–2

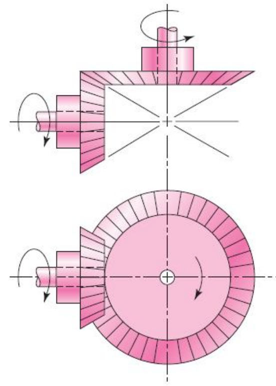
Helical gears are used to ...smit motion between ...allel or nonparallel shafts.



اجزاء سوارشونده روی محور- چرخدندها

Figure 13–3

Bevel gears are used to transmit rotary motion between intersecting shafts.



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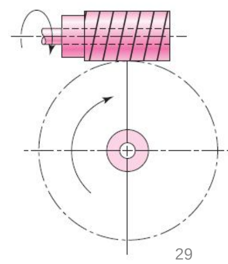
are used to transmit rotary motion between nonparallel and nonintersecting shafts.



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نامگذاري چرخدنده ساده

Figure 13–5

Nomenclature of spur-gear teeth.

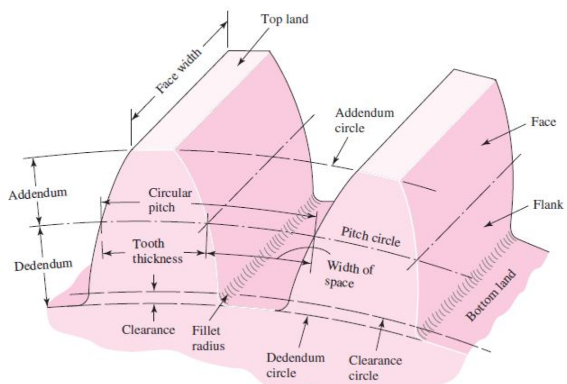

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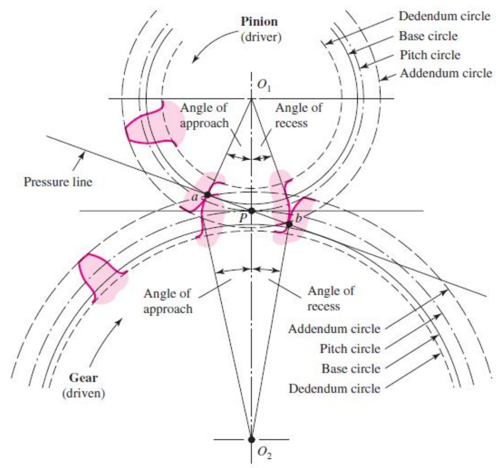
Figure 13–12

Tooth action.



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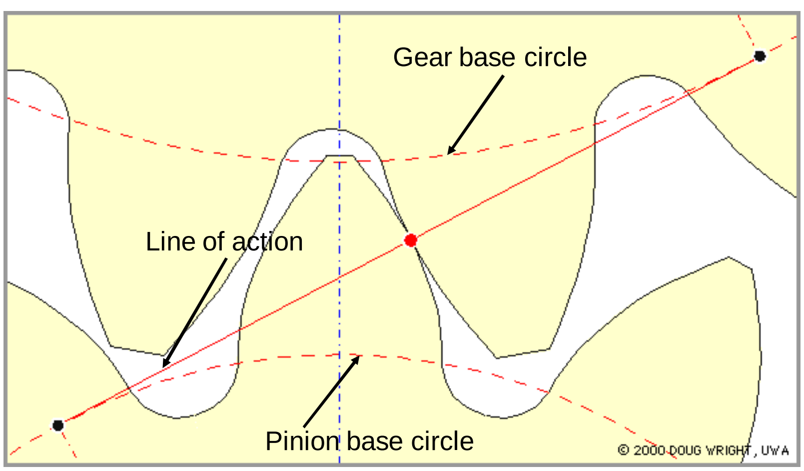

http://www.mech.uwa.edu.au/DANotes/gears/meshing/meshing.html

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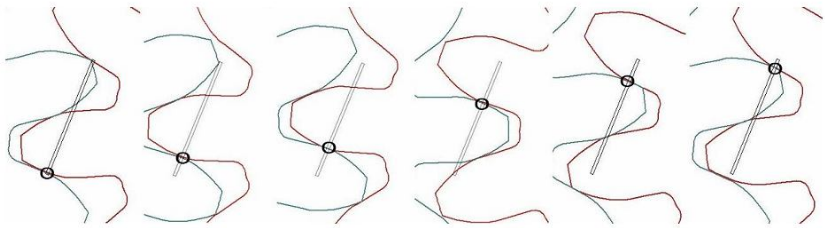
تماس دو چرخدنده ساده



نقطه تماس همواره بر روی خط عمل قرار دارد.

