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QUICK REVISION

FORMULA SHEET

for

GATE -ME THEORY OF MACHINES

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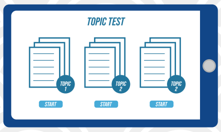
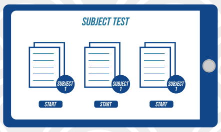
OUR COURSES

Online Test Series

Course Features



Topic Wise Tests



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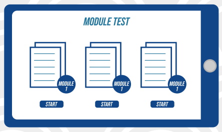

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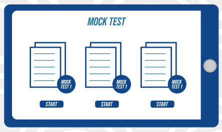

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Theory of Machines

Chapter 1: Simple Mechanism

Link: Smallest unit of machine, which should transfer relative motion.

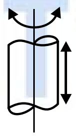
Spring: used as restoring forces, we cannot consider it as kinematic link.

Classification of Kinematic Pairs

A. **On the Basic of DOF**

Example:

1)(cylindrical pair) C – pair [DOF = 2]



(Spatial joint)

2)(Prismatic pair) P – pair [DOF = 1] (Planar joint)

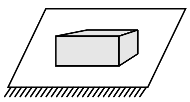
3)(Revolute pair) R – pair [DOF = 1] (Planar joint)

4)H – pair [DOF = 1] (Screw pair) (Spatial joint)

5)G – pair [DOF = 3] (Spherical pair) (Spatial joint)

6)E – pair [DOF = 3] (Planar or Evena pair)

Eg: Cube on surface



B. **According to type of contact:**

- Lower pair (surface contact)
  - Linear motion pairs

    DOF = 1 **Eg:** R, P, H pair
  - Surface motion pairs

    DOF > 1 **Eg:** C, E pair
- Higher pair (Point/Line contact, DOF =2)
- Wrapping pair when one link wrapped over the other link

All the surface motion pairs will violate Kutzbach equation.

C. **On the basic of type of closure**

- Self-closed pair (Permanent contact)

  Eg: Turning, sliding pair
- Forced closed pair (open pair)

  Eg: H.P in cam & follower

  Door closure

  Automatic clutch

D. **On the basis of motion between links**

- Completely constraint motion (Self-desired motion)

  Eg: P pair
- Successfully constraint motion (Forcefully desired motion)

  Eg; Foot step bearing
- Incompletely constraint motion (Unconstraint motion)

  Eg: C- pair

Degree of Freedom (DOF):

$3\,(n-1)-2J-h-F_r$

Where n = no. of link

J = No. of binary joint

h = no. of higher pair (Kutzbach equation) valid for R-pair; P pair.

- DOF < 0(Super structure)
- DOF = 0 Structure/Frame
- DOF > 1
  - DOF = 1 constraint/Kinematic mechanism
  - DOF > 1 unconstraint mechanism

F_r = redundant dof

Grubler's equation:

$3n - 2j = 4$

Here DOF = 1; h = 0

Minimum number of link to form mechanism = 4.

Modified Kutzbach Equation:

$DOF = 3[n - n_r - 1] - 2[J - J_r] - h - F_r$

Where:

n = No. of links

n_r = No. of redundant links

J_r = No. of redundant joints

J = No. of Binary joint

F_r = redundant dof

If there are n no. of links, then possible inversion will be $\leq n$

Grashof's Law:

A. $S + l \leq p + q$

S= shortest link

l = longest link

If S (Shortest link) fixed = Double crank mechanism.

If S is adjacent to fixed = Crank rocker

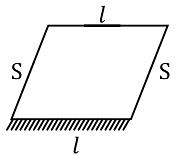
If S is opposite to fixed = Double Rocker or Lever mechanism.

B. $S + l > p + q$

Double rocker only

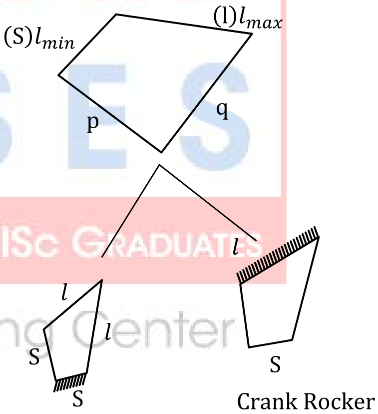
C. If $S + l = p + q$

- All links are equal length (Rhombus linkage)



Double crank only.

