Machinery Design

CHAPTER 5
DESIGN OF THREADED FASTENNERS AND JIONTS

5.1 BASIC CONCEPTS

Screws have been used as fasteners for a long time. Screw or thread joints are separable joints held together by screw fastenings, such as screws, bolts, studs and nuts, or by thread cut on the parts to be joined.

Screws engage the threads of nuts or of other parts.

A nut is a threaded fastening with internal thread. It is screwed on the bolt and is of a shape designed to be gripped by a wrench or by hand.

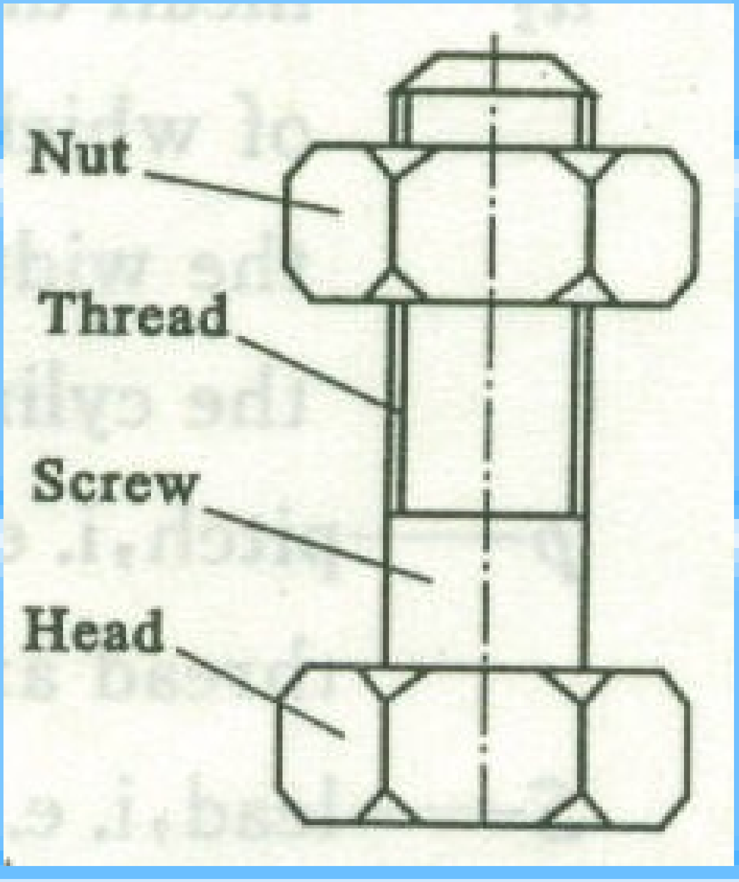


Fig.5.1 Screw (bolt) and nut

5.2 THREAD STANDARDS AND DEFINITIONS

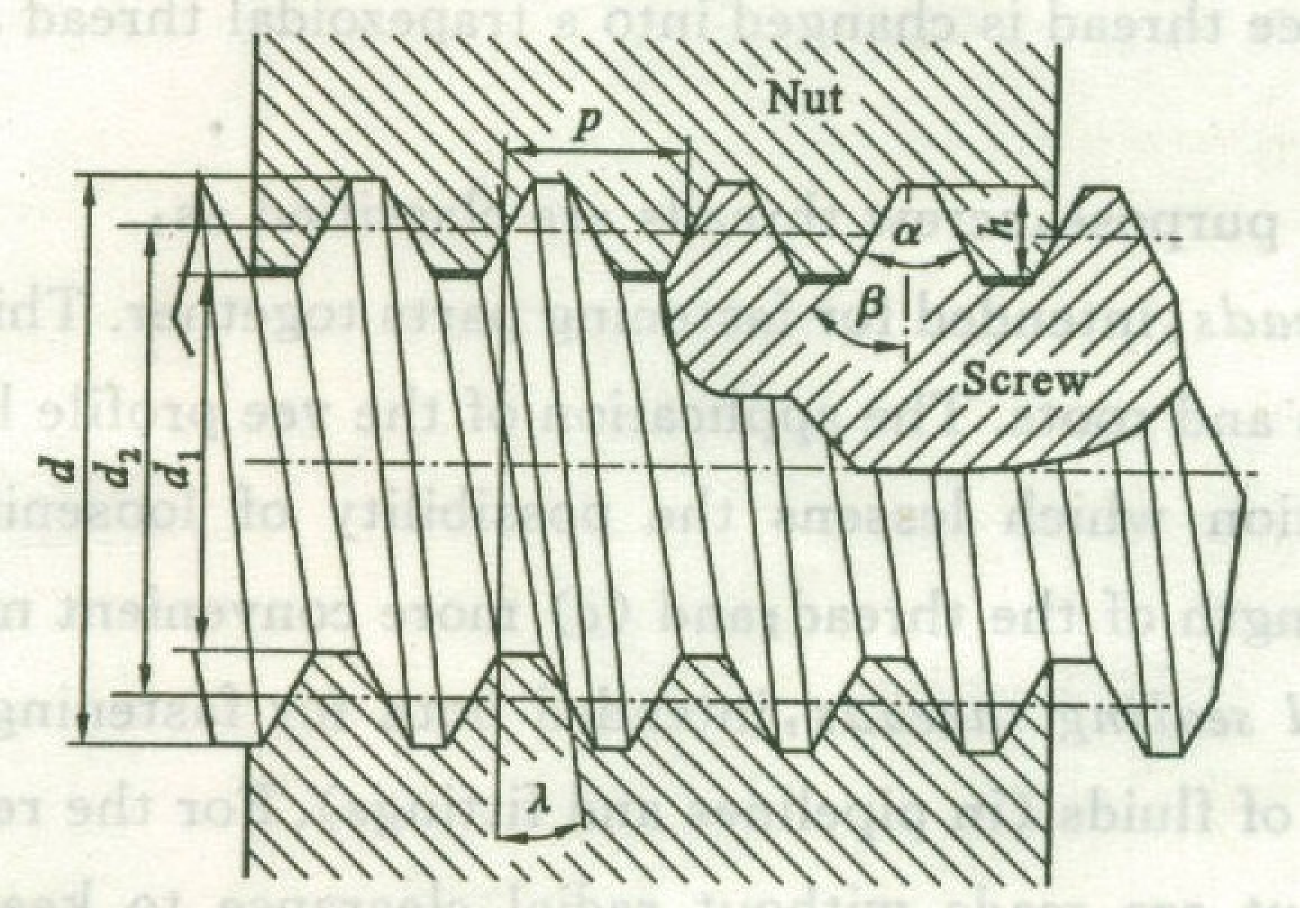


Fig.5.2 General screw-thread symbols

Where d — the largest diameter of a screw thread;

d_1 — minor diameter, i.e. the smallest diameter of a screw thread;

d_2 — mean diameter;

p — pitch;

s — lead;

α — Thread angle;

h — height of thread engagement;

λ — lead (or helix) angle.

According to their purpose, screw threads are classified as:

(1) Fastening threads;

(2) Fastening and sealing threads;

(3) Power threads

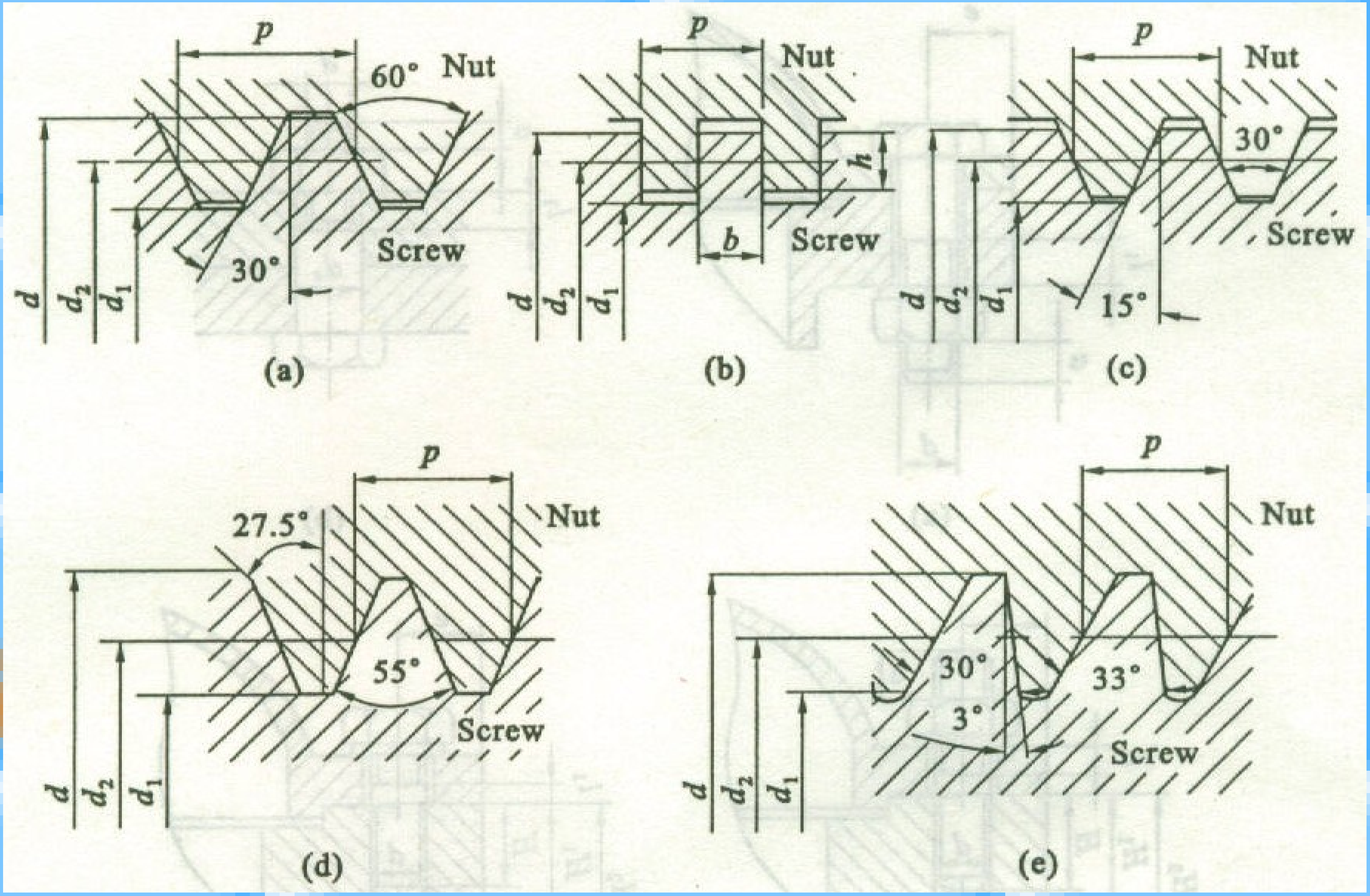


Fig.5.3 Principal type of screw threads

5.3 SCREW FASTENINGS

Depending upon the type of screw joint involved, screw fastenings are classed as:

(1) Screws with nuts, generally called bolts;

(2) Cap screws inserted into tapped holes in the parts being fastened;

(3) Studs, or stud-bolts, used with nuts and having threads on both ends.

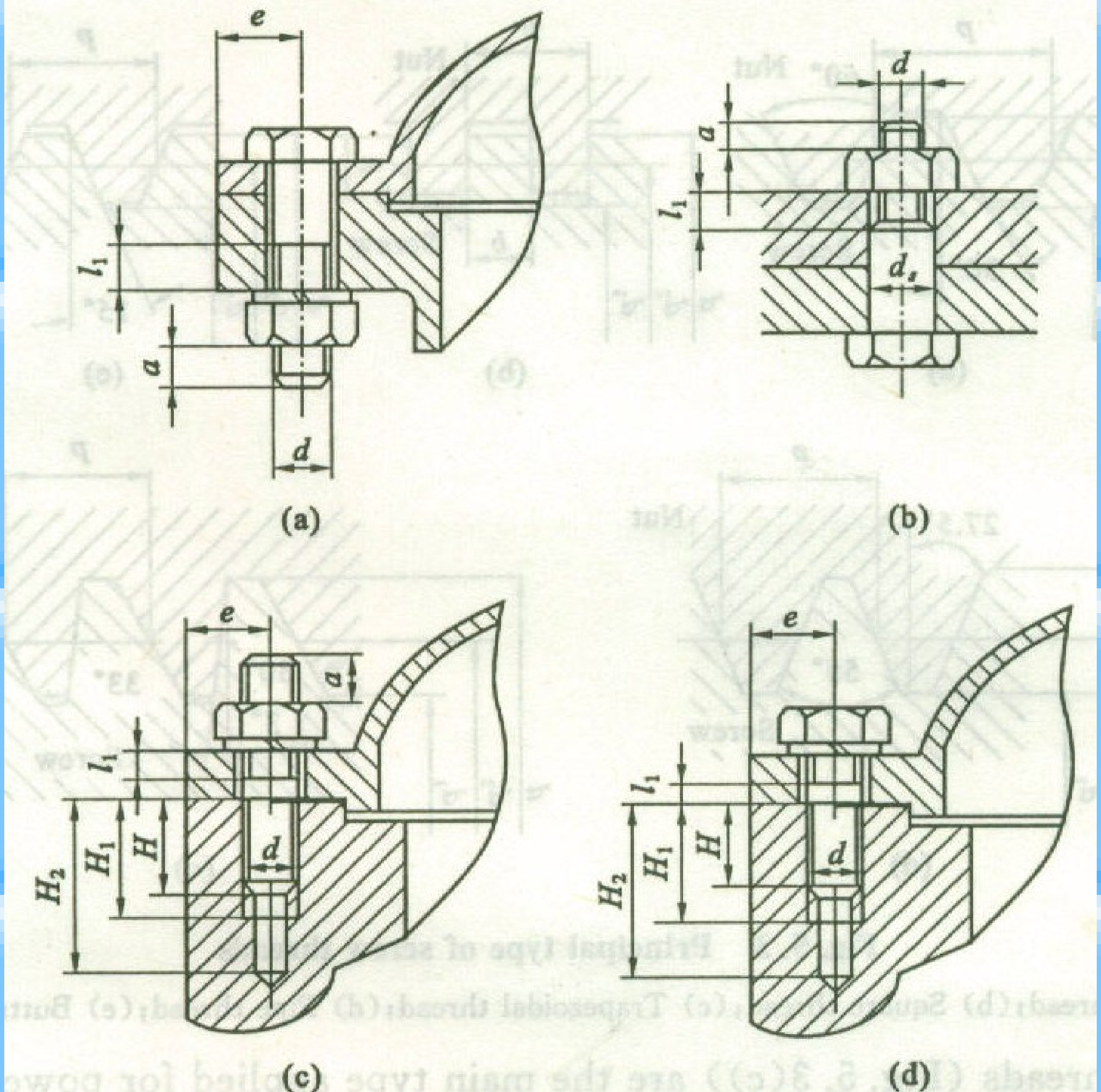


Fig.5.4 Principal types of screw joints

With respect to the shape of their heads, screw fastenings are divided into:

(1) Those in which the head is engaged externally by a tool (wrench , etc.);

(2) Those in which the head is engaged internally and from the end face;

(3) Those that prevent the screw fastening from turing.

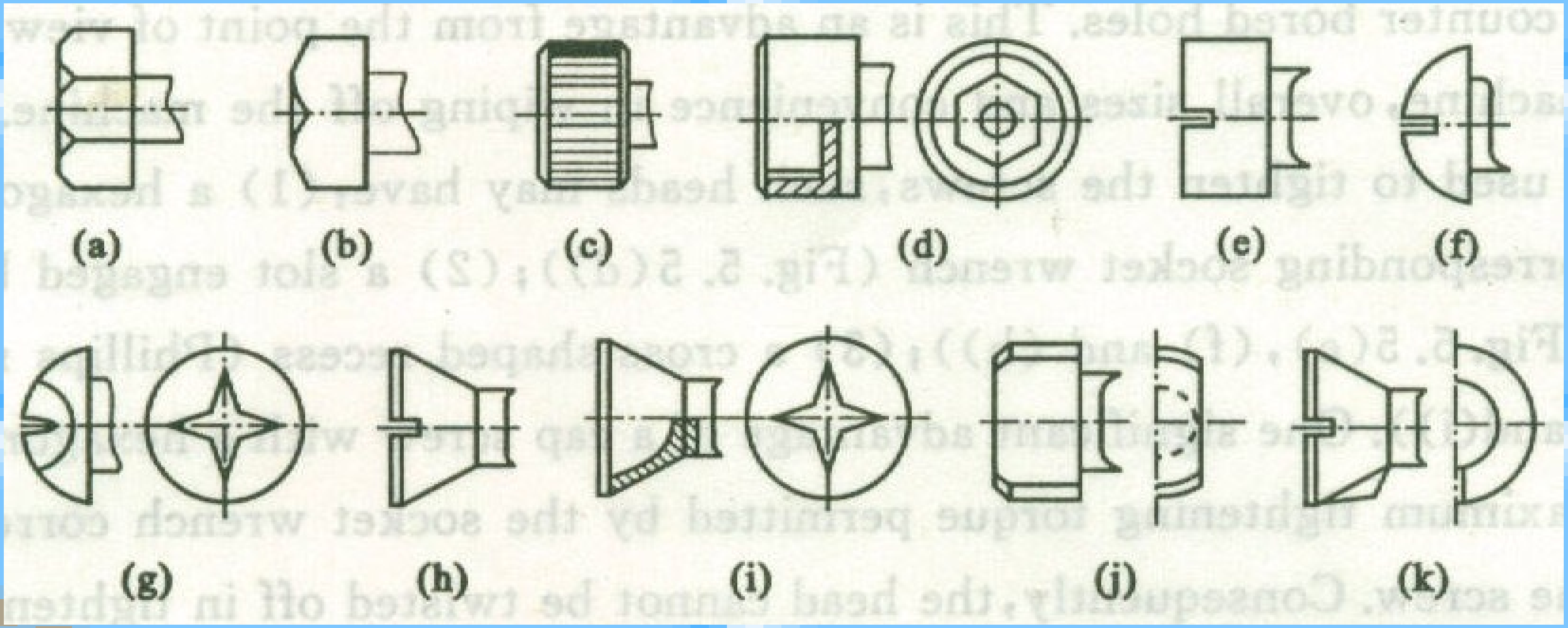


Fig.5.5 Heads of cap screws

Setscrews are another form of fastener; the usual use for them is to prevent relative circular motion between two parts such as shafts and pulleys. They may be used only where the torque requirements are low.