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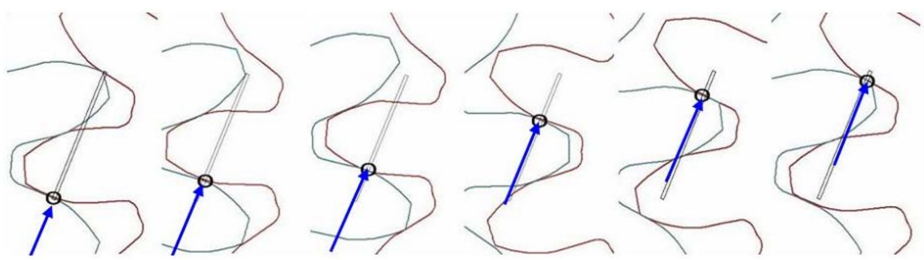
جهت نیرو در تماس بین دو چرخدنده ساده



راستای اعمال نیرو در جهت خط عمل قرار دارد.

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Figure 13–28

A two stage compound gear train.

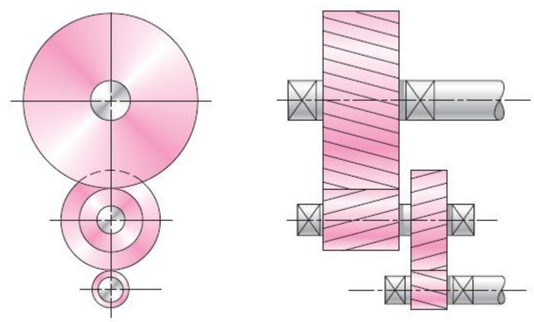
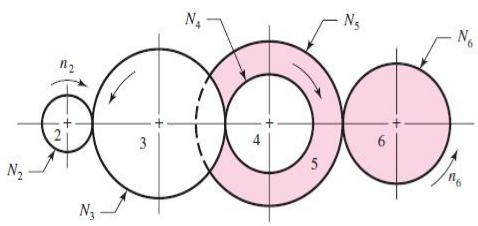


Figure 13–27

A gear train.



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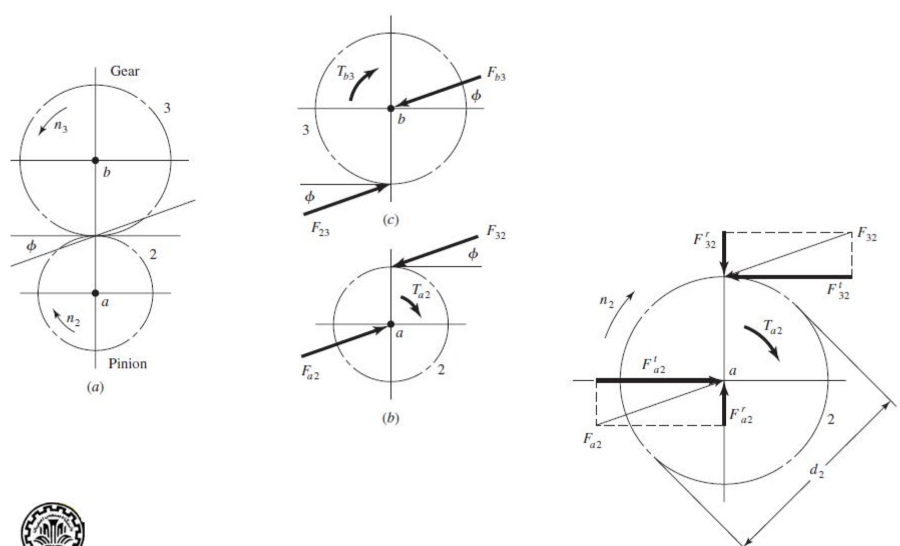
تحلیل نیروها در چرخدنده ساده



