PROGRESS THROUGH KNOWLEDGE	ANNA UNIVERSITY CHENNAI 600 025 OFFICE OF THE CONTROLLER OF EXAMINATIONS	Off:	22357273,22357274
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Letter No.24100/UG/C10/2015 Dated: 31/08/2015

From
Dr.G.V.UMA
Controller of Examinations
Anna University
Chennai – 600 025

To
The Principals of all Affiliated College/
Government College / Deans of University
Colleges under Anna University,
Chennai – 600 025 (Zone I to XXI)

Sir,

Sub: O/o COE Anna University, Chennai – 25 – Theory Examinations – Affiliated Colleges – Nov/Dec 2015 Exams – Permission to use the Code books for ME6503, Regulation 2013 – Reg.

****

The principals of all the affiliated colleges, offering B.E Mechanical Engineering is informed that the Approved Additional Pages are also permitted along with the PSG Design Data Book in the Examination 'ME6503 - Design of Machine Elements'

Yours faithfully,

31/8/15

CONTROLLER OF EXAMINATIONS

Encl: As above
Copy to All the Zonal Co-ordinator
Zone1 to 21

PSG DESIGN DATA BOOK

ADDITIONAL PAGES

(200)

ANNA UNIVERS TY :: CHENNAI - 25

Dr.G.RA MAIYAN, Ph.D.
CHAIRMAN
FACULTY OF MECHANICAL ENGINEERING

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Lr.No. 53 2/AC21/2003

Date:25th September,2003

To
The Director,
Academic Courses
Anna University
Chennai-25.

HOD
Mech

30/9

29/9/03

Sir,

In the curriculum of B.E. Mechanical Engineering, there is a course on "M.E. 331 Design of Machine Elements". The PSG Design Data Book is an approved the date book which is permitted in the examination. However the Principal, Sri Krishna College of Engineering and Technology, Coimbatore has stated that data are not available for certain items like Design of Pipe Joints, Design of Piston Assembly, Design of shaft couplings and has sent data including figures for use in the examination. The same has been scrutinized by me and found to be in order.

The Principal of all affiliated colleges, offering B.E. Mechanical Engineering may please be informed that the additional pages (copy to be sent) enclosed are also permitted along with the PSG Design Data Book in the Examination on "Design of Machine Elements".

Yours faithfully,

Sd/-
Chairman
Faculty of Mechanical Engineering

Encl: Data Sheets: 20 pages

ANNA UNIVERSITY:CHENNAI 600 025

Academic Courses

Endt No. 55/AU/AC01/2003

Date: 25.09.2003

Copy forwarded for information and necessary action

DIRECTOR
(Academic Courses)

To
All Principals offering B.E. Mechanical Engineering

CONTENTS

List of Figures

1. DESIGN OF PIPE JOINTS

$Di = 18.8\sqrt{W/v\rho}$

Di – Internal diameter of pipe
W- Discharge or flow rate in kg hr for gases & Tonnes /hr for Liquids
ρ- Density in Kg/m^3 for gases and $Tonne/m^3$ for liquid.
v – Flow velocity m/s.

1.1 Circular Flanged pipe joint:-

$D_1 = D_p - d_1$

D_1 - Diameter of the circle touching bolt holes.
D_p - Pitch circle diameter.
d_1 . Diameter of bolt hole.

$S = \pi/4\ (dc)^2\ \sigma_t\ .\ n$

S – Resistance to tearing of bolts
d_c – Core dia of bolts
n- No. of Bolts

$P_c = \pi D_p / n$

P_c – Circumferential Pitch of bolts
P_c should be between $20\sqrt{d_1}$ to $30\sqrt{d_1}$

$d = 0.75t + 10mm$
$n = 0.0275D + 1.6$
$t_f = 1.5t + 3mm$
$B = 2.3d$
$D_0 = D + 2t + 2B$
$D_p = D + 2t + 2d + 12mm$

d – Nominal diameter of bolts
n - No. of bolts
t_f - Thickness of flange
B - Width of flange
D_0 – Outside diameter of flange
D_p – Pitch circle diameter of bolts

1.2 Oval Flanged Pipe Joint:-

$F = \pi/4\ [(D_1)^2 - (D_2)^2]\ P$

F - Force required to separate the flanges
D_1 – Outside Diameter of Packing
D_2 – Internal Diameter of Pipe
P - Pressure of the passing fluid.

$F_b = \pi/4\ (d_c)^2\ \sigma_{tb}$

F_b – Force or load taken up by bolts
σ_{tb} – Allowable tensile stress of bolt material

$d = 0.75t + 10mm$
$t_f = 1.5t + 3mm$

d – Nominal diameter of bolts
t_f – Thickness of the flange

$D_o = D + 2t + 4.6d$ — D_o – Out side diameter of flange

$D_p = D_o - (3t + 20mm)$ — D_p – Pitch circle diameter

1.3 Square Flanged Pipe Joint:

$t = R[\sqrt{(\sigma_t + P) / (\sigma_t - P)}]$ — According to Lames equation

$F = \pi/4\ (D_1)^2\ P$ — F – Force required to separate the flanges

$F_1 = F / n$ — F_1 – Force on each bolt

L = Outside diameter of the pipe + 2 Diameter of bolt

$= D + 2t + 2d$

L - Minimum Length of diagonal for the square

$L_1 = L / \sqrt{2}$ — L_1 – Side of the square flange between center to center of bolt holes

$L_2 = L_1 + 2d$ — L_2 – Side of the flange for nuts & bolts without overhang

$d_m = D + 2t$ — d_m – Nominal or Major diameter of threads

2. DESIGN OF PISTON ASSEMBLY

2.1 Design of Cylinder:

$\sigma_l = D^2 . p / [(D_o)^2 - D^2]$

σ_l – Apparent Longitudinal stress in the cylinder
D_o -- Outside diameter of the cylinder in mm
D – Inside diameter of the cylinder in mm
p – Maximum pressure in engine in N/mm^2.
t -- Thickness of the cylinder wall in mm
1/m – Poisson ratio – usually taken as 0.25
σ_c -- Apparent circumferential stress in cylinder
l -- Length of the cylinder
σ_{nl} -- Net longitudinal stress
σ_{nc} – Net Circumferential stress
C – Allowance for reboring

$\sigma_c = D.p / 2t$

$\sigma_{nl} = \sigma_l - [\sigma_c / m]$
$\sigma_{nc} = \sigma_c - [\sigma_l / m]$
$t = (p.D/2\sigma_c) + C$

D in mm	75	100	150	200	250	300	350	400	450	500
C in mm	1.5	2.4	4.0	6.3	8.0	9.5	11	12.5	12.5	12.5

$t = 0.045D + 1.6mm$
$t_l = 0.03 D$ to $0.035D$

t_l – Thickness of dry liner

$t_{wl} = 0.032D + 1.6mm$ or t/3 for bigger cylinder. 3t/4 for smaller cylinder.

Water space between the outer cylinder wall and inner jacket wall = 10mm for 75mm cylinder to 75mm for 750mm cylinder or 0.08D + 6.5mm.

Length of the stroke (l) is generally 1.25D to 2D

Length of the Cylinder is 15% greater than the length of the stroke i.e. 1.15l

The Max. gas Pressure (p) may be taken as 9 to 10 times the mean effective pressure(P_m)

The flange thickness should be taken as 1.2t to 1.4t where t is the wall thickness of cylinder.

The nominal or major diameter of the stud or bolt (d) usually lies between $0.75t_f$ to t_f where t_f is the thickness of flange. A stud or bolt in no case should be less than 16mm diameter.

The pitch of the stud or bolt should be between $19\sqrt{d}$ to $28.5\sqrt{d}$.

