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:

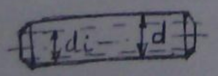
Detachable joints are classified as:

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.
$d_i \leq 0.68\,d$

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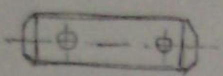
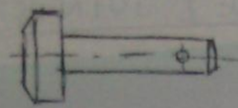
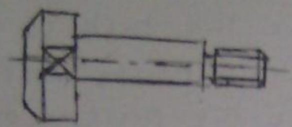


Straight bolt with pin holes



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

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

$\sigma_{ball} \simeq 10\ N/mm^2$ in bronze bushings

$\sigma_{ball} = 5\ N/mm^2$ 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 pivot pins in linkages, knuckle joints, piston pins, security pins

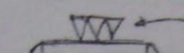
2. PINS

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

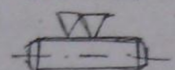
(i) Types

(a) Cylindrical pins

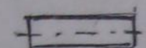
very smooth



Centering pin

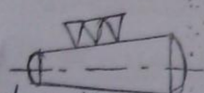


Coupling pin



Drifting (punching) pin

(b) Taper pins



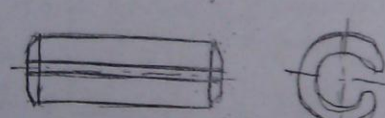
taper 1:50

Taper pin with thread

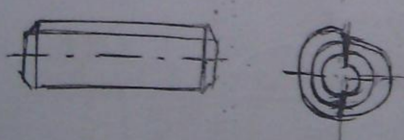
taper pin with internal thread

(c) Others : SPRING PINS

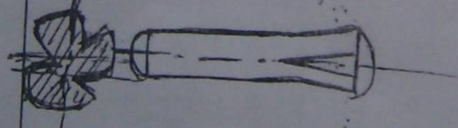
Split pin



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 55Si7

Split pins : St, MS, Cu, Al.

(iii) Allowable Stresses

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

Static $\sigma_{b\,all} \simeq 0.3\,S_{ut}$

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

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

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

Static ——— $\sigma_{B\,all} = 0.25\,S_{ut}$

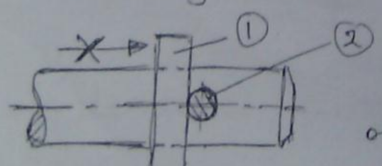
Reversed fatigue ——— $\sigma_{B\,all} = 0.15\,S_{ut}$

Alternating " ——— $\sigma_{B\,all} = 0.12\,S_{ut}$

(iv) Applications

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

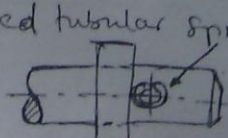
1. Securing against axial motion



① Washer

② Split pin

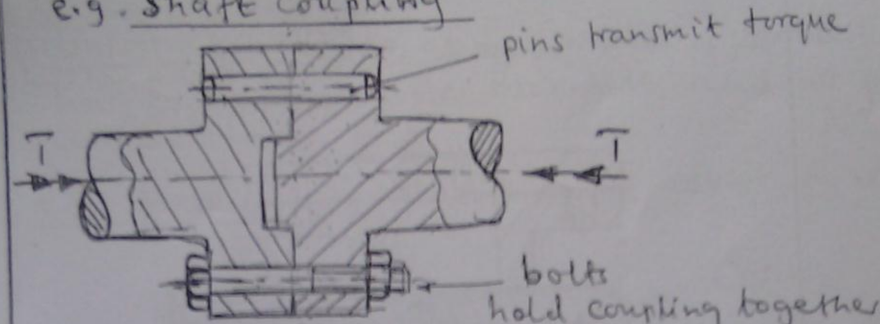
or Slotted tubular spring pin



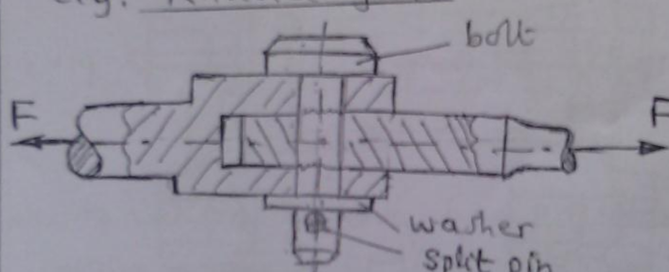
2. Shear pins

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

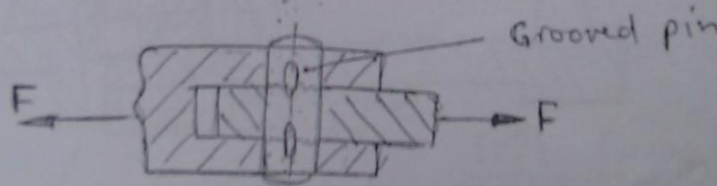
e.g. Shaft coupling



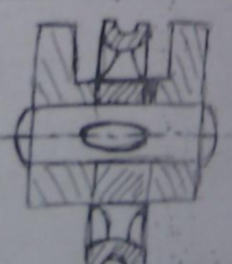
e.g. Knuckle joint



Analysis
(a) Shear
(b) Bearing



e.g. Axle for small pulley

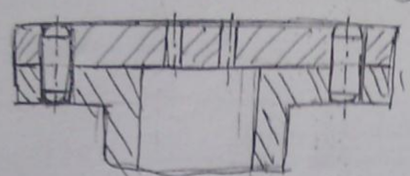


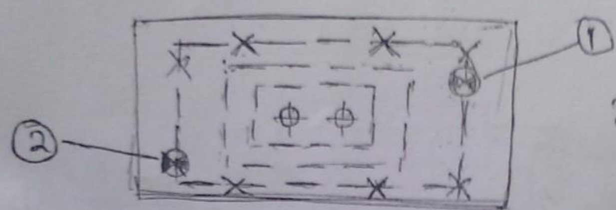
e.g. Locking of a nut

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.

