وزارة التعليم العالي والبحث العلمي
هيئة التعليم التقني
المعهد التقني / النجف
قسم تقنيات الميكانيك

الحقيبة التعليمية لمادة اجزاء مكائن للطلبة السنة الثانية /ميكانيك

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Types of stress

1- Compression stress — اجهاد الضغط
2- Ten sion stress — اجهاد السحب
3- Bending stress — اجهاد الانحناء
4- Shear stress — الاجهاد القص

F $\Rightarrow$ Force ; A $\Rightarrow$ area ; σ الاجهاد

$\sigma = \frac{f}{a}$ القانون العام $\Rightarrow$ N / mm^2 الوحدات

$\sigma\ c = \frac{fc}{a}$; $\sigma\ t = \frac{ft}{\sigma}$; $\tau\ s = \frac{fs}{a}$

$\sigma\ b = \frac{Mb * y}{I}$; Mb عزم الانحناء N.mm

I عزم القصور الذاتي mm^4

= N/mm^2 وحدات — y نصف قطر

*Subjected area to stress

A- For solid shaft

$A = \frac{\pi}{4}\ d^2$ العمود الأول

B- For Hollow shaft

$A = \frac{\pi}{4}\ (do^2 \_ di^2)$

Ex Find shear stress and tension stress for shaft (32mm) when shear force and tension force (400 N; 600N)?

$\tau\ s = \frac{Fs}{A}$; $A = \frac{\pi}{4}\ d^2$; $A = \frac{\pi}{4}\ (32)^2 = 772 mm^2$

τ s = $\frac{400}{772}$ = 0.05 N/mm^2

σ t = $\frac{Ft}{A}$ = $\frac{600}{772}$ = 0.86 N/mm^2

Ex **Find Bending strees when shaft subjected to force the diam of shaft (31mm) the length of shaft (82mm) the pollar Moment (120mm^4)**

σ b = $\frac{Mb * y}{I}$; Mb = F* L

= 200 * 82

= 16400 N. mm

σ b = $\frac{16400 * 18}{120}$

Ex **For Hollow shaft the outer diam. and inner diam. (78mm & 58mm) Find the shear - stress and compression stress the shear force and comp force (800N ; 600N) .**

A= $\frac{\pi}{4}$ (do^2 _ d i^2) ; A= $\frac{\pi}{4}$ (78^2 _ 58^2) = 427mm^2

τ s = $\frac{Fs}{As}$; τ s = $\frac{800}{427}$ = 1.8 N/mm

σ c = $\frac{Fc}{A}$ = $\frac{600}{427}$ = 1.6 N/mm

Ex **The ratio ($\frac{di}{do}$ = 0.8) and the outer diam (120mm) find the shear stress and tension stress when shear force ; and tension force (1600N ; 1200N)**

$\frac{di}{do}$ = 0.8

di= do * 0.8 = 120 * 0.8

= 96 mm

$A = \frac{\pi}{4}(do^2 - di^2)$

$= \frac{\pi}{4}(120^2 - 96^2) = 336mm^2$

$\tau s = \frac{Fs}{A}$; $\tau s = \frac{1600}{336} = 5.6\ N/mm^2$

$\sigma t = \frac{Ft}{A} = \sigma t = \frac{1200}{336} = 4.2\ N/mm^2$

Hock. Law قانون هوك

$\delta = \frac{F * L}{A * E}$

$\delta \Longrightarrow$ deflection in the length .

F $\Longrightarrow$ Force ; L $\Longrightarrow$ Length ; A $\Longrightarrow$ area ;

E $\Longrightarrow$ Modulu young معامل يونك

Ex For pipe the outer and inear diam (36- 30mm) at length (260mm) Find the reduction in the length if subjected to compression force (4300N) and Modulun younge ($E = 117 * 10^3\ N/mm^2$)

$A = \frac{\pi}{4}(do^2 - di^2)$; $A = \frac{\pi}{4}(36^2 - 30^2) = 311$

$\delta = \frac{F * L}{A * E} = \frac{4300 * 260}{311 * 117} = 0.0307\ mm$

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الاسبوع الثاني-الرابع

The General Law Bettwen (T; τ ; θ)

القانون العام بين العزم والاجهاد القاص وزاوية الالتواء

$\frac{T}{J} = \frac{\tau}{R} = \frac{G\,\theta}{L}$ وحدات

T $\Longrightarrow$ Torque N.m ; N.cm ; N.mm

J $\Longrightarrow$ Pollar Moment m^4 ; cm^4 ; mm^4

$\theta \Longrightarrow$ angle torsion rad

Deg * $\frac{\pi}{180} \Longrightarrow$ (rad)

G $\Longrightarrow$ Modlus of Rigidity N/mm^2

τ $\Longrightarrow$ shear stress ; L $\Longrightarrow$ Length ; R $\Longrightarrow$ radius of curvature

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*** Law of solid shaft** *قانون العمود الصلد*

$J = \frac{\pi}{32} d^4$; $A = \frac{\pi}{4} d^2$

***Law of Hollow shaft**

$J = \frac{\pi}{32} (do^4 - di^4)$; $A = (\pi/4) (do^2 - di^2)$

****when shaft subjected Torque M*** عندما يتعرض العمود الى عزم التواء

$d = 1.72 \sqrt[3]{\frac{T}{\tau}}$ $\frac{T}{J} = \frac{\tau}{R} = \frac{G\,\theta}{L}$

For solid shaft $\frac{\pi}{16} d^3 = \frac{T}{\pi}$

$\frac{T}{J} = \frac{\tau}{R}$ $\frac{T}{\frac{\pi}{32} d4} = \frac{\tau}{\frac{d}{2}}$ $\frac{T}{\tau} = \frac{\pi}{32} d^4 \times \frac{2}{d}$ $d = 1.72 \sqrt[3]{\frac{T}{\tau}}$

***For Hollow shaft**

$\frac{T}{J} = \frac{\tau}{R} = \frac{G\,\theta}{L}$ $\frac{T}{J} = \frac{\tau}{R}$; $J = \frac{\pi}{32} (do^4 - di^2)$; $R = \frac{do}{2}$

$$\frac{T}{\frac{\pi}{32}(do4 - di4)} = \frac{\tau}{\frac{do}{2}}$$

$$\frac{T}{\tau} = = \frac{\frac{\pi}{32} do\,4 \left(1 - \frac{di\,4}{do\,4}\right)}{\frac{do}{2}}$$

Let $K = \frac{\pi}{32} d^{o4} (1 - K^4) * \frac{2}{do}$

$$\frac{T}{\tau} = \frac{\pi}{16} do^3 (1 - K^4)$$

$$do^3 = \frac{\frac{T}{\tau}}{\frac{\pi}{16}(1-K)} = \frac{T}{\tau} * \frac{16}{\pi(1-K4)}$$

$$do = \sqrt[3]{\frac{16}{\pi}} * \frac{T}{\tau(1-K4)} = 1.72\sqrt[3]{\frac{T}{\tau(1-K4)}}$$

Ex For solid shaft (150mm) diam is subjected to torque (48+ 10^6 N.mm)Find Max shear stress and the torsion angle if Modulu Rigidity?