$t_h = D\ [\sqrt{cp / \sigma_c}]$

t_h – Cylinder head thickness
D – Cylinder bore in mm
p – Max. pressure inside the cylinder in N/mm^2
σ_c – Allowable circumferential stress in MP_a
c – a constant of value 0.1

$D_p = D + 3d$

D_p – The pitch circle diameter

2.2 Design of piston

Piston head or crown:

$t_H = \sqrt{3pD^2 / 16\sigma_t}$ in mm

t_H – Thickness of piston head (Based on strain)
p – Max. gas pressure or explosion pressure in N/mm^2
D – Cylinder Bore or outside diameter of the piston in mm:
σ_t – Permissible bending (tensile) stress of piston material in MP_a
For Grey CI = 35 to 40 MP_a
For Nickel CI & Aluminum alloy = 50 to 90 MP_a
For forged steel 60to 100 MP_a

$t_H = H / 12.56K\ (T_C - T_E)$ mm

t_H – Thickness of piston head (Based on heat transfer)

H – Heat flowing through the piston head in watts or kJ / s
K – Heat conductivity factor in W/m/°C
For Grey CI = 46.6W/m/°C
For Steel = 51.25 W/m/°C
For Aluminum alloys = 174.75 W/m/°C
T_C – Temperature at the center of piston in °C
T_E – Temperature at the edges of the pistonhead °C

The temperature difference ($T_C - T_E$) mat be taken as 220 °C for CI , 75 °C for aluminum.

$H = C . HCV . m . Bp$ (in kW)

Heat flowing through the piston head (H)

C – Constant representing the portion of the heat supplied to the engine. The value can be taken as 0.05
HCV = Higher calorific value of fuel in kJ / kg

45×10^3 kJ / Kg for diesel
47×10^3 kJ / Kg for petrol
m – mass of the fuel used in g . brake power /Sec.
B.p – Brake power of the engine per cylinder.

1. The larger of two values of t_H is taken into account
2. When t_H is 6mm or less no ribs need to be provided. When t_H is greater than 6mm. ribs should be provided for a thickness of $t_H/3$ to $t_H/2$.
3. For engines having length of stroke to cylinder bore (L/D) ratio Upto 1.5 a cup is provided in the top of the piston having a radius equal to 0.7D

2.3 Design of piston rings:

$t_1 = D\sqrt{3P_w/\sigma_t}$

t_1 – Radial thickness of the ring
P_w – Pressure of gas on the cylinder wall in N/mm^2
Value limited from $0.025 N/mm^2$ to $0.042 N/mm^2$
σ_t – Allowable bending (tensile) stress in Mpa
Value may be 85 Mpa to 110Mpa for CI rings

$t_2 = D/10n_R$

t_2 – The maximum axial thickness
n_R – No. of rings

$b_1 = t_H$ to $1.2t_H$

b_1 – Width of the top land

$b_2 = 0.75t_2$ to t_2

b_2 – Width of the other lands

Gap Between the free ends of the ring = $3.5\ t_1$ to $4\ t_1$
Gap when the rings in the cylinder = 0.002D to 0.004D

2.4 Piston Barrel:

$t_3 = 0.03D + b + 4.5$ mm

t_3 – Thickness of Barrel
b – radial depth of piston ring groove which is taken as 0.4mm larger than the radial thickness of piston ring (t_1)

$b = t_1 + 0.4$
$t_3 = 0.03D + t_1 + 4.9$ mm

The piston wall thickness t_4 towards the open end is decreased from $0.25t_3$ to $0.35t_3$

2.5 Piston Skirt:

$P = p . [\pi D^2 / 4]$

P - Max. Gas load on the piston

R = 1/10 of Max. Gas load

R – Max. side thrust on the cylinder

l = 0.65D to 0.8D

l – Length of the piston skirt

3. DESIGN OF SHAFT COUPLINGS

3.1 Muff Coupling:

$D = 2d + 13mm$
$L = 3.5d$
$T = \pi/16 . \tau_c [D^4 - d^4 / D]$
$l = 3.5d /2 = L/2$
$T = l.w.\tau. d/2$ (shear failure of key)
$T = l. [t/2]. \sigma_c.[d/2]$
(Crushing failure of key)

D – Outside dia of sleeve.
L – Length of the sleeve
d – Dia of the shaft
T – TORQUE transmitted by coupling
τ_c – 14Mp_a for CI
l – Length of the key in each shaft
w – Width of the key
t – Thickness of the key
τ - Shear stress in Key
σ_c – Crushing stress in key

3.2 Flange Coupling:

$D = 2d$, $L = 1.5d$

$D_1 = 3d$, $D_2 = 4d$
$t_f = d/2$
$n = 3$ for d upto 40mm
$= 4$ for d upto 100mm
$= 6$ for d upto 180mm
$t_p = d/4$.

d_1 – Nominal Diameter of bolts
d – Shaft dia
D – Outside dia
D_1 – Dia of bolt circle
n – number of bolts
t_f – Thickness of flange
t_p–Thickness of protecting flange
τ_s, τ_b, τ_k – Allowable shear stress for shaft, bolt and key the material respectively.
τ_c– Allowable shear stress for flange material.
σ_{cb}, σ_{ck}–Allowable crushing stress for bolt and key material respectively.
T-Torque transmitted by the coupling

3.3 Hub design:

$T = \pi/16 . \tau_c[D^4 - d^4/D]$

Key:

$T = l.w. \tau_k. d/2$
$T = l.(t/2). \sigma_c . (d/2)$

Flange:

$T = \pi. [D^2/2]. \tau_c. t_f$

Bolts:

$T = n. [\pi/4].d_1^2. \tau_b. [D_1/2]$
$T = n.d_1. t_f. \sigma_{cb}. [D_1/2]$

3.4 Flexible Coupling: [Bushed pin type]

$W = P_d . d_2 . l$

$T = P_d . d_2 . ln.(D_1/2$
$\tau = W/\{[\pi\backslash 4].d_1^2\}$
$M = W [\{l/2\} + 5mm]$

L – Hub Length
l = Length of the bush in the flange
d_2 – Dia of bush
P_b – Bearing pressure on bush

$\sigma_l = M/z$, $d_1 = 0.5d/\sqrt{n}$
$T = l w . \tau_k . [d/2]$
$T = l.(t/2) . \sigma_c . (d/2)$
$L = 1.5d$
$T = [\pi/2] D^2 \tau_c t_f$

n – Number of pins
D_1 – Dia of pitch circle of pins
$P_b \leq 0.5$ to $0.8 v/mm^2$.
W = load on each pin
M – bending moment on pin
σ_l = Bending stress in pin
τ - Shear stress in pin
z – Modulus of section of pins$[\pi/32]d_1^3$.
d – shaft dia
W – Width of key
t – Thickness of key
$t_f - d/2$ $t_p = d/4$
τ_c – Shear stress in flange
D – Dia of hub
D = 2d.
D_2 – PCD of pins
D_2 – (4 to 6) d
T- torque transmitted by the coupling

3.5 Marine Coupling:

Flange thickness = d/3
Taken of bolt – 1 is20 to 1 is 40
PCD of bolts $D_1 = 1.6d$
Out side dia of flange $D_2 = 2.2d$

Shaft dia	35 to55mm	56 to150mm	151to250mm	251to390mm	Above390mm
No. of Bolts	4	6	8	10	12

d = shaft diameter
$t = [\pi/4]\, d_1^2 \tau_b\, n\, [D_1/2$
τ_b = shear stress in bolts