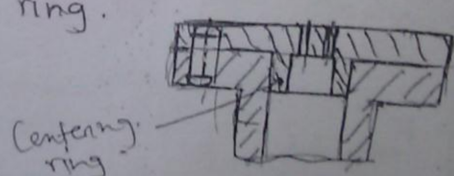
e.g. Casing and cover of non-circular shap and with no centering ring.



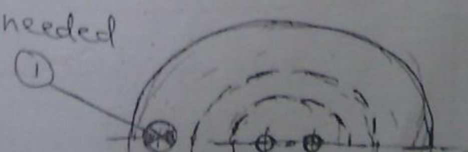


2 pins needed

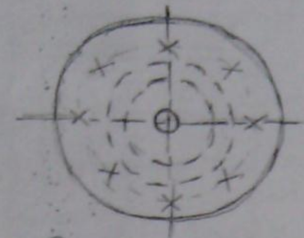
e.g. Casing and cover of circular shape, cover with centering ring.



One pin only is needed



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



No centering pin needed.

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

e.g. Machining (Casing and cover - fig. below)

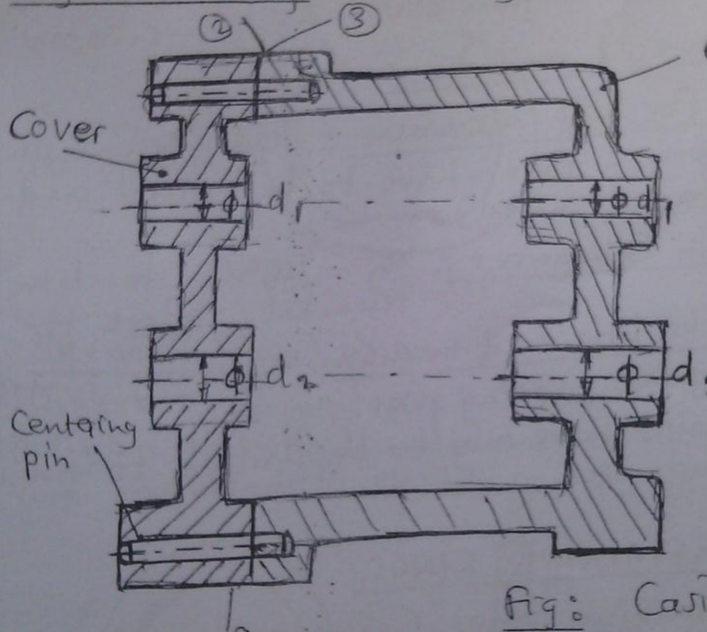


Fig: Casing and cover

Operation Sequence

(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.

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⑥ Drill holes for centering pins and mou... pins.

⑦ Drill holes for shaft 1 and 2, and machi... to size ϕd_1, ϕd_2.
The holes in cover and casing will then be well aligned.

4. <u>Safety pins</u>

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.

<u>Purpose</u>: to protect more expensive parts from overload.

<u>General remarks:</u>

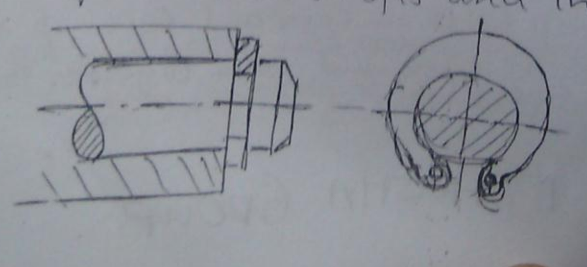
i) <u>For high accuracy</u>: cylindrical and taper pins. Bores have to be drilled and reamed (taper reamed).

<u>Pin tolerances</u>: m - tight, h - bore often reamed, <u>Bore</u> H

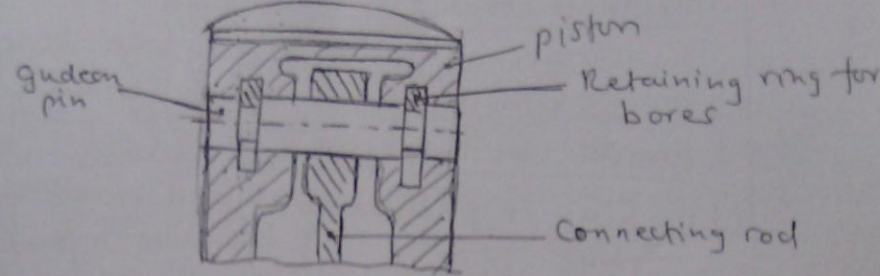
ii) <u>For low and medium accuracy</u>: grooved pins, spring pins.
Holes have only to be drilled to nominal diameter of the pin.

3. <u>RETAINING RINGS</u>

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).

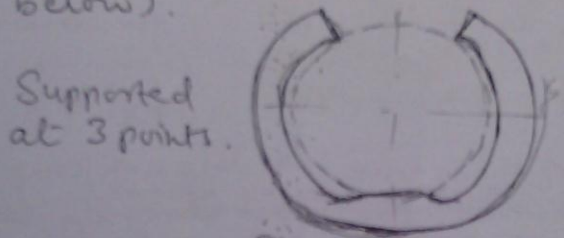


fig. Circlip.