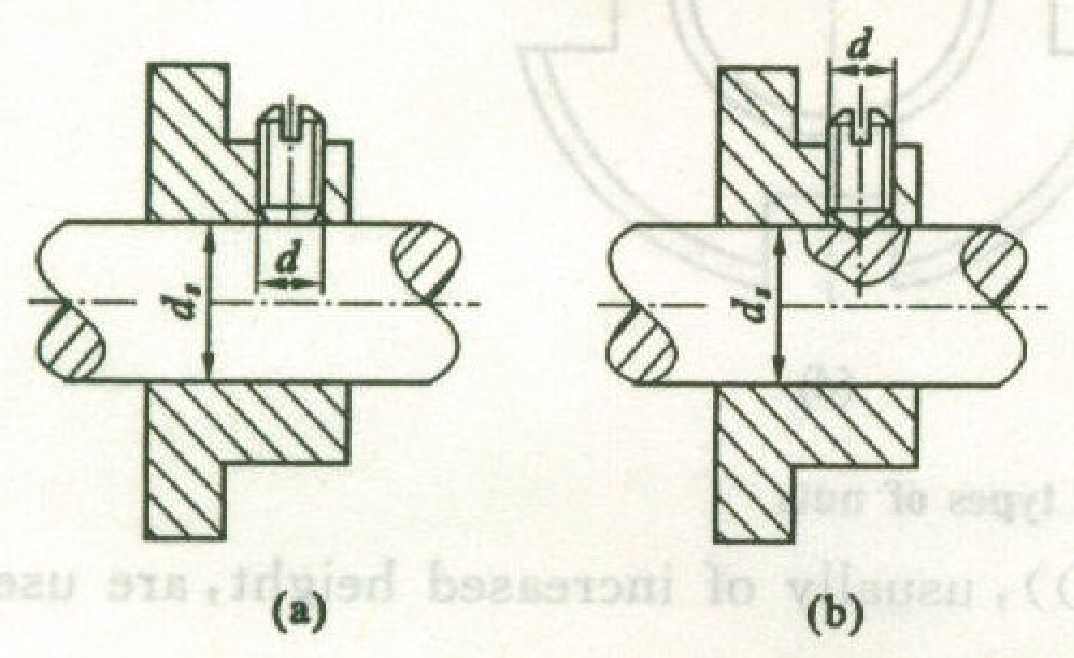


Fig.5.7 Applications of setscrews

The types of points of setscrews are follows:

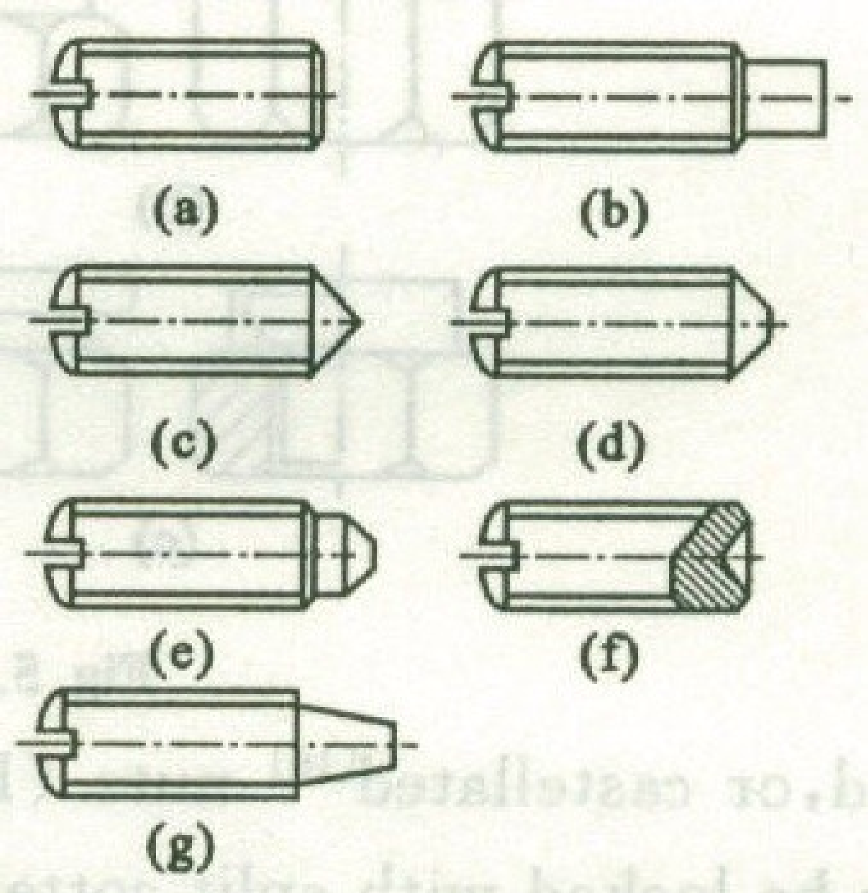


Fig.5.8 Setscrews with various points

The main types of nuts are as follows:

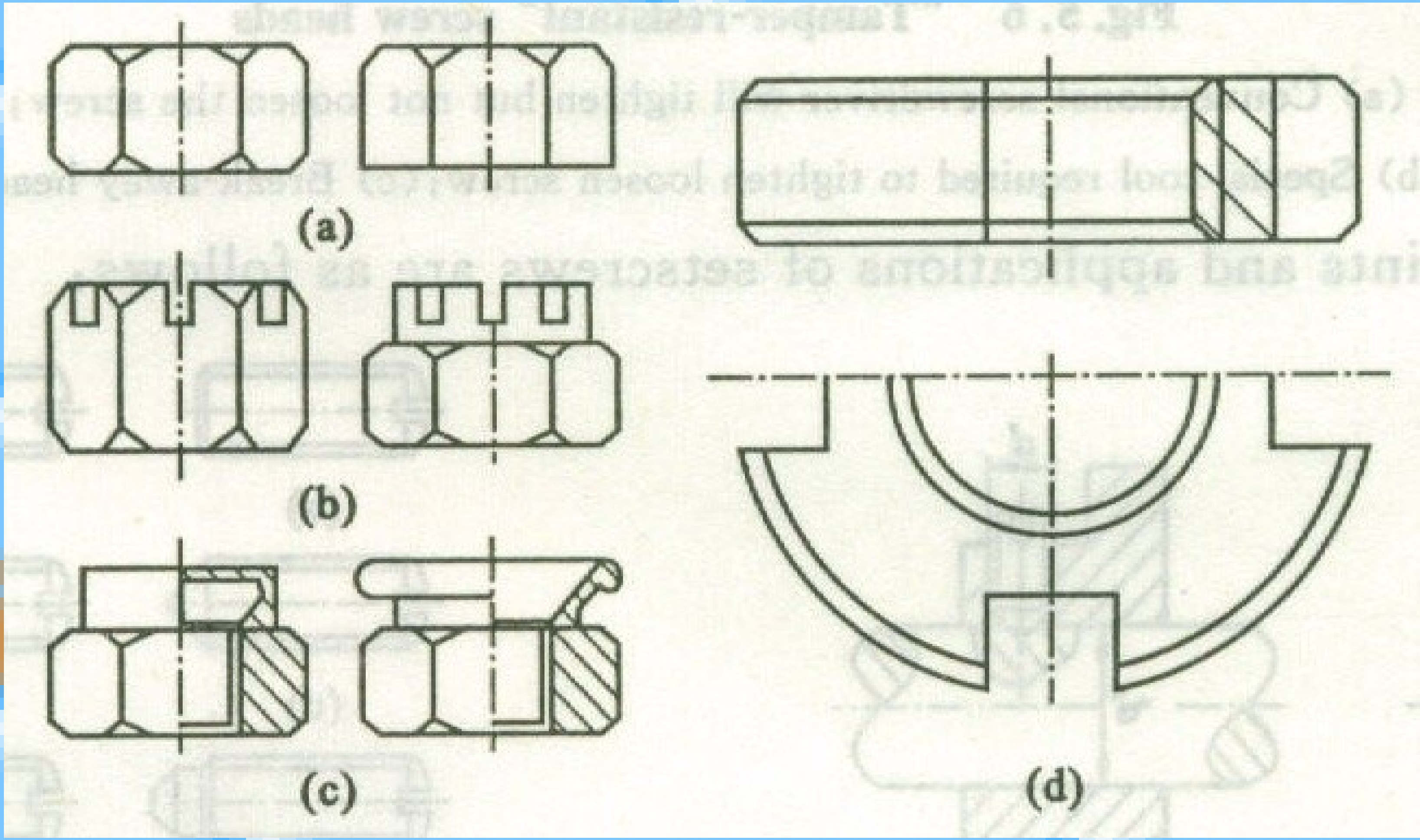
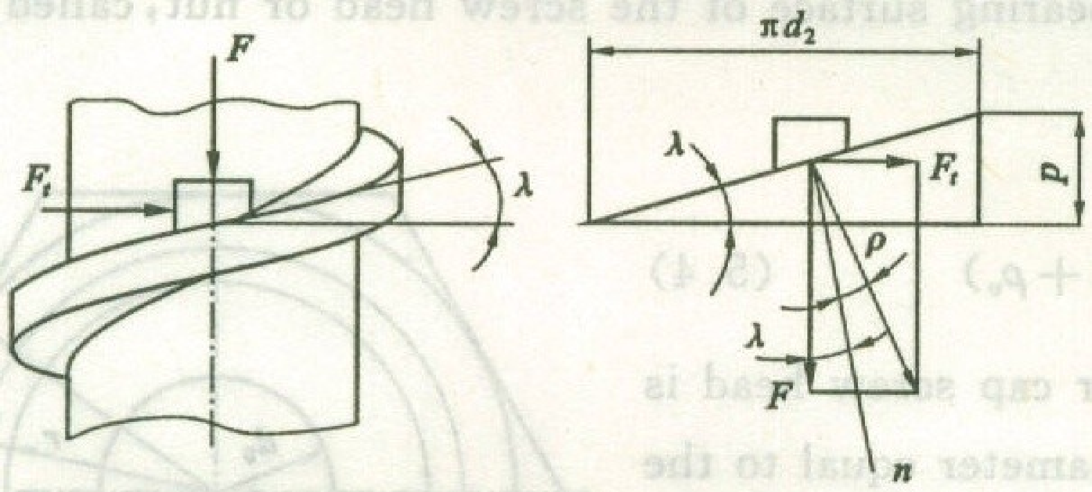