(34* 10^3 N/mm) with length 600mm

$$J = \frac{\pi}{32} d^4 = \frac{\pi}{32}(150)^4 = 49.6 * 10^6 \text{ mm}^2 ; \frac{T}{J} = \frac{\tau}{12} ; \tau = \frac{T}{J} * R$$

$$= \frac{48 * 10\ 6}{496 * 10\ 6} * \frac{150}{2}$$

τ **= 7.25 N/mm²**

$$\frac{T}{J} = \frac{G\theta}{L} \qquad \theta = \frac{T}{J} * \frac{L}{G} = \frac{48 * 10\ 6}{49 * 10\ 6} * \frac{600}{34 * 10\ 3}$$

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Ex For pipe the outer and innear diam (80mm; 64mm) is subjected to torque (42.8 * 10^6 N.mm) Find shear stress and the length shaft if torsion angt (20°) and Modulu of Rigidity (28* 10^3 N.mm)

$$J = \frac{\pi}{32}(do^4 - di^4) ; J = \frac{\pi}{32}(80^4 - 60^4)$$

$$J = 40.6 * 10^6 \text{ mm}^4$$

$$\frac{T}{J} = \frac{\tau}{R} ; \tau = \frac{T}{J} * R$$

$$\tau = \frac{42.8 * 10\ 6}{40.6 * 10\ 6} * \left(\frac{80}{2}\right) = 6.4 \text{ N/mm}^2 .$$

$\frac{T}{J} = \frac{G\,\theta}{L}$; L = $= \frac{40.6}{42.8} * 28 * 10^3 * 20 * \frac{\pi}{180}$

محاضرة الاسبوع الخامس والسادس

Key الخابور

تعريف الخابور : وهي قطعة من الحديد المطروق تستخدم لربط قطعتين معاً بحي4ث يمنع أي حركة بينهما

Types of Keys *أنواع الخوابير*

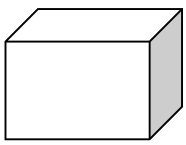
1- الخوابير المماسية

2- الخوابير المدورة

3- الخوابير الغاطة وهذه على عدة أنواع :

أ الخابور المربع squer - Key

$b = \frac{D}{4} = t$

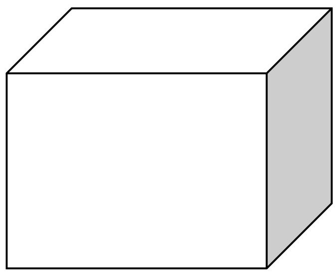
t 

b

ب الخابور المستطيل Rectangular – Key

$b = \frac{D}{4}$

$t = \frac{2}{3}\, b$

t 

b

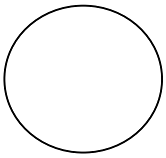
جـ خابور ذي الدقن Tauqer with gib head

$B= \frac{D}{4}$; $T = \frac{D}{6}$

$L = 1.57 * D$

Spline – shaft المحددات

$b = \frac{D}{4}$



D= 1.25 – d

Length of Key

1- Rectangular Key *المستطيل*

$L = \frac{\pi}{8} * \frac{D^2}{b}$

When d= 20

$b = \frac{D}{4} = \frac{20}{4}\ 5$

$L = \frac{\pi}{8} * \frac{D^2}{b} \quad = \frac{\pi}{8} * \frac{400}{5} = 31.4$

2- Squar Key

$L = 1.57 * D$

D = 10

$L = 1.57 * 10 \ = 15.7$

عند تعرض العمود الى إجهاد سحق

$L = \frac{\pi}{4} * \frac{D^2}{t}$ d= 20; $t = \frac{2}{3} * b$

$= \frac{2}{3} * 5 = 3.34$

$L = \frac{\pi}{4} * \frac{20^2}{3.34}$

L = طول الخابور الرابع في حالة السحق

<u>Ex</u> Design Rectangular Key befor and after at diam (50mm)Find all diamension?

$b = \frac{D}{4} = \frac{50}{4} = 12.5$

$t = \frac{2}{3} b = \frac{2}{3} * 12.5 = 8.3$

$L = \frac{\pi}{8} * \frac{D^2}{b}$

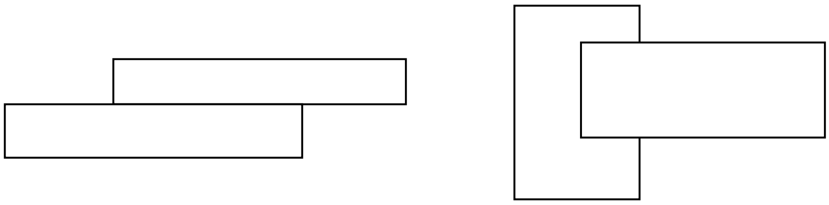
$= \frac{\pi}{8} * \frac{50^2}{12.5} = 78.5$

$L = \frac{\pi}{4} * \frac{D^2}{t} = \frac{\pi}{4} * \frac{50^2}{8.3}$

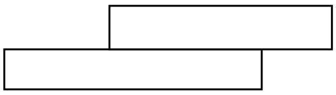
Welding اللحام

1- Lap. Welding Joint لحام تراكبي

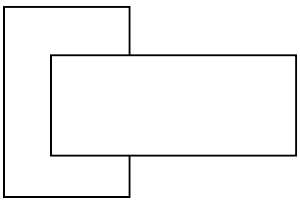
A- Single Lap welding. Joint



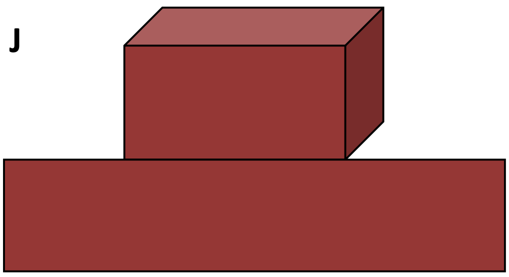
B- Double Lap welding. J



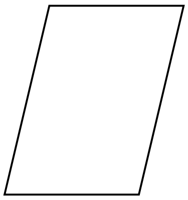
C- Paralled Lap. Welding. J



D- Double Lap. W. J



E- Cylider Lap.w.J



Bolt & Nut البرغي والصامولة

Law of Bolt and nut

h = 0.56 D **K = 0.7D**

do = di + 2h **H = 0.8D**

di = do – 2h **e = 1.72D**

S = 1.5D ; b = 2D

$$\sigma_t = \frac{ft}{n\,(At)}$$

σ_t اجهاد سحب n = عدد البراغي

Ft $\Longrightarrow$ قوة السحب

$$= \frac{ft}{n\left(\frac{\pi}{4}\,d^2\right)}$$

$$\tau s = \frac{fs}{n\left(\frac{\pi}{4}\,d^2\right)}$$

<u>Ex</u> For Bot find all diamension with tension stress and shear stress to the 6 Bolt if the diam of Bolt (18mm)at ft ; fs (400N; 600N)