Seeger (Snap) rings

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

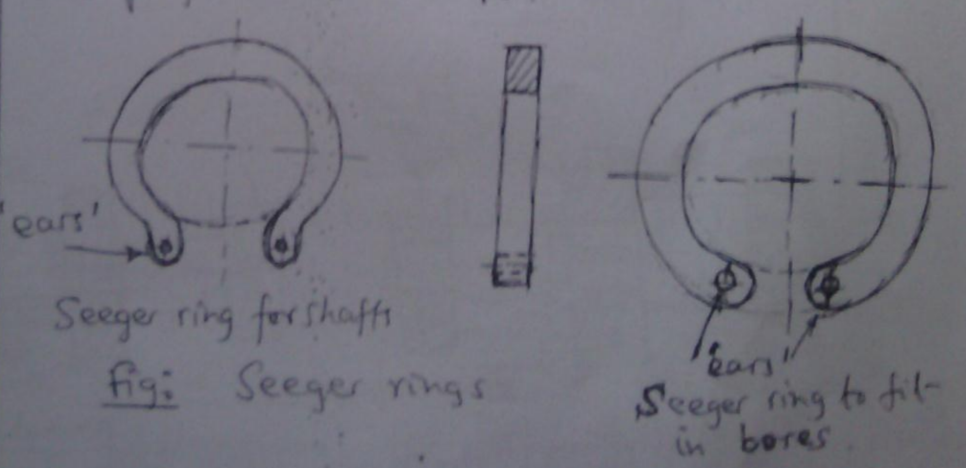


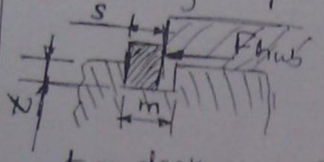
fig: Seeger rings

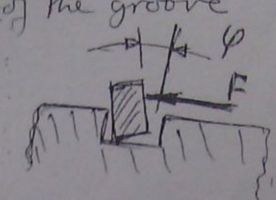
Design factors

Seeger-rings transmit very high axial loads 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

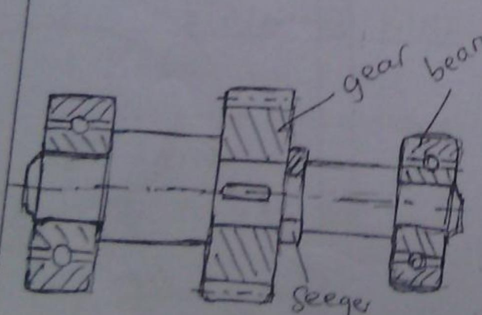
$\therefore \quad t/s = 0.17 \ldots 0.67$ for DIN 471/72

and $t/s = 0.87 \ldots 1.7$ 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.

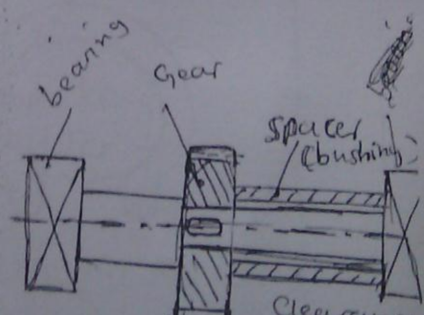
(2) At higher speeds, the ring may come out of the groove due to inertia forces.

(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

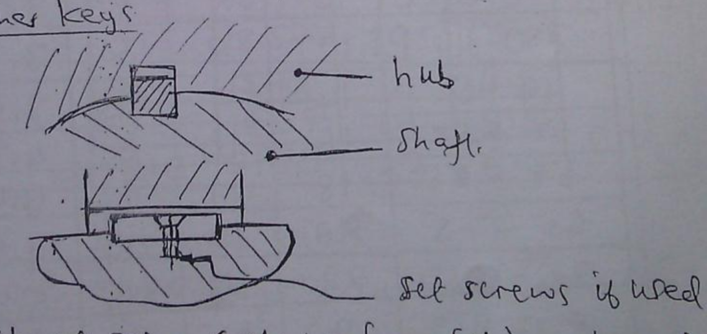
Keys are used to prevent relative (rotary) motion between shafts and machine elements such as gears, pulleys, sprockets, cams, levers etc.
A key is a demountable machinery part, which when assembled into keyseats, provides a positive means of transmitting torque between shaft and hub.

(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.

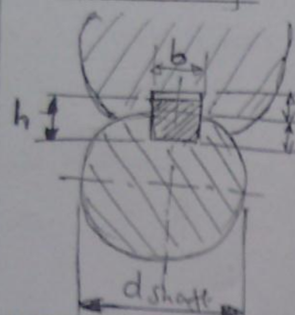
(a) Feather keys



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 $d_{sh} \le 160$ mm
Rectangular for $d_{sh} > \phi\ 160$ mm
*In other standards

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

Dimensioning

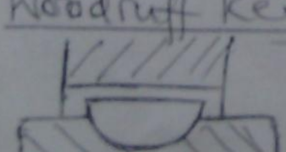


t_2 = keyway depth in hub
t_1 = keyway depth in shaft

Specified as $b \times h \times l$
where l = length of key

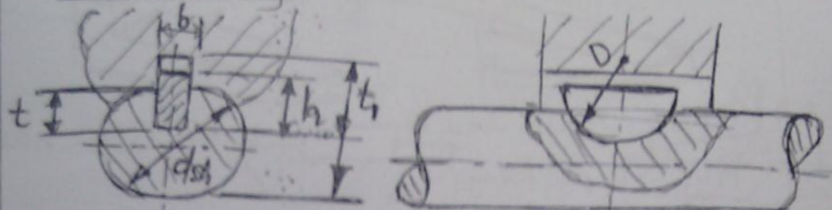
d	b	h	t_1	t_2	Length l
6 – 8	2	2	1.2	1	6 – 20
8 – 10	3	3	2	1.3	6 – 36
10 – 12	4	4	2.5	1.8	8 – 45
12 – 17	5	5	3	2.3	10 – 56
17 – 22	6	6	3.5	2.8	14 – 70
22 – 30	8	7	4	3.3	18 – 90
30 – 38	10	8	5	3.3	22 – 110
38 – 44	12	8	5	3.3	28 – 140
44 – 50	14	9	5.5	3.8	36 – 160
50 – 58	16	10	6	4.3	45 – 180
58 – 65	18	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



They are used for small ~~t~~orques only.