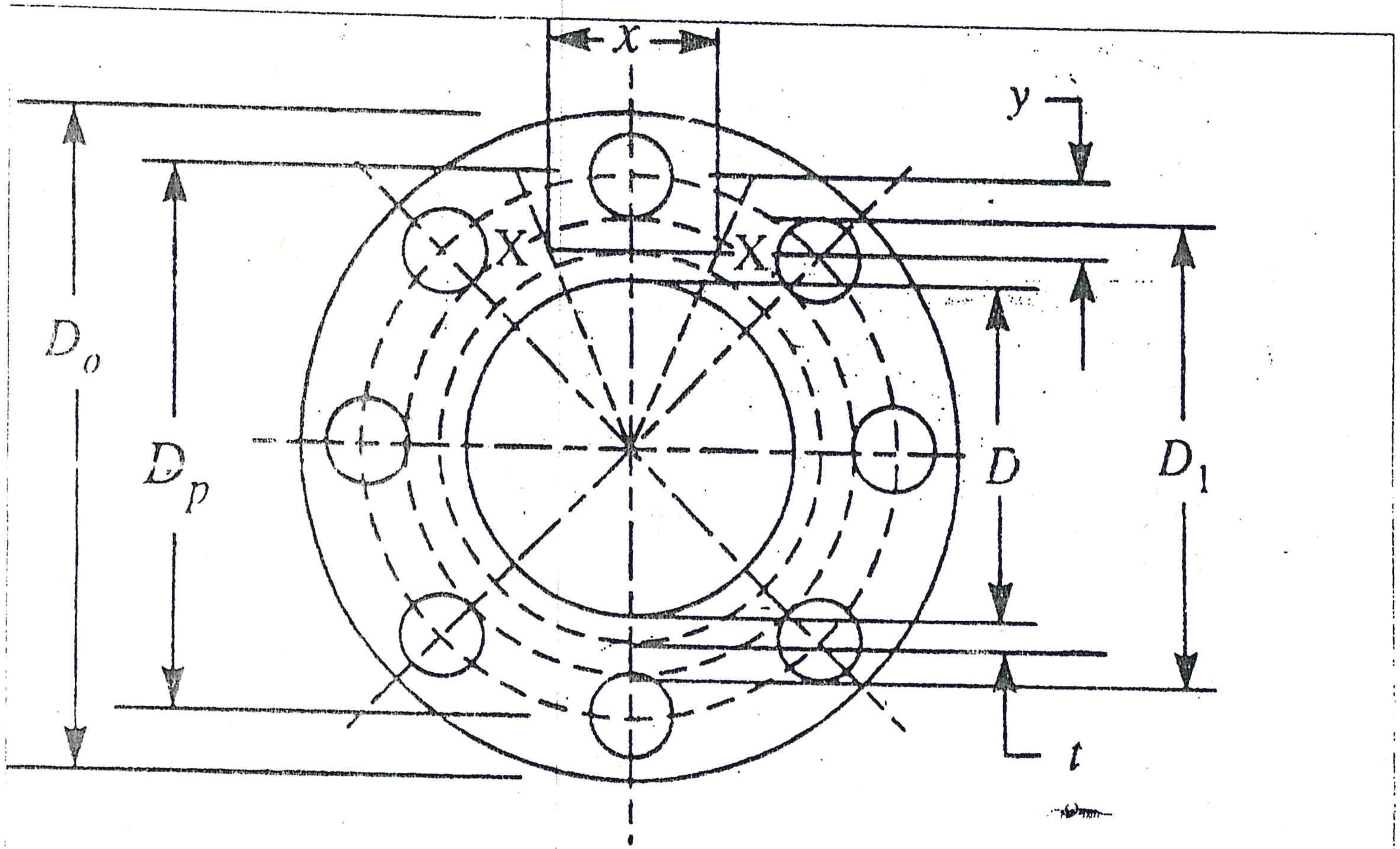

Fig 1.1 CIRCULAR FLANGED PIPE JOINT

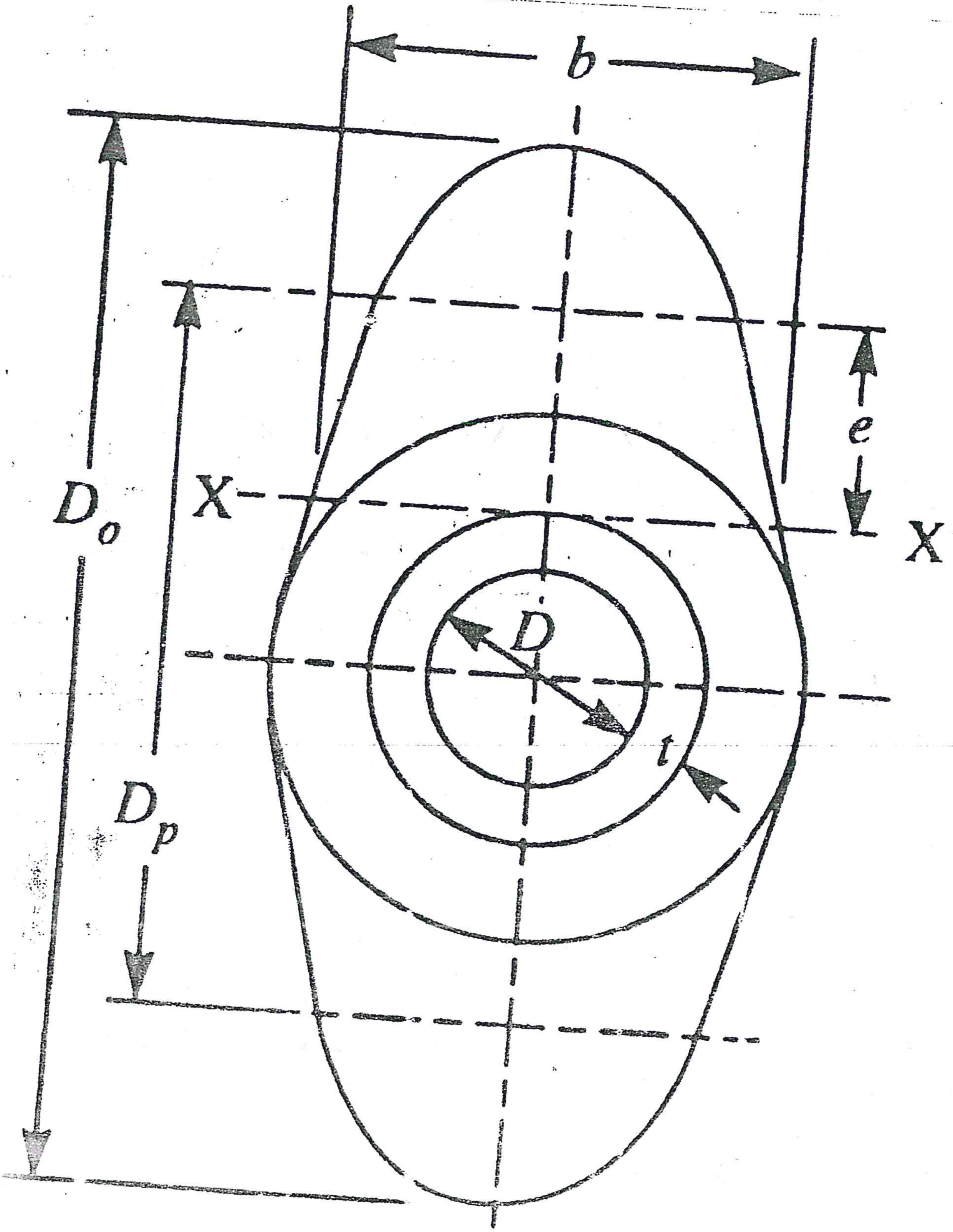

Fig. 1.2 OVAL FLANGED PIPE JOINT

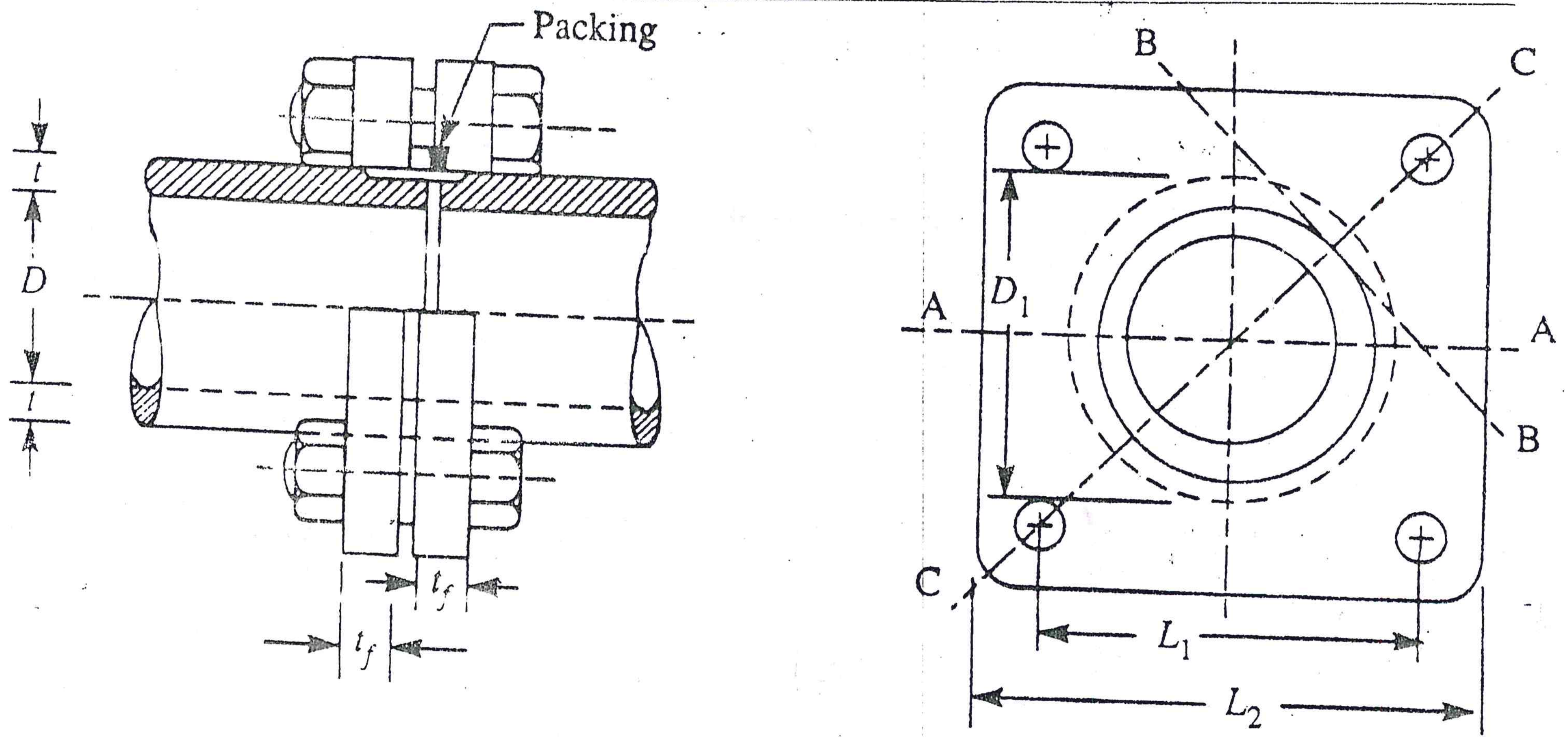

Fig. 1.3 Square flanged pipe joint.