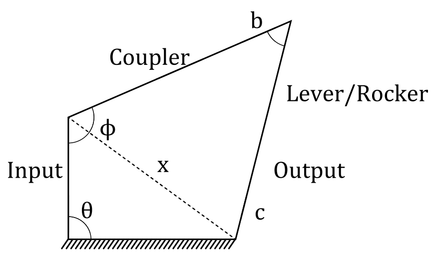
- Two links of equal length



Double Crank

Crank Rocker

Transmission Angle (Y): angle between coupler and lever/Rocker



$\left.\begin{array}{l} Y_{min} \text{ at } \theta = 0° \\ Y_{max} \text{ at } \theta = 180° \end{array}\right\} \left(\begin{array}{l} \text{but before this} \\ \text{check } x < b + c \end{array}\right)$

Toggle Positions $\phi = 180°$ and $\phi = 0°$

Mechanical Advantage (MA):

$$= \frac{F_{output}}{F_{input}} = \frac{(Torque)_{output}}{(Torque)_{input}}$$

$$= \frac{\omega_{input}}{\omega_{output}} \times \eta_{mech}$$

MA = ∞ at Toggle positions.

ω = Angular velocity

Simple Mechanism:

A. Four bar mechanism

B. Single slider crank mechanism

C. Double slider crank mechanism

A. Four Bar Mechanism:

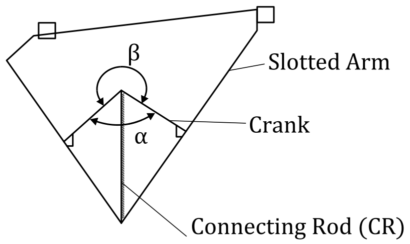
Inversions:

- Double crank mechanism
  Eg: Coupling rod of locomotives
- Crank rocker mechanism
  Eg: Beam engine
- Double rocker mechanism
  Eg: Watt's indicator

B. **Single Slider Crank Mechanism**

- Cylinder fixed
  Eg: Compressor, engine
- Crank fixed:
  Eg: Whitworth QRM, Rotary IC engine
- Connecting Rod (CR) Fixed:
  Eg: Crank slotted lever QRM
  Oscillatory cylinder engine
- Piston/Slider Fixed:
  Eg: Hand pump, Pendulum pump

C. **Crank Slotted Lever Mechanism (CR is fixed):**

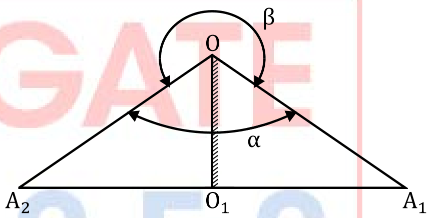


$$QRR = \frac{\beta}{\alpha} > 1 \text{ (Always} > 1)$$

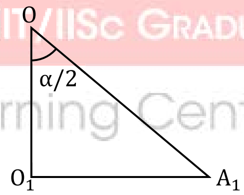
$$\text{Quick Return ratio(QRR)} = \frac{\text{Cutting time}}{\text{Ruturn Time}}$$

$$\text{Stroke length} = \frac{2 \times \begin{pmatrix}\text{Length of}\\ \text{slotted bar}\end{pmatrix} \begin{pmatrix}\text{Crank}\\ \text{length}\end{pmatrix}}{\text{Length of CR}}$$

Whitworth QRM: (Crank fixed)



$$QRR = \frac{\beta}{\alpha}$$



$$\cos\left(\frac{\alpha}{2}\right) = \frac{\text{crank length}}{\text{CR length}}$$

$$\text{Stroke length} = A_1A_2 = 2\,(O_1A_1)$$

D. **Double Slider Crank Mechanism:**

- Slotted bar fixed: Elliptical trammels
- One slider fixed: Scotch yoke mechanism.
- CR fixed: Oldham's coupling.

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Chapter 2: Motion Analysis

Number of Instantaneous Centre

$$= n_{C_2} = \frac{n(n-1)}{2}$$

Where n = No. of Links

Kennedy's Theorem:

For relative motion between No. of links in a mechanism any three links, their I-centre Must lie on a straight line.

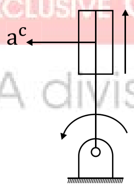
Theorem of Angular Velocity:

- I_{mn}
  - Can be treated on link m
  - Can be treated on link n

$$V_{I_{nn}} = \omega_n(I_{mn}I_{1m}) = \omega_n(I_{mn}I_{1n})$$

If I_{1n} and I_{1m} both are on same side of I_{mn} then sense of rotation is same for ω_m and ω_n

Coriolis Acceleration (a^c):



$$a^c = 2(\vec{\omega} \times \vec{V})$$

Where, V = Velocity of slider

ω = angular velocity of link on which slider is sliding.