Fig.5.10 Principal types of nuts

5.4 SREWING-UP TORQUE, EFFICIENCY AND SELF-LOCKING CONDITIONS

$$T = T_1 + T_2$$

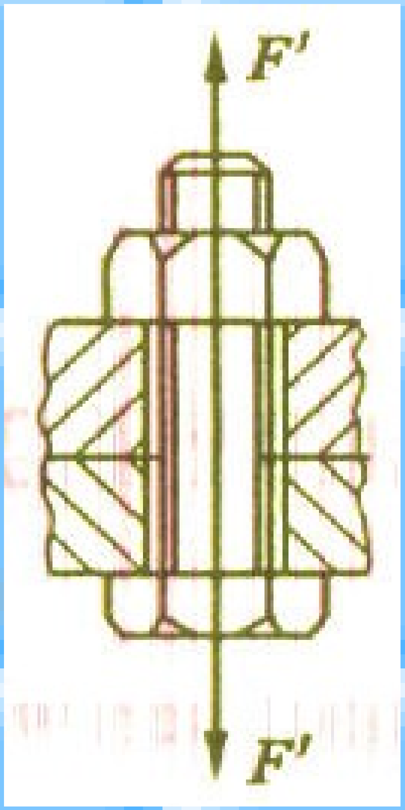


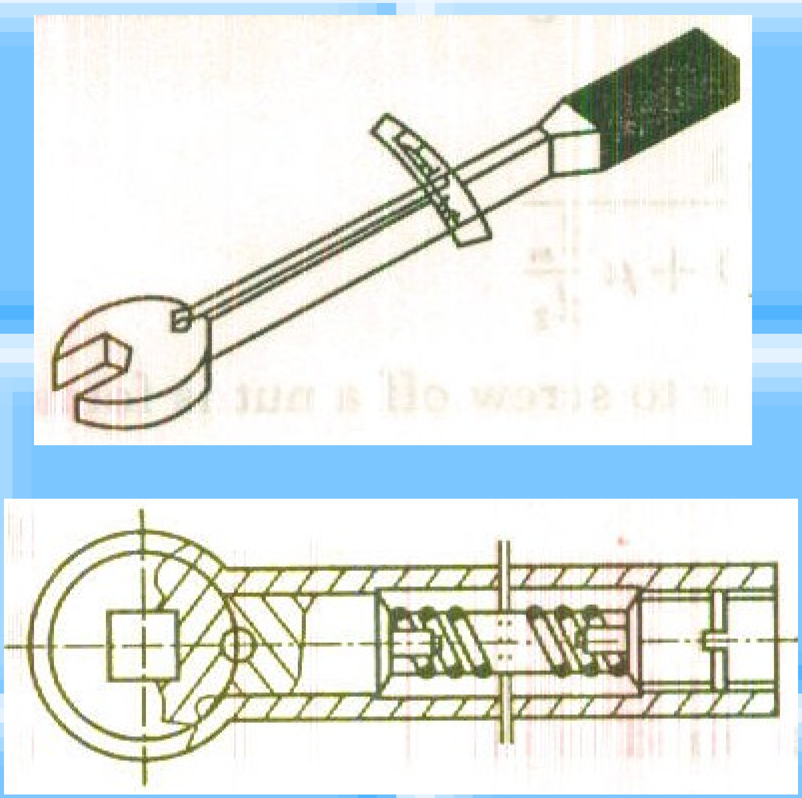
$$T \approx 0.2Fd$$

$$\eta = \frac{tg\lambda}{tg(\lambda + \rho_v)}$$

$$\lambda < \rho_v$$

5.5 BOLT TIGHTENING AND INTIAL TENSION





F′----initial force

Fig.5.13 Torque-wrenches

5.6 PREVENTING UNINTENTIONAL UNSCREWING OF SCREW JOINTS

Locking can be accomplished by the following measures:

(1) By supplementary friction;

(2) by using special locking devices;

(3) by plastic deformation or welding on.

