Design of fits and clams pressure connections

Limits and Fits

The designer is free to adopt any geometry of fit for shafts and holes that will ensure the intended function. There is sufficient accumulated experience with commonly recurring situations to make standards useful. There are two standards for limits and fits in the United States, one based on inch units and the other based on metric units.

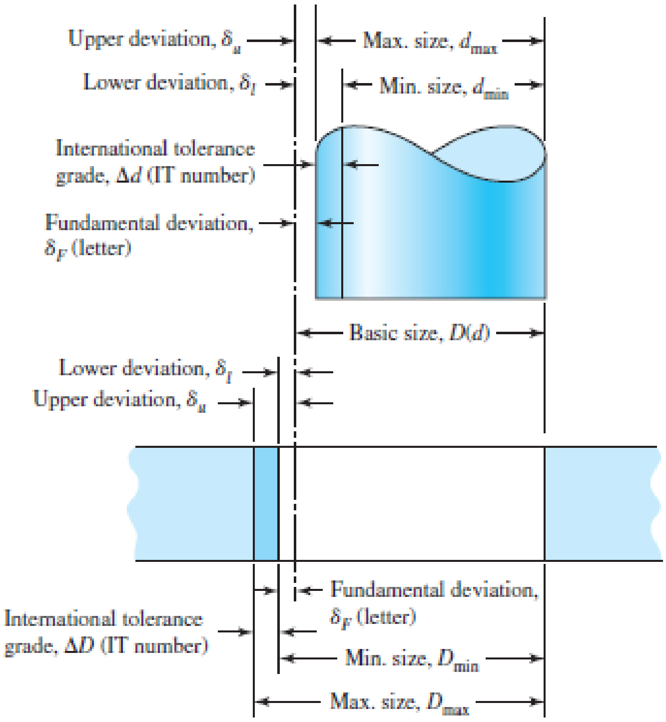
These differ in nomenclature, definitions, and organization. No point would be served by separately studying each of the two systems. The metric version is the newer of the two and is well organized, and so here we present only the metric version but include a set of inch conversions to enable the same system to be used with either system of units.

In using the standard, capital letters always refer to the hole; lowercase letters are used for the shaft. The definitions illustrated in Fig. 5.1 are explained as follows:

• Basic size is the size to which limits or deviations are assigned and is the same for both members of the fit.
• Deviation is the algebraic difference between a size and the corresponding basic size.
• Upper deviation is the algebraic difference between the maximum limit and the corresponding basic size.
• Lower deviation is the algebraic difference between the minimum limit and the corresponding basic size.
• Fundamental deviation is either the upper or the lower deviation, depending on which is closer to the basic size.
• Tolerance is the difference between the maximum and minimum size limits of a part.
• International tolerance grade numbers (IT) designate groups of tolerances such that the tolerances for a particular IT number have the same relative level of accuracy but vary depending on the basic size.
• Hole basis represents a system of fits corresponding to a basic hole size. The fundamental deviation is H.
• Shaft basis represents a system of fits corresponding to a basic shaft size. The fundamental deviation is h. The shaft-basis system is not included here.

Figure 5.1

Definitions applied to a cylindrical fit.



The standard uses tolerance position letters, with capital letters for internal dimensions (holes) and lowercase letters for external dimensions (shafts). As shown in Fig. 5.1, the fundamental deviation locates the tolerance zone relative to the basic size.