h= 0.65D **e = 1.72D** **do = di + 2h**

K = 0.7D **S = 1.5D** **di =do – 2h**

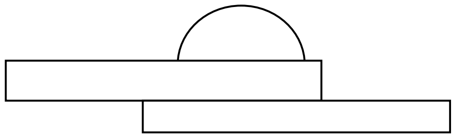
H = 0.8D **b = 2D**

محاضرة الاسبوع السابع والثامن

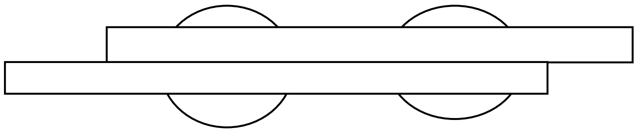
Rivet Joint

1- **Lap. Riveted Joint**

A- **Single Lap. R. J**

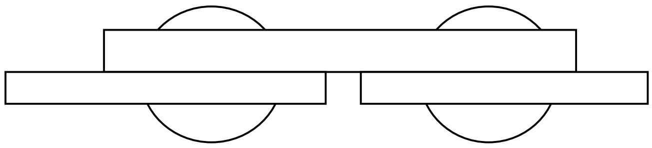


B- **Double Lap. R. J**

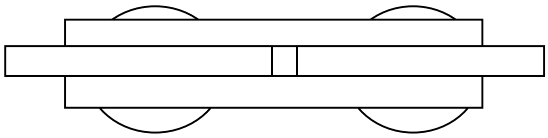


2- **Butt Rivet Joint**

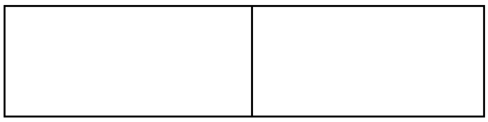
A- **Single Butt. Joint**



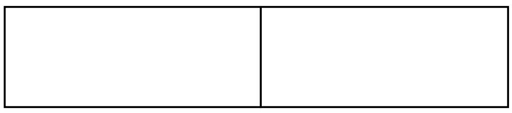
B- **Double Butt Joint**



C- **Multiple Rivet. J**



D- **Zig. Zag Butt R. J**



Type of failure فشل قصي

1- **Shearing failure**

2- **Crushing failure**

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Law of Riveted stress

1- Shearing stress

$$\tau = \frac{Fs}{n\left(\frac{\pi}{4}\ d\ 2\right)}$$

$\tau =$ اجهاد تماس

Fs = قوة قص ; n = عدد القطع المعرضة

2- Beanding stress اجهاد انحناء

$\sigma b =$ اجهاد انحناء ; t = السمك

Fb = قوة الانحناء ; d = قسط البرشام

n = عدد القطع

3- Tension stress

$\sigma t =$ الشد والسحب ; Ft = قوة

P = الخطوة ; d = قسط البرشام

4- Efficieney of Rivet كفاءة البرشام

$$\delta = \frac{fs;Fb;Ft \text{ اقل قيمة}}{f} \times 100\%$$

f =

$$\delta = \frac{fs;Fb * Ft \text{ اقل قيمة}}{\sigma t * p * t} \times 100\%$$

<u>Ex</u> Single lap joint rivet the thickness(1.5cm); the diam of rivet (20mm). Find the efficiency of Rivet when pitch (6cm)

2 ; $\sigma b = 1600\frac{N}{cm^2}$; $\sigma t = \frac{1200N}{c\square}$ 2

$$\tau = \frac{Fs}{n\left(\frac{\pi}{4}d\ 2\right)}$$; Fs = N * $\pi/4$ * d² * τ

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Fs =900 * l * $\frac{\pi}{4}\left(\frac{20}{10}\right)2 = 2827\ N$

Fb = 4800N

Ft = (p – d) $* t * h * \sigma t$ = (6 – 2)$* 1.5 * 1 * 1200 = 7200N$

$\delta = \frac{Fs;Fb;Ft}{\sigma t * p * t} * 100\%$ $= \frac{2827}{1200 * 6 * 1.5} * 100\%$ = 26%

Ex Find the efficiency of Rivet at thickness(t= 8mm); $\tau = \frac{30N}{m□^2}$ the width 70mm diam of Rivet (18mm) at Joint three picea; $\sigma b = 50$ N/mm²; $\sigma t = 60$

$\tau s = Fs/(n(\pi/4\ d^2))$; Fs= $d^2 * n$

Fs= $30 * \frac{\pi}{4} * 18^2 * 3 = 22892N$

Fb= $\sigma b * (t * d) n$

= $50 * (8 * 18) * 3 = 7200N$

Ft= $\sigma t * (p - d) * n * t$

$60 * (70 - 18) * 3 * 8 = 21600N$

$\delta = \frac{Fs;Fb;ft \text{ اقل قيمة}}{\sigma t * p * t} * 100\%$

$= \frac{7200}{60 * 70 * 8} * 100\% = 21\%$

الاسبوع التاسع والعاشر

Spring

Spring are used at

1- Appling force or Controling and brake and clutch
2- Reducing the effect of shocks
3- Storing energy
4- Measuring force

Type of Spring

1- Helical spring نابض حلزوني

a- Compression helical spring

b- Tension helical spring

2- Leaf spring نابض ورقي

3- Torsional spring نابض التواء

Free Length of spring

Lf = Ls + $\delta +$ E   Ls =(n+d)   الطول المضغوط

E = (n – 1) 0 – 1 خلوص النابض   ; δ الاستطالة

Spring index(C) دليل النابض

C = $\frac{dm}{d}$; dm = متوسط القطر d = قطر النابض

dm = di + d ; dm = do – d

d = $\frac{do - di}{2}$

*Stiffness spring معدل متانة النابض

Ks = $\frac{F}{\delta}$ F = force ; $\delta =$ الاستطالة

*Pitch(p)

P = $\frac{Lf}{(n-1)}$

*Shesr stress of spring

$$\tau = \frac{8 * f * dm}{\pi d^3} * K \qquad K = \frac{4c - 1}{4c - 4} + \frac{0.615}{c}$$

*Deflection of spring

$$\delta = \frac{8 * n * F * dm^3}{G * d^4}$$

; G = معامل الصلادة N/mm²

Ex : Helical spring the mean diam (25mm) and wire diam(3mm) if the shear st (44* 10^6 N/mm^2)and deflection(25mm)& G= 86.2* 10^4 N/mm^2

Find: 1- max loading force   2- No of turn   3- Pitch

C = $\frac{dm}{d} = \frac{25}{3} =$ **8.3mm**

K = $\frac{4c-1}{4c-4} + \frac{0.615}{c}$ $= \frac{8.3*4-4}{4*8.3-1} + \frac{0.615}{8.3} =$ **1.17**

$\tau = \frac{8*F*dm*K}{\pi d^3}$

F = $\frac{\pi d^3 * \tau}{8*dm*K}$ = $\frac{\pi 3^3 * 44*10^6}{8*25*1.17}$ **= 240N**

$\delta = \frac{8*n*dm^3*F}{G*d^4}$

n= $\frac{\delta*G*d^4}{8*dm^3*F} = \frac{25*86.2*10^4*3^4}{8*25^3*246}$ **n = 10**

p = $\frac{Lf}{(n-1)}$ = $\frac{(n+d)+\delta+(n-1)*0.1}{(n-1)}$ = $\frac{10*3+25(10-1)*0.1}{(10-1)}$ **P =2.5mm**

Ex: helical spring the diam of wire (6mm) the main diam97.5mm); shear st.σ = 350*10^3 N/mm^2 ; G= 840N/mm^2; No of coil(30turn)