Direction: Rotate the $\vec{V}$ in the same sense as of $\vec{\omega}$.

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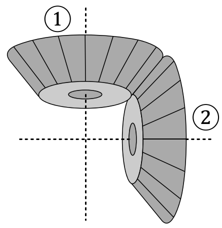
Chapter 3: GEARS

Positive Drive: Slip is not possible. Eg: (gears)

Negative Drive: Slip is possible, Eg: Belt, rope, Chain.

Mitre Gear:



Similar Gear at 90°

$$\frac{\omega_1}{\omega_2} = 1$$

Circular Pitch:

$$(P_c) = \frac{\pi D}{T}$$

Where D = Diameter of gear /pinion

T = No. of teeth

Module (m):

$$= \frac{D}{T} \text{ (Module is same for mating gears)}$$

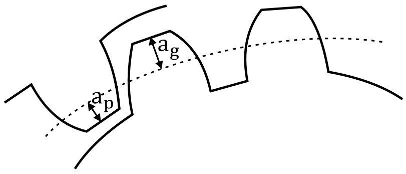
Diametral Pitch $(P_d): \frac{T}{D}$

$$P_d \cdot P_c = \frac{T}{D} \times \frac{\pi D}{T} = \pi$$

Working depth = Addendum + Dedendum- clearance.

= sum of addendum of both mating gears.

Tooth space – Tooth thickness of mating gear = Backlash



Where a_g = Addendum of gear

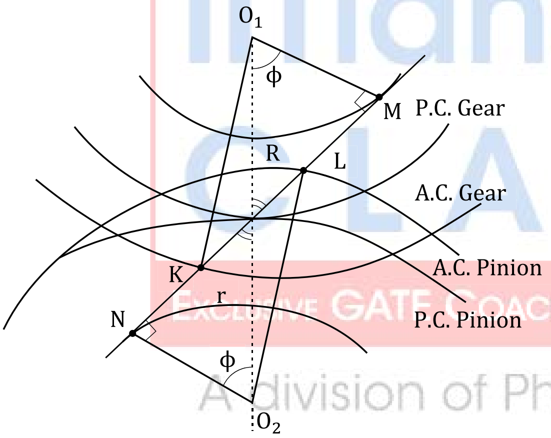
a_p = Addendum of Pinion

Law of Gearing:

Line of action must always pass through the fixed point (pitch point) on the line joining the centres of rotation of gear.

$V_{sliding} = V_s = (\omega_1 + \omega_2)QP$

= Sum of angular Velocity × distance between pitch point and point of contact.



A.C = Addendum circle

Path of contact = Path of approach (KP) + path of recess (PL)

$$KP = \sqrt{R_A^2 - R^2\cos^2\phi} - R\sin\phi$$

$$PL = \sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi$$

→ Arc of contact $= \dfrac{\text{Path of contact}}{\cos\phi} = \dfrac{KL}{\cos\phi}$

– Angle turned by pinion

$= \dfrac{\text{Arc of contact}}{r}$ (in radian)

→ Angle turned by Gear

$= \dfrac{\text{Arc of contact}}{R}$ (in radian)

→ Contact ratio $= \dfrac{\text{Arc of contact}}{\text{circular pitch }(P_c)} \geq 1$

→ Gear ratio $= G = \dfrac{T}{t}$

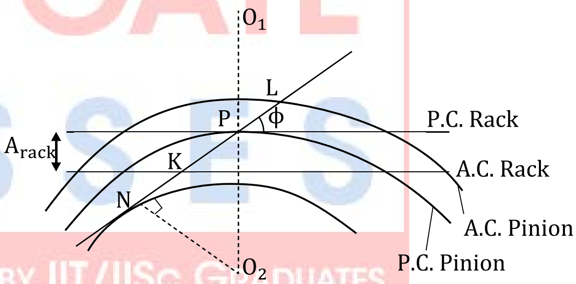
Where T = No. of teeth on gear

t = No. of teeth on pinion.

