1 — 12

TEMPORARY (DETACHABLE) JOINTS

Introduction

Detachable joints have the task to allow for the disassembly of machine parts without causing any damage to the parts concerned. The various types of detachable joints are the bolts and pins, retaining rings, keys, splines, threaded bolts and nuts etc.

Note: (1) It is difficult to differentiate between bolt and pin. The function only will tell.
(2) Threaded bolts and nuts will be dealt separately in another topic.

Classification:

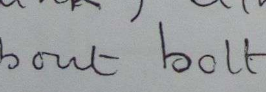
Detachable joints are classified as:

1. Form Locking elements :- e.g. bolts, pins, splines, retaining rings.
2. Force Locking elements : e.g. Threaded bolts and nuts, and force and shrink fits.
3. Mixed type elements : e.g. keys.

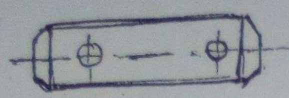
1. Bolts

Are used for the connection of two components ('link') either both or only one being rotary about bolt axis. It is the oldest element used

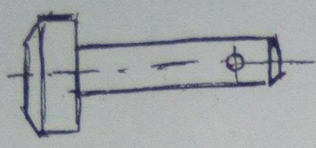
(i) Types



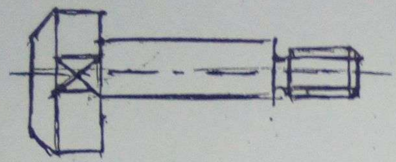
Straight bolt. May be hollow.
$d_i \leq 0.68\,d$



Straight bolt with pin ho



Bolt with head and hole



Clamp bolt

(ii) Materials

St 50, St60, C35 for general purpose

C56, 15Cr3, 16 MnCr5 for special purpose

(iii) Allowable Stresses

(a) Bearing stresses ($\sigma_{b\,all}$)

For bolts in linkages and in bushings

$\sigma_{b\,all} \simeq 10 \ldots 14 \ N/mm^2$ in steel bushings, ground, hardened or inhardened

$\sigma_{b\,all} \simeq 10 \ N/mm^2$ in bronze bushings

$\sigma_{b\,all} = 5 \ N/mm^2$ in Cast-Iron

(b) Bending Stresses ($\sigma_{B\,all}$)

$$\sigma_{B\,all} = 80 \ N/mm^2$$

(c) Shearing stresses (τ_{all})

$$\tau_{all} = 60 \ N/mm^2$$

(iv) Applications

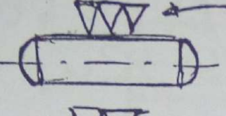
Used as axles, as pivote pins in linkages, knuckle joints, piston pins, security pins

2. PINS

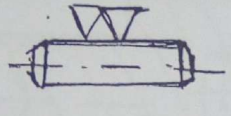
Are used for connecting, fastening, holding, centering, securing etc of machine parts.

(i) Types

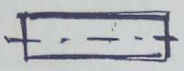
(a) Cylindrical pins


very smooth

Centering pin.

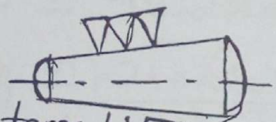


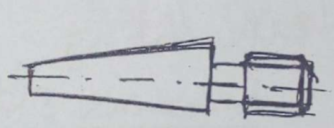
Coupling pin



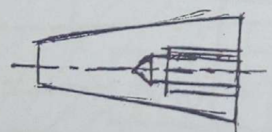
Drifting (punching) pin

(b) Taper pins


taper 1:50.

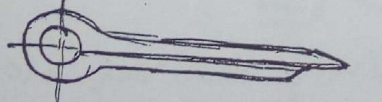


Taper pin with thread

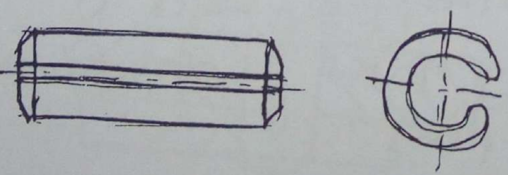


taper pin with internal thread.

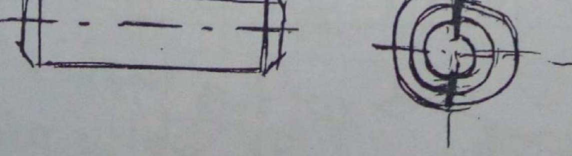
(c) Others: SPRING PINS



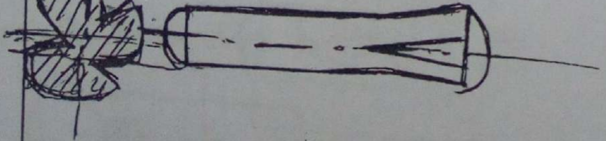
Split pin



Slotted tubular Spring pin



- Spiral wrapped pin



Grooved pins [in various forms.]

(ii) **Materials**

Cylindrical and taper pins : St 50 k, 9S.

Slotted pins : 45 S 20 k, MS 60 Pb

Surface and bore copper plated or nickel plated or similar (against corrosion)

Slotted spring pins : Spring steel 55 Si 7

Split pins : St, MS, Cu, Al.

(iii) **Allowable Stresses**

(a) **Bearing stresses** (σ_{ball})

Static $\sigma_{ball} \simeq 0.3\, S_{ut}$

Reversed fatigue $\sigma_{ball} \simeq 0.2\, S_{ut}$

Alternating " $\sigma_{ball} \simeq 0.15\, S_{ut}$

(b) **Bending stresses** (σ_{Ball})

Static ———— $\sigma_{Ball} = 0.25\, S_u$

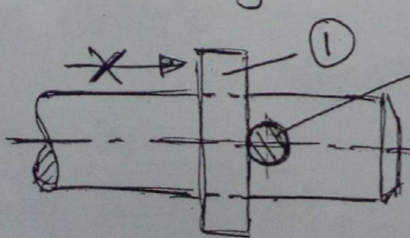
Reversed fatigue —— $\sigma_{Ball} = 0.15\, S$

Alternating " —— $\sigma_{Ball} = 0.12\, S$

(iv) **Applications**

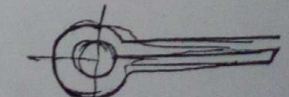
1. Securing against axial motion
2. Shear pins (or bolts)
3. Locating pins (dowel pins)
4. Safety pins