Dimensioning



Dimensioned as $\underline{b \times h}$

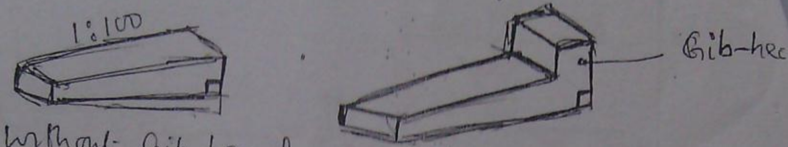
d (Shaft)	b	h	l	D	t	t_1
6–8	2	2.6	6.76	7	1.8	d+1
		3.7	9.66	10	2.9	
8–10	3	3.7	9.66	10	2.5	d+1.4
		5.0	12.65	13	3.8	
		6.5	15.72	16	5.3	
10–12	4	5.0	12.65	13	3.5	d+1.7
		6.5	15.72	16	5.0	
		7.5	18.57	19	6.0	
12–17	5	6.5	15.72	16	4.5	d+2.2
		7.5	18.57	19	5.5	
		8.0	21.63	22	6.0	

(2) Keys in Strained joints

These 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 sunk-keys can be with or without a gib-head. (Fig. below).

The Not-Sunk keys are Flat, Saddle

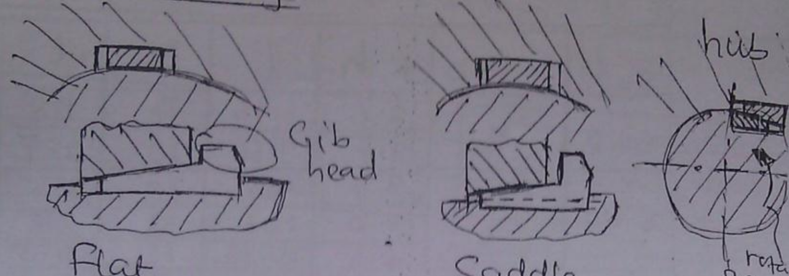
and tangential keys. They sit straight on shaft. Hence advantage of no keyway. However they carry small torques.



without Gib-head — with Gib-head

Fig: Taper keys

(a) Not-sunk keys



Flat — Saddle — Tangentia

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 $d_{shaft} \le \phi 160$ 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.

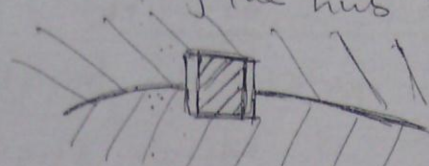
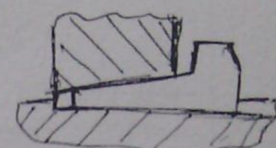


fig: Sunk-key



with Gib-head

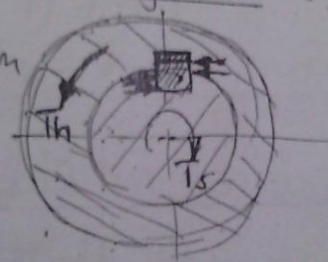
(ii) Transmission of Load

(a) Unstrained joint e.g. feather key

T_s = torque on shaft.

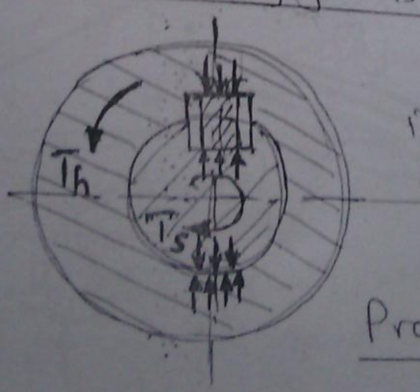
T_h = Torque on hub

$T_s = T_h = T$



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)

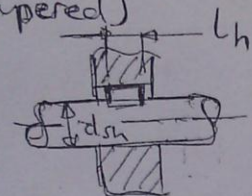
$n = 1.5$ for steady torque

$n = 2.5$ " 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)

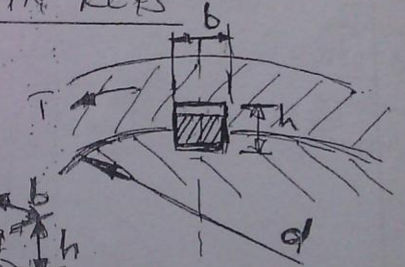


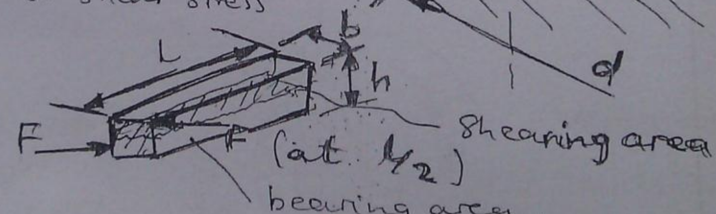
$$L_h = (1 \ldots 1.25)\, d_{shaft}$$

* (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 $\sigma_b = \dfrac{F}{A_b} \leq \sigma_{ball}$; $F = \dfrac{T}{r}$, $r = \dfrac{d}{2}$

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

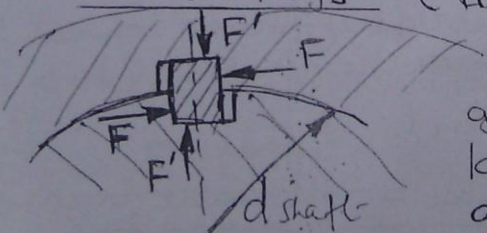
(b) Woodruff Key

Same equations above apply with

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

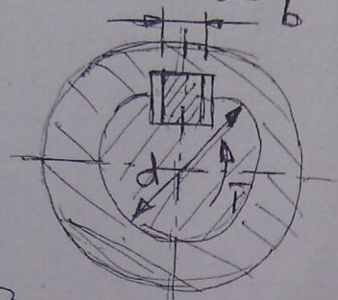


Fig: Assembly

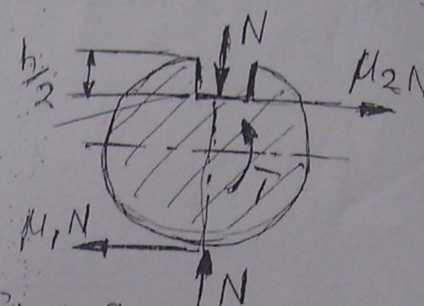
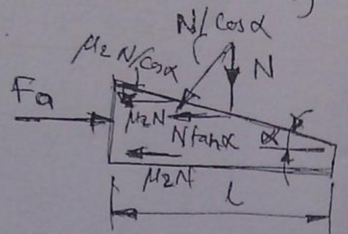


Fig: Shaft loading



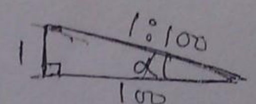


Fig: Key loading

(1) Torque transmitted (T)

From shaft loading

$$T = \tfrac{1}{2}\mu_1 N d + \tfrac{1}{2}\mu_2 N\,(d - \underbrace{h/2}_{\text{neglected}})$$

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

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

From experience:

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

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

(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)

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

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).