→ Velocity ratio $= \dfrac{\omega_p}{\omega_g} = \dfrac{T}{t} > 1$

$\Rightarrow \dfrac{\omega_g}{\omega_p} = \dfrac{t}{T} < 1$

→Path of contact in rack and pinion



A_{rack} = Addendum of rack

A.C pinion ⇒ Addendum circle of pinion

P.C. pinion ⇒ pitch circle of pinion

$K.L = kP + PL$

$$= \frac{A_{rack}}{\sin\phi} + \sqrt{r_A^2 - r^2\cos^2\phi} - r\sin\phi$$

Methods to Prevent Interference:

- Under cut gears
- Increase ϕ(Pressure angle)
- By stubbing the teeth
- Increasing No. of teeth

Minimum No. of teeth required for pinion?

Teeth on Gear:

$$\Rightarrow T_{min} = \frac{2A_G}{\sqrt{1 + \frac{1}{G}\left(\frac{1}{G} + 2\right)\sin^2\phi} - 1}$$

Teeth on Pinion:

$$t_{min} = \frac{2A_P}{\sqrt{1 + G(G+2)\sin^2\phi} - 1}$$

A_G = Fractional addendum of gear

A_P = Fractional Addendum of pinion

Always make gear safe first if $A_G = A_P$

For Rack and Pinion

$$t_{min} = \frac{2A_R}{\sin^2\phi}$$

In all, $\left.\begin{matrix} A_G \\ A_P \\ A_R \end{matrix}\right\}$ These are fractional addendum

$m \cdot A_P$ = A of pinion

$m \cdot A_G$ = A of Gear

$m \cdot A_R$ = A of Rack

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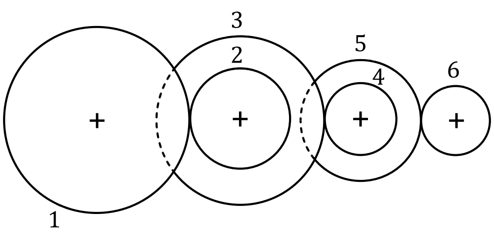
Chapter 4: GEAR TRAINS

$$\text{Speed ratio} = (SR) = \frac{\omega_1}{\omega_2} = \frac{\omega_{driver}}{\omega_{driven}}$$

$$\text{Train value} = \frac{1}{SR}$$

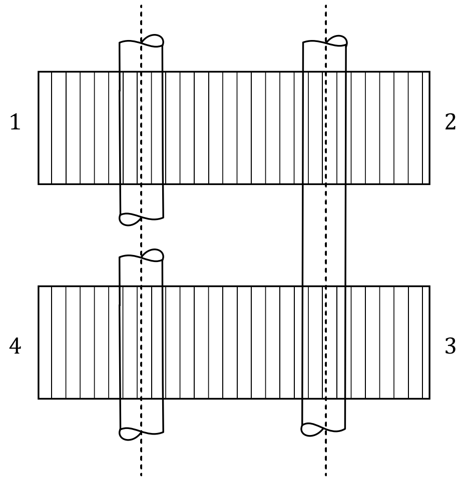
Compound Gear Train:



$$SR = \frac{\omega_1}{\omega_6} = \frac{T_2T_4T_6}{T_1T_3T_5}$$

T = No. of teeth on gears.

Reverted Gear Train:



m = module of 1 , 2

m′ = module of 3, 4

$$SR = \frac{\omega_1}{\omega_4} = \frac{T_2T_4}{T_1T_3}$$

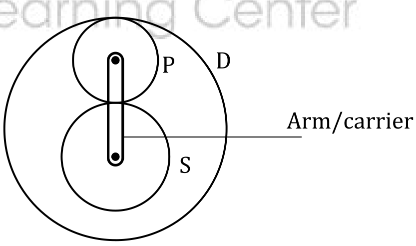
$$r_1 + r_3 = r_3 + r_4$$

$$m\left(\frac{T_1}{2} + \frac{T_2}{2}\right) = m'\left(\frac{T_3}{2} + \frac{T_4}{2}\right)$$

If speed reduction is same

$$\frac{T_2}{T_1} = \frac{T_4}{T_3}$$

Planetary Gear Train (Epicyclic):



$$r_s + 2r_p = r_D$$

$$T_s + 2T_p = T_D$$

$$\Rightarrow T_{input} + T_{output} + T_{fixing} = 0$$

$$\rightarrow \eta_{GT}[T_{input} \cdot \omega_{input}] + T_{output} \times \omega_{output} = 0$$