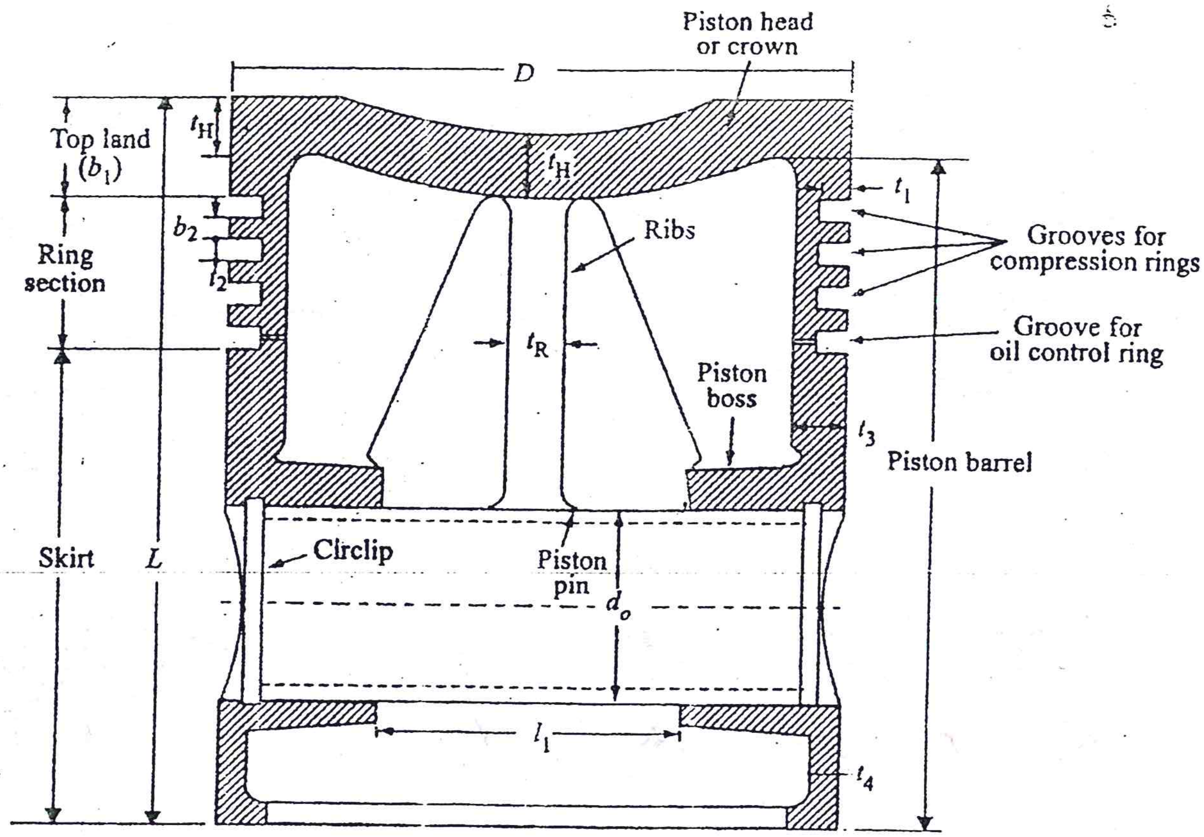

Fig 2.1 Piston for I.C. engines

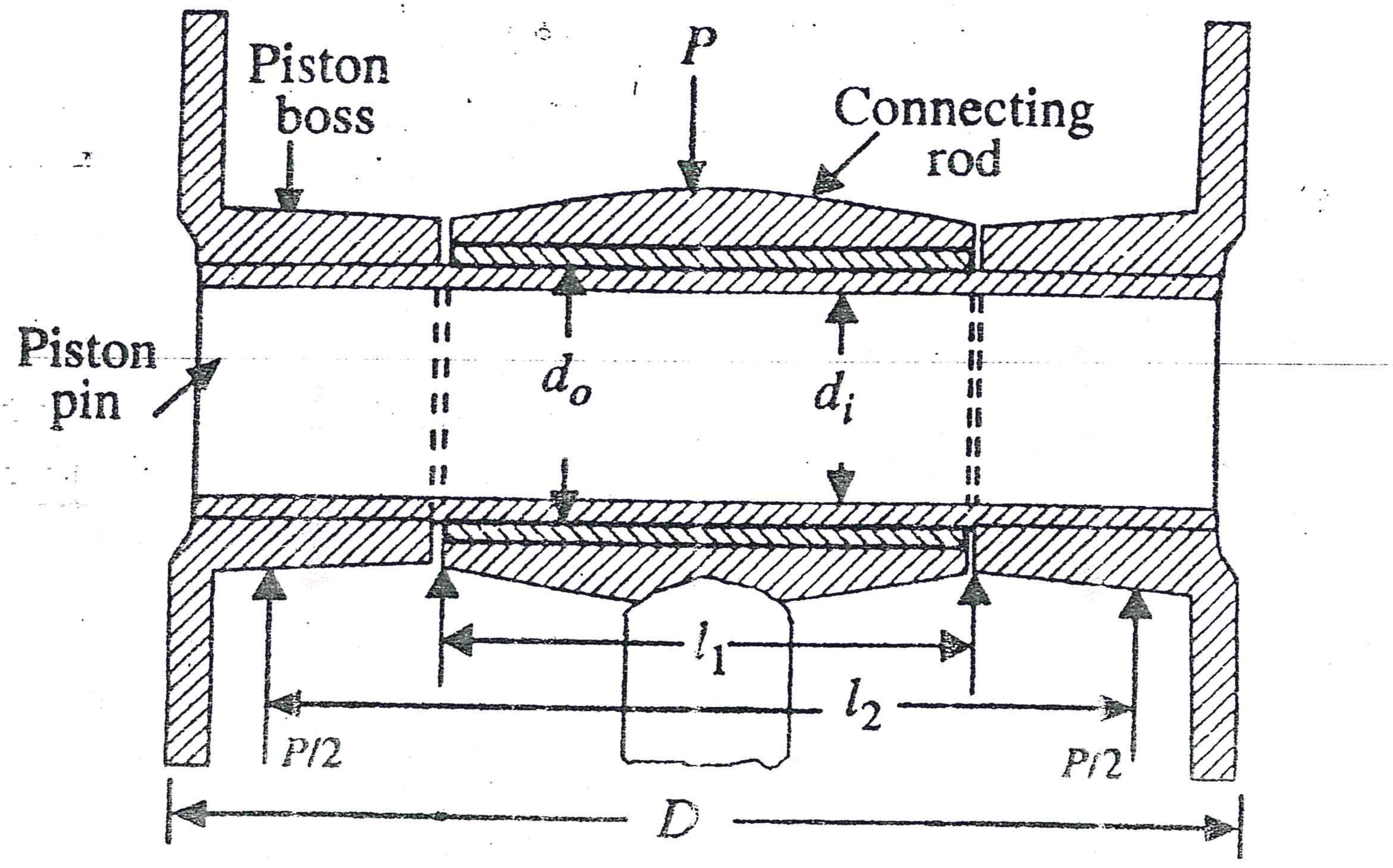

Fig. 2.2 Piston Pin