Find: 1- loading force 2- deflection 3- Free length

4- Solid length &E

C = $\frac{Dm}{d} = \frac{7.5*10}{6} = 12.5$

K = $\frac{4c-4}{4c+4} + \frac{0.615}{c}$ $= \frac{4*12.5-4}{4*12.5-1} + \frac{0.615}{17.5} = 2.17$

$\tau = \frac{8*F*Dm*K}{\pi d^3}$

350 = $\frac{8*F*75*2.17}{\pi(6^3)}$

∴ **F =198N**

2- $\delta = \frac{8*n*F*dm^3}{G*d^4} = \frac{8*30*198*(75)^3}{840*6^4}$ **= 0.303**

Lf = (n+d) +$\delta + E$

= (30 + 6) + 0.303 +(n – 1) $* 0.1)$ = 41

3- Ls = (n* d) = (30* 6) = 180mm

E = (n -1)* 0.1

= (30 – 1)* 0.1 = 2.9mm

Ex: Helical spring diam of wire(6mm)and outer diam (75mm); 950N/mm^2 shear stress. (350) Find

1- Loading force 2- deflection per no of

3- stiffness of spring

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Dm = do – d = 75 -6= 69mm

C = $\frac{dm}{d}$ = $\frac{69}{6} = 11.5$

K = $\frac{4C-1}{4C-4} + \frac{0.615}{C}$ = $\frac{4+11.5-1}{4+11.5-4} + \frac{0.615}{11.5} = 1.12$

$\tau = \frac{8 * f * dm * K}{\pi d3}$; F = $\frac{\tau + \pi d3}{8 * dm * k}$ F = $\frac{350 * \pi * (6\,3)}{8 * 69 * 1.1} =$ 289N

$\tau = \frac{8 * n * dm\,3 * F}{G * d\,4} = \frac{8 * 69\,3 * 289}{950 * 6\,4} = \frac{\delta}{h}$

$\frac{\delta}{n} = 9.2 *$ 10^{-3} mm

Ks = $\frac{F}{\delta} = \frac{289}{9.2 * 10\ -3}$

Ex: For Helical spring the ratio between innear out diam(0.6); the shear stress(640N/mm^2) and 360N/mm^2 at outer diam (120mm) Find; No of Cool24

1- Stiffness of spring 2- Pitch

$\frac{di}{de} = 0.6; di = 120 * 0.6 = 72$

d = $\frac{do - di}{2}$ = $\frac{120 - 72}{2} = 24mm$

dm =do – d =120 -24 =96mm

$\tau = \frac{8 * F * dm * k}{\pi d^3}$; C = $\frac{dm}{d} = \frac{96}{24} = 4$

K = $\frac{4C - 1}{4C - 4} + \frac{0615}{C}$; K = $\frac{4 * 4 - 1}{4 * 4 - 4} + \frac{0.615}{4} = 0.8$

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F = $\frac{\tau * \pi * d^3}{8 * dm * K}$ = $\frac{640 * \pi * (21.1)^3}{8 * 96 * 4} = 105N$

$\delta = \frac{8 * n * dm^3 * F}{G * d^4} = \frac{8 * 24 * 96^3 * 105}{360 * 24^4}$

Ks = $\frac{f}{\delta} = \frac{105}{0.02}$ = 0.02

P = $\frac{Lf}{n - 1}$

الاسبوع الحادي عشر والثاني عشر

Belts and Pulleys

*Types of belts

1- Flal belt

2- Vee-belt

3- Circular belt

Speed Condition نسبة السرع

A- Single Condition

B- Compound condition

$\frac{N_2}{N_1} = \frac{d_1}{d_2}$ $\frac{N_4}{N_1} = \frac{d1 * d_3}{d2 * d_4}$

Ex : It is the drive shaft at (N1 = 620 r. p. m) by means belt from parallel shaft having apuling(d1= 30cm)& running at(240 r. p. m)

N1 =620; d1= 30; N2= 240

N1 d1 =N2 d2

620 * 30 = 240* d2

d2 = $\frac{620 * 30}{240} = 77cm$

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Ex: For compound Coneltion the diam(d1= 50cm; d2= 25cm ; d3= 70; d4= 28) and rotation speed at driver shaft(N1= 180r. p. m) find the rotation speed in driven shaft ?

$\frac{N4}{N1} = \frac{d1 * d3}{d2 * d4}$; N4 = N1 * $\frac{d1 * d3}{d2 * d4}$

N4 =180 * $\frac{50 * 70}{25 * 28} = 900 r.p.m$

Power transmitted by belt

P =(T1 –T2) * V ; P =power ; watt = $\frac{N.m}{s}$

T1 = Max tension Force

T2 = Min tension force

V = velocity

PA = (T1 – T2) VA

PB = (T1 –T2)VB

ملاحظ: إذا أعطيت قيمة T بالكيلو غرام يضرب بقيمة التعجل الأرضي(9.81)

Ex: The tension in two side(100Kgf; 80Kgf)and the velocity 75m/s Find the power in hours power

$P = \frac{T1 - T2}{750} * V$; $P = \frac{100 - 80}{750} * 75$ = 2H. p

Slip of belt الانزلاق في السيور

A- For single conclition

$\frac{N2}{N1} = \frac{d1}{d2}\left(1 - \frac{st}{10V}\right)$ st =(s1+s2)

B- For compound coneltion $\frac{N4}{N1} = \frac{d1 * d3}{d2 * d4}\left(1 - \frac{st}{100}\right)$

-22-

Ex: An engine running(150r. p. m) for driver shaft the diam of pulley (d1=750; d2= 450; d3= 900) Find the final rotation speed if the silp in two side 2 L

S1 =0.02 ; S2 =0.02 ; St = 0.04

$\frac{N4}{N1} = \left(\frac{d1 * d3}{d2 * d4}\right)\left(1 - \frac{st}{100}\right)$

N4 = N1($(d1 * d3)/(d2 * d4))(1 - 0.04)$ = 150 $\left(\frac{750 * 900}{450 * 150}\right)(0.96) = 1440$

Ratio of tension in belt

$\frac{T1}{T2}$ = m$^{\theta}$; T1 = max tension force ; T2 = max

M or μ = Cofficient of friction ; $\theta =$ Angle of contact

Or 2.3 Log $\frac{T_1}{T_2}$=e$^{\mu\theta}$

V = $\frac{\pi DN}{60}$ تحويل السرعة الدورانية الى خطية

*length of belt

A-open sys

L = [$\pi(r1 + r2) + (r1 - r2)2/x + 2X$]

L= [$\pi(r1 + r2) + (r1 + r2)2/x + 2X$] ; r1; r2 أنصاف اقطار

-23-

Ex: Find the force power in A&B transmitted by belt running over pulley(60cm; 40cm) at rotation speed(200r. p. m; 120r. p. m)with coffieient of fricition(0.25) and angle (160°) and max tension force(250N)?