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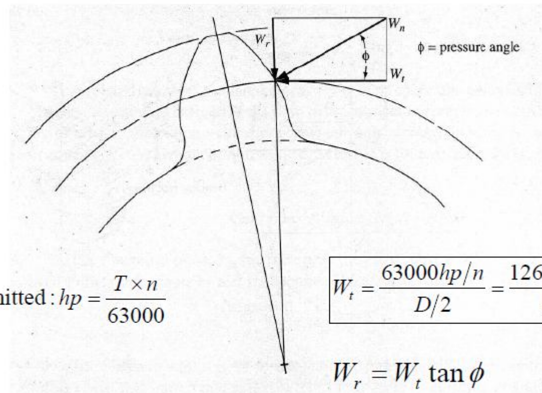
نیروهای وارده بر چرخدنده ساده



$W_t = \frac{T}{D/2}$

Power transmitted : $hp = \frac{T \times n}{63000}$

$T : lb \cdot in$

$n : rpm$

$$W_t = \frac{63000hp/n}{D/2} = \frac{126000hp}{nD} \text{ (lb)}$$

$$W_r = W_t \tan\phi$$

$$W_n = W_t / \cos\phi$$

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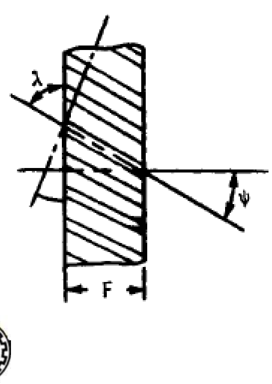
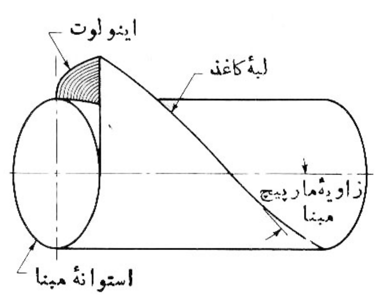
چرخدنده مارپیچ موازي

يك جفت چرخدنده خارجي: زاويه مارپيچ يكسان ولي در خلاف جهت يكد يگر

$$\Sigma = \Psi_1 \pm \Psi_2 \; ; \quad \Sigma = 0 \rightarrow \Psi_1 - \Psi_2 = 0 \rightarrow \Psi_1 = \Psi_2$$

شكل دندانه: مارپيچ وار اينولوت





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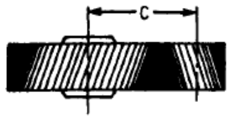
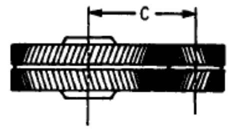
انواع چرخدنده هاي مارپيچ موازي



(a)



(b)

(c)

(a) Single-helix gear

(b) Double-helix gear

(c) Types of double-helix:
Left: conventional
Centre: staggered
Right: continuous or herringbone

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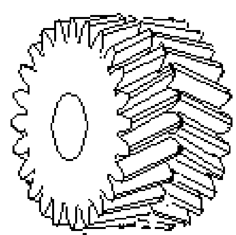
چرخدنده های جناقی

- To avoid axial thrust, two helical gears of opposite hand can be mounted side by side, to cancel resulting thrust forces



- Herringbone gears are mostly used on heavy machinery.

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چرخدنده مارپيچ موازي--مشخصه ها

نحوه تماس: خط - نقطه – خط:

- درگیری تدریجی دندانه ها
- انتقال آرام بار
- توانایی انتقال بارهای سنگین در سرعتهای بالا
- اهمیت کم نسبت تماس

زاویه مارپیچ: ایجاد مولفه نیرویی محوری (علاوه بر شعاعی)

با افزایش ψ ایجاد سروصدا کاهش می یابد (انتقال قدرت یکنواخت تر)

با افزایش ψ نیروی محوری افزایش می یابد

عموما 15, 23, 30و 45°

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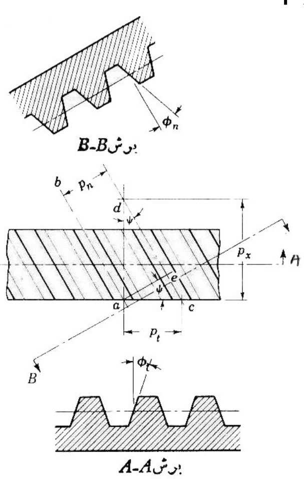
نامگذاری چرخدنده های مارپیچ

گام دایره ای:

- گام دایره ای عرضی (گام دایره ای) - Pt
- گام دایره ای عمودی - Pn
- گام دایره ای عرضی (گام محوری) - Px

مدول:

- مدول عرضی (مدول)- m
- مدول عمودی - mn



$$P_n = P_t \cos\Psi$$

$$P_x = \frac{P_t}{\tan\Psi}$$

$$\cos\Psi = \frac{\tan\Phi_n}{\tan\Phi_t}$$

$$P_t = \pi \cdot m$$

$$P_n = \pi \cdot m_n$$

$$m_n = m\cos\Psi$$

شکل ۱۴-۳ نامگذاری چرخدنده‌های مارپیچ.