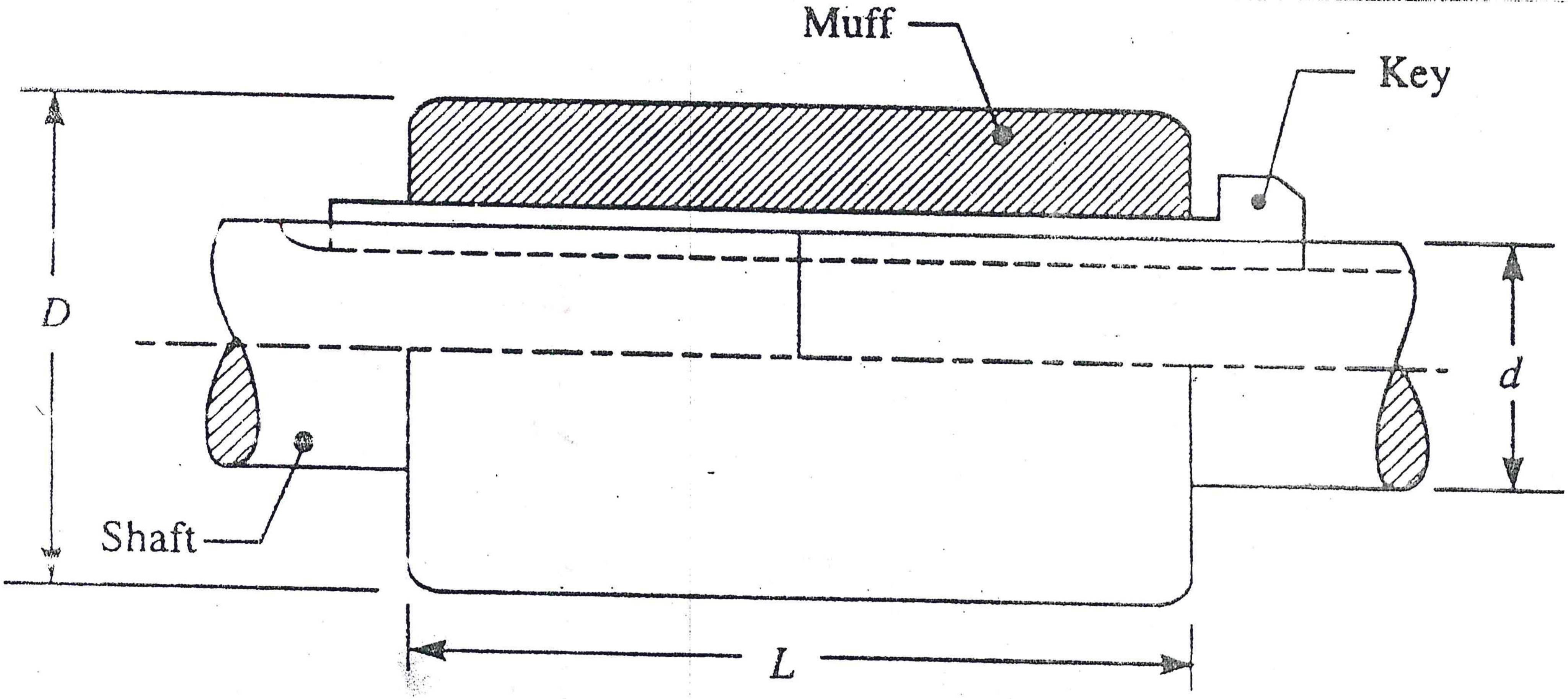

Fig. 3.1 Sleeve or muff coupling.

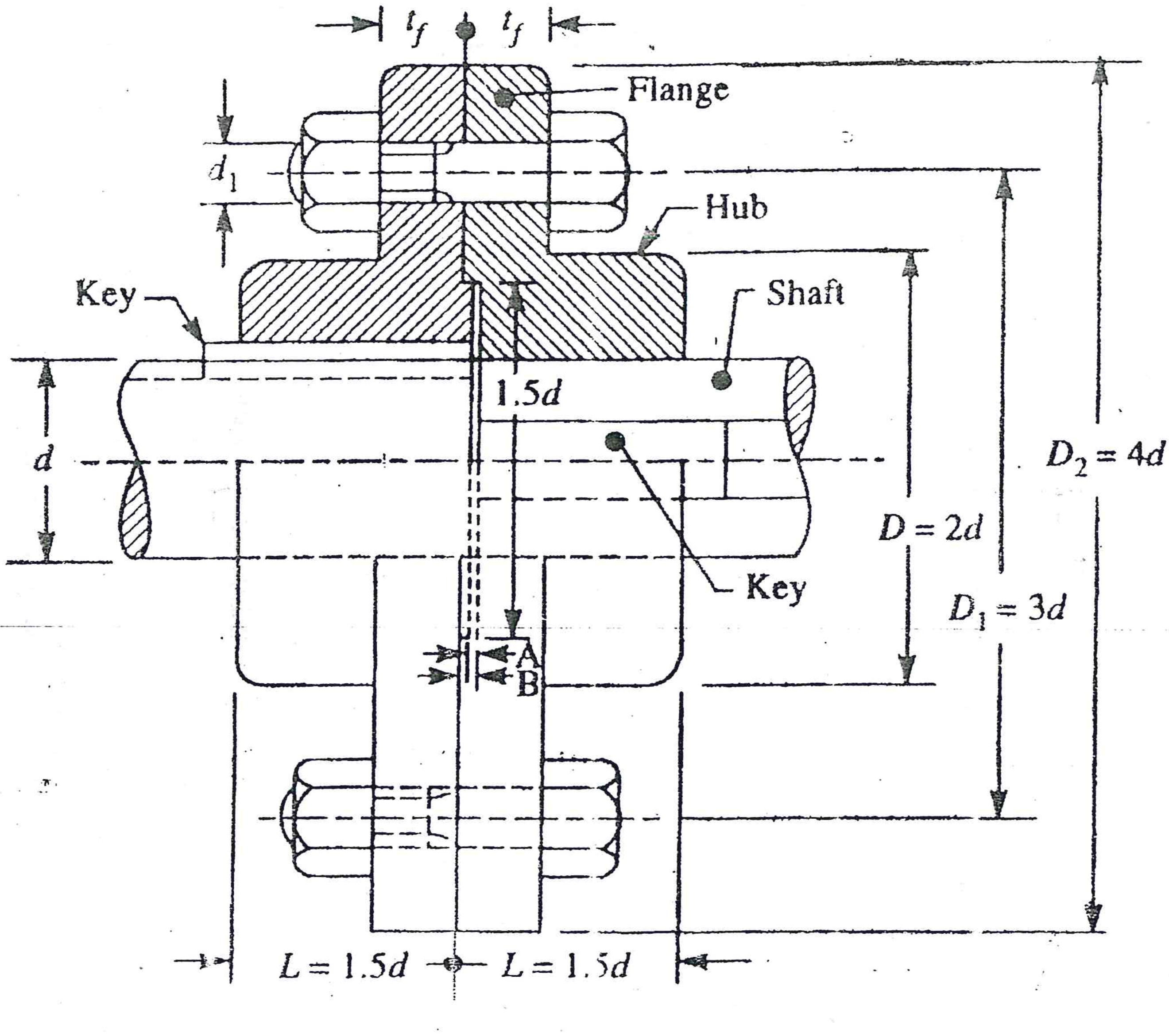

Fig 3.2 Unprotected type flange coupling.