η_{GT} = Efficiency of gear train

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Chapter 5: GYROSCOPE

Gyrosope Couple $(C_g) = L \times \frac{d\theta}{dt}$

$c_g = I.\omega.\omega_p$

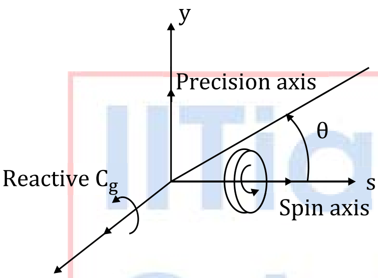
$L =$ Angular momentum of rotor $= I.\omega$

$\omega =$ Angular Velocity of rotor

$I =$ Mass moment of Inertia of rotor

$\omega_p =$Angular Velocity of axis on which rotor is rotating.

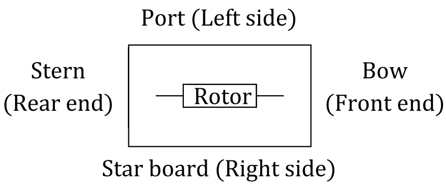
Calculation of Reactive Gyroscopic Couple:



Reactive c_g direction

$=$ Spin axis $(\omega) \times$ Preission axis (ω_p)

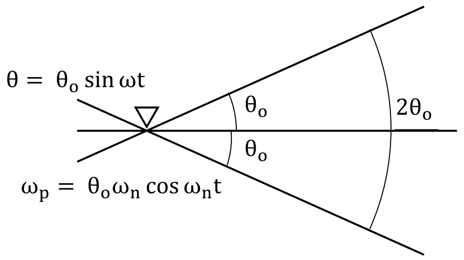
Gyroscopic Effects in Water Boat:



Motion

1. **Pitching:**

   Up and down movement of bow and stern



$2\,\theta_o =$ Total included angle

$\omega_n = \frac{2\pi}{T}$ (Natural frequency)

$(\omega_p)_{max} = \theta_o \times \omega_n = \theta_o \times \frac{2\pi}{T}$

T = Time period of pitching

$C_g = I\omega(\omega_p)_{max}$

2. **Steering:**

   Left or right turn of ship

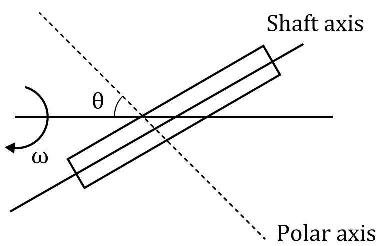
   $C_g = I\omega\omega_p$

   $\omega_p = \frac{V}{R} = \frac{\text{Velocity of boat}}{\text{Radius of curvature /circle}}$

3. **Rolling:**

   No couple formed as precision axis and spin axis coincide.

Gyroscopic couple by bearing due to misalignment of disc:



$G = \frac{m}{8}\omega^2 r^2 \sin(2\theta)$

$\theta =$ misalignment in degree

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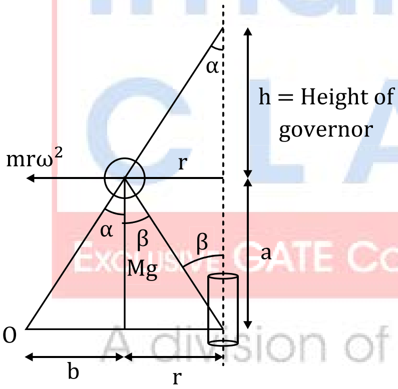
Chapter 6: GOVERNORS

- Governor
  - Inertia Governor
  - Centrifugal Governor
    - Pendulum Type

      Eg: Watt Governor
    - Gravity Controlled
      - Porter
      - Proell
    - Spring Controlled
      - Hartnell
      - Hartung
      - Pickering
      - Wilson Hartnell

1. Watt Governor:



$$\sum M_o = 0$$

$$mg\,b = mr\omega^2(a)$$

$$mr\omega^2 = mg\left(\frac{b}{a}\right)$$

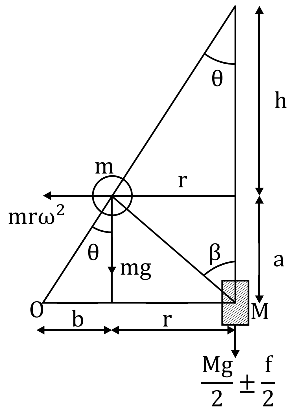
$$r\omega^2 = g\tan\alpha$$

$$r\omega^2 = g \times \frac{r}{\omega}$$

$$\omega^2 = \frac{g}{h}$$

$$N^2 = \frac{895}{h}$$

2. Porter Governor:



$$\sum M_o = 0$$

$$\omega^2 = \frac{2mg + (Mg \pm f)(1+k)}{2mh}$$

$$k = \frac{\tan\beta}{\tan\theta}$$

f = friction force on sleeve.