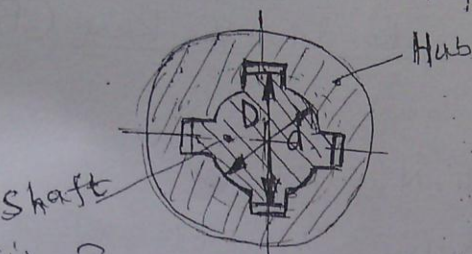
5. SPLINES

Splines are keys that form an integral part with the shaft. They also permit axial

motion between shaft and hub in addi-
to preventing relative rotary motion.

<u>Number of splines</u> : 4, 6, 8, 10, ----20

<u>Materials</u> : 37Cr4, 41Cr4, 42CrMo4



D = Major diameter
d = minor diameter

(i) <u>Types of profiles</u>

(1) Straight Sided Splines

(2) Involute Spline

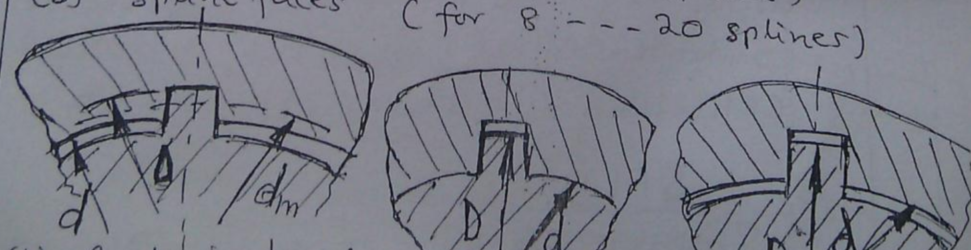
(3) Serrated

<u>Note</u>: Involute are cut similar to gear teeth. They have lower stress concentration than straight sided splines.

(ii) <u>Centering</u> (example - straight-sided spline

Splines are centred by : (figs below)

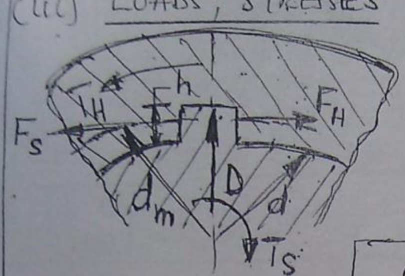
(1) Major diameter (heavy duty and in special case
(2) Minor diameter (for 6, 8, 10 splines)
(3) Spline faces (for 8 --- 20 splines)



(1) By Major diameter (2) By Minor Diameter (3) By Spline faces

<u>Figs</u>: 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

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

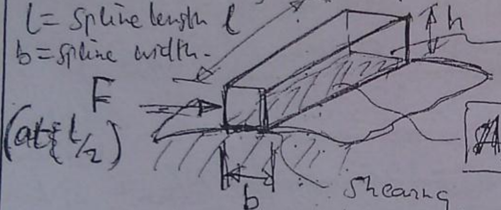
$$\boxed{d_m = mZ}$$ ← for involute splines,
m = module,
Z = no. of splines.

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 $\sigma_b = \frac{F}{A_b} \leq \sigma_{ball}$, σ_{ball} = 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 $\tau = F/A_s \leq \tau_{all}$, 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.

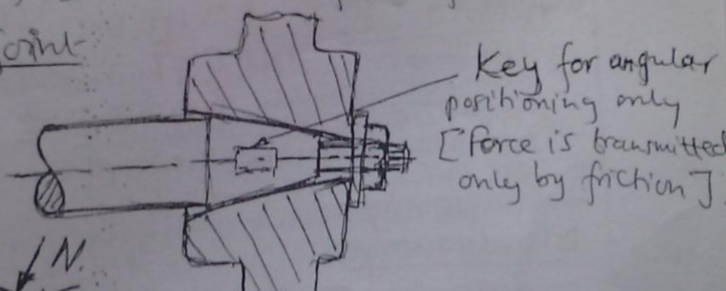
Applications

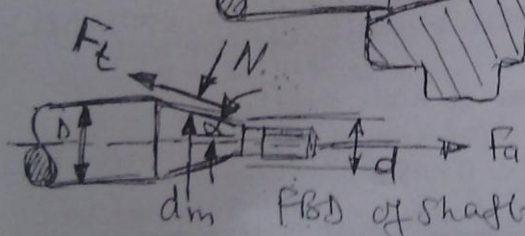
e.g. car transmission – shifting gears, coupling (gears, pulleys)

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)

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

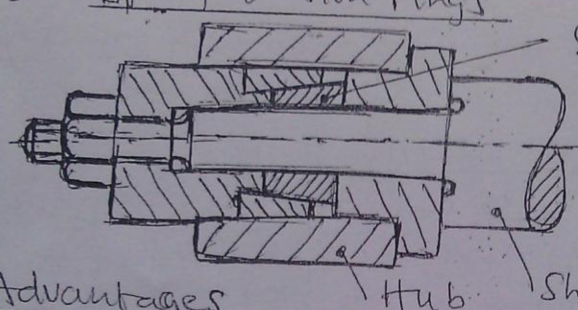
α = inclination angle, with $\mu = \tan\varphi = \frac{\sin\varphi}{\cos\varphi}$, φ = friction angle