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

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

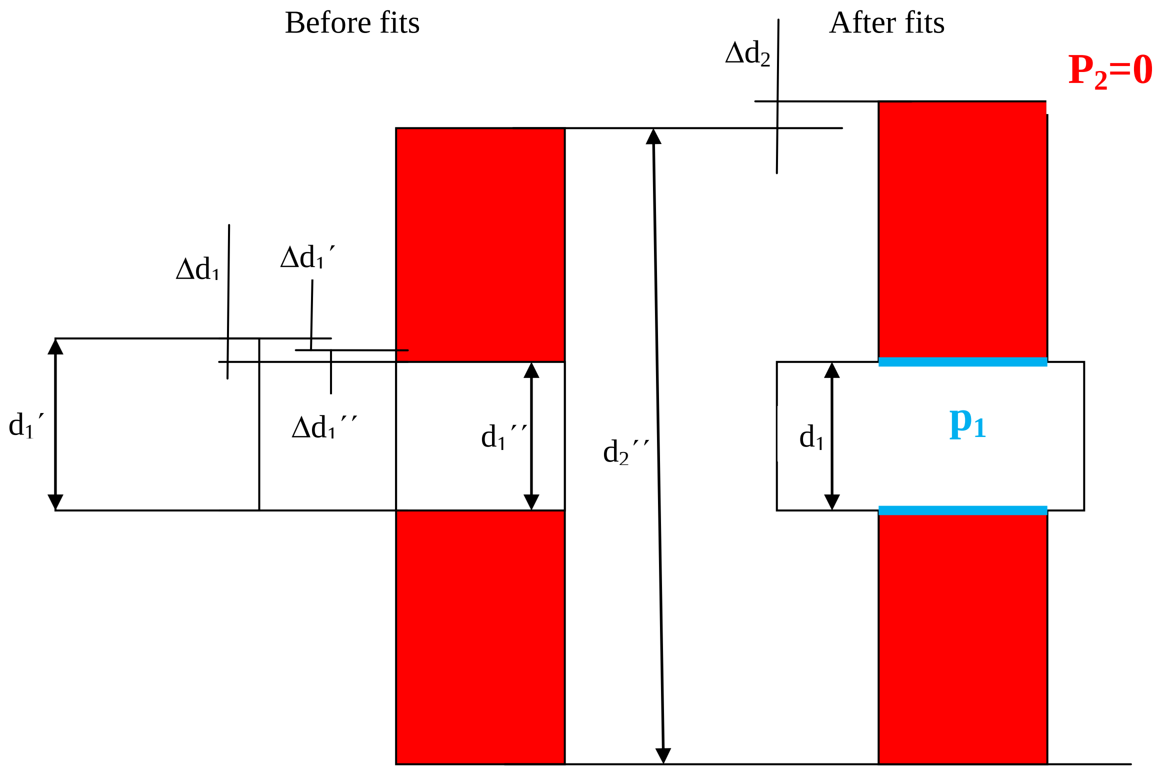
Locational interference fit: for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements - H7/p6

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

Stress and Torque Capacity in Interference Fits

Interference fits between a shaft and its components can sometimes be used effectively to minimize the need for shoulders and keyways. The stresses due to an interference fit can be obtained by treating the shaft as a cylinder with a uniform external pressure, and the hub as a hollow cylinder with a uniform internal pressure.