Case 1:

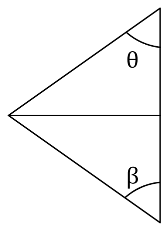
If f (friction) = 0

$$\omega^2 = \frac{2mg + [Mg][1+k]}{2mh}$$

Case 2:

If f = 0

$$\omega^2 = \left(\frac{m+M}{m}\right)\frac{g}{h}$$

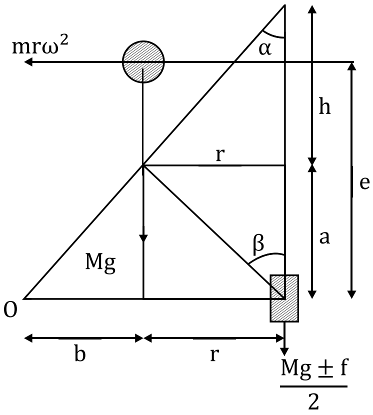


$\theta = \beta$

arms are of equal length

M = Mass of Sleeve

3. **Proell Governor:**



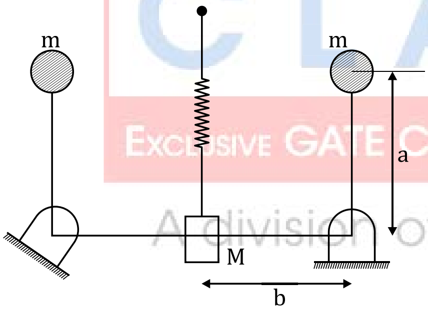
$\sum M_o = 0$

$$\omega^2 = \frac{a}{e}\left[\frac{2\ mg + (Mg \pm f)(1+k)}{2\ mh}\right]$$

If $f = 0$, and $k = 1$

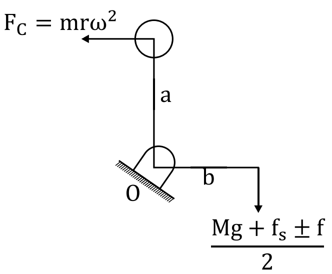
$$\omega^2 = \frac{a}{e}\left[\frac{m+M}{m}\right] \times \frac{g}{h}$$

4. **Hartnell Governor:**



$$mr_1\omega_1^2(a) = \frac{Mg + fs_1 \pm f}{2}\ (b)$$

$$mr_1\omega_1^2(a) = \frac{Mg + fs_2 \pm f}{2}\ (b)$$

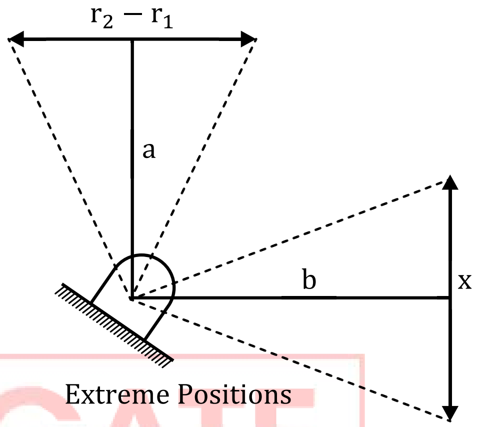


f_s = spring force, f = frictional force

If $f = 0$

$$\frac{r_1\omega_1^2}{r_2\omega_2^2} = \frac{Mg + f_{s_1}}{Mg + f_{s_2}}$$

Total sleeve movement $(x) = \frac{b}{a}(r_2 - r_1)$



Extreme Positions

$$(f_{c_2} - f_{c_1})\frac{2a}{b} = f_{s_2} - f_{s_1}$$

Additional compression of spring

$$(x_2 - x_1) = \frac{b(r_2 - r_1)}{a}$$

Stiffness of spring $(k) = \frac{2(f_{c_2} - f_{c_1})}{(r_2 - r_1)}\left(\frac{a}{b}\right)^2$

$$\text{Sensitiveness} = \frac{\text{Range of speed}}{\text{Mean speed}}$$

$$= \frac{N_2 - N_1}{N} = \frac{2(N_2 - N_1)}{(N_2 + N_1)}$$

$$\text{Sensitivity} = \frac{1}{\text{Sensitiveness}} = \frac{N_1 + N_2}{2(N_2 - N_1)}$$

Sensitivity $= \infty$ for isochronous governor

Hunting: An excessively fast to & fro motion of sleeve between stopper.