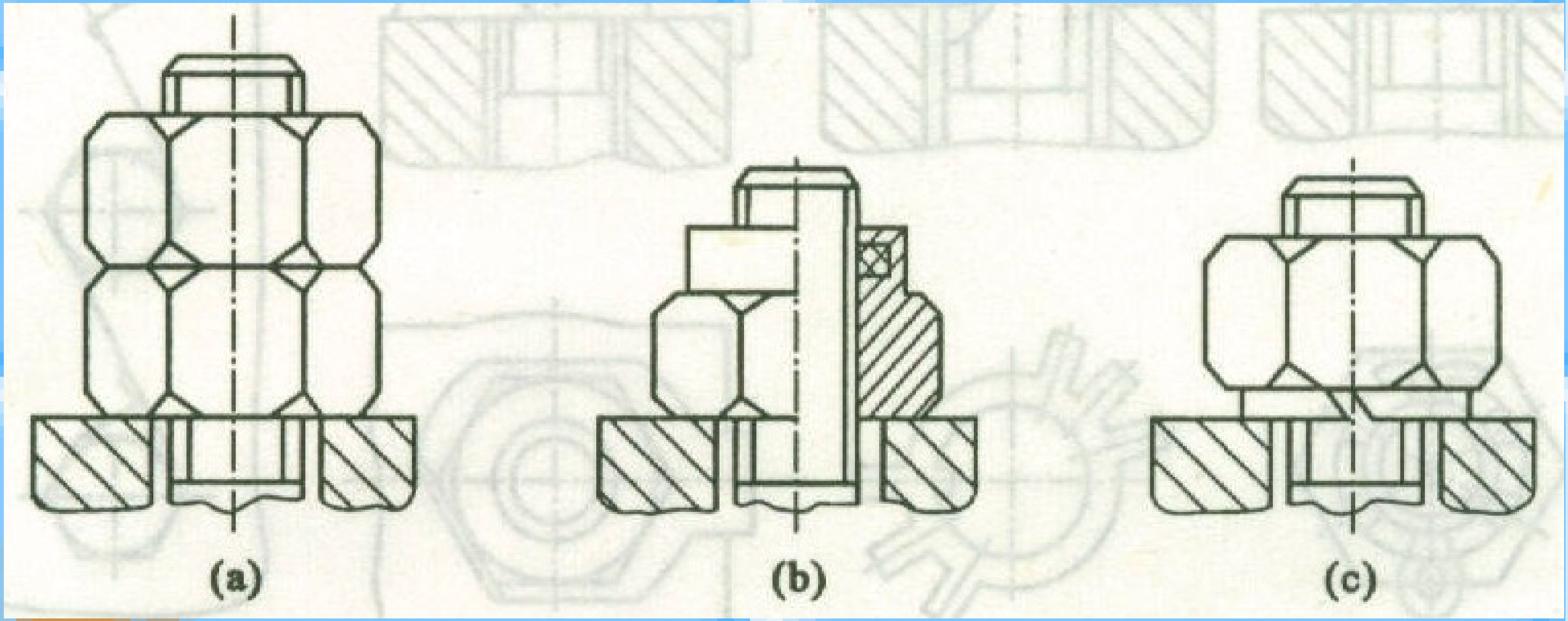


Fig.5.14 Locking devices based on the application of supplementary friction

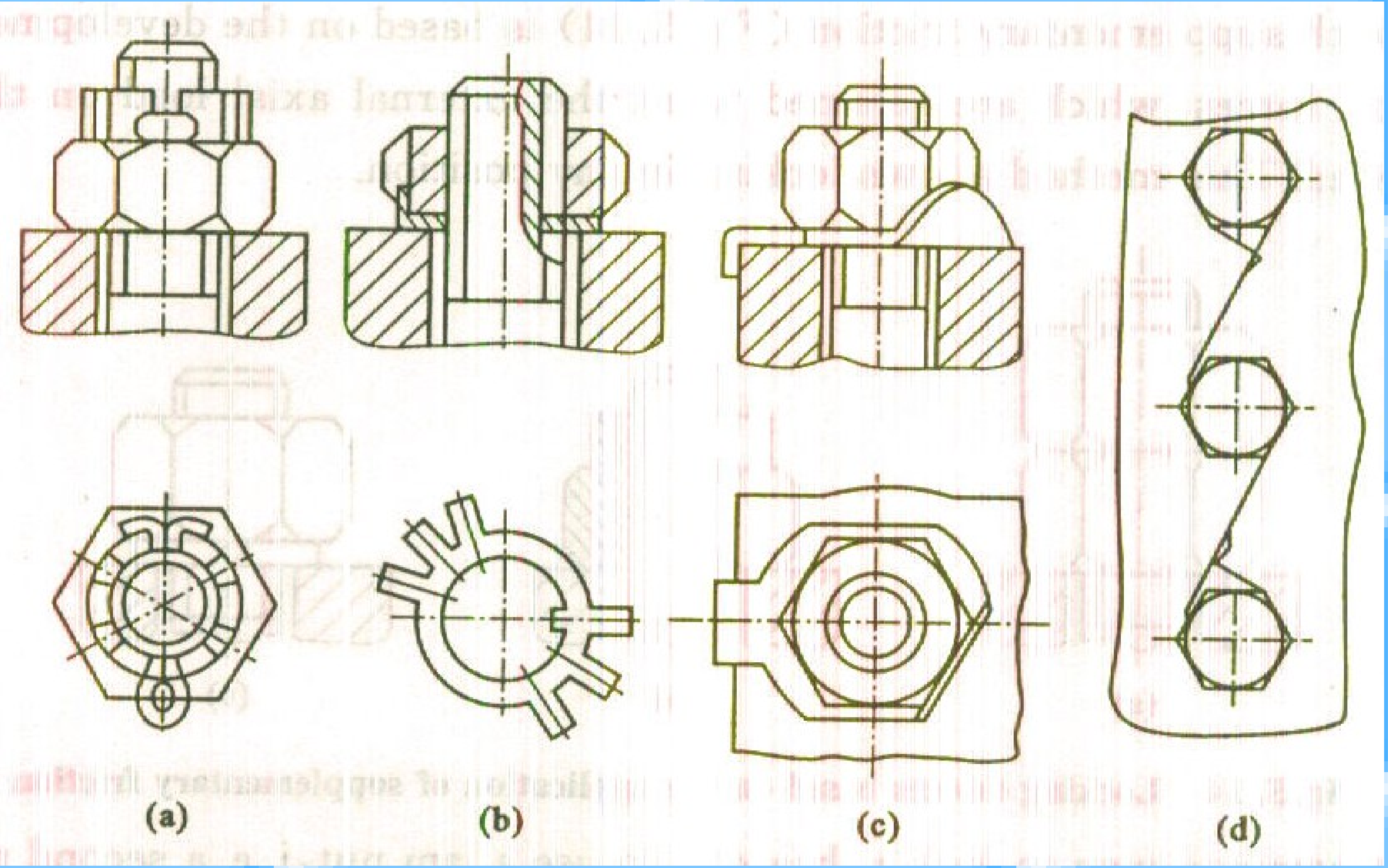


Fig.5.15 Locking devices using special locking elements

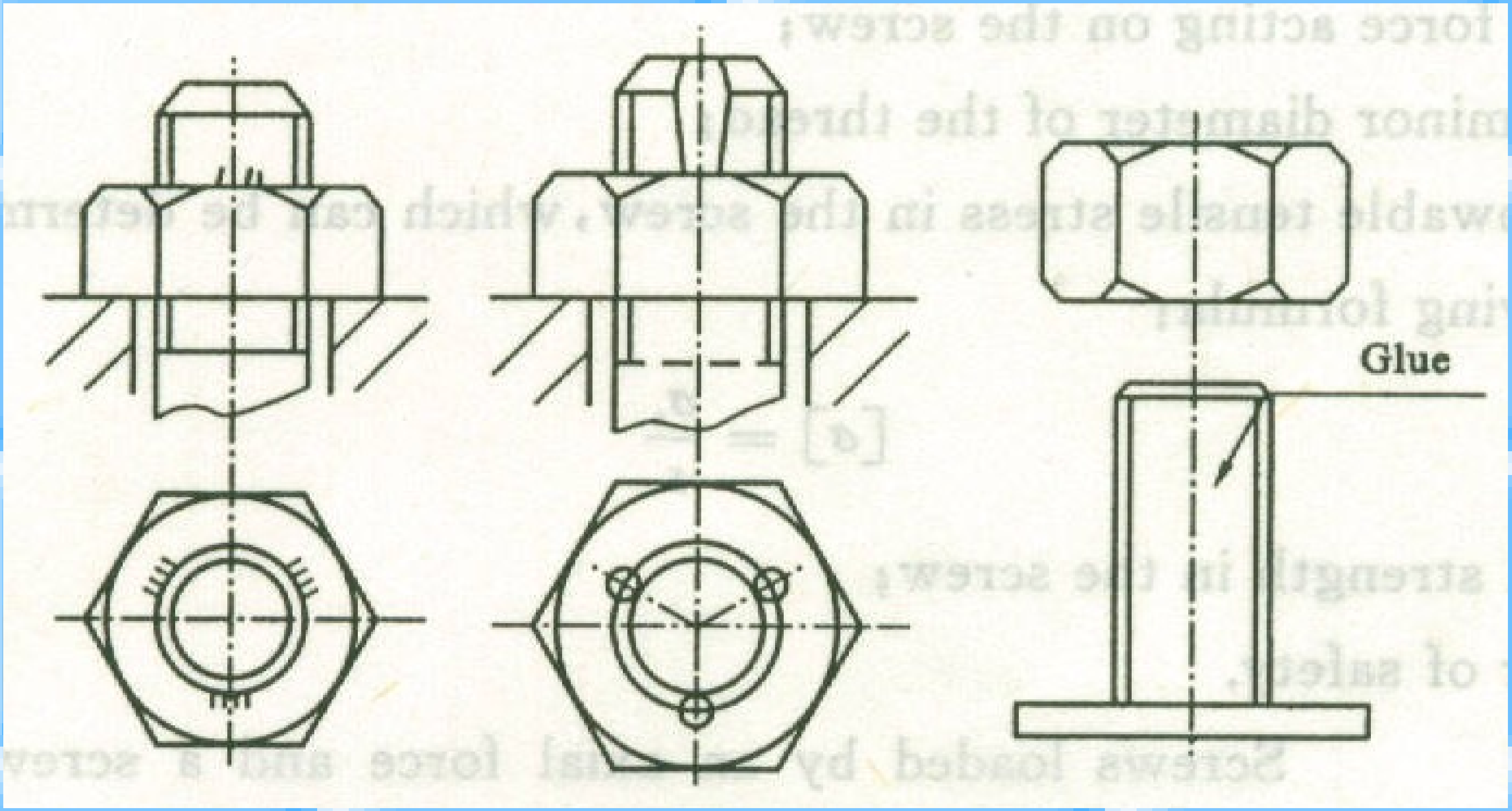


Fig.5.16 Permanent locking

5.7 SCREW AND THREAD ELEMENT DESIGN FOR STEADY LOADS

1. Screws without an initial preload

The nominal tensile stress in the screw is

$$\sigma = \frac{F}{\frac{\pi d_1^2}{4}} \le [\sigma]$$

From which the minor diameter is

$$d_1 \ge \sqrt{\frac{4F}{\pi[\sigma]}}$$

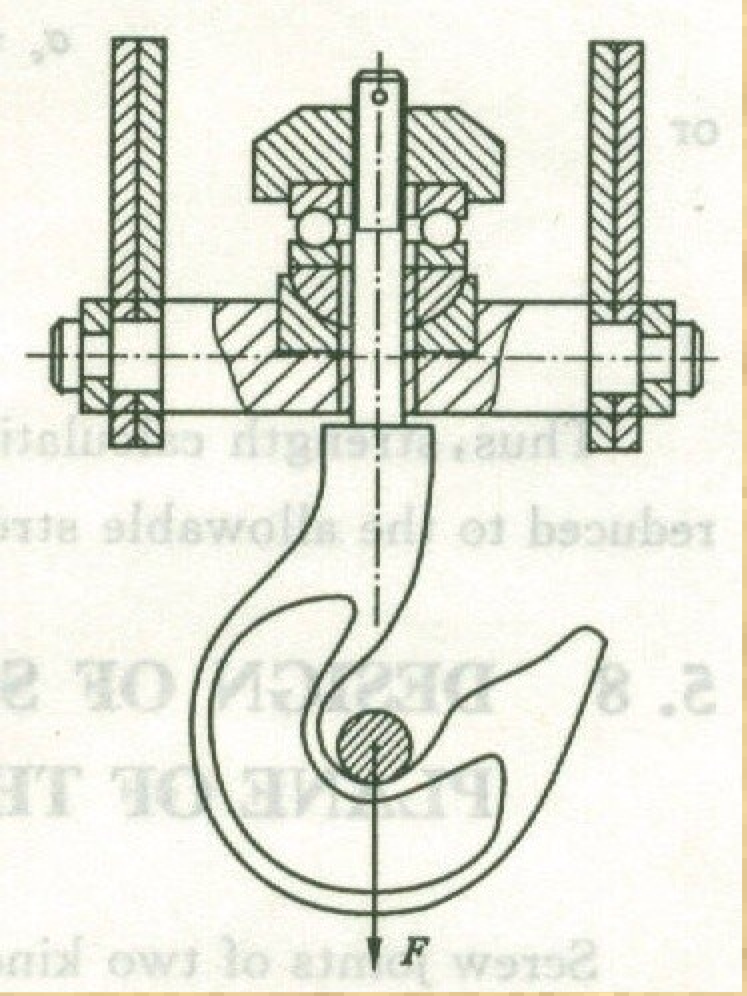


Fig.5.17 Hoisting hook

2. Screws loaded by an axial force and a screwing-up torque

The equivalent nomal stress in a screw due to tension and torque is

$$\sigma_e = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{\sigma^2 + 3(0.5\sigma)^2} \approx 1.3\sigma$$

or

$$\sigma_e = \frac{1.3F'}{\frac{\pi d_1^2}{4}} \leq [\sigma]$$

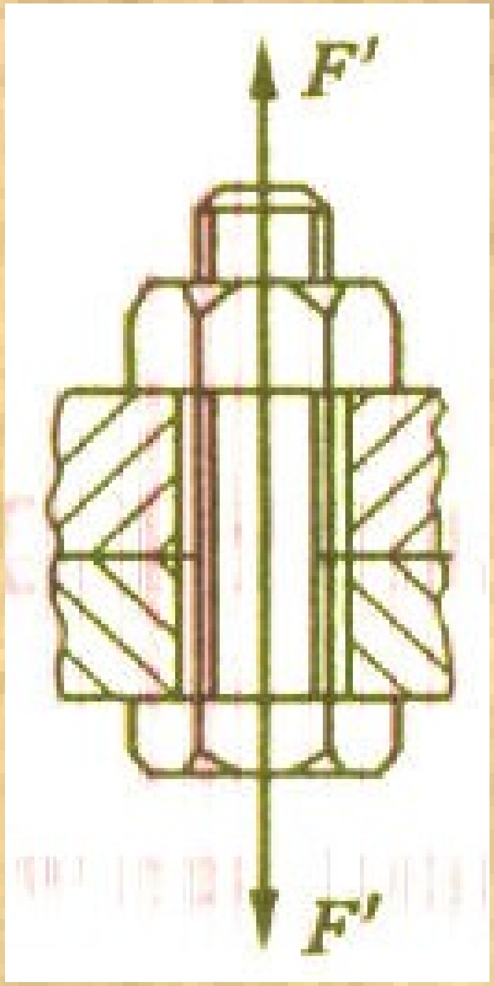


Fig.5.18 Bolted joints in tension

5.8 DESIGN OF SCREW JOINTS SUBJECT TO LOADS ON THE PLANE OF THE JOINT

Screw joints of two kinds are employed:

(a) With screws inserted in holes with a clearance;

(b) With screws fitting into reamed holes without appreciable clearance.