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

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

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

(2) <u>Spring tension rings</u>



possible to use elements in serie

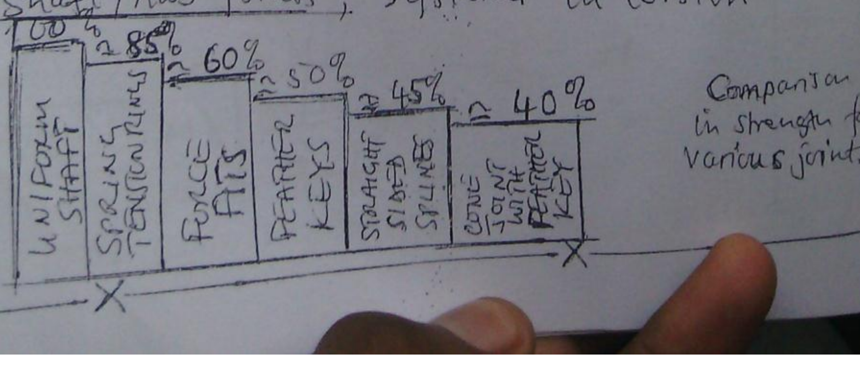
Advantages

- 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. <u>Endurance Limit reduction factors for various Shaft/hub joints; systems in torsion</u>



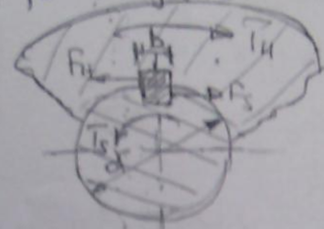
Comparison in strength for various joints

①—⑤

Questions & Solutions — Keys, Splines

Q1 A steel shaft with a yield strength of 510 MPa has a diameter of 36 mm. The shaft rotates at 600 rpm and transmits 30 kW through a gear (made from steel). Select an appropriate key for the joint.

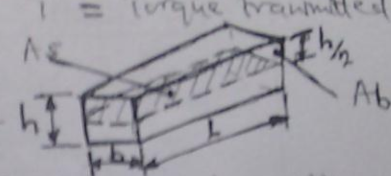
Solun



$T_H = T_s = T$

T_H = hub torque (gear torque)
T_s = shaft torque.
T = Torque transmitted

Assuming a feather key



A_s = shearing area, $A_s = b \times l$; b = width, l = length

A_b = bearing area, $A_b = \frac{h}{2} \cdot l$, h = key depth

Now $d = 36$ mm $\therefore r = d/2 = 18$ mm

$$T = \frac{P}{\omega}, \quad P = 30\,\text{kW}, \quad n = 600\,\text{rpm}$$

$$\therefore T = \frac{30 \times 10^3}{\pi \times 600/30} = \underline{477.5\ \text{Nm}}$$

Key section chosen

$b = 10$ mm ; $h = 8$ mm (for $d = 30 - 38$ mm) from tables

Key material: $S_y = 440$ MPa $< S_{y\,shaft} = 510$ MPa

Factor of safety: $n = 2.5$ (nothing stated about nature of loading.)

Bearing failure:

$$\sigma_{ball} = \frac{S_y}{n} = \frac{440}{2.5} = \underline{176\ \text{MPa}}$$

$$\therefore T_{max} = A_b \cdot \sigma_{ball} \cdot r = \frac{h}{2} l\, \sigma_{ball} \cdot r$$

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$\therefore$ $l_{min} = 37.7$ mm take $l = 3$ mm

Shear Failure:

$S_{sy} = 0.5 S_y$ $\therefore S_{sy} = 220$ MPa.

$\tau_{all} = \frac{S_{sy}}{n}$ $\therefore \tau_{all} = \frac{220}{2.5} = 88$ MPa

Now $T_{max} = A_s . \tau_{all} . r = b . l . \tau_{all} . r$

$\therefore$ $l_{min} = 30.2$ mm take $l = 31$ mm

For Stability:

$L \simeq (1 \ldots 1.25) d_{shaft}$ to prevent hub from rocking on shaft.

$\therefore$ Choose $l = 1.25 d = 1.25 (36) = 45$ mm.

$\therefore$ Take $l = 45$ mm

$\therefore$ Feather key $10 \times 8 \times 45$ Ans. Chosen.

Q.2 A woodruff key 5×6.5 is used to key a gear on a 17 mm diameter shaft made from steel with $S_u = 625$ MN/m² and $S_y = 530$ MN/m². The key is made from the same material as the shaft. Determine the torque capacity of the shaft. Calculate the torque capacity of the key, using a factor of safety of 1.5 based on the yield strength of the material.

Solun:

Torque capacity of the shaft

with keyway $\tau_{all} = 0.75 (0.18) S_u$

$\tau_{all} = 0.75 (0.30) S_y$ by ASME CODE

or

$\therefore \tau_{all} = 0.75(0.18)625 = 84.37\ MPa$

or $\tau_{all} = 0.75(0.30)530 = 119.25\ MPa$

$\therefore$ Take $\tau_{all} = 84\ MPa$

Now $\tau_{max} = \dfrac{16T}{\pi d^3} = \tau_{all}$ solid shaft

$\therefore T_{max} = \dfrac{\pi d^3}{16}\tau_{all}$

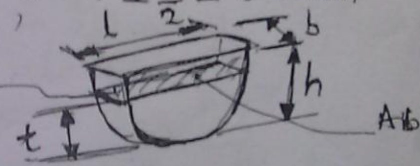
Hence $T = \dfrac{\pi(17)^3(84)\times10^{-3}}{16} = 81.03\ Nm$

$\therefore T_s = 81\ Nm$ Ans.