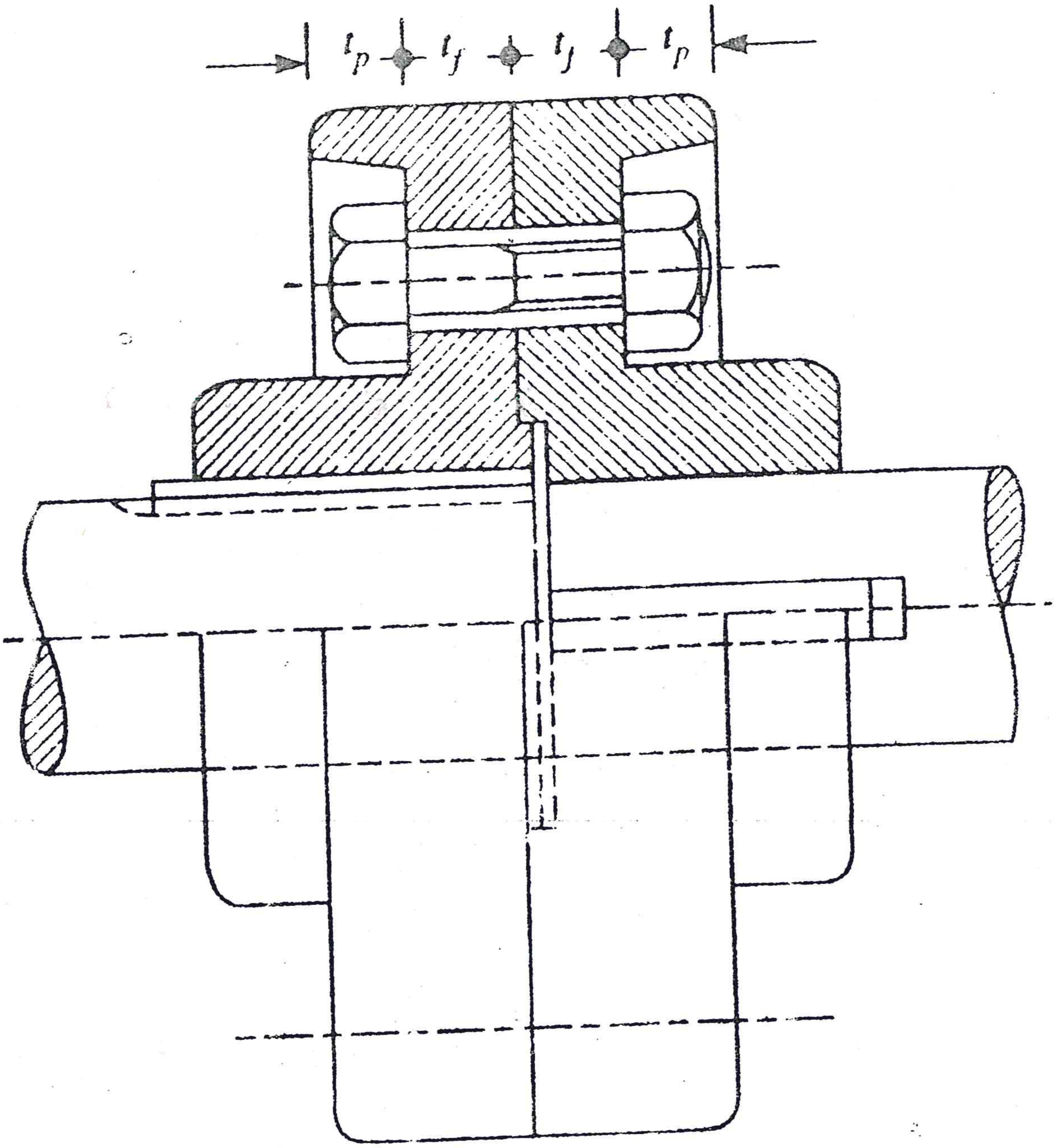

Fig 3.3 Protective type flange coupling.

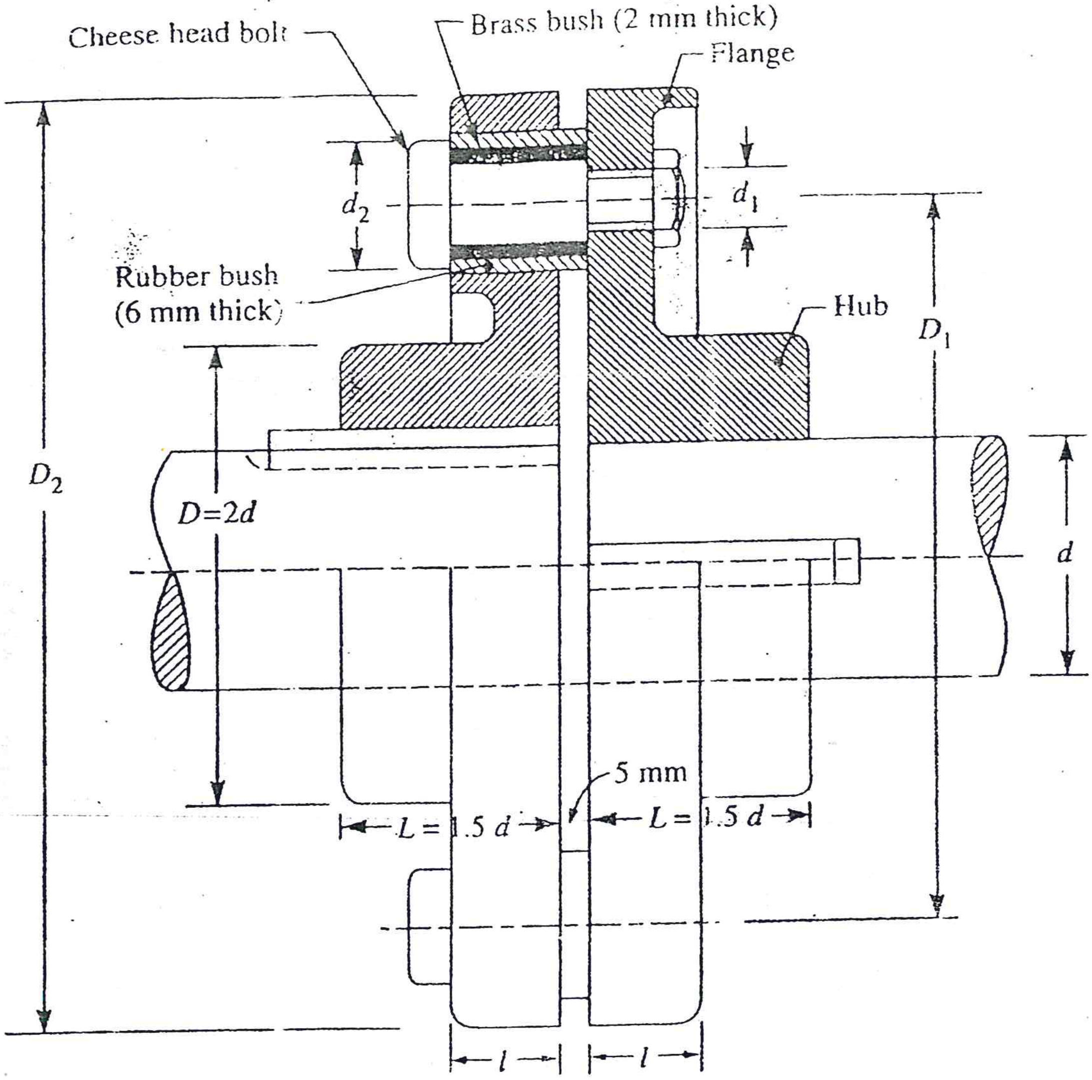

Fig 3.4 Bushed-pin flexible coupling.