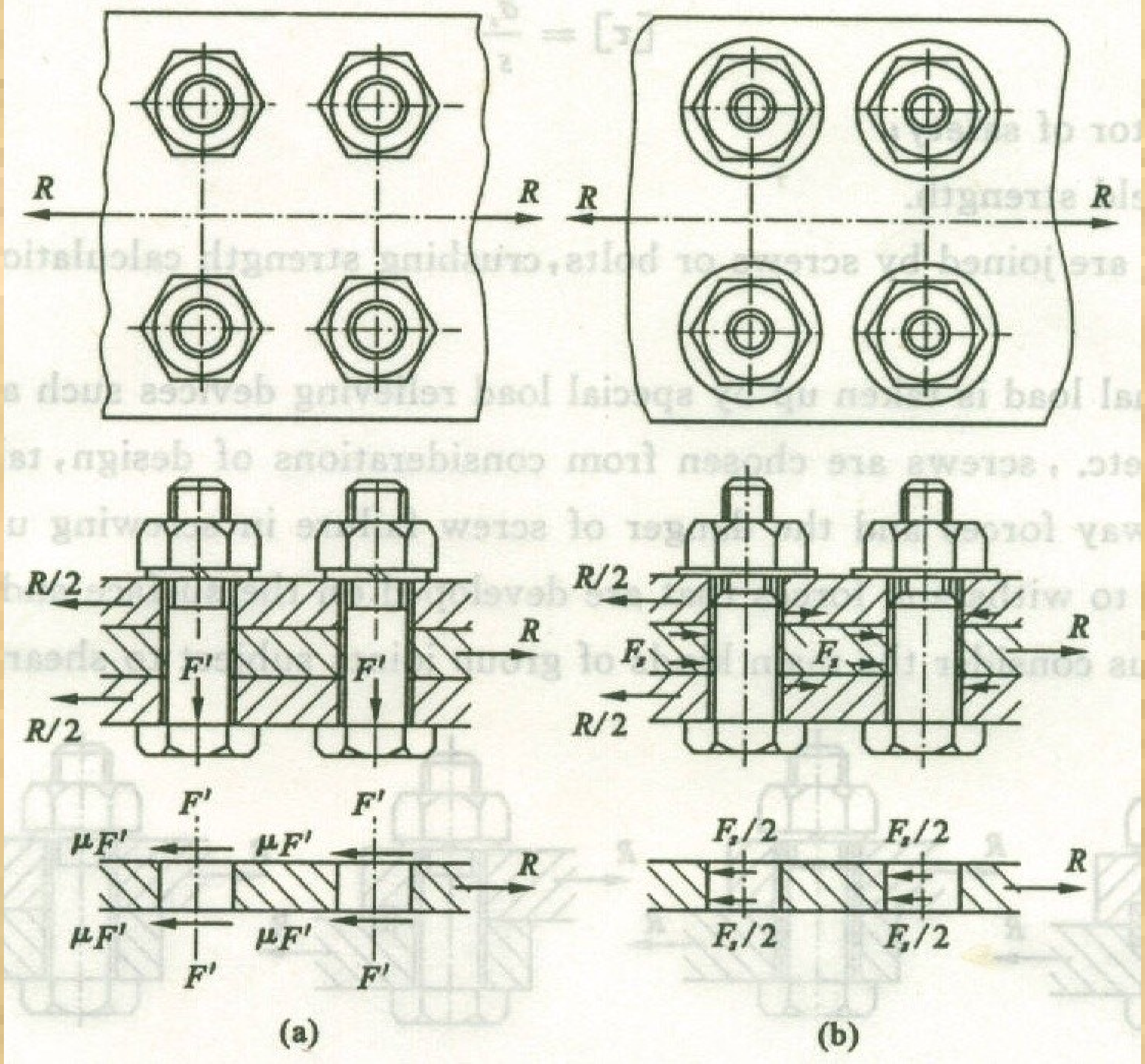


Fig.5.19 Screw joints subject to loads on the plane of the joints

1. If screws are installed with a clearance, they must develop a friction force on the plane of the joint, which exceeds the external shear force.

The required screwing-up force in this case is

$$F' = \frac{KR}{\mu m z} \leq [\sigma]$$

The required screw diameter can be calculated on the basis of the corresponding screwing-up force.

2. If screws are fitted into reamed holes they are checked in shear. Then the strength condition of the screw is

$$\sigma_p = \frac{F_s}{d_0 h} \le [\sigma_p]$$

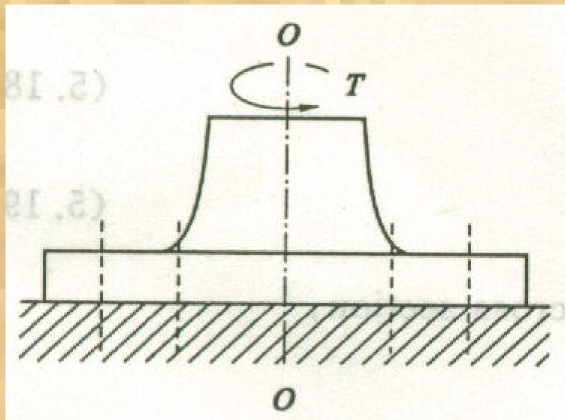
$$\tau = \frac{4F_s}{\pi d_0^2 m} \le [\tau]$$

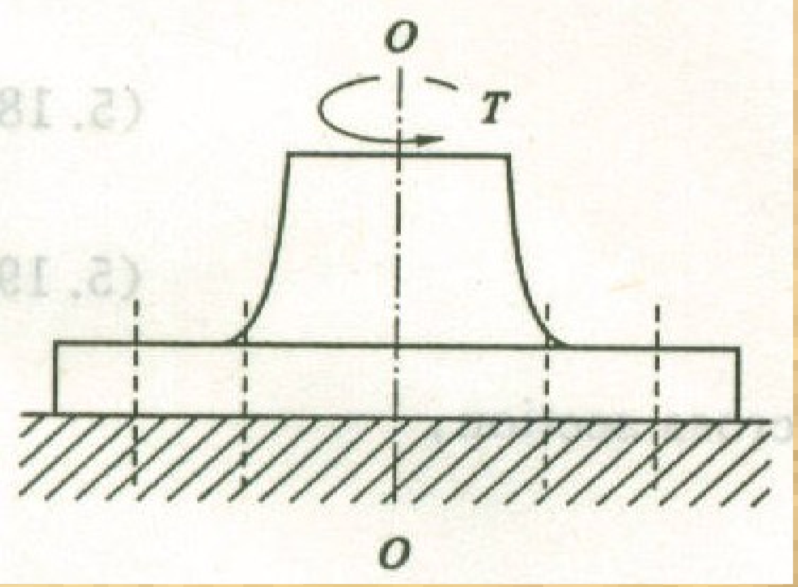
$$F_s = \frac{R}{z}$$

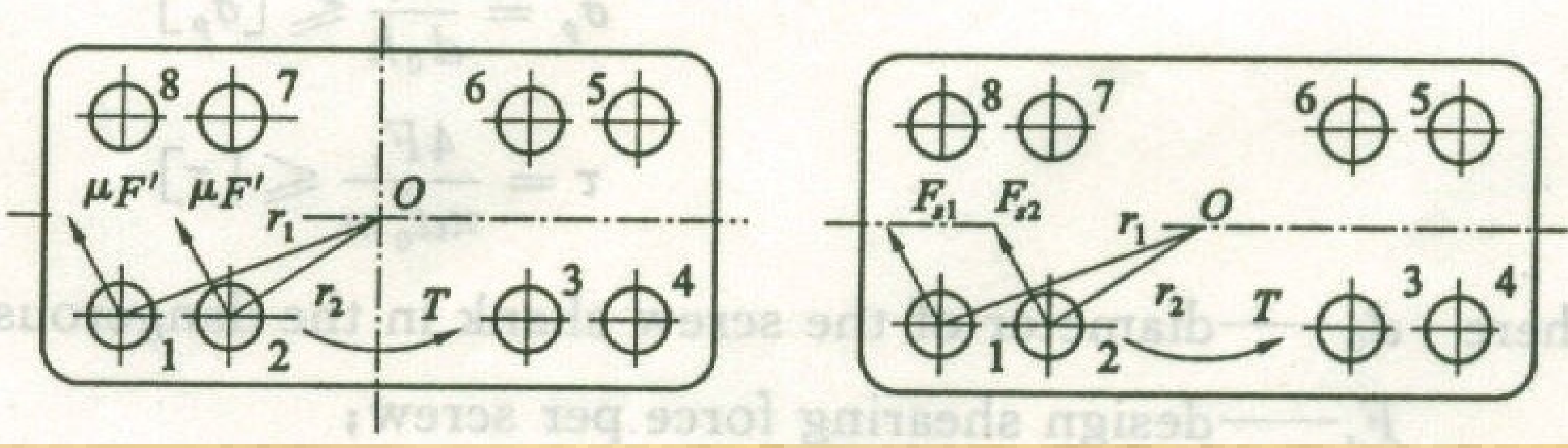
3. joints loaded by the moment T developed by the forces acting on the plane of the abutting surfaces of the joint when the screws are installed with clearance in their holes