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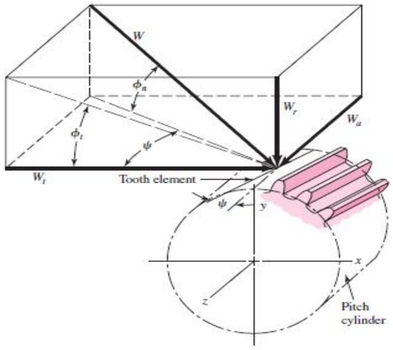
تحلیل نیروهای چرخدنده های مارپیچ



$$W_r = W \sin \phi_n$$
$$W_t = W \cos \phi_n \cos \psi$$
$$W_a = W \cos \phi_n \sin \psi$$

otal force

adial component

W_t = tangential component, also called transmitted load

W_a = axial component, also called thrust load

$$W_r = W_t \tan \Phi_t \qquad W_a = W_t \tan \Psi \qquad W = \frac{W_t}{\cos \Phi_n \cos \Psi}$$

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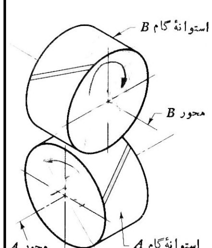
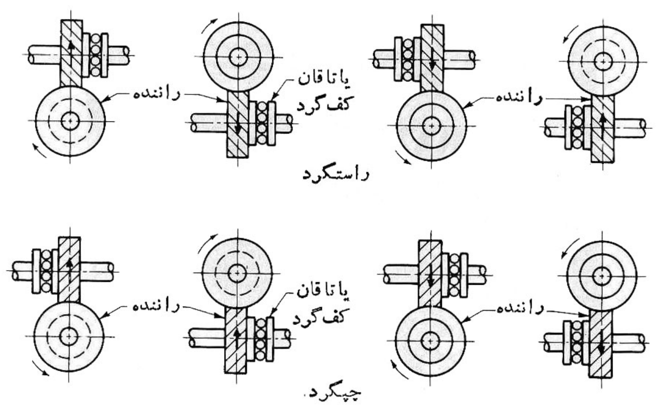
چرخدنده مارپیچ ضربدری





شکل۱۴-۱۱ بار محوری، چرخش، و رابطهٔ چپگردی یا راستگردی چرخدنده‌های مارپیچ ضربدری.

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چرخنده حلزونی

شکل۱۴-۱۲ حلزون و چرخندهٔ حلزونی با پوش یک‌جانبه.

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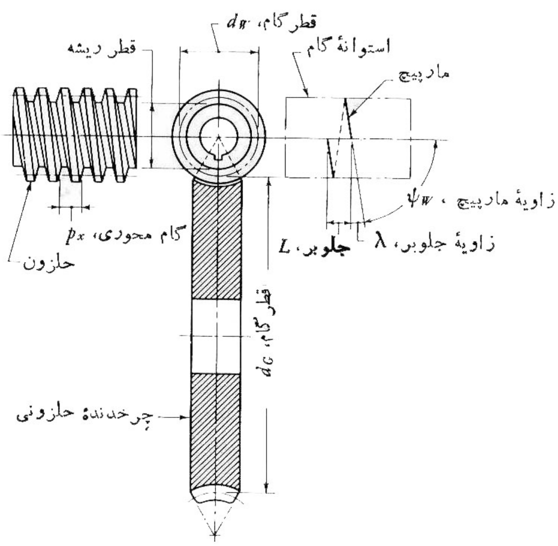
WORM AND WORM GEAR



شکل۱۴-۱۳ نامگذاری مجموعهٔ چرخندهٔ حلزونی با پوش یک‌جانبه.