1. **Securing against axial motion**

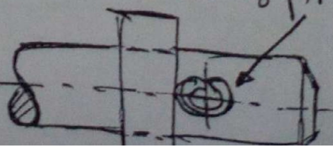


① Washer

② Split pin



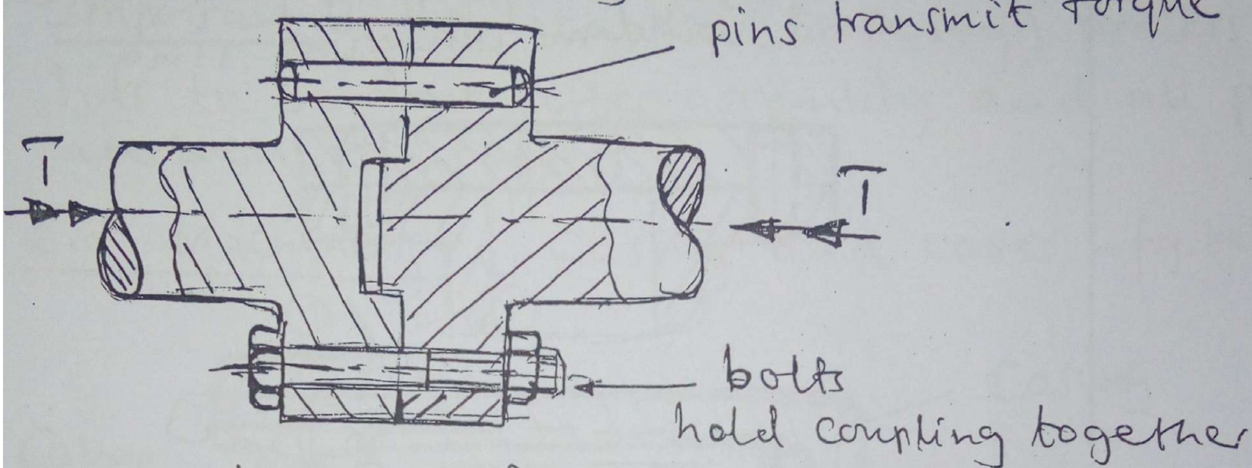
or slotted tubular spring pin



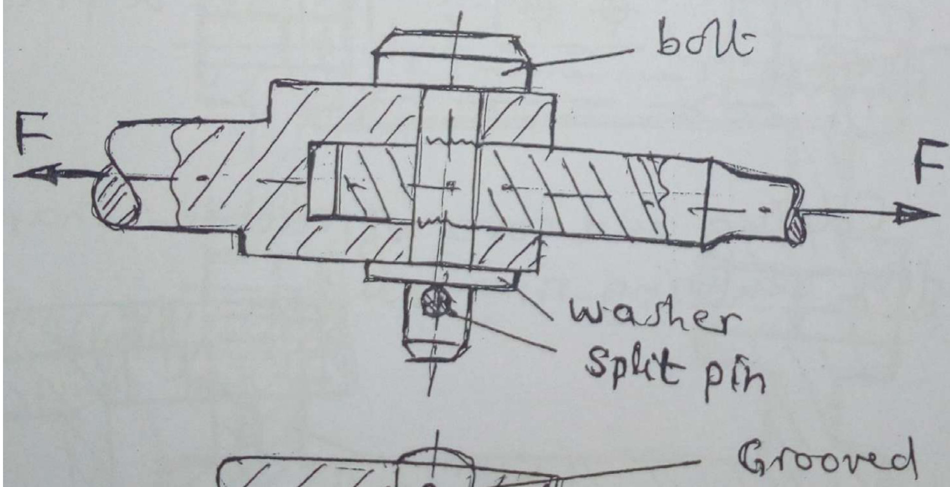
2. Shear pins

All pins (bolts) mentioned can be used to transmit shear forces in a large number of varieties.

e.g. Shaft coupling



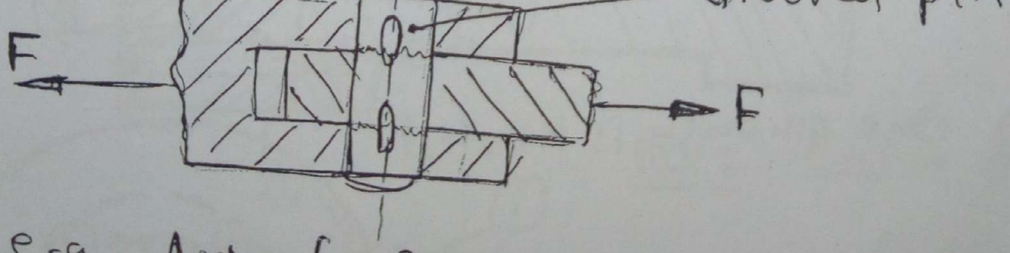
e.g. Knuckle joint



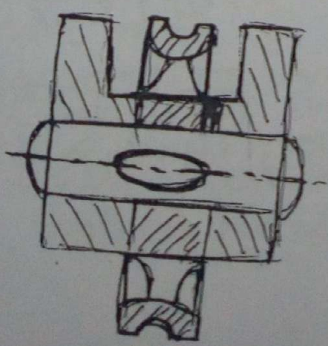
Analysis

(a) Shear

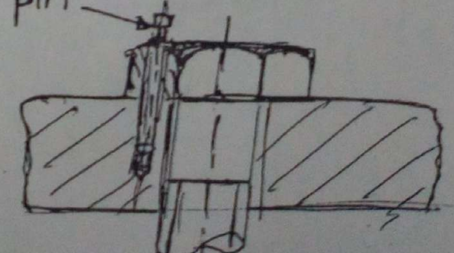
(b) Bearing



e.g. Axle for small pulley



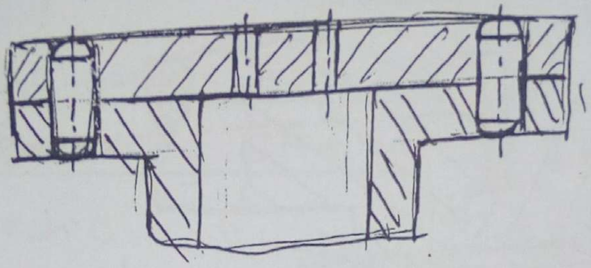
e.g. Locking of a nut

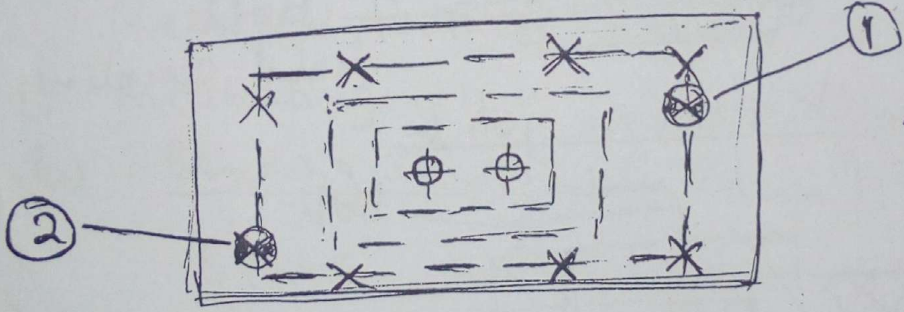


3. Locating pins

Locating means to fix the positions of two mating parts relatively to each other; after disassembling they should have the same position after reassembling them again.

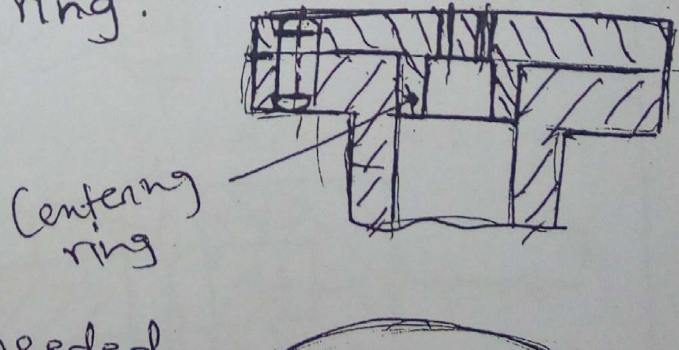
e.g. Casing and cover of non-circular shap and with no centering ring.



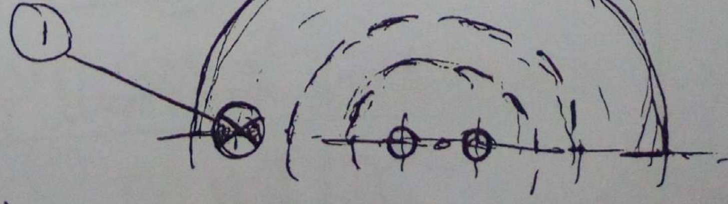


2 pins needed

e.g. Casing and cover of circular shape, cover with centering ring.