The holding power condition is

$$F' = \frac{KT}{\mu(r_1 + r_2 + \cdots + r_z)} = \frac{KT}{\mu \sum_{i=1}^{z} r_1}$$







a) b)

Fig.5.21 Joints subject to shearing moments

4. When the screws are installed without clearance in their holes, the condition of equilibrium is

$$T = F_{s1} r_1 + F_{s2} r_2 + \cdots + F_{sz} r_z$$

According to the condition that the forces are proportional to the displacements

$$\frac{F_{s1}}{r_1} = \frac{F_{s2}}{r_2} = \cdots = \frac{F_{sz}}{r_z}$$

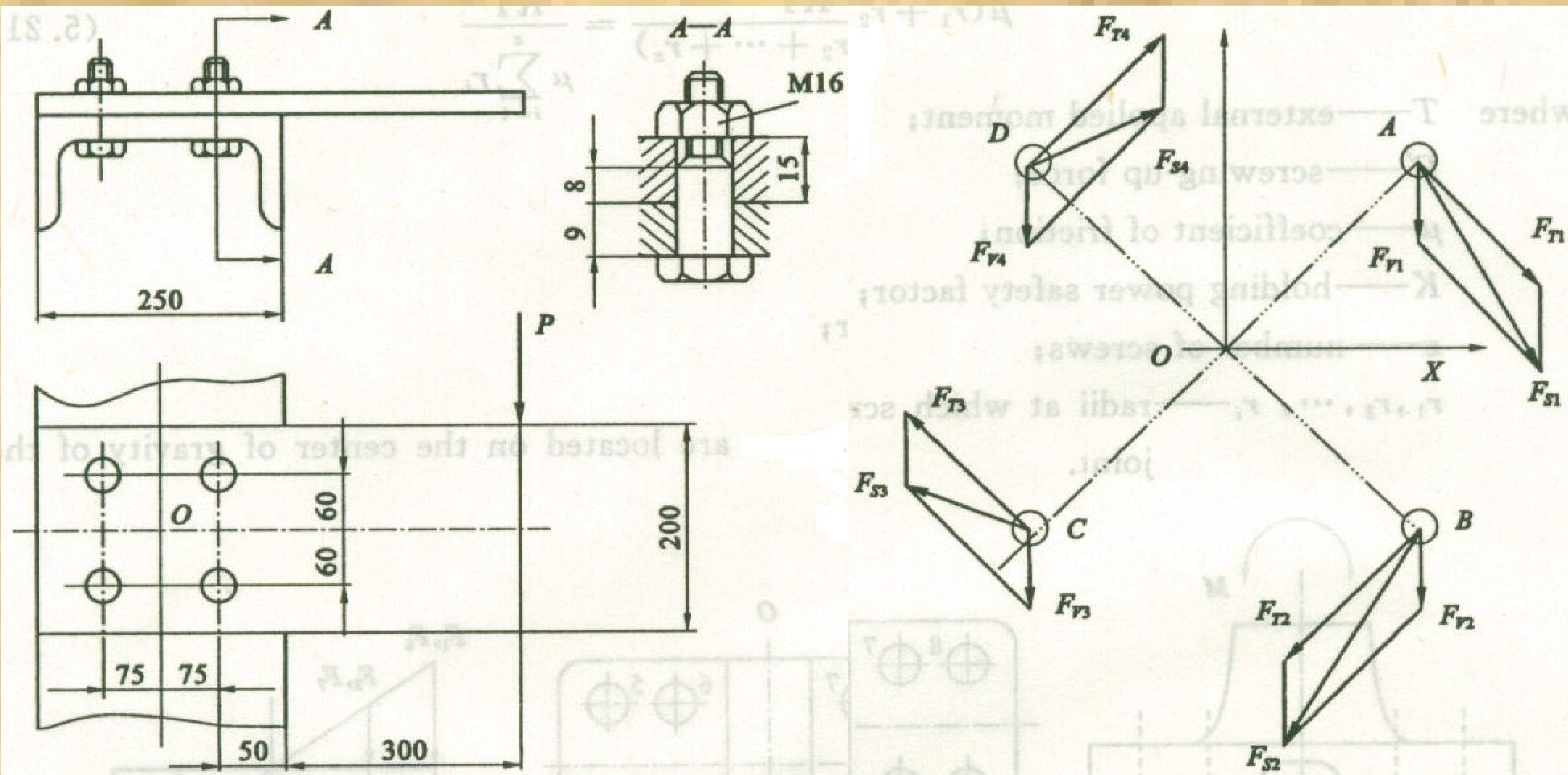
So the shearing force on the most heavily loaded screw is:

$$F_{s1} = F_{s4} = F_{s5} = F_{s8} = \frac{T r_1}{\sum_{i=1}^{8} r_i^2}$$

Example 5.1 Shown in Fig.5.23(a) is a 15-by 200-mm rectangular steel bar cantilevered to a 250-mm steel channel by using four bolts. Based on the external load of 16 kN, we find:

(1) The resultant load on each bolt.

(2) The maximum bolt shear stress.



a) b)

Fig.5.23 Dimentions in millimeters

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(1) The sheer reaction V would pass through O and the moment reaction T would be about O. These reactions are

V=16kN T=16×425=6800Nm

The distance from the centroid to the center of each bolt is

$$r=\sqrt{60^2+75^2}=96.0mm$$

The primary shear load per bolt is

$$F_v=\frac{V}{z}=\frac{16}{4}=4kN$$

The secondary shear forces are equal,

$$F_T=\frac{Tr}{4r^2}=\frac{T}{4r}=\frac{6800}{4\times 96.0}=17.7kN$$

The resultants are obtained by using the parallelogram rule.

$$F_A = F_B = 21.0 kN$$

$$F_C = F_D = 13.8 kN$$

(2) Bolts A and B are critical because they carry the largest shear load.