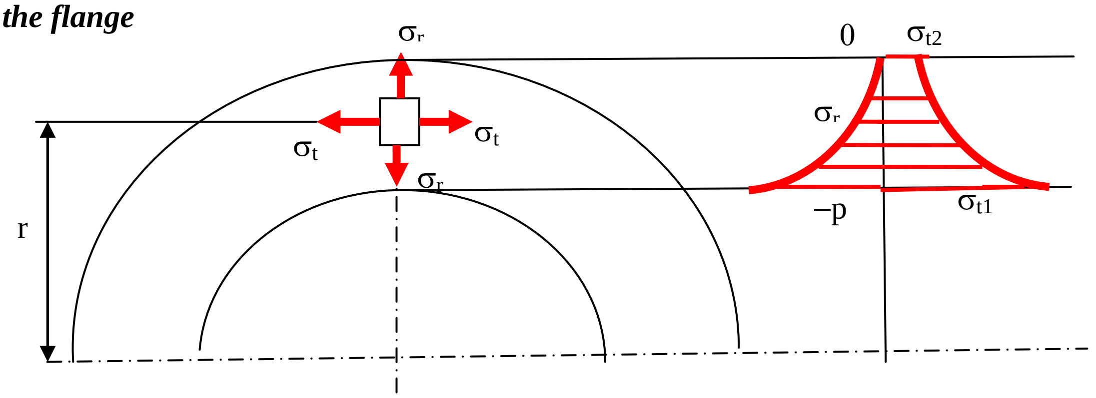


Calculate the Interference

Total tolerance

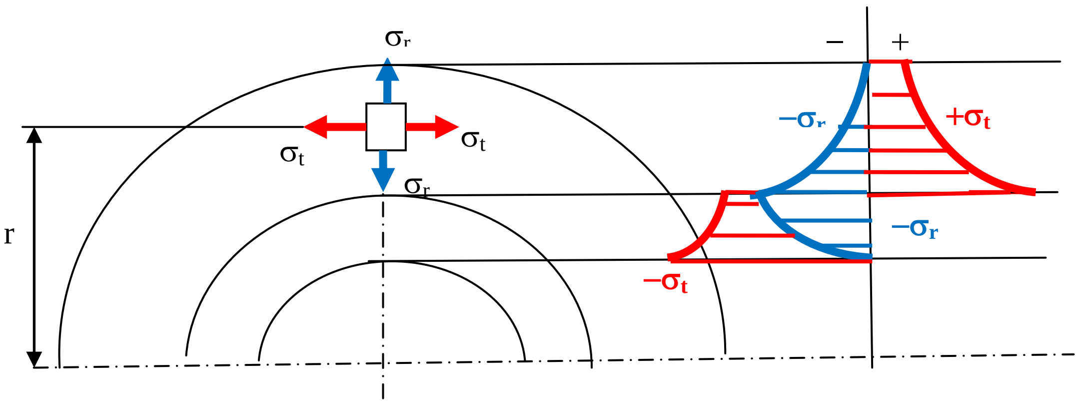
Deformations of flange

Stress in the flange



Basic conditions for stress in fits
Radial Stress

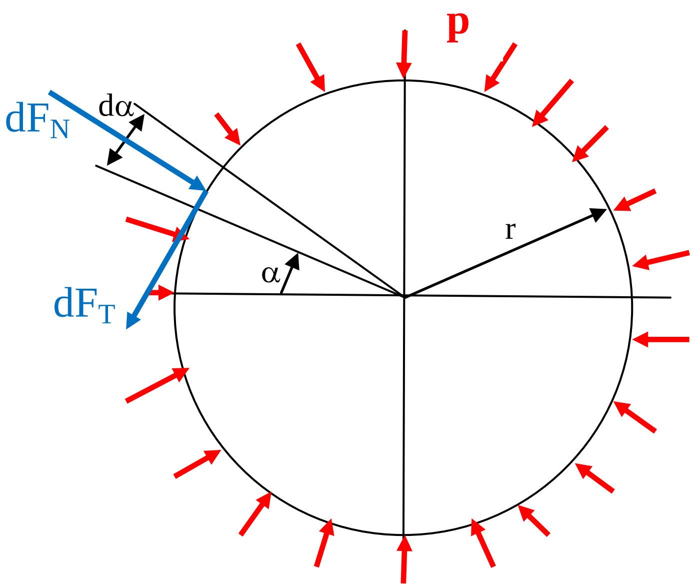
Stress in the flange and hollow shaft



d_0 – minimal diameter of hollow shaft
d1 – maximal diameter of shaft = diameter of hole of flange
d2 – maximal diameter of flange

Constant´s of fits of hollow shaft

Torque Capacity in Interference Fits



Basic condition

$$M_k \leq M_T$$

M_k – load torque [Nm]
M_T – frictions torque [Nm]

***Montage – Assembly of* Fits**

The assembly of the fits joint can be done by cold pressing, hot pressing or subcooling pressing.

Cold pressing

The journal of shaft is pushed into the hole of the pressed part by a mechanical press. Relatively large pressing forces are required for the assembly of such a connection, and therefore this method is used for smaller diameters and smaller overlaps.

The montage force:

If:

Δd_z – interference of surface – this interference is roughness dependent,

$$\Delta d_z = (R_{zh}+R_{zd})/2$$

R_{zh}, R_{zd} – the minimal and maximal value of roughness

Hot pressing

The temperature of the shaft will correspond to the ambient temperature "t_o" during pressing. To ensure pressing, the hole of the pressed part is heated to the mounting temperature "t_n" so as to reduce the pressing force.