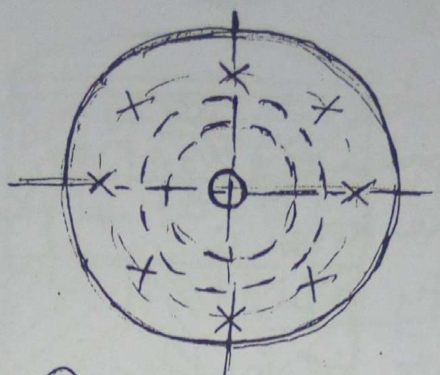


One pin only is needed



e.g. Casing and cover of circular shape, cover with centering ring and a feature in centr

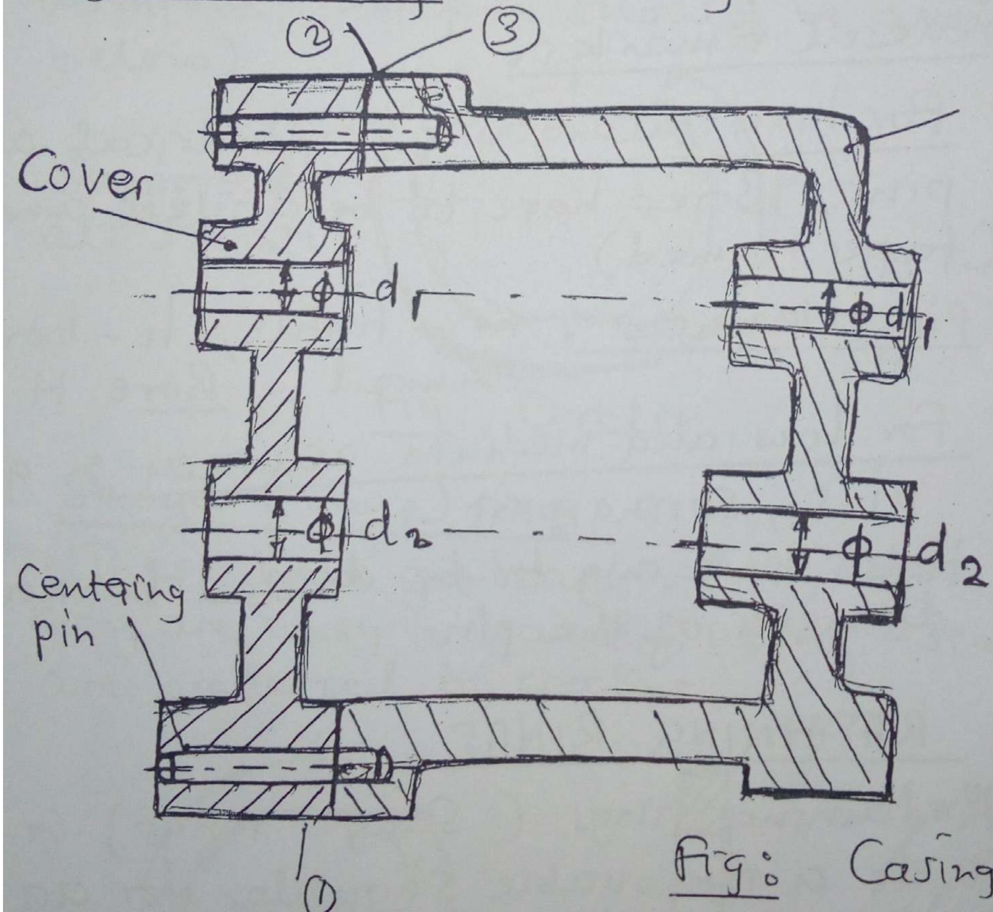
(4)



No centering pin needed.

<u>Important:</u> The locating (centering) process has to be done after assembly and all parts are well adjusted.

<u>e.g. Machining</u> (Casing and cover – fig. below).



<u>Fig:</u> Casing and cover

<u>Operation sequence</u>

① Mill base ② Mill face of cover ③ Mill case face.

④ Drill holes in cover and case $\phi < d_1, d_2$

⑤ Bolt case and cover together through drilled holes of d_1 and d_2.

⑥ Drill holes for centering pins and pins.

⑦ Drill holes for shaft 1 and 2, and mach to size ϕd_1, ϕd_2.

The holes in cover and casing will then be well aligned.

4. Safety pins

All sorts of pins can be used as safety pins. These safety pins have a predetermined breakin point, being the weakest part in an assembly.

Purpose: to protect more expensive parts from overload.

General remarks:

i) For high accuracy: cylindrical and taper pins. Bores have to be drilled and reamed (taper reamed).

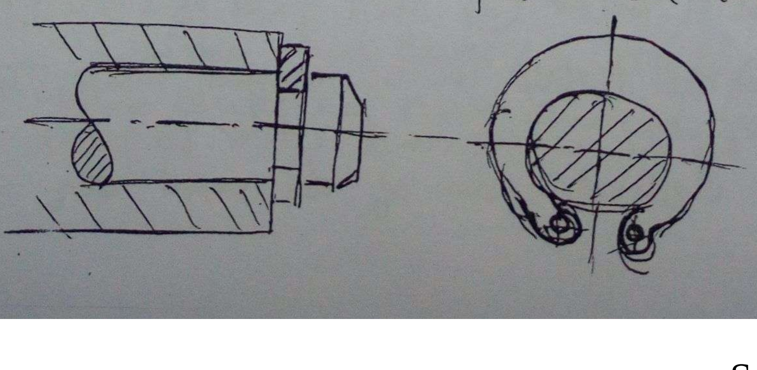
Pin tolerances: m – tight, h – bore often reamed, Bore H

ii) For low and medium accuracy: grooved pins, spring pins.

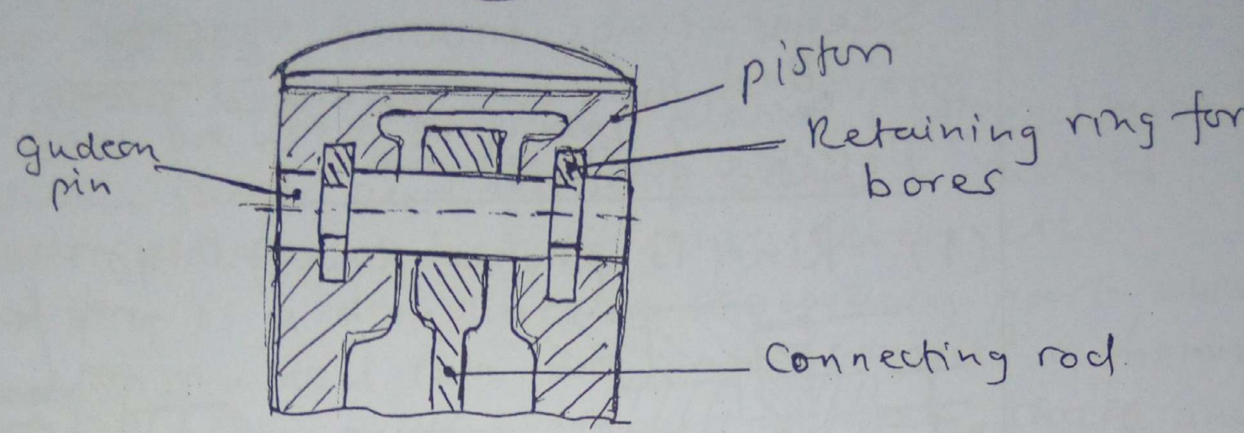
Holes have only to be drilled to nominal diameter of the pin.

3. RETAINING RINGS

Retaining rings (Seeger rings) or circlips provide a removable shoulder for accurately locating, retaining, and locking components on shafts and in bores.



Retaining ring for shafts



Circlips

Are the type of retaining rings which have no 'ears'. They are difficult to mount and dismount, therefore should be avoided. (fig. below).

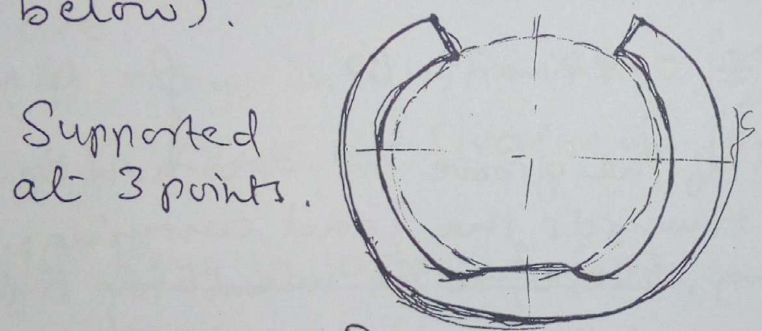


fig. Circlip.

Seeger (Snap) rings

These have got 'ears'. (fig. below). Therefore they are easy to handle using nose pliers. They are preferred to circlips.