VA= $\frac{\pi DN}{60} = \frac{\pi\left(\frac{60}{100}\right)*200}{60} =$ **6.28m/s**

VB = $\frac{\pi\left(\frac{40}{100}\right)*120}{60} =$ **2.5m/s**

T2 =124N

PA = $\frac{T1-T2}{750}*$ **VA =** $\frac{250-124}{750}*$ **6.28 = 1.05 H.P**

PB = $\frac{T1-T2}{750}*VB$ **=** $\frac{\square}{*2.5} = 0.42\ H.P$

Ex: Find the length of belt in open and Cross sys the diam in drive and driven (480cm; 80cm) the distance between the center of pulley(1200cm)

r1= $\frac{d1}{2} = \frac{480}{2} = 240\,;\qquad r2 = \frac{80}{2} = 40$

L open = [$\pi(r1+r2)+(r1-r2)2/x+2x]$

=[$\pi(240+40)+\ (240-40)2/1200+2*1200]$ **= 3313cm**

L cross= [$\pi(r1+r2)+\frac{(r1+r2)2}{x}+2x$

=[$\pi(240+40)+\frac{(240+40)2}{1200}+2x1200$ **= 3344.5cm**

Ex: Find the force power transmitted by Belt running over pulley (60cm)diam. At rotation speed(200r. p. m); 0.25; angle(160°) and max tension force(250N)?

V = $\frac{\pi dN}{60} = \frac{\pi\left(\frac{60}{100}\right)}{60} * 200 = \frac{6.28m}{\square}$

T2 = $\frac{250}{0.25 * 160 * \frac{\pi}{80}}$ **= 124N**

P = $\frac{T1 - T2}{750}$ * **v =** $\frac{750 - 124}{750}$ * **6.28 = 10.5H.P**

Ex: Compute the power for pulleys(A & B) the diam the driven and driving(140mm; 80mm) with rotation speed91200; 800 r. p. m); M=0.25and max tension force(650N)and Angle(110°)?

VA = $\frac{\pi DN}{60} = \frac{\pi\left(\frac{140}{1000}\right) * 1200}{60} =$ **(2.5m/s)**

VB = $\frac{\frac{80}{1000} * 800}{60}$ **=10.5m/s**

310N PA =

PB= $\frac{T1 - T2}{750} * VB = \frac{640 - 310}{750} * 10.5$ **= 5.6H.P**

EX: Deter mine the efficiency of rivet with thickness(6mm); shear stress(150N/mm²)when tension and Bending stress9180; 240N/mm²)?

$\tau s = \frac{fs}{n\left(\pi/4 \;\; d2\right)}$ **;** $\tau s * n * \frac{\pi}{4}$ **d^2**

fs= 150* $2 * \frac{\pi}{4}$ * **(12^2) =**

-25-

fb= $\sigma b * \frac{\pi}{4} * n(t * d)$ **= 240*** $\frac{\pi}{4} * 2(6 * 12)$

ft= $\sigma t * (p - d) * t * n = 180 * (36 - 12) * 6 * 2$

$$\delta = \frac{ft; fb; fs}{\sigma t} * 100\%$$

EX: For helical spring the ratio($di/do = 0.8$) with outer diam(84mm); No of coil(18turn)

$\tau = 6880$ **n/mm^2 ; G =8700N/mm^2**

Find : 1- loading force 2- deflection 3- pitch

$\frac{di}{do} = 0.8$ **; di=do*** 0.8 **=84*** $0.8 = 67.2$

$\tau = \frac{8 * f * dm * K}{\pi d^3}$ **; f=** $\frac{\pi d^3 * \tau}{8 * dm * K}$ **=** $\frac{\pi * 8.5^3 * 6880}{8 * 75.5 * 9.5}$ **f=25N**

d= $\frac{do - di}{2} = \frac{84 - 67.2}{2} = 8.5$

dm= di + d =67.2+8.5 = 75.5

C= $\frac{dm}{d} = \frac{75.5}{8.5} = 9.5$

K = $\frac{4c - 1}{4c - 4} + \frac{0.615}{c} = \frac{4 * 9.5 - 1}{4 * 9.5 - 4} + \frac{0.615}{9.5} = 3.2$

$$\delta = \frac{8 * f * dm^3 * n}{G * d^4} = \frac{8 * 25 * 75.7^3 * 18}{8700 * (8.5)^4}$$

P = $\frac{Lf}{(n - 1)}$

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EX: Find shear stress and diam after stress with outer diam and innear diam(46mm: 32mm)at power(48H.p) and rotation speed(840r. p. h) the Contant K=0.2 when pipe subjected to tosio stress?

P =

T = $\frac{p * 3600}{2\pi N} = \frac{48 * 750 * 3600}{2\pi * 840} = 24567N$

J= $\frac{\pi}{32}$ (do^4- di^4) =$\frac{\pi}{32}$(46^4 -32^4)

$$\frac{T}{J}=\frac{T}{R} \quad ; \quad \tau=\frac{T}{J}*R=\frac{24567}{J}*\left(46/_2\right)$$

d= 1072 $\frac{T}{\tau(1-k\,4)}$

الاسبوع الخامس عشر والسادس عشر

Clutch *النابض(الفاصل)*

Define: It is adevic mechanical between the engine and gear. Box to transmitte the power and treatment the vibration the shaft and another

Type of clutch *أنواع القوابض*

1- **Friction clutch** *قابض احتكاكي*

2- **Direct clutch** *قابض مباشر*

3- **Fluid clutch** *قابض هيدروليكي*

4- **Cone clutch** *قابض مخروطي*

5- **Cenlerfugal clutch** *قابض مركزي*

*System of disc clutch

A – uniform pressure

T = n $*m*m*\frac{2}{3}*\left(\frac{r1\ \ 3-r1\ \ 3}{r1\ \ 2-r2\ \ 2}\right)$

n= (No of driving+ No. Of disc(for drive) -1 N.m ; N. cm ; N. mm T = torque ; n= No of disc friction *عدد الأوجه المحتكة*

m= coefficient of Friction ; w= weight

r1, r2= Radii of friction disc

D. uniform wear التآكل المنتظم

T =n

EX: A single plate friction clutch For uniform wear the power (30 H.P) at rotation speed 2100r. p. m and axial load(145N) the external diam(250cm)Find the internal diam with m= 0.35?

P =

T = $\frac{30 * 750 * 60}{2\pi * 2100} = 102.3\ N.m$

T =n $* m * w * \frac{r1 + r2}{2}$

R1 = $\frac{25/2}{100} = 0.125m$

102.3 = 2 $* 0.35 * 145 * \frac{0.125 + r2}{2}$

R2= 1.89m ; D2= 3.78

EX: single plate driction clutch mean Radiun(7.5cm) the loading force(2.5K N); m=0.3 and N= 3000r. p. m Find the Torque and power at uniform wear

P =
$2\pi N\ T$

P = $\frac{2\pi * 3000 * 112.5}{60}$

= $\frac{6.28 * 3000 * 112.5}{60} = 35325\ \frac{N.m}{s}$ **=47.1 H.P**

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Ex: For uniform pressure; No of disc in driving and driver(3; 2) external and internal diam (240mm; 120mm) and power(25KW)

N= 1575r. p. m ; m= 0.25 Find the total Load?