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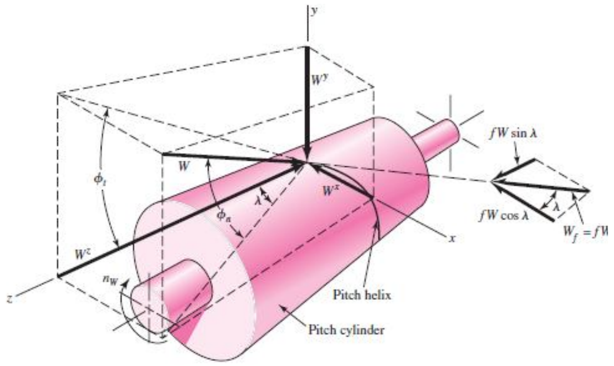
آنالیز نیرویی حلزون و چرخ حلزون



$$W^x = W \cos \phi_n \sin \lambda$$
$$W^y = W \sin \phi_n$$
$$W^z = W \cos \phi_n \cos \lambda$$

$$W_{Wt} = -W_{Ga} = W^x$$
$$W_{Wr} = -W_{Gr} = W^y$$
$$W_{Wa} = -W_{Gt} = W^z$$

$$W^x = W(\cos \phi_n \sin \lambda + f \cos \lambda)$$
$$W^y = W \sin \phi_n$$
$$W^z = W(\cos \phi_n \cos \lambda - f \sin \lambda)$$

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Bevel gears

- **Bevel gears** are useful when the direction of a shaft's rotation needs to be changed
- They are usually mounted on shafts that are 90 degrees apart, but can be designed to work at other angles as well
- The teeth on bevel gears can be **straight**, **spiral** or **hypoid**
- locomotives, marine applications, automobiles, printing presses, cooling towers, power plants, steel plants, railway track inspection machines, etc.

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Straight and Spiral Bevel Gears

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Bevel Gears Types

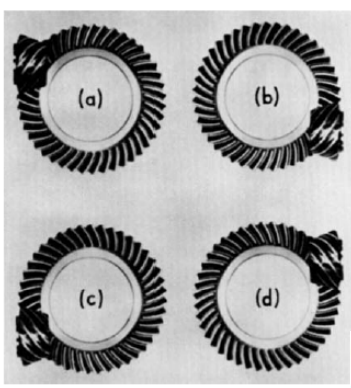
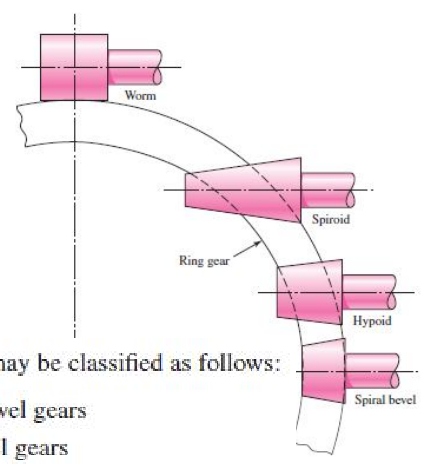


FIGURE 34.15 Hypoid offset. To determine the direction o offset, always look at the gear with the pinion at the right. Thu the gear sets of (*a*) and (*b*) are both offset *below* center; simila reasoning shows that (*c*) and (*d*) are offset *above* center. (*Glea son Machine Division.*)



Bevel gears may be classified as follows:

- Straight bevel gears
- Spiral bevel gears
- Zerol bevel gears
- Hypoid gears
- Spiroid gears

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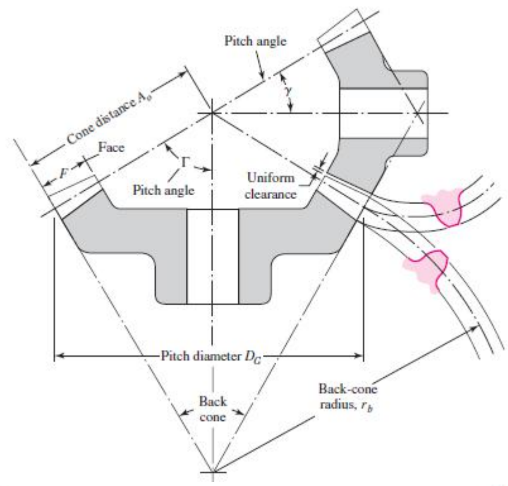
نامگذاری چرخدنده مخروطی