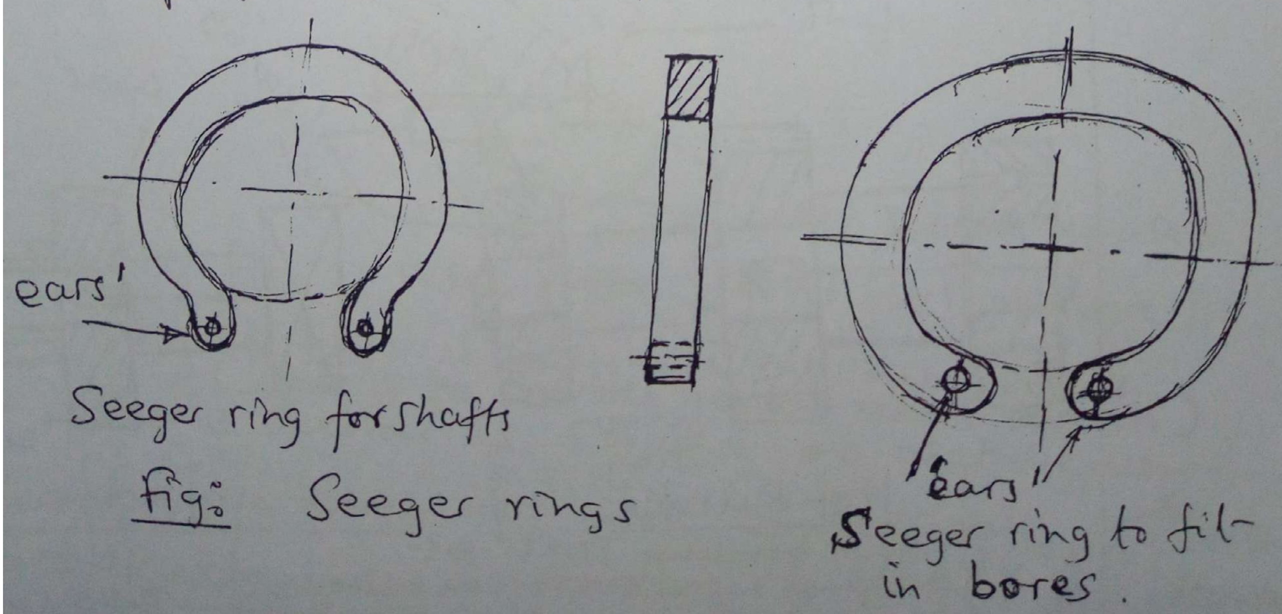


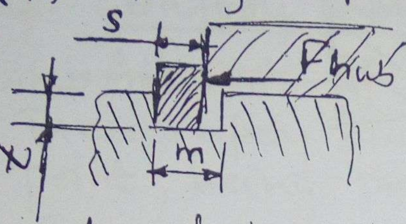
fig: Seeger rings

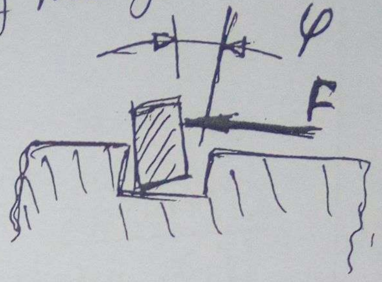
Design factors

Seeger-rings transmit very high axial loa[d] provided they are mounted properly.

Possible failures are:

(1) Ring is pushed out of the groove





t = depth of groove
m = width of groove , s = thickness of ring

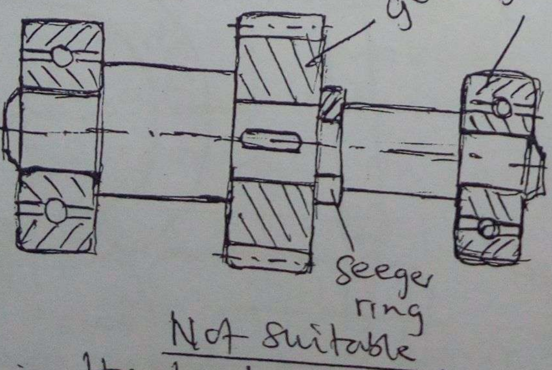
$\therefore \; t/s = 0.17 \dots 0.67$ for DIN 471/7[2]

and $t/s = 0.87 \dots 1.7$ for DIN 6799

The width of the groove 'm' does not contribute anything towards the load carrying capacity of the ring. Therefore no need for tight groove.

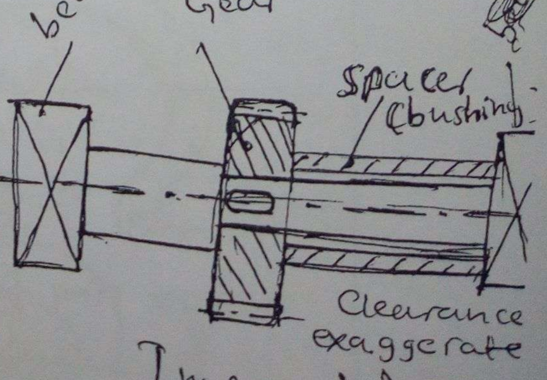
(2) At higher speeds, the ring may come out of the groove due to inertia forces.

(3). Groove is sharp edged, therefore causes notch effect; try to avoid using seeger ring at points of high stresses.



Not suitable

i.e. Higher bending stress at centre and more stress concentration due to groove of ring



Improved Design

No groove
$\therefore$ No stress concentration

4. KEYS

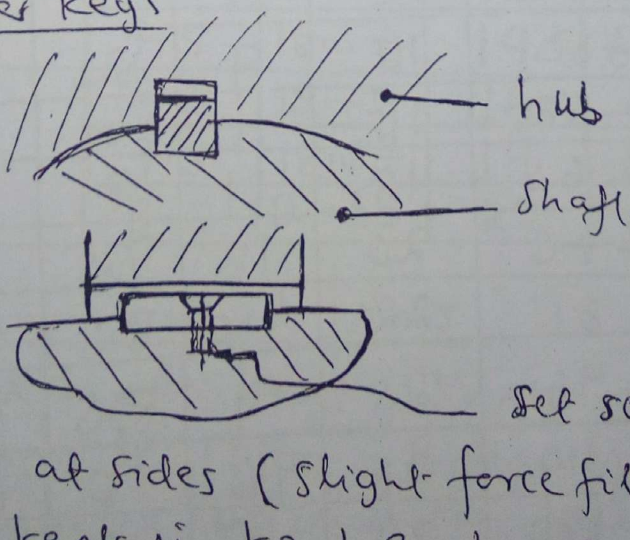
Keys are used to prevent relative (rotary) motion between shafts and machine elements such as gears, pulleys, sprockets, cams, levers etc.
A key is a demountable machinery part, which when assembled into keyseats, provides a positive means of transmitting torque between shaft and hub.

(i) Types of Key Joints

Generally there are two types of key joints. These are:
(1) Keys in unstrained joints
(2) Keys in strained joints.

(1) Keys in unstrained joints – These are feather keys and woodruff keys. They are form locking.

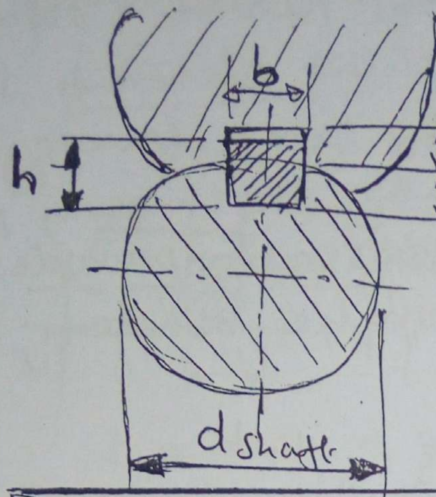
(a) Feather keys



May fit at sides (slight force fit) or loose fit, may be kept in keyhole by set-screws. (If it is to provide relative axial motion)
Cross section is square or rectangular
Square keys for dsh ≤ 160 mm
Rectangular for dsh > ϕ 160 mm
* In other standards

Key size : $b \simeq 0.25\, d_{sh}$ for square k

Dimensioning



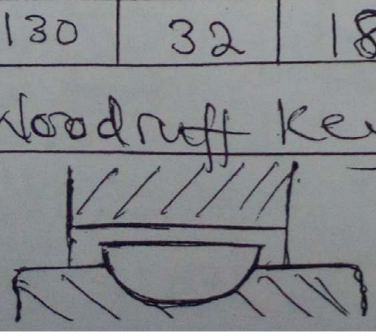
t_2 = keyway depth in hub
t_1 = keyway depth in shaft