N= (No of dis in driving+ No of disk in driven)-1

= (3+2)-1 = 5-1= 4

P= $2\pi N T$

T= $\frac{25 * 1000 * 60}{2\pi * 1575}$ = 151.9

T= n$* m * w * \frac{2}{3}\left(\frac{r1^3 - r2^3}{r1^2 - r2^2} \right)$

151.6= 4$* 0.25 * w * \frac{2}{3}\left(\frac{(0.12)^3 - (0.06)^3}{(0.12)^2 - (0.06)^2}\right)$

151.6 $* 3/2 = w((0.001728 - 0.000216)/(0.0144 - 0.0036) = w(0.001512/0.01081) \quad w = 1625N$

Core clutch قابض مخروطي

1- Uniform pressure

T= n

Cosec θ $= \frac{1}{\sin\theta}$

2- Uniform wear

T =n

EX: For uniform wear at cone clutch the angle (30°) the external and internal(250mm; 150mm)the friction disc in driving and drive (3; 2)the axial force (10K n) m=0.25 and angule speed= 100 rad/ s

N= (3+2)-1= 4 T = n

=4$* 0.25 * (10 * 10^3) * \frac{0.25 + 0.15}{2} * \frac{1}{\sin 30} = 2750\ N.m$

P=

w= $\frac{2\pi N}{60}$

P= 100$* 2750 = 275000 = 36606$

EX: Find No of friction dis wear the diam for driving and driven(180mm. 160mm) at power(8 H.P) at rotation speed(1400r. p. h) with m =0.3?

- **A- Uniform pressure**
- **B- Uniform wear**
- **C- Loading force 1200N**

T= $\frac{8 * 750 * 3600}{2\pi * 1400} = \frac{21600}{87.9} = 2454.5N$

T= n$* m * w * \frac{r1\ 2 - r2\ 3}{r1\ 2 - r2\ 2}$

2454= n$* 0.3 * 200 * ((0.18)3 - (0.16)3)/((0.18)2 - (0.16)2)$

n= $\frac{2454}{0.3 * 200} * \frac{0.18\ 2 - 0.16\ 2}{0.18\ 3 - 0.16\ 3} = \frac{0.0324 - 0.0256}{0.005832 - 0.004096} = \frac{2454}{0.3 * 1200} * \frac{0.0068}{0.00173}$ **= 2.6 ≈ 3**

T= n$* m * w * \frac{r1 + r2}{2}$

2454=n$* 0.3 * 1200 * \frac{\frac{90}{1000} * \frac{80}{1000}}{2}$ n= $\frac{2454}{0.3 * 1200} * \frac{2}{0.17} = 80$

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EX: For uniform wear at cone clutch the external and internal diam (120mm; 80mm) the friction disc in driving and driven (4; 2) the axial(2500N) at coefficient of friction (0.25)with power(40 H.P) and angular speed (100) radls? Find the angle?

P= cosec= $\frac{300}{125}$

40$* 750 = 100 * T$ Cosec= 2.4

T=4$* 75 = 200 N.m$ $\frac{1}{sine}$ =2.4

T=n =0.416

Ex: A spur gear of Module(4) and No of teeth(32). Determine all diamension?

DP= M *T = 4 $* 32 = 128mm$

B=1.2H

=1.2*4= 4.8

DO= DP+ 2M = 128+2 $* 4 = 136mm$

DR= DP _ 2.4M =128- 2.4 $* 4 = 118.4mm$

H= 2.2M = 2.4 $* 4 = 8.8mm$ **Pe=** $\pi\ DP$

R= $\frac{1}{8} * dp = \frac{1}{8} * 128 = 16mm$

H= 0.065 P **P=** $\frac{H}{0.65} = \frac{8.8}{0.65} = 13.5$

EX: Bevel gear the ratio between root diam and out diam(0.8) with the outer diam(140) Find all diamension?

$\frac{di}{do} = 0.8\ ;\ \ di = do * 0.8;\ \ di = 140 * 0.8 = 112mm$

do=dp+2M ………1

di= dp-2.4M……..2

dp= dp – 2M ……..3 **di= do – 2M – 2.4M**

4.4M= do – di= 140 – 112 **M=** $\frac{18}{4.4} = 4$ **dp= 140 - 2** $* 4 = 132$

T= $\frac{dp}{M} = \frac{132}{4} = 33$ **R=**

R= $\frac{1}{8}\ dp = \frac{1}{8} * 132 = 16.5$

***ration of speed in gears** نسبة السرع بالتروس

C= r1+r2

N1 * d1= N2 d2

$\frac{N1}{N2} = \frac{d2}{d1}$; $\frac{N1}{N2} = \frac{T2}{T1}$

C =distance between gears

r1, r2= raolun of gears

N1, N2 rotation speed

A- Advantage of gear

1- Transmitted exact velocity ratio

2- High efficiency

3- Tranmitted a large power

B- Dis advantage of gear

1- The manufaction of gear required aspical toof

2- Any error in tooth caven vibration and noise

3- Any effect in gear is demage in teeth

Lewise equation معادلة لويس

Wt= $\sigma b * b * \pi * m * y$

Wt = tangential load حمل مماسي

$\sigma b =$ Bending stress جهاد انحناء

b= teath width عرض السن

M= modul

y= Lewise factor عامل لويس

T = Wt* $Cs * rs$

T= torque عزم Cs= عامل الحزم

Wt= حمل مماسي rs= أصغر نصف قطر

Power = T * W

T = عزم ; W= سرعة زاوية Angular velocity rad/s

EX: By using Lewise equation Find the No. of teeth at gear connected with diam(0.6m) when 1- Lewise factor(0.1) 2- width teeth 10M

3- Torque 3 Kn. M; c= 45cm; cs= 1.3

$\sigma b = 2500 \frac{N}{mm}{}^{2}$

r2= $\frac{d2}{2} = \frac{0.6 * 100}{2} = 30cm$

r1+r2= C = 45-30=r1 ∴ r1= 15cm

T= Wt* $r2 * Cs$

Wt= $\frac{T}{rs * Cs} = \frac{3 * 1000}{\frac{15}{100} * 1.3}$ **=15384.6 N**

Wt= $\sigma b * b * m * \pi * y$

15384.6 =2500* $10M * M * \pi * 0.1$

$M^2 = \frac{15384.6}{2000 * \pi * 0.1} = \frac{15384.6}{2500} = 1.9$

M= 1.39= 1.5

D1= Dp1= M* $T1$

30 =1.5* $T1$ **= T1= 20**

D2= Dp2= M* $T2$

$\frac{15}{1.5} = T2 = 10$

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Gearing التروس

Type of gearing أنواع التروس

1- **External gearing**
2- **Internal gearing**
3- **Rack and pinion**

Types of gear أنواع المسننات

1- **Spur gear ترس عدل**

2- Bevel gear ترس مخروطي

3- Rack and pinian جريدة

4- Worm gear ترس دودة

5- Helical gear ترس حلزوني

*Diagran of gear

Law of gear

1- DP= M* T

DP= pitch diam قطر الخطوة M = Modul

T= No of teeth

Do=Dp+ 2A

2- Do= Dp+ 2M

Do= out diam

3- Dr= Dp- 2.4M

A= M ; B= 1.2A

4- H= 2.2* M ارتفاع السن

5- Circular pitch (Pt)