The montaging temperature:

Montage clearance: $v = (0{,}2\text{-}0{,}4)10^{-3}.\ \sqrt{d}$ - mm

α_n – coefficient of temperature deformations, for steel – $1{,}1.10^{-5}K^{-1}$

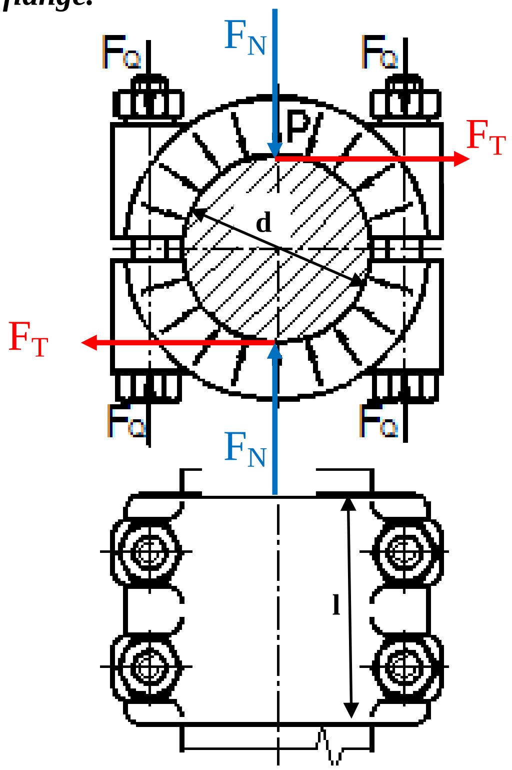
The subcooling pressing

The analogy of this pressing method is similar to the previous case, except that the shaft is subcooled.

The assembling temperature:

Design of clam for high pressure connections

A. The clam with divided flange.



A1. The rigid type of divided flange

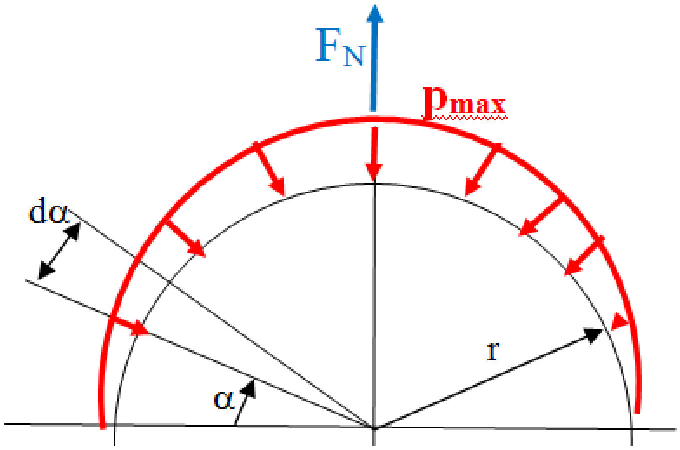
Pressure **p= const.**

Basic condition

Forces in screw

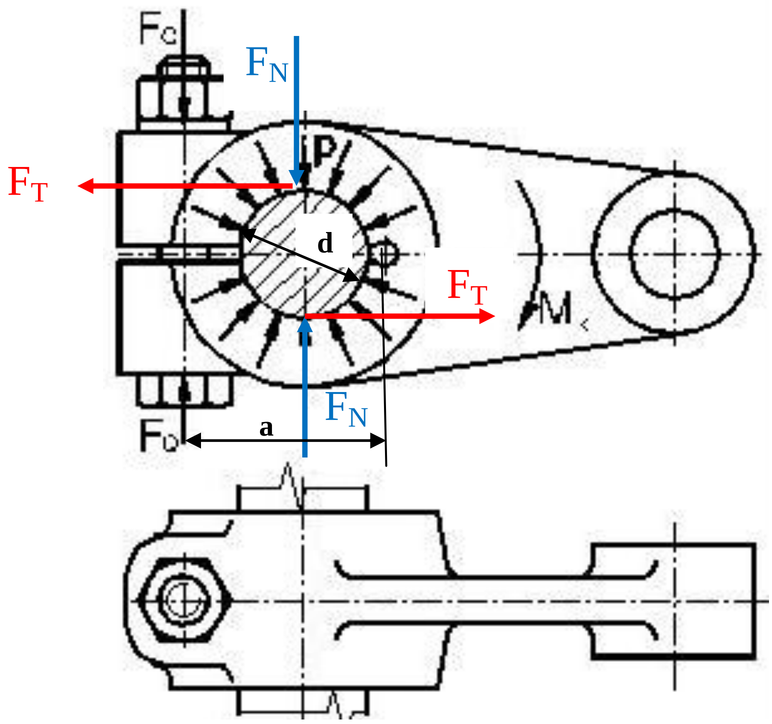
A2. The flexible type of divided flange

Pressure $\mathbf{p = p_{max} . cos\alpha}$



Basic condition

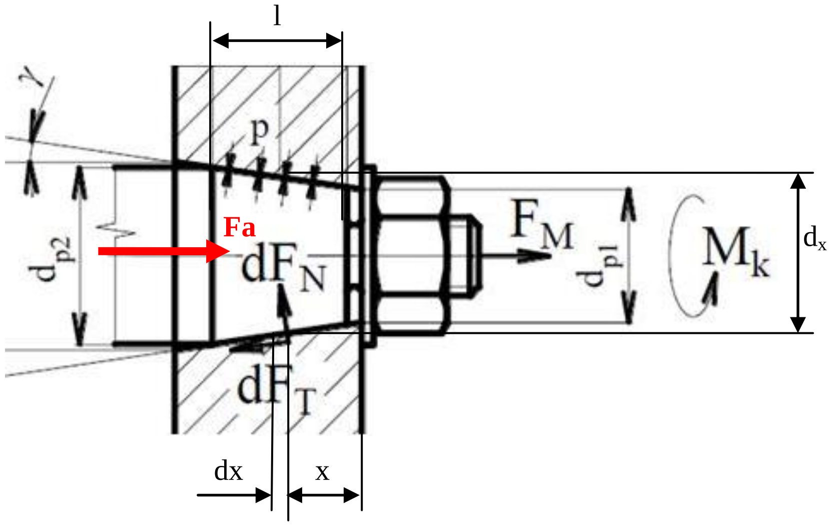
B. The clam with cutting flange



Basic condition

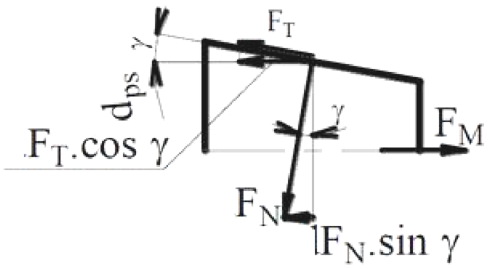
C. Conical connections

The conical connections of the shaft with the hub are perceived as frictional connections. The amount of friction of the abutment of the bearing conical surfaces in which the specific pressure is generated in the joint by means of the axial force "Fa". This axial force can be created by the mounting force "F_M", namely: tensile force in the nut or in the fastening screw, or in another way depending on the method of conical connection of the shaft to the hub.



We assume contact along the entire length of the conical joint, then we can express the normal force:

From the condition of balance of forces when we replace the effect of pressure with the axial force in the joint it follows:



Montague axial force:

If the joint transmits torque, then the friction torque in the joint must be greater than the load torque. For the friction moment we write the condition of load capacity:

The recommended conicity K is 1: 5 to 1:10. A lower value of conicity increases the demands on the release force and the method of disassembly. It is usually chosen: ***tanγ = 1:2K.***

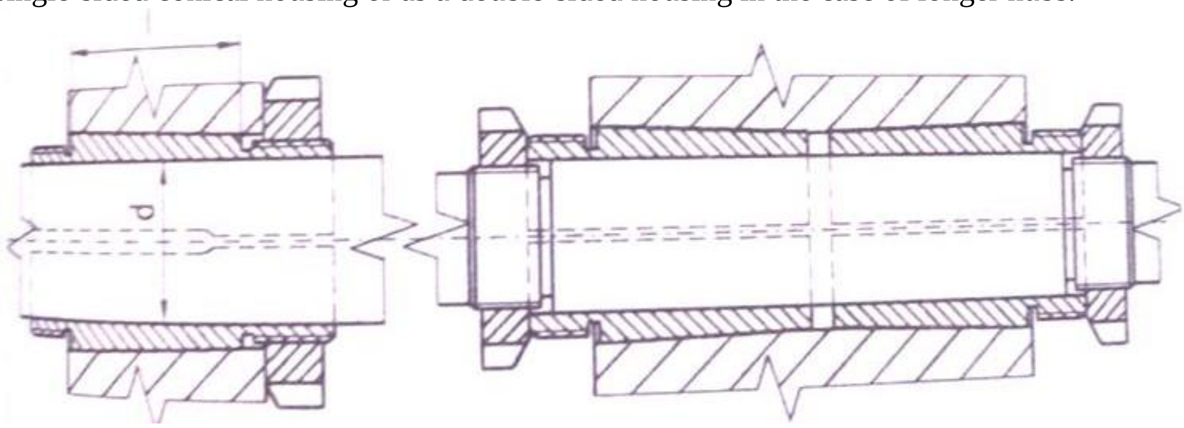
Press of conical connections:

If:
n – safety coefficient, usually n=2
p_D – allowable stress

It applies to the transmission of axial force by means of a conical joint:

D. Conical and tapered locking elements

Tapered clamping sleeves or tapered clamping or spacer rings are widely used for their simple assembly procedure. The embodiment can be made according to FIG. As a single-sided conical housing or as a double-sided housing in the case of longer hubs.

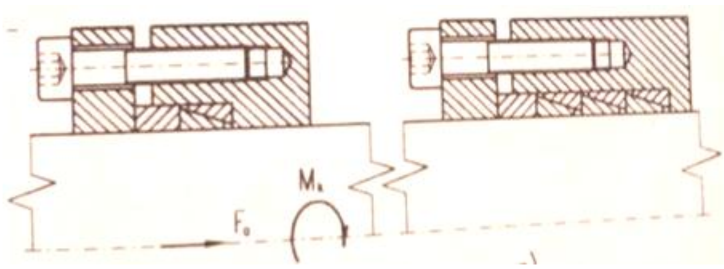


The taper of the bushings ranges from 1:10 to 1:15 and tolerance H8/j7 mounting is recommended.

The transmitted torque can be expressed:

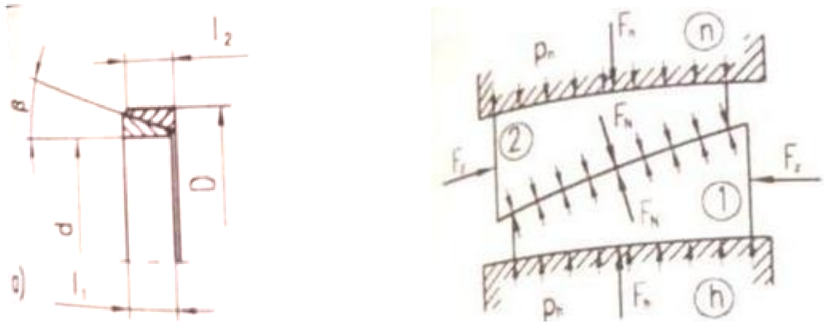
if:
n – safety parameter, usually n=2
d – diameter of shaft
l – length of sleeves

For the transmission of large torques which have an alternating character, it is suitable to use spacer clamping rings whose surfaces are conical.



The generally is conical angle β more 15°.

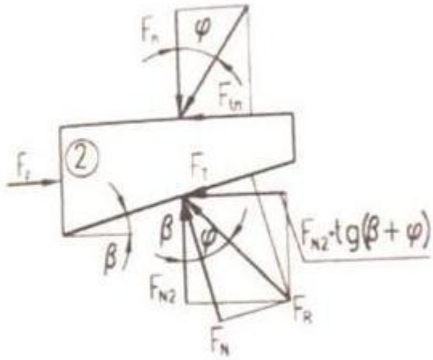
Force analyzes



Force equilibrium:

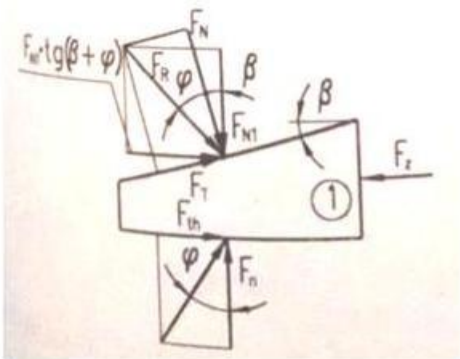
$$F_n = F_h = F_{N1} = F_{N2}$$

Forces analyzes of ring no.2



Force equilibrium in horizontal plane of ring no.2:

Forces analyzes of ring no.1:



Force equilibrium of ring no.1:

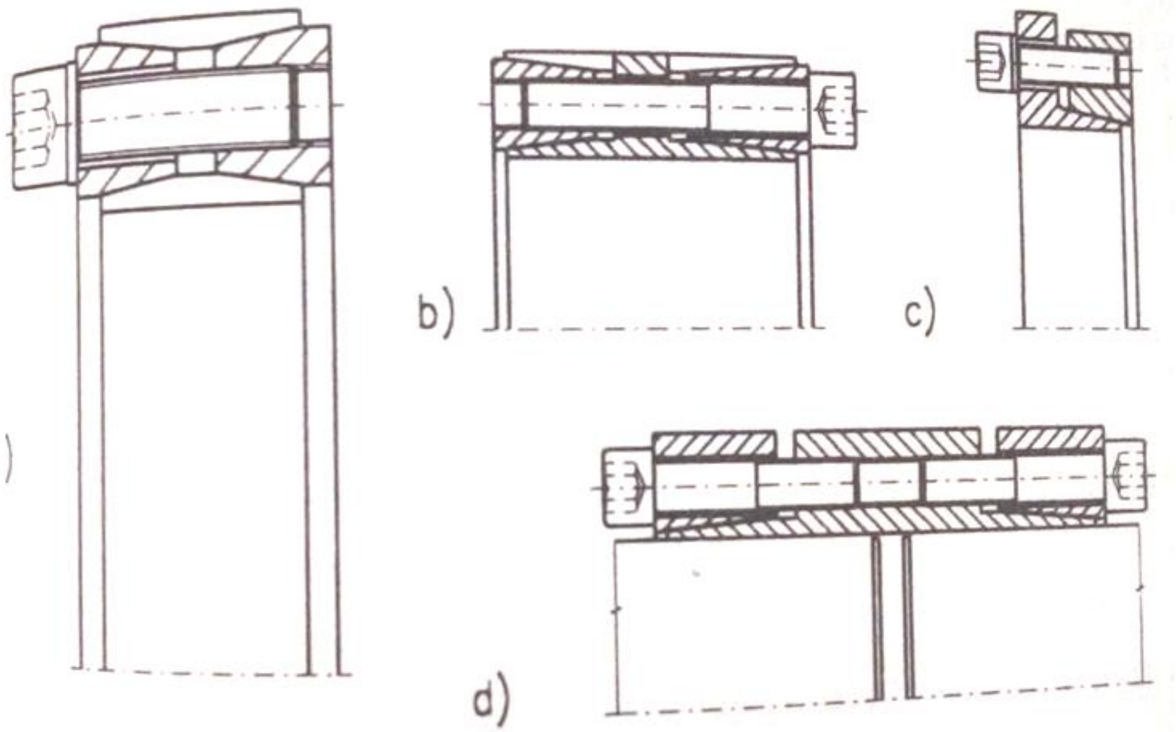
Press between inner ring no.1 and shaft:

Press between outer ring no.2 and hub:

p_D – 210MPa shaft – ring
p_D – 90MPa ring – hub; steel – steel

Practical applications of tapered rings

It is possible to use several pairs of rings for heavy load transfer, a typical example being rings of the Locking Elements Ringfeder type fig. However, a maximum of 4 rings are used.



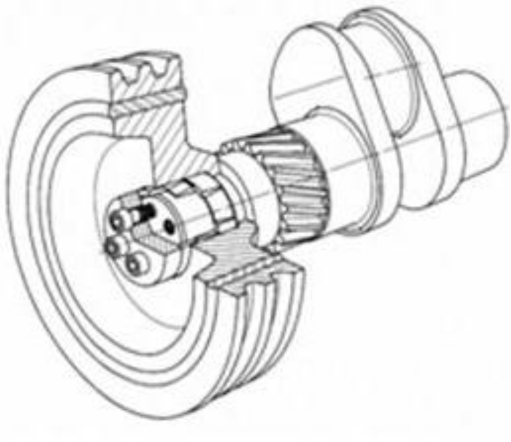
Then the load on the pairs of rings will be distributed according to the geometric series as follows:

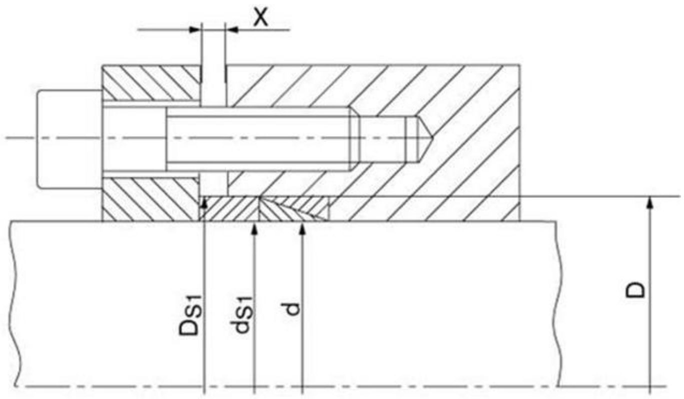
$$M_{kn} = M_{k1} + M_{k1}.q^1 + M_{k1}.q^2 + M_{k1}.q^3 \ldots + M_{k1}.q^{n-1}$$

$$q = \frac{tan\beta}{2.f + \tan\beta}$$

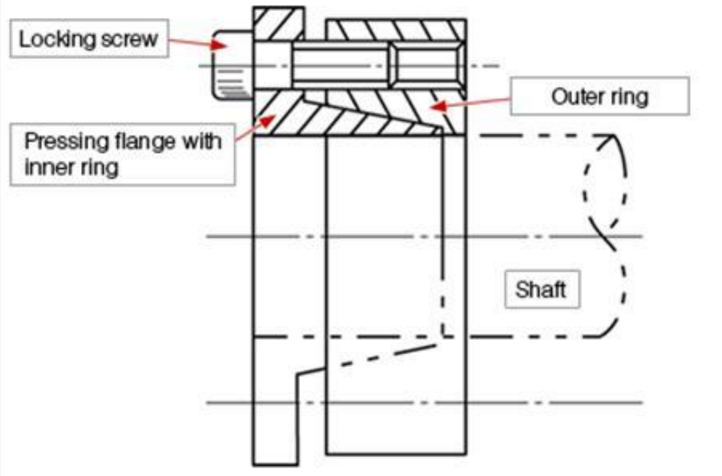
For the ring β = 16°42´a the coefficient of friction f = 0,15 will be q = 0,5, then for the application of four pairs of rings we get:

$$M_{k4} = M_{k1} + \frac{1}{2}.M_{k1} + \frac{1}{4}M_{k1} + \frac{1}{8}M_{k1}$$



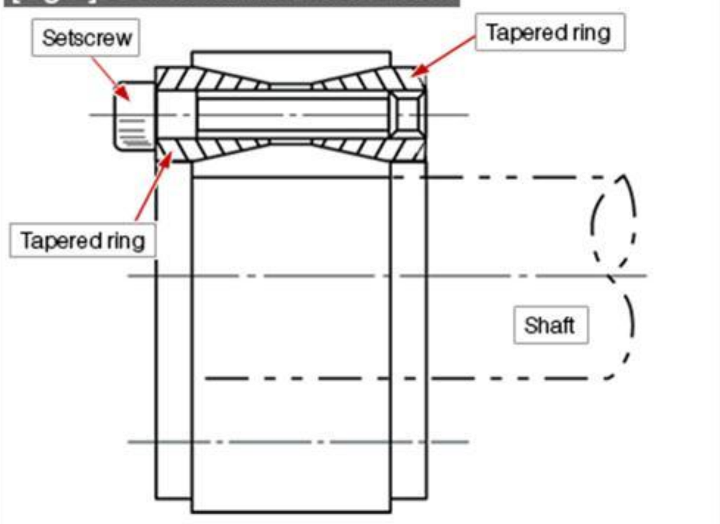


[Fig. 1] Single conical Mecha-lock



[Photo 1]

[Fig. 2] Double conical Mecha-lock



[Photo 2]