$$\tan \gamma = \frac{N_P}{N_G} \qquad \tan \Gamma = \frac{N_G}{N_P}$$

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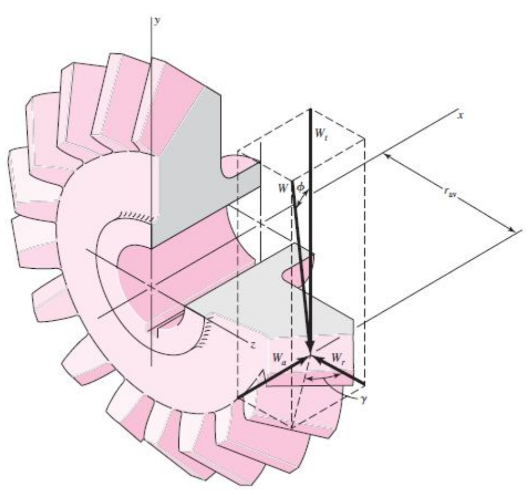
آنالیز نیرویی چرخدنده مخروطی ساده



$$W_t = \frac{T}{r_{av}}$$

$$W_r = W_t \tan \phi \cos \gamma$$

$$W_a = W_t \tan \phi \sin \gamma$$

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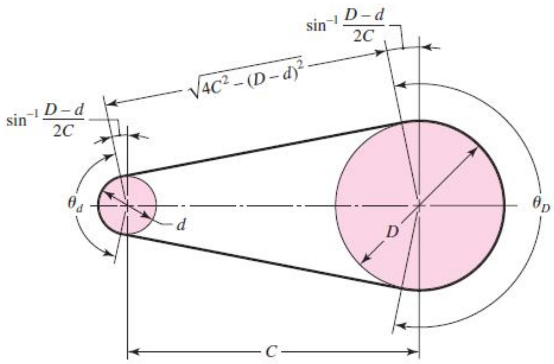
تسمه و پولی



$$\theta_d = \pi - 2\sin^{-1}\frac{D-d}{2C}$$

$$\theta_D = \pi + 2\sin^{-1}\frac{D-d}{2C}$$

$$L = \sqrt{4C^2-(D-d)^2} + \tfrac{1}{2}(D\theta_D + d\theta_d)$$

(*a*)

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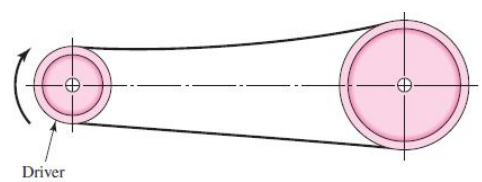
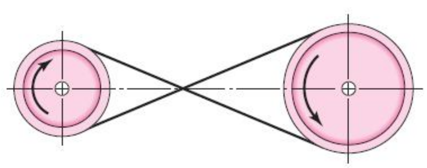
آرایش تسمه



(*a*)



(*b*)

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Setscrews

- Setscrews resist axial and rotational motion
- They apply a compressive force to create friction
- The tip of the set screw may also provide a slight penetration
- Various tips are available

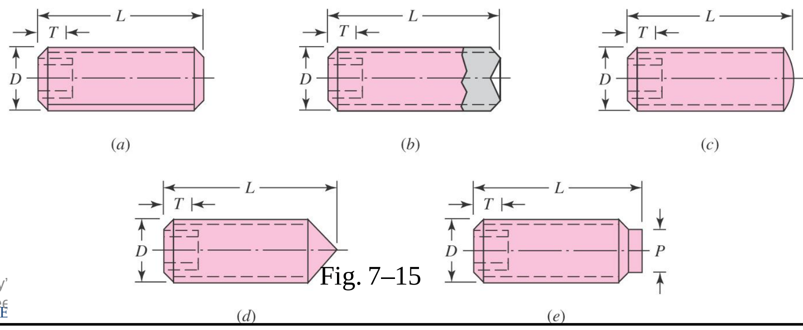

Fig. 7–15

Dep. of Mech. E

Setscrews

Table 7–4

Typical Holding Power (Force) for Socket Setscrews*

Source: Unbrako Division, SPS Technologies, Jenkintown, Pa.