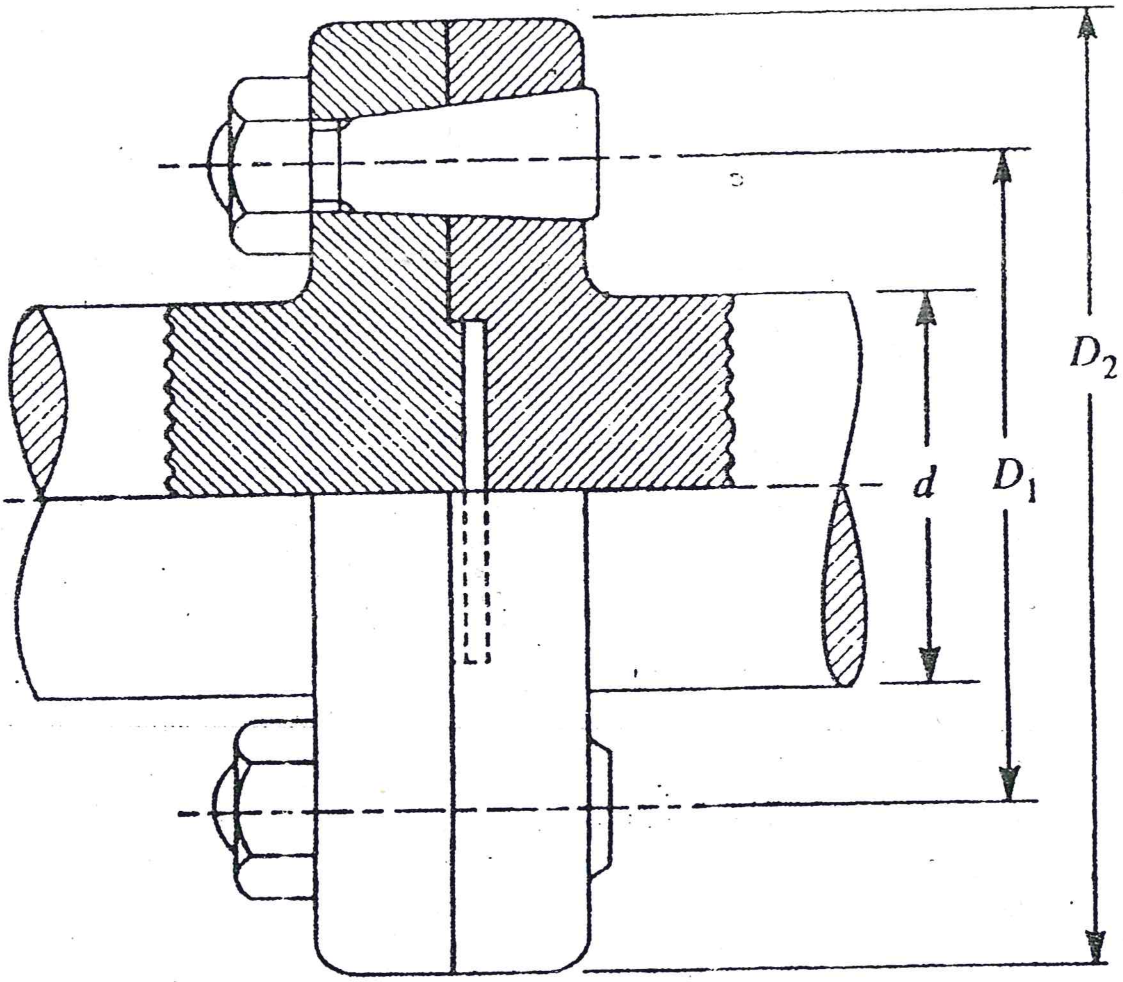

Fig 3.5 Marine type flange coupling.

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ANNA UNIVERSITY : : CHENNAI - 600 025

ADDITIONAL DATA SHEETS APPROVED FOR UNIVERSITY EXAMINATION

with effect from November 2005

Designation of plain carbon steels:

IS Designation	
Old Designation	New Designation
C 10	10 C 4
C 30	30 C 8
C 40	40 C 8
C 35 Mn 75	35 C 8
C 45	45 C 8
C 50	50 C 4
C 60	60 C 4

Suggested Factors of safety:

Type of load	Steel / Ductile material		CI
	Based on σ_u	Based on σ_y	
Static load	3-4	1.5 - 2	5-6
Repeated, mild shock	6	3	7-8
Reversed Mild shock	8	4	10 - 12
Heavy shock	10-15	5-7	15-20

Variable loading :

Endurance limit of the machine member: (With Factors considered)

$(\sigma_{-1}) = (\sigma_{-1}.A.B.C) / K_f$

Where, A = load factor

= 1 for reversed bending

= 0.7 for reversed axial load

= 0.6 for reversed torsion

B = Size Factor

If D < 10mm then B = 1

If 10 mm < D < 50mm then B = 0.9

If 50 mm < D < 100mm then B = 0.8

If 100 mm < D < 150mm then B = 0.7

C = surface finish factor

= 0.9 for ground or cold rolled surface

= 0.7 for machined surface

= 0.6 for cold rolled surface

Relation between Endurance limit and ultimate strength:

Steel	$\sigma_{-1} = 0.5\ \sigma_u$
Cast Steel	$\sigma_{-1} = 0.4\ \sigma_u$
Cast Iron	$\sigma_{-1} = 0.35\ \sigma_u$
Non Ferrous metals and alloys	$\sigma_{-1} = 0.3\ \sigma_u$

Basquin Equation $A = \sigma_f \cdot L^B$.

$\sigma_f = 0.9\ \sigma_u$ for $L = 10^3$ cycles

σ_{-1} (*endurance limit*)= $0.5\ \sigma_u$. for $L = 10^6$ cycles

A,B= constants

Endurance strength for loading with different values and parts:

$$\sigma_{fn} = K_t\ \sigma_a\ \sigma_u / (\sigma_u - \sigma_m)$$

Where, σ_a, σ_m = variable and mean stresses for any 'part' of the loading cycle

$n = 1, 2, 3 \ldots\ldots n$

Miner's equation $(N_1 / L_1) + (N_2 / L_2) + \ldots\ldots = 1$ or

$(\alpha_1 / L_1) + (\alpha_2 / L_2) + \ldots\ldots = 1/ L_e$

Where, L_e = expected life.

N= Number of actual cycles

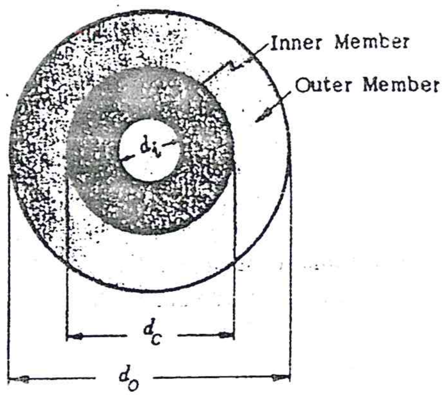
L= life at that particular part of the load

Reliability factors:

Reliability %	Kr
50	1
90	0.897
95	0.868
99	0.814
99.9	0.753
99.99	0.702

STRESSES DUE TO INTERFERENCE FITS may be calculated by considering the fitted parts as thick-walled cylinders, as shown in Fig. 3-2, by the following equations:

$$p_c = \frac{\delta}{d_c\left[\dfrac{d_c^2 + d_i^2}{E_i(d_c^2 - d_i^2)} + \dfrac{d_o^2 + d_c^2}{E_o(d_o^2 - d_c^2)} - \dfrac{\mu_i}{E_i} + \dfrac{\mu_o}{E_o}\right]}$$



where

p_c = pressure at the contact surface, N/m^2
δ = the total interference, m
d_i = inside diameter of the inner member, m
d_c = diameter of the contact surface, m
d_o = outside diameter of outer member, m
μ_o = Poisson's ratio for outer member
μ_i = Poisson's ratio for inner member
E_o = modulus of elasticity of outer member, N/m^2
E_i = modulus of elasticity of inner member, N/m^2