Pc= $\pi\, Dp$

Arc of teeth : R= $\frac{1}{8}\, DP$

EX: By lewise equation compute the M of teeth if shall diam(80cm) y=0.3.2; Cs= 1.8 ; T= 1120 N.m ;

$\sigma b = \frac{360N}{c\square^{2}}$; b= 8 M

R1= $\frac{d1}{2} = \frac{80}{2} = 40cm$

r1+r2= 140

r2= 140- 40= 100cm

T= Wt* $Cs * rs$

1120= Wt* $18 * \frac{40}{100}$

Wt= 1555.5 N

Wt= $\sigma b * b * M * y * \pi$

1555= 360* $5 * 8$ M²* $0.32 * 3.14$

$M^2 = \frac{1555}{9034}$; $M = 0.5$

Dp= 2 r1 = M T1

T1= $\frac{2 * 40}{0.5}$ =160

T2= $\frac{2 * 100}{0.5} =$ 400

Lewise equation معادلة لويس

Wt= $\sigma b * b * m * y * \pi$

Wt= الحمل المحوري ; b= ; y

$\sigma b =$ اجهاد الانحناء ; M=

T= Wt* $cs * rs$; T= عزم ; C1= عامل الخدمة

Power=

Wt= power= Wt

EX: By lewise equation compute the No. of teeth if rotation speed for driven and driving(260, 90 r. p. m) when Torqu 210V N. m ; X= 44cm; y= 0.1 ; Cs= 2.4 ;

$\sigma b = \frac{1500n}{m\square}$ ² ; b=8M?

$\frac{N1}{N2} = \frac{d2}{d1}$; $\frac{90}{26} = \frac{d2}{d1}$; $\frac{r2}{r1} = 0.34$

r2= r1* 0.34

r1+12= 44

r1+r1* $0.34 = 44$

r1= $\frac{44}{1.34} = 32.8cm$

r2= 0.34* $32.8 = 11.16\ cm$

T=wt* $Cs * rs$

Wt= $\frac{2100}{2.4 * \frac{11.2}{100}} = 7812.5\ n$

Wt= $\sigma b * b * m * y * \pi$

7812.5= 1500* $8M * m * 0.1 * 3.14 = 3768$ **M²**

M²= $\frac{7812.5}{3768}$

M= 1.439 ≈ 1.5

Dp= MT

Dp1= 2 r1 = MT1

= 2* $32.8 * 10 = 1.5\pi$

$$\pi = \frac{2 * 328}{1.5} = \frac{656}{1.5}$$

Weeks No. 19-20

Design of Journal Bearings

Introduction:

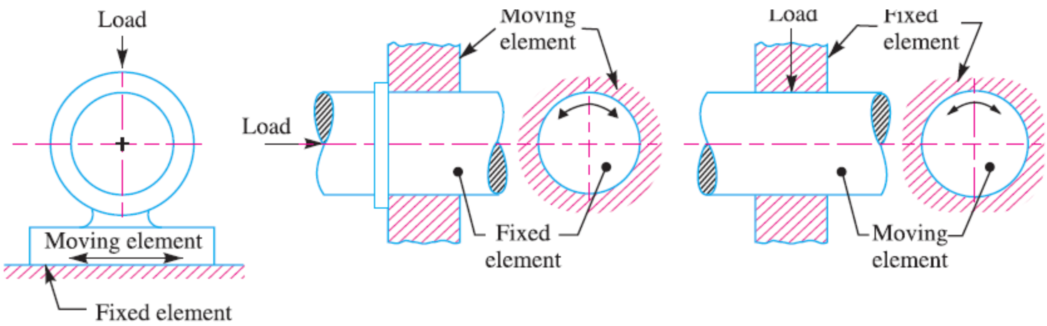
A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. A little consideration will show that due to the relative motion between the contact surfaces, a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are in direct contact, there will be rapid wear. In order to reduce frictional resistance and wear and in some cases to carry away the heat generated, a layer of fluid (known as lubricant) may be provided. The lubricant used to separate the journal and bearing is usually a mineral oil refined from petroleum, but vegetable oils, silicon oils, greases etc., may be used.

26.2. Classification of Bearings

Though the bearings may be classified in many ways, yet the following are important from the subject point of view:

1. **Depending upon the direction of load to be supported.** The bearings under this group are classified as: **(a)** Radial bearings, and ***(b)*** Thrust bearings. In radial bearings, the load acts

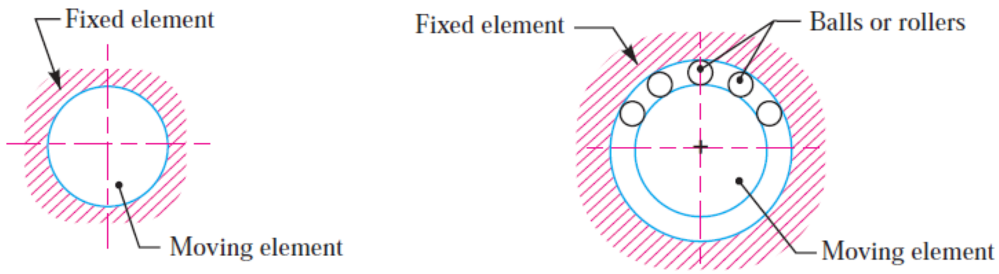
perpendicular to the direction of motion of the moving element as shown in Fig. 26.1 (a) and (b). In thrust bearings, the load acts along the axis of rotation as shown in Fig. 26.1 (c).
Note: These bearings may move in either of the directions as shown in Fig. 26.1.



(*a*) Radial bearing. (*b*) Radial bearing. (*c*) Thrust bearing.

Fig. 26.1. Radial and thrust bearings.

2. ***Depending upon the nature of contact.*** The bearings under this group are classified as : ***(a)*** Sliding contact bearings, and ***(b)*** Rolling contact bearings. In sliding contact bearings, as shown in Fig. 26.2 (a), the sliding takes place along the surfaces of contact between the moving element and the fixed element. The sliding contact bearings are also known as plain bearings.



(*a*) Sliding contact bearing. (*b*) Rolling contact bearings.

Fig. 26.2. Sliding and rolling contact bearings.

In ***rolling contact bearings,*** as shown in Fig. 26.2 (*b*), the steel balls or rollers, are interposed between the moving and fixed elements. The balls offer rolling friction at two points for each ball or roller.

Terms used in Hydrodynamic Journal Bearing

A hydrodynamic journal bearing is shown in Fig. 26.7, in which O is the centre of the journal and O′ is the centre of the bearing. Let D = Diameter of the bearing,

d = Diameter of the journal, and l = Length of the bearing.

The following terms used in hydrodynamic journal bearing are important from the subject point of view :

1. ***Diametral clearance.*** It the difference between the diameters of the bearing and the journal. Mathematically,

diametral clearance, $c = D - d$

Note : The diametral clearance (c) in a bearing should be small enough to produce the necessary velocity gradient, so that the pressure built up will support the load. Also the small clearance has the advantage of decreasing side leakage. However, the allowance must be made for manufacturing tolerances in the journal and bushing. A commonly used clearance in industrial machines is 0.025 mm per cm of journal diameter.