Size, in	Seating Torque, lbf · in	Holding Power, lbf
#0	1.0	50
#1	1.8	65
#2	1.8	85
#3	5	120
#4	5	160
#5	10	200
#6	10	250
#8	20	385
#10	36	540
$\frac{1}{4}$	87	1000
$\frac{5}{16}$	165	1500
$\frac{3}{8}$	290	2000
$\frac{7}{16}$	430	2500
$\frac{1}{2}$	620	3000
$\frac{9}{16}$	620	3500
$\frac{5}{8}$	1325	4000
$\frac{3}{4}$	2400	5000
$\frac{7}{8}$	5200	6000
1	7200	7000

- Resistance to axial motion of collar or hub relative to shaft is called *holding power*
- Typical values listed in Table 7–4 apply to axial and torsional resistance
- Typical factors of safety are 1.5 to 2.0 for static, and 4 to 8 for dynamic loads
- Length should be about half the shaft diameter

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Keys and Pins

- Used to secure rotating elements and to transmit torque

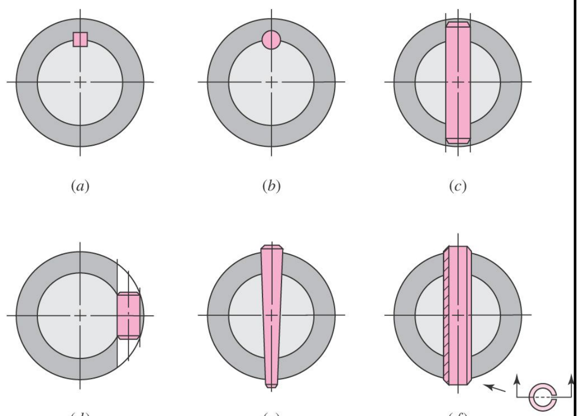

Fig. 7–16

Tapered Pins

- Taper pins are sized by diameter at large end
- Small end diameter is

$$d = D - 0.0208L \qquad (7\text{–}35)$$

- Table 7–5 shows some standard sizes in inches

Size	Commercial Maximum	Commercial Minimum	Precision Maximum	Precision Minimum
4/0	0.1103	0.1083	0.1100	0.1090
2/0	0.1423	0.1403	0.1420	0.1410
0	0.1573	0.1553	0.1570	0.1560
2	0.1943	0.1923	0.1940	0.1930
4	0.2513	0.2493	0.2510	0.2500
6	0.3423	0.3403	0.3420	0.3410
8	0.4933	0.4913	0.4930	0.4920

Table 7–5

Keys

- Keys come in standard square and rectangular sizes
- Shaft diameter determines key size

Shaft Diameter		Key Size		
Over	To (Incl.)	w	h	Keyway Depth
$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{64}$
$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{3}{64}$
		$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{16}$
		$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$
		$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$
$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{32}$
$1\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$
		$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$
$2\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{32}$
		$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{16}$
$2\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$
		$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$

ey's Mechanical Engineering Design

Table 7–6

Dep. of Mech. Eng.

Keys

- Failure of keys is by either direct shear or bearing stress
- Key length is designed to provide desired factor of safety
- Factor of safety should not be excessive, so the inexpensive key is the weak link
- Key length is limited to hub length
- Key length should not exceed 1.5 times shaft diameter to avoid problems from twisting
- Multiple keys may be used to carry greater torque, typically oriented 90º from one another
- Stock key material is typically low carbon cold-rolled steel, with dimensions slightly under the nominal dimensions to easily fit end-milled keyway

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Gib-head Key

- Gib-head key is tapered so that when firmly driven it prevents axial motion
- Head makes removal easy
- Projection of head may be hazardous

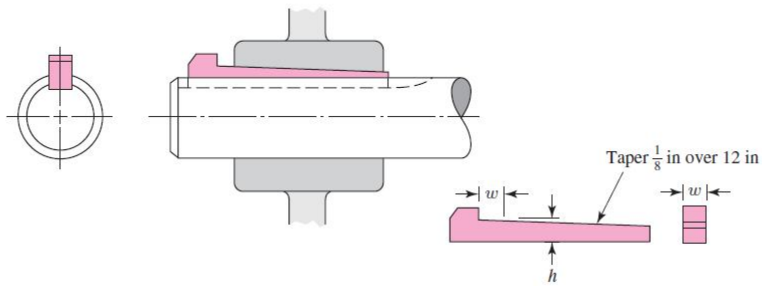

Fig. 7–17 (a)

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Woodruff Key

- Woodruff keys have deeper penetration
- Useful for smaller shafts to prevent key from rolling
- When used near a shoulder, the keyway stress con
tha