If the outer and inner members are of the same material, the above equation reduces to

$$p_c = \frac{\delta}{\dfrac{2d_c^3(d_o^2 - d_i^2)}{E(d_c^2 - d_i^2)(d_o^2 - d_c^2)}}$$

After p_c has been determined, then the actual tangential stresses at the various surfaces, in accordance with Lamé's equation, for use in conjunction with the maximum shear theory of failure, may be determined by:

On the surface at d_o, $\quad s_{to} = \dfrac{2p_c d_c^2}{d_o^2 - d_c^2}$

On the surface at d_c for the outer member, $\quad s_{tco} = p_c\left(\dfrac{d_o^2 + d_c^2}{d_o^2 - d_c^2}\right)$

On the surface at d_c for the inner member, $\quad s_{tci} = -p_c\left(\dfrac{d_c^2 + d_i^2}{d_c^2 - d_i^2}\right)$

On the surface at d_i, $\quad s_{ti} = \dfrac{-2p_c d_c^2}{d_c^2 - d_i^2}$

Stress concentration Factors in welds:

Reinforced Butt Weld	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T – Butt joint with sharp corner	2

Screw Fastenings:

Gasket constant : C = 1 for soft gasket

= 0.6 for copper – asbestos gasket

= 0.1 for lead gasket

= 0 when no gasket is used.

Knuckle Joint:

Proportions:

Diameter of the rod = d

Diameter of the pin, $d_1 = d$

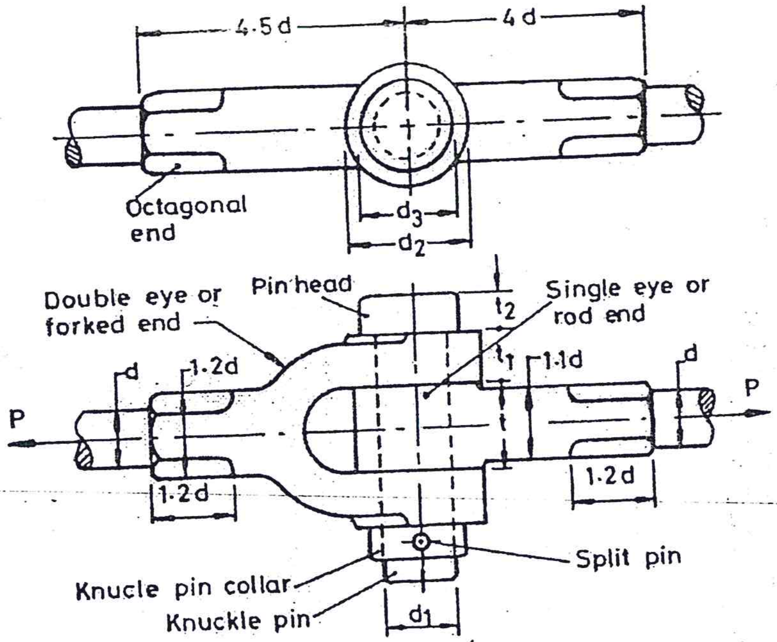
Outer Diameter of the eye, $d_2 = 2d$

Dia of the pin head, $d_3 = 1.5d$

Thickness of the eye, $t = 1.25d$

Thickness of the fork, $t_1 = 0.75d$

Thickness of the pin head, $t_2 = 0.5d$



KNUCKLE JOINT

Sleeve and Cotter Joint:

Proportions:

Diameter of the rods to be connected = d

Diameter of the enlarged end, d1 = 1.3 d

Outside Diameter of the sleeve, d2 = 2.5d

Distance of the slot from the sleeve end, a = 1.3 d

Length of the sleeve, L = 8d

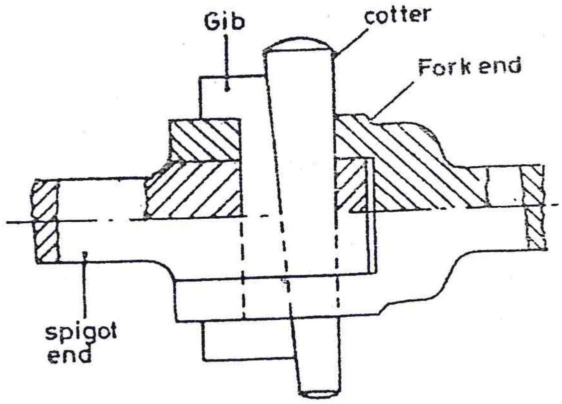
Length of the cotter, l = 4d

Width of the cotter, b = 1.3 d

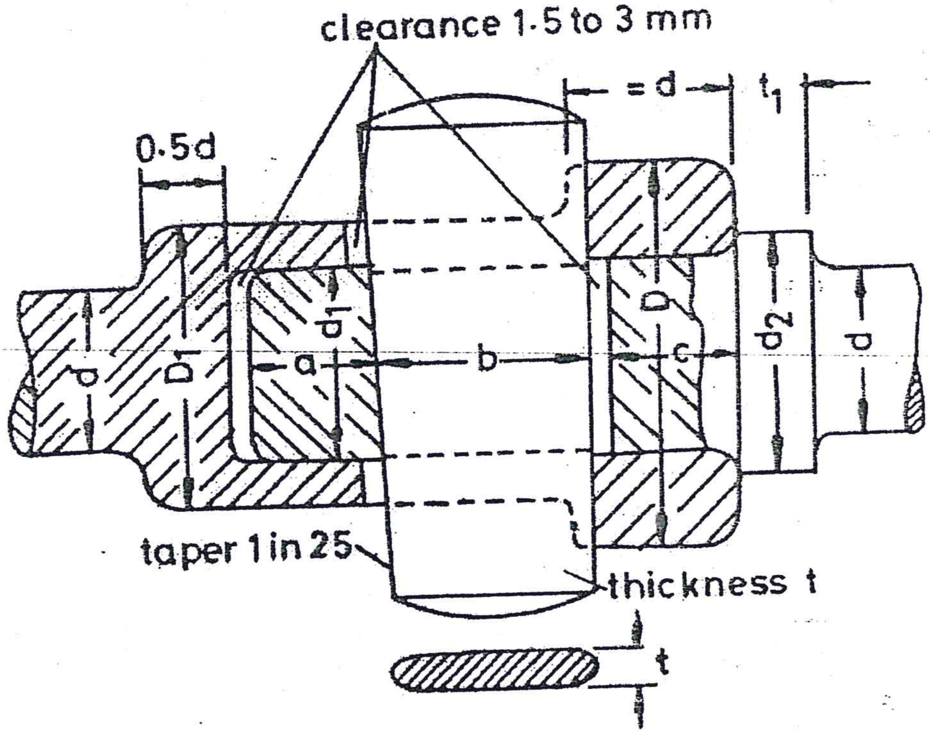
Distance of the slot from the rod end, $c = 1.4d$

Length of the enlarged end of the outside sleeve, $e = 0.5\ d$

Thickness of the cotter, $t = 0.3d$



GIB AND COTTER JOINT



COTTERED JOINT FOR RODS

Splined Connections:

Torque carrying capacity of Splined Connections, $T = p.\ A.\ r_m$

Where, p = permissible bearing pressure, $< 7\ N/mm^2$

A = total area of the splines = h.l.N

h = height of the splines = $(D - d) / 2$

l = length of the hub

n = no. of splines

r_m = mean radius = $(D + d) / 2$

Units of viscosity :

Absolute viscosity, $1\ Pa - s = 10^3\ cP$

$= Ns/m^2 = kg / m\text{-}s$

Kinematic viscosity, 1 Stoke $= 1\ cm^2 /s$

Leaf Spring:

Nip, $h = 2PL^3 / (nEbt^3)$

Where, n = total no. of leaves

b = width of leaves

t = thickness of leaves

Load on the clip bolts required to close the gap:

$$P_b = 2n_e . n_g . P / n (2 n_g + 3 n_e)$$

where, n_e = no. of extra full length leaves

n_g = no. of graduated leaves

APPROVED DATA SHEETS

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CONTROLLER OF EXAMINATIONS

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