2. ***Radial clearance.*** It is the difference between the radii of the bearing and the journal. Mathematically, radial clearance:

$$c_1 = R - r = \frac{D - d}{2} = \frac{c}{2}$$

3. ***Diametral clearance ratio.*** It is the ratio of the diametral clearance to the diameter of the journal. Mathematically, diametral clearance ratio:

$$= \frac{c}{d} = \frac{D - d}{d}$$

4. ***Eccentricity.*** It is the radial distance between the centre (*O*) of the bearing and the displaced centre (*O′*) of the bearing under load. It is denoted by *e*.
5. ***Minimum oil film thickness.*** It is the minimum distance between the bearing and the journal, under complete lubrication condition. It is denoted by h_0 and occurs at the line of centres as shown in Fig. 26.7. Its value may be assumed as c / 4.
6. ***Attitude or eccentricity ratio.*** It is the ratio of the eccentricity to the radial clearance. Mathematically, attitude or eccentricity ratio:

$$\varepsilon = \frac{e}{c_1} = \frac{c_1 - h_0}{c_1} = 1 - \frac{h_0}{c_1} = 1 - \frac{2h_0}{c} \quad \ldots (\because\ c_1 = c/2)$$

7. ***Short and long bearing***. If the ratio of the length to the diameter of the journal (i.e. l / d) is less than 1, then the bearing is said to be short bearing. On the other hand, if l / d is greater than 1, then the bearing is known as long bearing. Notes :
   1. When the length of the journal (l) is equal to the diameter of the journal (d), then the bearing is called square bearing.
   2. Because of the side leakage of the lubricant from the bearing, the pressure in the film is atmospheric at the ends of the bearing. The average pressure will be higher for a long bearing than for a short or square bearing. Therefore, from the stand point of side leakage, a bearing with a large l / d ratio is preferable. However, space requirements, manufacturing, tolerances and shaft deflections are better met with a short bearing. The value of l / d may be taken as 1 to 2 for general industrial machinery. In crank shaft bearings, the (l / d) ratio is frequently less than 1.

محاضرات الاسابيع 21-24

CAM MECHANISMS:

1. **Types, characteristics, and motions:**
2. **Basic cam motions.**
3. **Layout and design;manufacturing considerations**
4. **Force and torque analysis:**

CAM MECHANISM TYPES, CHARACTERISTICS, AND MOTIONS:

Cam-and-follower mechanisms, as linkages, can be divided into two basic groups:
1. Planar cam mechanisms
2. Spatial cam mechanisms

In a planar cam mechanism, all the points of the moving links describe paths in parallel planes. In a spatial mechanism, that requirement is not fulfilled. The design of mechanisms in the two groups has much in common. Thus the fundamentals of planar cam mechanism design can be easily applied to spatial cam mechanisms, which is not the case in linkages. Examples of planar and spatial mechanisms are depicted in Fig. 40.1. Planar cam systems may be classified in four ways:

(1) according to the motion of the follower—reciprocating or oscillating;

(2) in terms of the kind of follower surface in contact—for example, knife-edged, flat-faced, curved-shoe, or roller;

(3) in terms of the follower motion—such as dwell-rise-dwell-return (D-R-D-R), dwell rise- return (D-R-R), rise-return-rise (R-R-R), or rise-dwell-rise (R-D-R); and (4) in terms of the constraining of the follower—spring loading (Fig. 40.1«) or positive drive (Fig. 40.16). Plate cams acting with four different reciprocating

followers are depicted in Fig. 40.2 and with oscillating followers in Fig. 40.3. Further classification of reciprocating followers distinguishes whether the centerline of the follower stem is radial, as in Fig. 40.2, or offset, as in Fig. 40.4. Flexibility of the actual cam systems requires, in addition to the operating speed, some data concerning the dynamic properties of components in order to find discrepancies between rigid and deformable systems. Such data can be obtained from dynamic models. Almost every actual cam system can, with certain simplifications, be modeled by a one-degree-of-freedom system, shown in Fig. 40.5, where m_e denote, respectively, the input (coming from the shape of the cam profile) and the

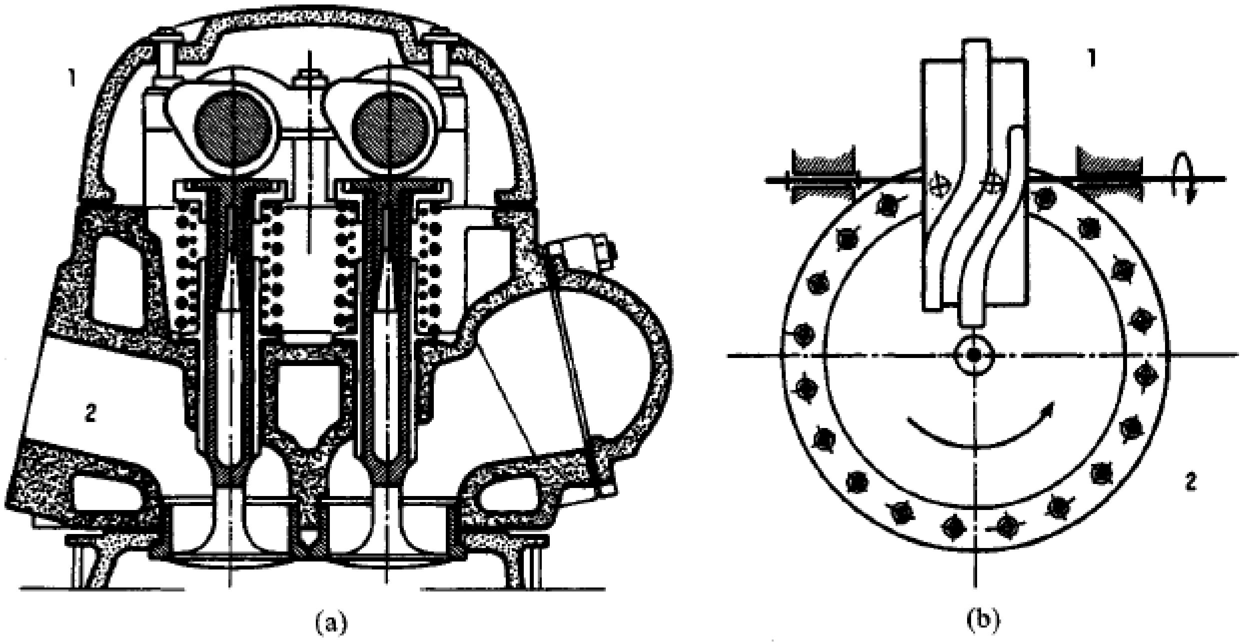


FIGURE 40.1 (*a*) Planar cam mechanism of the internal-combustion-engine D-R-D-R type; (*b*) spatial cam mechanism of the 16-mm film projector R-D-R type.

output of the system. The equivalent mass m_e of the system can be calculated from the following equation, based on the assumption that the kinetic energy of that mass equals the kinetic energy of all the links of the mechanism:

$$m_e = \sum_{i=1}^{i=n} \frac{m_i v_i^2}{\dot{s}^2} + \sum_{i=1}^{i=n} \frac{I_i \omega_i^2}{\dot{s}^2}$$

where ra, = mass of link *i*
VI = linear velocity of center of mass of z'th link
Ii = moment of inertia about center of mass for ith link
co, = angular velocity of *ith* link
5 = input velocity