Torque capacity of key

Bearing failure

$T_{max} = A_b \cdot \sigma_{ball} \cdot r$; $r = \dfrac{d}{2} = 8.5\ mm$



$A_b = (h-t) \cdot l$

Key 5×6.5

$\therefore b = 5mm$, $h = 6.5mm$, $t = 4.5mm$, $l = 15.72\ mm$ from tables

$\therefore A_b = (6.5-4.5)15.72 = 31.44\ mm^2$

$\sigma_{ball} = \dfrac{S_y}{n}$, $n = 1.5$ $\therefore \sigma_{ball} = \dfrac{530}{1.5} = 353\ MPa$

$\therefore T_b = 31.44(353)(8.5)\times10^{-3}\ Nm = 943\ Nm$

Shearing failure

$T_{max} = A_s \cdot \tau_{all} \cdot r$

$A_s = b \cdot l = 5(15.72) = 78.6\ mm^2$

$r = 8.5\ mm.$

$\tau_{all} = \frac{S_{sy}}{n}$, $n = 1.5$, $S_{sy} = 0.5 S_y$

$\therefore \tau_{all} = \frac{S_y}{2n} = \frac{530}{2(1.5)} = \underline{176.6 \text{ MPa}}$

$\therefore T_s = 78.6(176.6) \times 8.5 \times 10^{-3}$ Nm

$= \underline{117.98 \text{ Nm}}$

Least of the two $\therefore$ $\underline{T_k = 94.3 \text{ Nm}}$

$\therefore$ Torque capacity of the key is $\underline{\underline{T_k = 94.3 \text{ Nm}}}$

Q.3 A splined connection in an automobile transmission consists of 10 splines cut in a 56 mm diameter shaft. The height of each spline is 5mm, and the keyways in the hub are 45 mm long. Determine the power that may be transmitted at 2500 rpm, if the allowable normal pressure on the splines is limited to 4.8 MN/m².

Soln:

Note: Based on bearing failure only

$T_{Nmax} = A_b \cdot \sigma_{ball} \cdot N \cdot r_m \cdot \varphi$

$\varphi = 0.75$ assumption, $N = 10$ splines.

$A_b = h \times l$, $h = 5$mm, $l = 45$mm

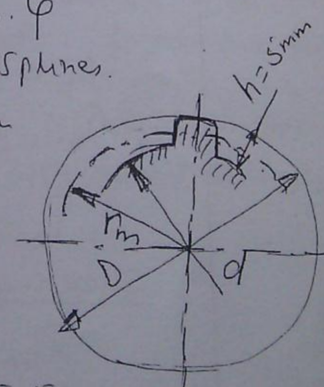
$\therefore A_b = 5 \times 45 = 225 \text{ mm}^2$

$\sigma_{ball} = 4.8 \text{ N/mm}^2$

$D = 56$ mm

$\therefore d = D - 2h = 46$ mm

$\therefore r_m = \frac{D+d}{4} = \frac{56+46}{4} = \underline{25.5 \text{mm}}$



$\therefore T_{Nmax} = 225(4.8)(10)(25.5)(0.75) \times 10^{-3}$ Nm

$= \underline{206.55 \text{ Nm}}$

Power transmitted

$P = T \cdot \omega$; $\omega = \frac{\pi n}{30}$; $n = 2500$ rpm

$\therefore P = 206.55 \times \pi \left(\frac{2500}{30}\right) \times 10^{-3}$ kW

$= 54$ kW

$\therefore$ The powe that may be transmitted is 54 kW Ans.

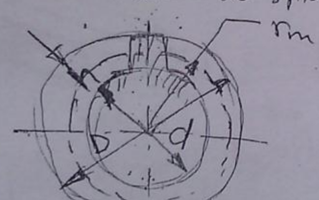
Q. 4

A splined connection $10 \times 72 \times 78$ (steel hub) is used for an automobile transmission. The splines have a length of 65 mm. Assume high shockload with a peak of 1750 Nm.

Detemine the suitability of the connection if $\sigma_{all} = 35$ MN/m^2.

Solun: Spline $10 \times 72 \times 78$

i.e. $N = 10$ splines, $d = 72$ mm, $D = 78$ mm

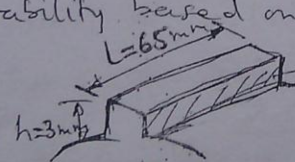


$\therefore h = \frac{D - d}{2}$

$= \frac{78 - 72}{2} = 3$ mm

Note:

Suitability based on bearing failure only.



A_b

$\therefore A_b = l \times h$

$= 65 \times 3 = 195$ mm^2

$\therefore T_{Hmax} = A_b \cdot \sigma_{ball} \cdot r_m \cdot N \cdot \varphi$

Assuming $\varphi = 0.75$, $r_m = \frac{D + d}{4} = \frac{78 + 72}{4} = 37.5$ mm

~~$T_{Hmax} = 195$~~

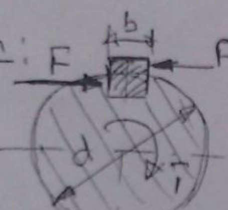
$\therefore T_{N max} = 195 \, (35)(37.5)(10)(0.75) \times 10^{-3}$ Nm
$= 1919.53$ Nm

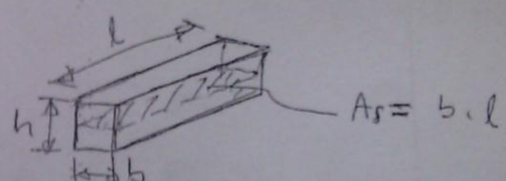
Now applied load is $T = 1750$ Nm i.e. < than max

$\therefore$ The connection is suitable Ans.

Q.5. A key is sometimes used as a shear pin for economical reasons in case of overloads. A shaft made from steel with $S_u = 660$ MN/m² and $S_y = 395$ MN/m² is transmitting power in torsion. The shaft is 50 mm in diameter. Determine the dimensions of a suitable key, if the key is to have 60% of the maximum strength capacity of the shaft. Material for the key $S_u = 540$ N/mm² and $S_y = 370$ N/mm².

Soln:





$d = 50$ mm

From tables, for $d = 50$ mm
$b = 14$ mm, $h = 9$ mm.