The shear-stress area is

$$A_s = \frac{\pi d_0^2}{4} = \frac{\pi 17^4}{4} = 227 mm^2$$

So the shear tress is

$$\tau = \frac{F_{s\max}}{A_s} = \frac{21.0 \times 10^3}{227} = 92.51 MPa$$

5.9 DESIGN OF SCREW JOINTS SUBJECT TO THE OVERTURNING MOMENT

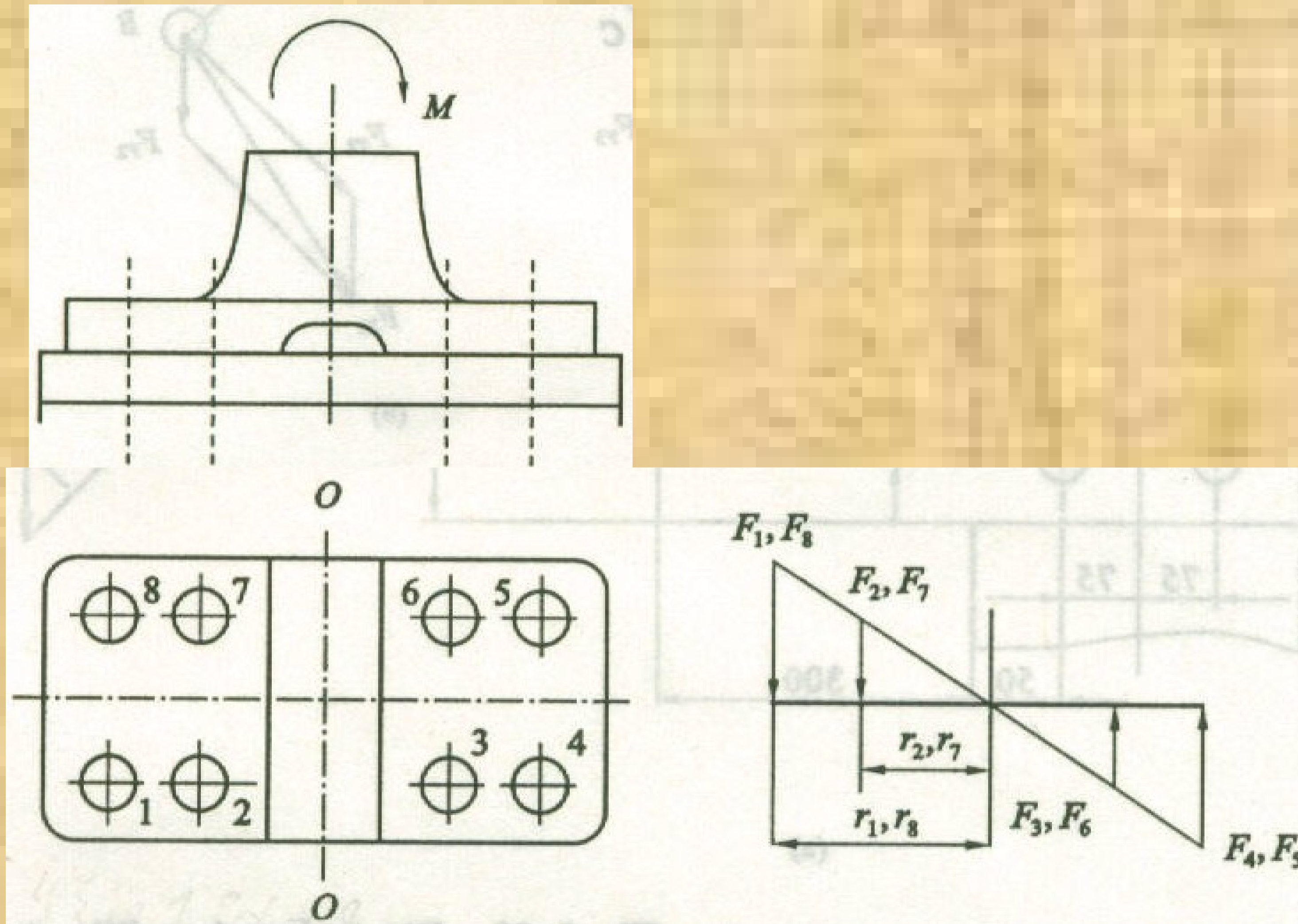

Fig.5.22 Joints subject to the overturning moment

The condition of equilibrium is

$$M = F_1 r_1 + F_2 r_2 + \cdots + F_z r_z$$

According to the condition that the forces are proportional to the displacements

$$\frac{F_1}{r_1} = \frac{F_2}{r_2} = \cdots = \frac{F_z}{r_z}$$

So the force on the most heavily loaded screw is

$$F_1 = F_8 = \frac{M r_1}{\sum_{i=1}^{8} r_i^2}$$

5.10 BOLT TENTION WITH EXTERNAL JOINT-SEPARATING FORCE

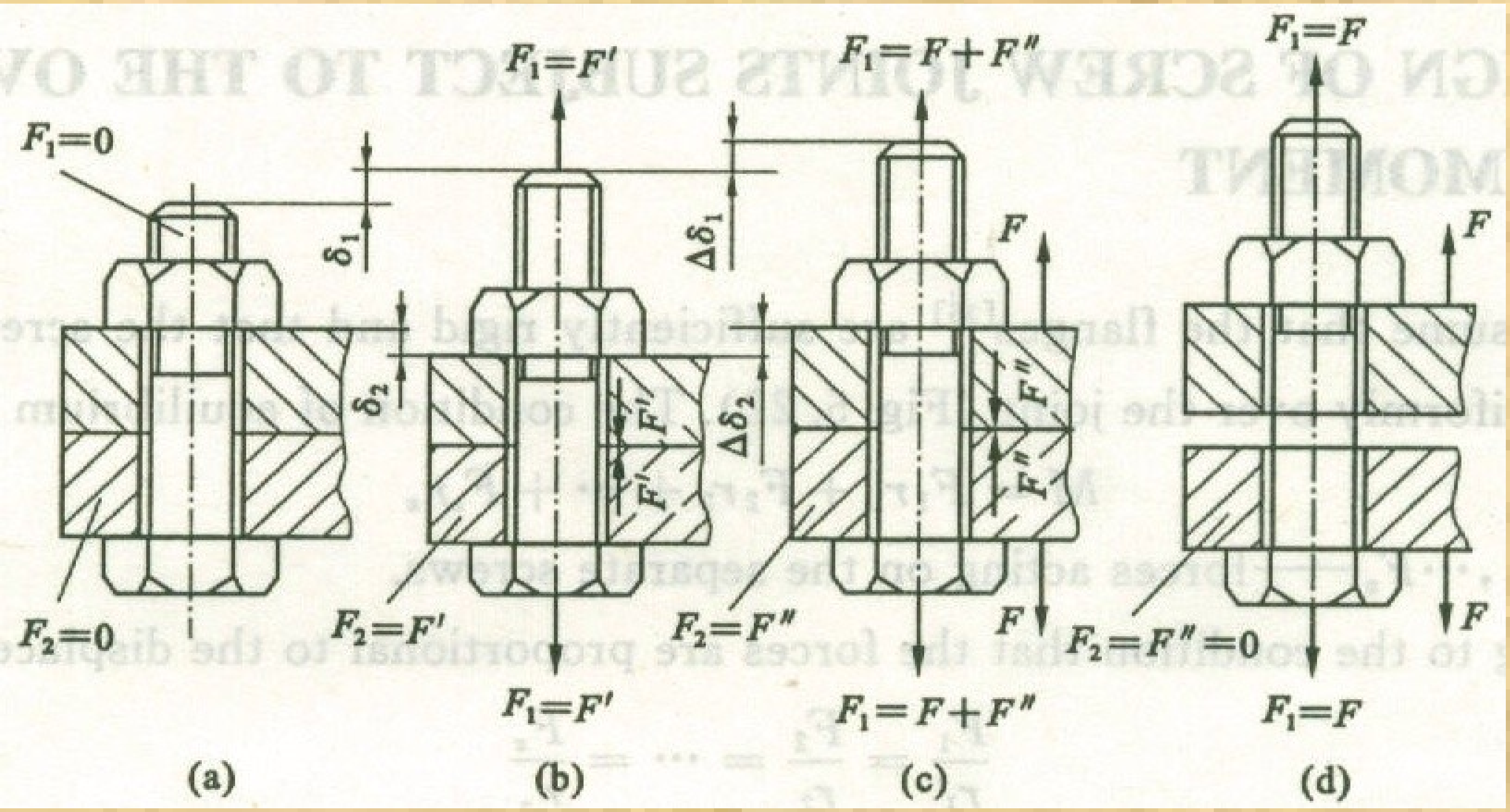

Fig.5.24 Study of bolt tensile loading

The resultant load on the bolt is

$$F_1 = F + F''$$

or

$$F_1 = F' + \frac{K_1}{K_1 + K_2} F$$

The resultant load in the connected members is

$$F_2 = F' - \frac{K_2}{K_1 + K_2} F$$

Fig. 5.25 is a plot of the force –deflection characteristics.

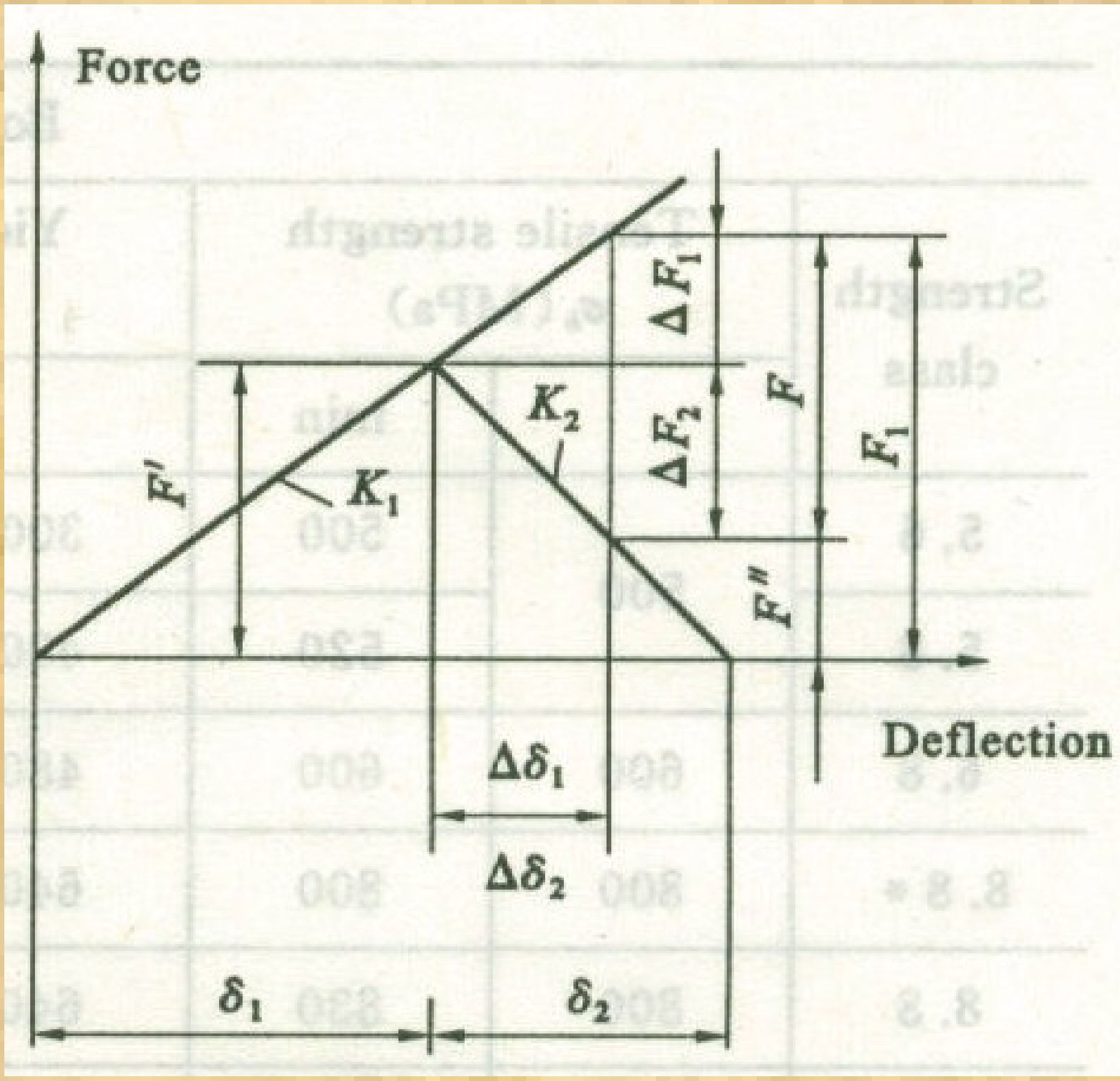

Fig.5.25 Plot of the force-deflection characteristics

Example 5.2 In Fig. 5.24(c), let $K_2=8K_1$. If the preload is $F'= 5000N$ and the external load is $F=5500N$, what are the resultant tension in the bolt and the compression in members?

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The resultant bolt tension is

$$F_1 = F' + \frac{K_1}{K_1 + K_2} F$$

$$= 5000 + \frac{K_1}{K_1 + 8K_1} 5000 = 5611N$$

The resultant compression of the members is

$$F_2 = F' - \frac{K_2}{K_1 + K_2} F$$

$$= 5000 - \frac{8K_1}{K_1 + 8K_1} 5500 = 111N$$

The member s are still in compression , hence there is no separation of the parts.