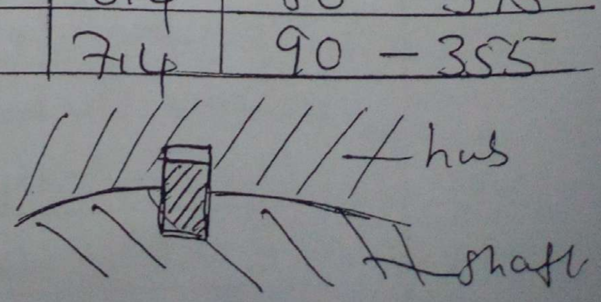
Specified as $b \times h \times l$

where l = length of key

d	b	h	t_1	t_2	Length l
6 – 8	2	2	1.2	1	6 – 20
8 – 10	3	3	2	1.3	6 – 36
10 – 12	4	4	2.5	1.8	8 – 45
12 – 17	5	5	3	2.3	10 – 56
17 – 22	6	6	3.5	2.8	14 – 70
22 – 30	8	7	4	3.3	18 – 90
30 – 38	10	8	5	3.3	22 – 110
38 – 44	12	8	5	3.3	28 – 140
44 – 50	14	9	5.5	3.8	36 – 160
50 – 58	16	10	6	4.3	45 – 180
58 – 65	18	11	7	4.4	50 – 200
65 – 75	20	12	7.5	4.9	56 – 220
75 – 85	22	14	8.5	5.9	63 – 250
85 – 95	25	14	8.5	5.9	70 – 280
95 – 110	28	16	10	6.4	80 – 315
110 – 130	32	18	11	7.4	90 – 355

(b) Woodruff Key





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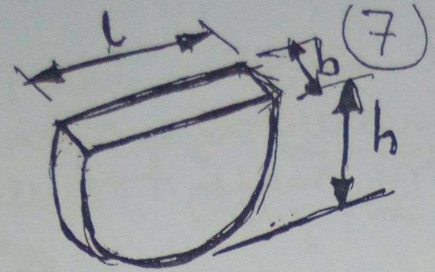
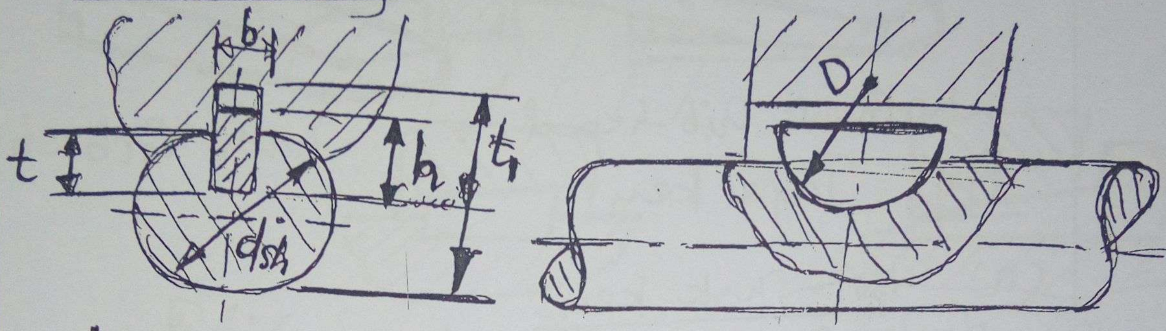


They are used for small torques only.

<u>Dimensioning</u>



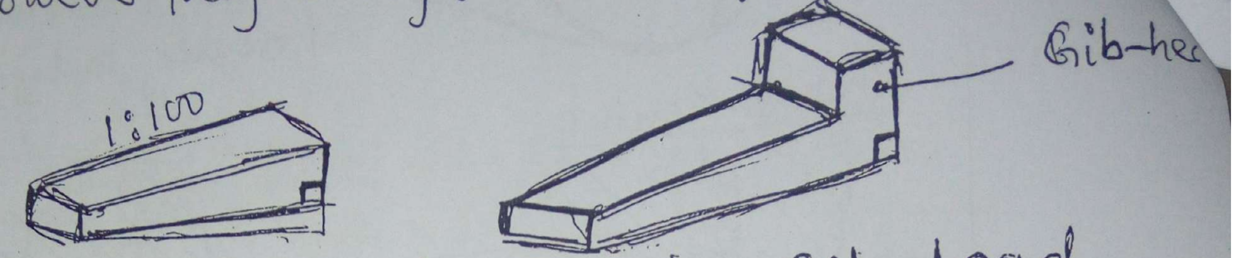
Dimensioned as <u>b x h</u>

d shaft	b	h	l	D	t	t_1
6 – 8	2	2.6	6.76	7	1.8	d + 1
		3.7	9.66	10	2.9	
8 – 10	3	3.7	9.66	10	2.5	d + 1.4
		5.0	12.65	13	3.8	
		6.5	15.72	16	5.3	
10 – 12	4	5.0	12.65	13	3.5	d + 1.7
		6.5	15.72	16	5.0	
		7.5	18.57	19	6.0	
12 – 17	5	6.5	15.72	16	4.5	d + 2.2
		7.5	18.57	19	5.5	
		8.0	21.63	22	6.0	

(2) <u>Keys in Strained joints</u>

There are taper keys, sunk or Not-sunk keys. Taper is 1:100. Bottom of shaft not tapered only the hub and the key hub face. The <u>sunk-keys</u> can be with or without a gib-head. (Fig. below).

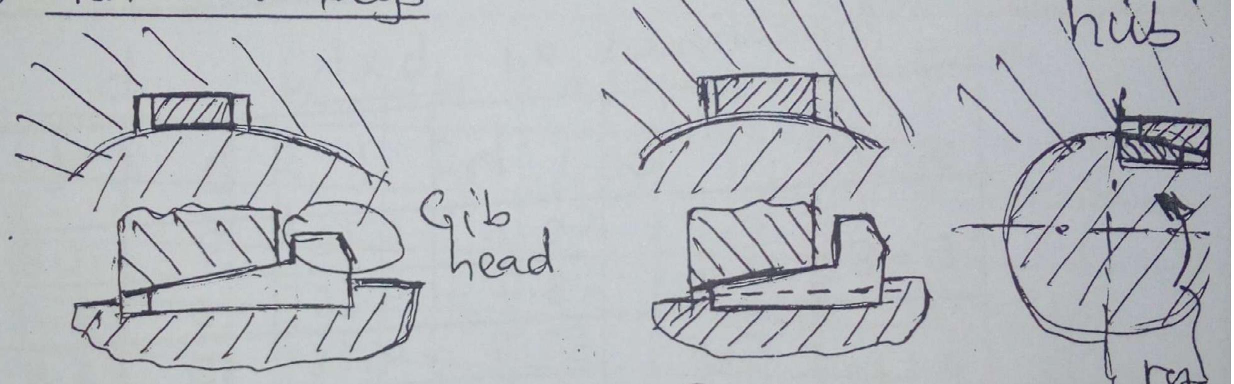
The <u>Not-sunk keys</u> are Flat, Saddle and tangential keys. They sit straight on shaft. Hence advantage of no keyway. However they carry small torques.



without Gib-head   with Gib-head

Fig: Taper keys

(a) Not-sunk keys



Flat   Saddle   eg Tangent

Taper 1:100

NB Direction of rotation is important.

Fig: Not-Sunk keys

(b) Sunk - Keys

Are most widely used and standardized. Taper is 1:100. American standards provide

Square keys: for $d_{shaft} \leq \phi 160$ mm

and

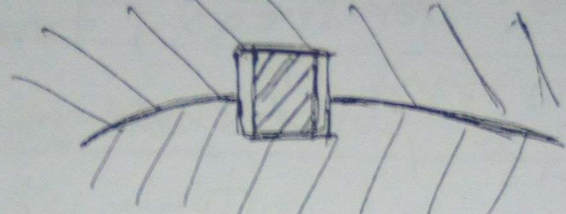
Rectangular keys: for $d_{sh} > 160$ mm dia

Key size: $b \simeq 0.25\, d_{shaft}$. (Square key

Can also be with or without Gib-head.