For shaft

$\tau_{all} = 0.75\,(0.18)\,S_u$ or $\tau_{all} = 0.75\,(0.3)\,S_y$

$\therefore \tau_{all} = 0.75\,(0.18)\,660$ or $\tau_{all} = 0.75\,(0.3)\,395$
$= 89.1$ N/mm² $\quad = 88.875$ N/mm²

$\therefore$ Take $\tau_{all} = 88$ N/mm² by ASME CODE

Torque capacity of shaft T_s:

$$\tau_{max} = \frac{16T}{\pi d^3} = \tau_{all} \qquad \therefore T_{max} = \frac{\tau_{all} \cdot \pi d^3}{16}$$

$$\therefore T_s = \frac{88(\pi)(50)^3}{16} \times 10^{-3}\ Nm = \underline{2.16\ kNm}$$

Now torque capacity of key $T_k = 0.6\, T_s$

$$\therefore T_k = 0.6(2.16) = \underline{1.296\ kNm}$$

The key is to serve as a shear pin

∴ Shearing only

$$\therefore T_k = \tau_{all} \cdot A_s \cdot r$$

$$A_s = b \cdot l = 14 \times l\ mm^2$$

$$\tau_{all} = \frac{S_{sy}}{n} = \frac{S_y}{2n} \quad , n = 1 \text{ safety device}$$

$$\therefore \tau_{all} = \frac{370}{2(1)} = \underline{185\ N/mm^2}$$

$$r = d/2 = 25\ mm$$

$$\therefore T_k = \tau_{all} \cdot r \cdot (14 \times l)$$

$$\therefore l_{max} = \frac{T_k}{\tau_{all} \cdot r\,(14)}$$

$$\therefore l_{max} = \frac{1296 \times 10^3}{185\,(25)(14)}\ mm = \underline{20.01\ mm}$$

∴ key length l = 20 mm for shearing

∴ Dimensions for key are $\underline{\underline{14 \times 9 \times 20\ mm}}$ Ans.

Q. 6

Determine the power capacity ratio for the two systems:

(a) a 26 mm diameter shaft with a 6 x 6 x 50 mm key.

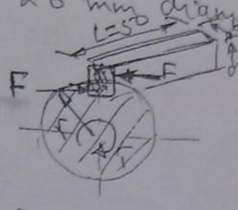
(b) a 26 mm diameter shaft with a 6 mm diameter pin key.

The pin key is perpendicular to the axis of the shaft and passes through the centre of the shaft.

Assume that only pure torque is transmitted, and that the material of the shaft is the same as used for key and pin. Assume a 25% reduction in torque capacity of the shaft due to the keyway, and a stress concentration factor of 1.75 due to the hole in the shaft.

Soln:

(a) a 26 mm diameter shaft with key 6x6x50 mm



$b = h = 6\,mm$

$$r = \frac{d}{2} = \frac{26}{2} = 13\,mm$$

If Given S_y, $n = 1$

Shaft

$$\tau_{max} = \frac{16T_s}{\pi d^3} = \tau_{all}, \quad \tau_{all} = \frac{S_{sy}}{n} = \frac{S_y}{2n}$$

$$\therefore \tau_{all} = \frac{S_y}{2} \quad \text{i.e. } n=1$$

$$\therefore T_{maxS} = \frac{\tau_{all} \cdot \pi d^3}{16} = \frac{0.5 S_y \pi (26)^3}{16}$$

TAKE ASME CODE

$$\tau_{all} = 0.75(0.3)S_y = 0.225 S_y$$

$$\therefore T_{maxS} = \frac{\tau_{all} \cdot \pi d^3}{16} = \frac{\pi (26)^3 (0.225) S_y}{16}$$

$$\therefore T_s = 776.4\, S_y$$ i.e. 25% reduction assumption given

Key:

Check bearing only. (Shear not likely to occur)

$\therefore$ Bearing failure

$T_{Kmax} = \sigma_b \ldots$

$$r = \frac{d}{2} = \frac{26}{2} = 13\,m$$

$$A_b = \frac{h}{2} \cdot L$$

$b = h = 6\,mm.$

$$\therefore A_b = \frac{6}{2}(50) = 150\,mm$$

$\sigma_{ball} = \frac{S_y}{n} = \frac{S_y}{1}$ for $n=1$

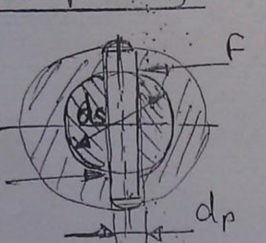
$\therefore T_{kmax} = S_y (150)(13)$

$\underline{T_k = 1950\, S_y}$

$\therefore$ Power capacity ratio for key : shaft $= 1950 : 776$

$\therefore$ Power ratio $\underline{\text{key : shaft} = 2.51 : 1 \text{ Ans.}}$

(b) <u>With pin key</u>



$d_p = 6$ mm dia of pin

$d_s = 26$ mm

$\therefore$ Pin in double shear

Hub diameter not known

$\therefore$ Bearing failure not checked

<u>Shear of Pin.</u>

$T_{maxp} = \tau_{all} . A_p . d_s$; i.e. $T = F.d_s$

$F = \tau_{all} . A_p$

$A_p = \frac{\pi d^2}{4} = \pi \frac{(6)^2}{4}$ mm^2,

$\tau_{all} = \frac{S_{sy}}{n} = \frac{S_y}{2n} = \frac{S_y}{2}$

$\therefore T_p = \frac{S_y}{2} \times \frac{\pi}{4} (6)^2 (26) = \underline{367.56\, S_y}$

$\therefore \underline{T_p \triangleq 368\, S_y}$

<u>Shaft torque capacity T_s</u>

$\tau_{all} = 0.3\, S_y$ ASME code

Now $\tau_{max} = K_{ts} . \tau$, $K_{ts} = 1.75$

$\tau_{max} = \frac{16 T}{\pi d^3} = \tau_{all}$

$$\therefore \frac{1.75 \times 16 \tau_s}{\pi (26)^3} = 0.3 S_y$$

$$\therefore \tau_s = \frac{0.3 S_y (\pi)(26)^3}{1.75 (16)} = \underline{591.6 S_y}$$

Again power ratio of pin : Shaft = 368 : 591.6

$\therefore$ Power ratio of pin : Shaft $= \underline{\underline{1 : 1.61}}$ Ans.

—— X —— X ——