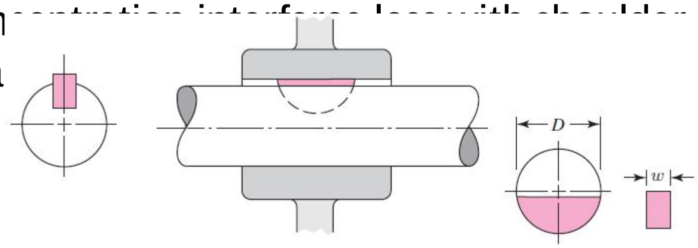

Fig. 7–17 (b)

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Retaining Rings

- Retaining rings are often used instead of a shoulder to provide axial positioning

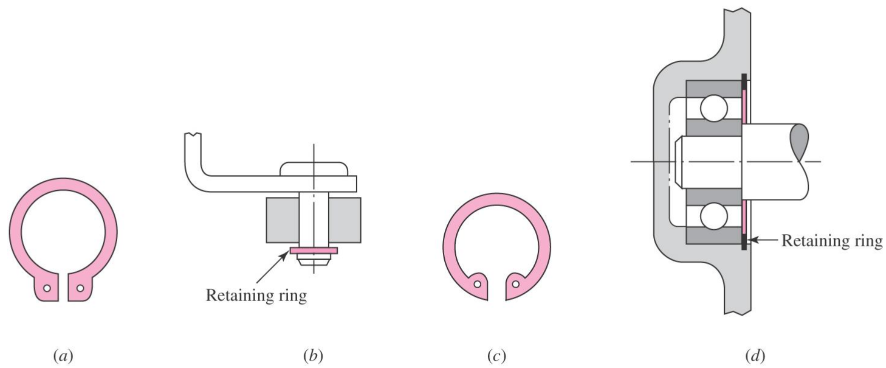

Fig. 7–18

Retaining Rings

- Retaining ring must seat well in bottom of groove to support axial loads against the sides of the groove.
- This requires sharp radius in bottom of groove.
- Stress concentrations for flat-bottomed grooves are available in Table A–15–16 and A–15–17.
- Typical stress concentration factors are high, around 5 for bending and axial, and 3 for torsion

Limits and Fits

- Shaft diameters need to be sized to “fit” the shaft components (e.g. gears, bearings, etc.)
- Need ease of assembly
- Need minimum slop
- May need to transmit torque through press fit
- Shaft design requires only nominal shaft diameters
- Precise dimensions, including tolerances, are necessary to finalize design

Nomenclature for Cylindrical Fit

- Upper case letters refer to hole
- Lower case letters refer to shaft
- *Basic size* is the nominal diameter and is same for both parts, *D=d*
- *Tolerance* is the difference between maximum and minimum size
- *Deviation* is the difference between a size and the basic size

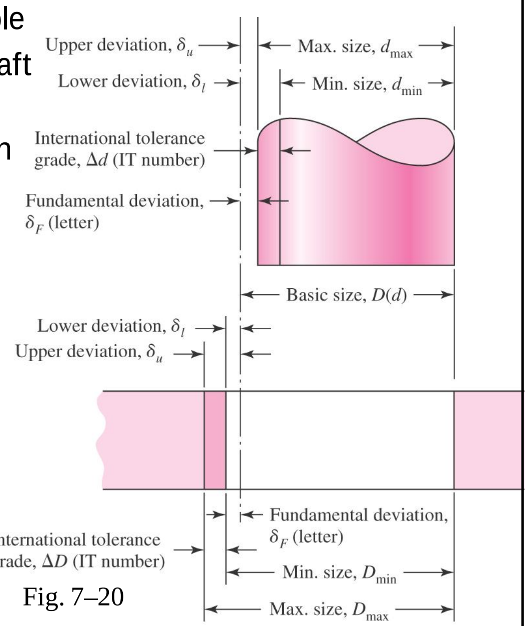

Fig. 7–20

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Description of Preferred Fits (Clearance)

Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6

Table 7–9

Dep. of Mech. Eng.

Shigley's Mechanical Engineering Design

Description of Preferred Fits (Transition & Interference)

Type of Fit	Description	Symbol
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

Table 7–9

Dep. of Mech. Eng.

Shigley's Mechanical Engineering Design

Shaft Design for Stress

- Stresses are only evaluated at critical locations
- Critical locations are usually
  - On the outer surface
  - Where the bending moment is large
  - Where the torque is present
  - Where stress concentrations exist