Fig. below shows a sunk-key joint

Keyway at the bottom of the shaft is not tapered. Only the hub keyway is tapered.



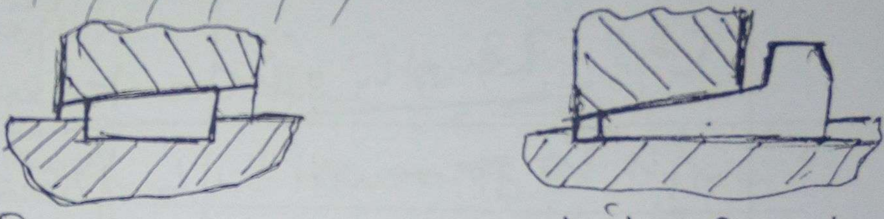
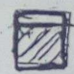
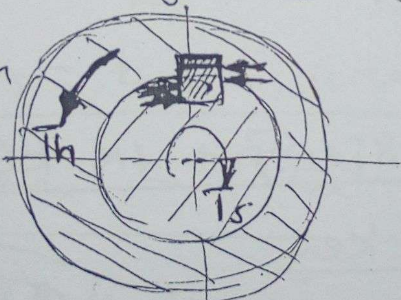


Fig: Sunk-key   with Gib-head

(ii) Transmission of Load

(a) Unstrained joint e.g. feather key





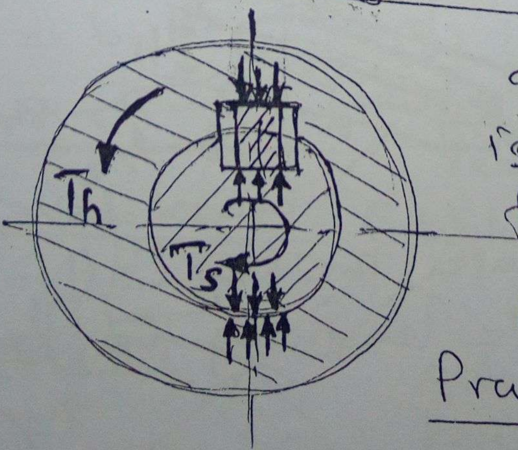
T_s = torque on shaft.

T_h = Torque on hub

$T_s = T_h = T$

Load is transmitted via lateral sides.

(b) Strained key joints



Theoretically torque is transmitted through friction only, i.e. no contact at lateral sides.

Practice: Mixed.

(iii) General design rules

(1) Factors of safety (n)

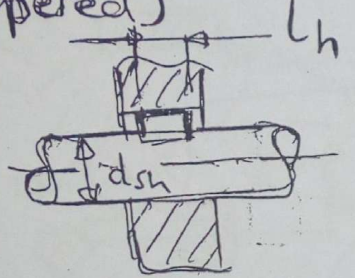
$n = 1.5$ for steady torque

$n = 2.5$ " Minor shock loads

$n = 4.5$ " High shock loads

(2) Length of the key into hub (l_h)

To prevent hub from rocking on the shaft when using straight keys (i.e. not tapered)

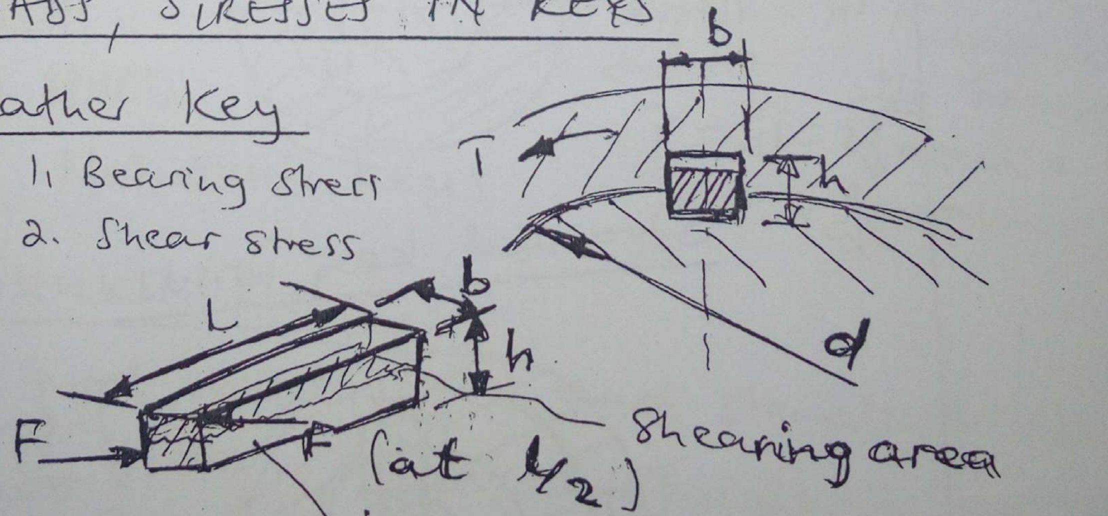


$$l_h = (1 \ldots 1.25)\, d_{shaft}$$

* (iv) LOADS, STRESSES IN KEYS

(a) Feather Key

Check: 1. Bearing stress
2. Shear stress



In most cases, the key will fail in bearing. Now if l = length of the key we have

(1) Bearing

Stress $\sigma_b = \dfrac{F}{A_b} \le \sigma_{b\,all}$; $F = \dfrac{T}{r}$, $r = \dfrac{d}{2}$

A_b = pressing area $= \frac{1}{2} h \cdot l$ of keyway

σ_{ball} = allowable bearing stress (pressure)

$\therefore$ $\boxed{T_{max} = \sigma_{ball} \cdot A_b \cdot r}$ (1) Torque Capacity in bearing.

r = shaft radius.

Allowable bearing stresses. σ_{ball}

hub/shaft material	$\sigma_{ball} = \left(\frac{S_y}{n}\right)$
Steel/Steel	100 – 150 N/mm^2
Cast Iron/Steel	$\leq$ 80 N/mm^2

(2) Shearing

Stress $\tau = \frac{F}{A_s} \leq \tau_{all}$; $F = \frac{T}{r}$

Now $A_s = b \cdot l$, area of shearing

$\therefore$ $\boxed{T_{max} = \tau_{all} \cdot A_s \cdot r}$ (2) Torque Capacity

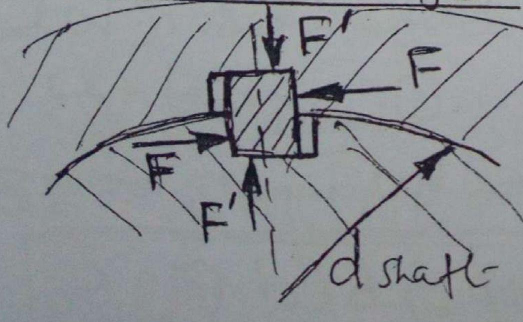
(b) Woodruff Key

Same equations above apply with

$$\boxed{\begin{aligned} A_s &= b \cdot l \\ A_b &= (h-t) \cdot l \end{aligned}}$$ (3)

hub keyway area. See dimensions.

(c) Parallel keys (tight-fit)



Neglect pretension as given by F' and design keys in the same manner as Feather keys.

(d) Taper Keys

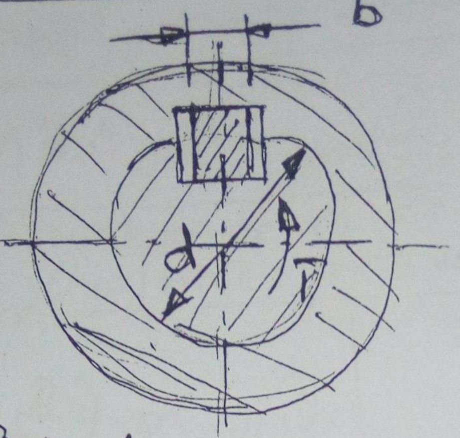


Fig: Assembly

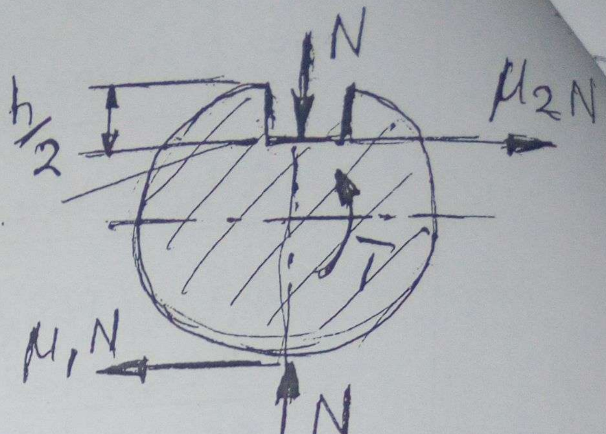
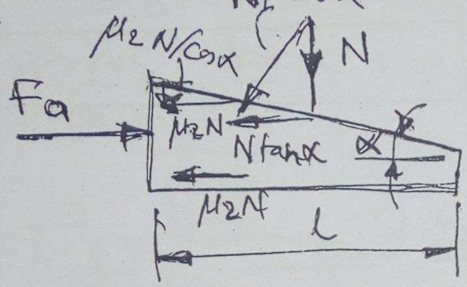


Fig: Shaft Loading



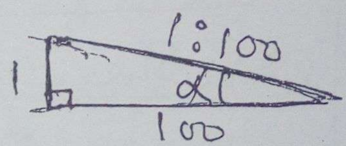


Fig: Key loading

(1) Torque transmitted (T)

From shaft loading

$$T = \frac{1}{2}\mu_1 N d + \frac{1}{2}\mu_2 N (d - h/2)$$

neglected

$$\therefore T = \frac{1}{2} N d (\mu_1 + \mu_2)$$

where μ_1 = coefficient of friction between shaft and hub

μ_2 = coefficient of friction between key and ~~shaft~~ key seats (hub and shaft)

But $N = b \cdot l \cdot \sigma_{ball}$

where σ_{ball} = allowable bearing stress

Hence

$$\boxed{T = \frac{1}{2}(\mu_1 + \mu_2)\, b l d\, \sigma_{ball}} \quad (4)$$

From experience:

$\mu_1 = 0.25$, $\mu_2 = 0.10$ (key become greased due to corrosion)

$\therefore$ $\boxed{T = 0.175\, d\, b\, l\, \sigma_{ball}}$ (5)

(2) Axial force to drive the key (F_a)

From key loading

$F_a = 2\mu_2 N + N\tan\alpha$

or $F_a = b\, l\, \sigma_{ball}\, (2\mu_2 + \tan\alpha)$

Using $\mu_2 = 0.10$, $\tan\alpha = 0.01$ (taper 1:100)

$\therefore$ $\boxed{F_a = 0.21\, b\, l\, \sigma_{ball}}$ (6)

Note: Consider strength of hub additionally otherwise it may burst.

(V) Advantages and Disadvantages

Advantages: Generally keys are simple to assemble and disassemble. They are also reliable and inexpensive.

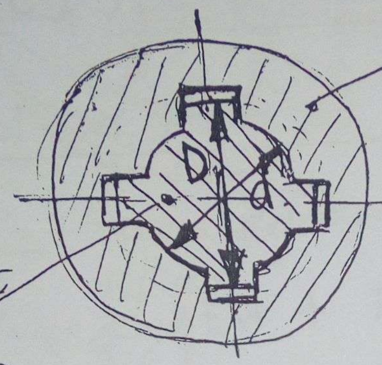
Disadvantages: The disadvantage with using keys are the strength reduction (due to notch effect) and the difficulty in centering (accuracy).

5. SPLINES

Splines are keys that form an integral part with the shaft. They also permit axial motion between shaft and hub in ... to preventing relative rotary motion.

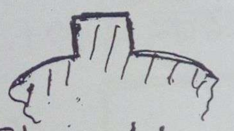
Number of splines : 4, 6, 8, 10, - - - - 20

Materials : 37Cr4, 41Cr4, 42CrMo4

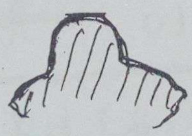


D = Major diameter
d = minor diameter

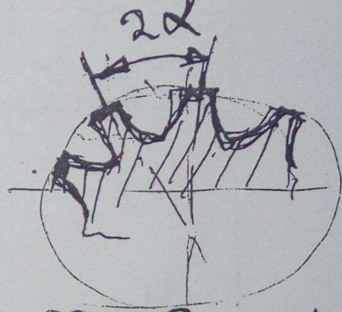
(i) Types of profiles



(1) Straight Sided Splines



(2) Involute Spline



(3) Serrated

Note: Involute are cut similar to gear teeth. They have lower stress concentration than straight-sided splines.

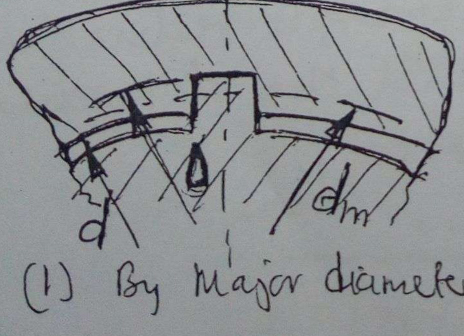
(ii) Centering (example - Straight-sided splin...

Splines are centred by : (Figs below)

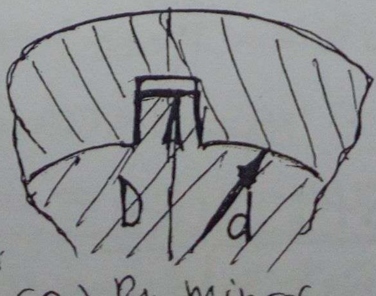
(1) Major diameter (heavy duty and in special cas...

(2) Minor diameter (for 6, 8, 10 splines)

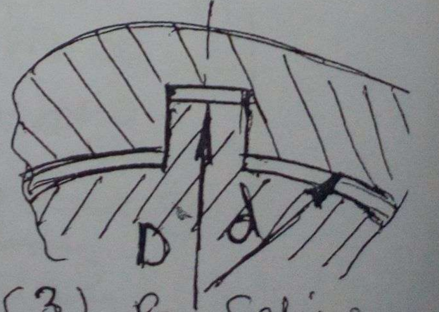
(3) Spline faces (for 8 - - - 20 splines)



(1) By Major diameter



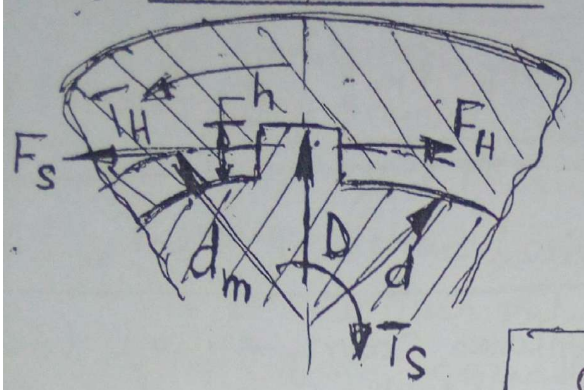
(2) By Minor Diameter



(3) By Spline faces

Figs: Centering of Splines

(iii) LOADS, STRESSES



T_H = hub torque

T_S = Shaft torque

$T_H = T_S = T$ transmitted torque.

d_m = mean diameter

$$\boxed{d_m = \frac{D+d}{2} \quad (1) \qquad d_m = mZ}$$

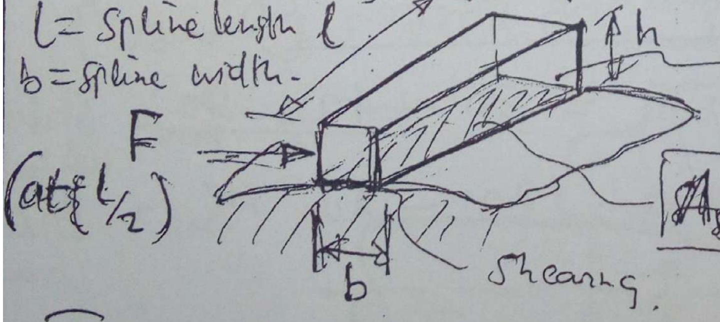
$d_m = mZ$ ← for involute splines,
m = module,
Z = no. of splines.

h = Spline depth

$$\boxed{h = \frac{D-d}{2}} \quad (2)$$

F = Transmitted force, s = shaft, h = hub

Considering the spline



$$\boxed{A_b = h \times l} \quad (3)$$

Bearing area

$$\boxed{A_s = b \cdot l} \quad (4)$$

Shearing area.

Therefore the spline is pressed on the keyway of hub (bearing failure), and may also shear at the root (Shearing) due to transverse force 'F'. Bending is neglected at the root though 'F' act at ($h/2$) since depth 'h' is small to give a big bending moment.
In general the spline is more likely to fail in bearing.

a) Bearing

Stress $\sigma_b = \frac{F}{A_b} \leq \sigma_{b\,all}$, $\sigma_{b\,all}$ = allowable bearing stress

and $F = T/r_m$, T = transmitted torque ...
r_m = mean radius = $\frac{d_m}{2}$

$\therefore \boxed{T_{max} = \sigma_{ball} \cdot A_b \cdot r_m}$ (5) for 1 spline

for N splines $\boxed{T_{N max} = \sigma_{ball} \cdot A_b \cdot r_m \cdot N}$ (5)

Generally not all splines carry equal load. (due to workmanship). Thus introducing a loading factor φ (In general $\varphi = 0.75$ — i.e. 75% of splines carry load)

$\therefore \boxed{T_{N max} = \sigma_{ball} \cdot A_b \cdot r_m \cdot N \cdot \varphi}$ (6)

Torque capacity in bearing

If all splines carry load $\therefore \varphi = 1$ or 100%.

Note: With clearance effective depth 'h' should be taken for $A_b = h \cdot l$.

Allowable bearing stresses (σ_{ball})

Steel $\sigma_{ball} = 80 - 120\ N/mm^2$ splines unhardened

$\sigma_{ball} = 120 - 200\ N/mm^2$ splines hardened.

(b) Shearing

Stress $\tau = \frac{F}{A_s} \leq \tau_{all}$, τ_{all} = allowable shear stress

with $F = T/r_m$

$\therefore \boxed{T_{N max} = \tau_{all} \cdot A_s \cdot r_m \cdot N \cdot \varphi}$ (7)

torque capacity in shearing

(iv) General Remarks

Advantages

- High torques to be transmitted
- Proper centering possible (due to accuracy)

Disadvantages

- High local stress concentrations in the recentering corners of grooves (especially straight-sided)
  NB. Still lower than keys.
- Non-uniform load distribution. General assessment is 75% of splines only carry load. NB $\varphi = 1$ if all splines carry equal load.
- Special cutting and measuring tools needed (in case of involute splines, same as for involute gear).
- Danger of distortion if hardened.

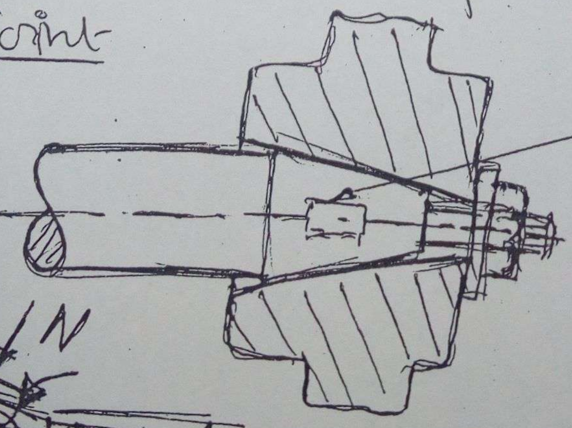
Applications

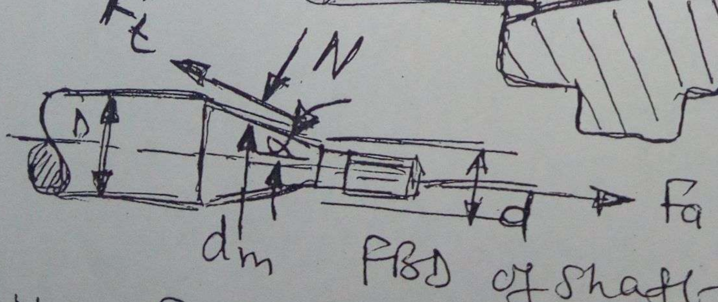
e.g. car transmission - shifting gears, coupling (gears, pulleys)

6. FRICTION LOCKING JOINTS

These are (1) Cone joints (2) Spring tension rings (3) Shrink and force fits.

(1) Cone joint





FBD of shaft

Now $F_t = \Sigma \Delta F_t$, $N = \Sigma \Delta N$

$F_a - F_t \cos\alpha - N \sin\alpha = 0$, with $F_t = \mu N$

$\therefore \boxed{T = F_t \cdot \frac{d_m}{2}}$ (1) Transmitted torque

Tightening force (F_a)

$$F_a = \frac{2T}{\mu d_m}(\mu \cos\alpha + \sin\alpha)$$

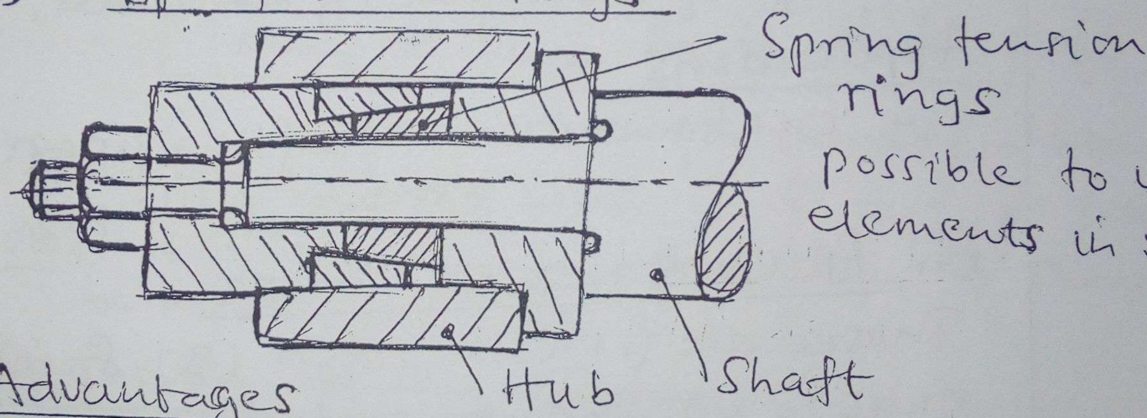
α = inclination angle, with $\mu = \tan\varphi = \frac{\sin\varphi}{\cos\varphi}$, φ = friction angle

$$F_a = \frac{2T}{\mu d_m}\left(\frac{\sin\varphi\cos\alpha + \sin\alpha\cos\varphi}{\cos\varphi \simeq 1}\right)$$

$$\therefore \boxed{F_a = \frac{2T}{\mu d_m}\sin(\alpha+\varphi)} \quad (2)$$

If a certain torque is to be transmitted, the hub has to be pressed with F_a. (use torque wrench).

(2) Spring tension rings



possible to use elements in serie

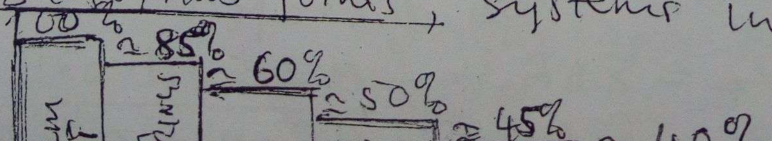
Advantages

- Minimum reduction in endurance limit
- Repeated assembly and disassembly possible without impairing reliability.
- Accurate mounting at any part of the shaft
- Less stringent tolerances (than in press fits).
- Simple adjustment of angular position.

Disadvantages

- Special parts of high precision
- Increased diameter of hub in order to accomodate rings.

7. Endurance Limit reduction factors for various Shaft/hub joints; systems in torsion



Comparison in strength for various joints