Isochronism: A governor is said to be isochronous if its equilibrium speed is constant for all radii of rotations.

Range of speed $= N_{max} - N_{min} = 0$

This means ω is same for all sleeve position which is not possible in case of Watt and porter.

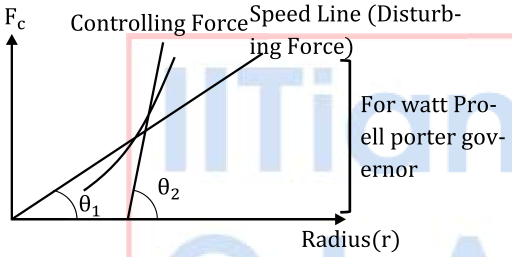
For Hartnell governor: it can be made isochronous governor

$$\omega_1 = \omega_2 = \omega$$

$$\frac{r_2}{r_1} = \frac{Mg + F_{s_2}}{Mg + F_{s_1}}$$

Stability of Governor:

Stable governor: Slope of restoring force > Slope of disturbing force.

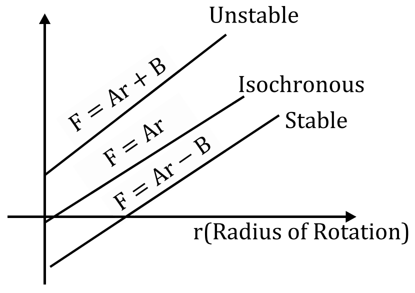


$\theta_2 > \theta_1$ for stable governor

$$\frac{dF(r)}{dr} > m\omega^2$$

If r ↑⇒ N ↑ (Stable governor)

For Spring controlled governor



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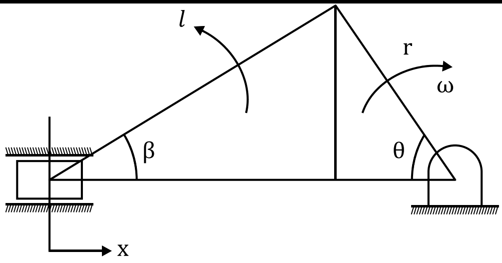
Chapter 7: DYNAMICS & FLYWHEEL



n = obliquity ratio

r = radius of crank

l = length of connecting rod

$$n = \frac{l}{r}$$

$$l \sin\beta = r \sin\theta$$

$$x = r\left[1 - \cos\theta + n - \sqrt{n^2 - \sin^2\theta}\,\right]$$

$$\left.\begin{aligned} V &= r\omega\left[\sin\theta + \frac{\sin 2\theta}{2n}\right] \\ a &= r\omega^2\left[\cos\theta + \frac{\cos 2\theta}{n}\right] \end{aligned}\right\} \text{Approx equation}$$

↓

$$\left.\begin{aligned} \omega_{cR} &= \frac{\omega\cos\theta}{n} = \frac{d\beta}{dt} \\ \alpha_{cR} &= \frac{-\omega^2 \sin\theta}{n} \end{aligned}\right\} \text{Valid when n is large}$$

V= Velocity of piston

a = acceleration of piston

ω_{cR} = Angular Velocity of connecting rod

α_{cR} = Angular acceleration of connecting rod

Inertia force on piston $= m\,a = F_i$

$$= m\left[r\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n}\right)\right]$$

$$F_{primary} = mr\omega^2 \cos\theta$$

$$F_{secondary} = mr\omega^2\left(\frac{\cos 2\theta}{n}\right)$$

m = mass of piston

F = net force on piston

$$= F_p - F_i + F_g - F_f$$

Where; $F_p = P_1A_1 - P_2A_2$

$F_g = Mg$

$F_f =$ Kinetic friction

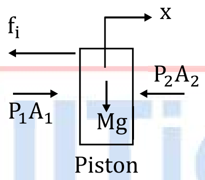
$P_1 =$ pressure of gas at cover end

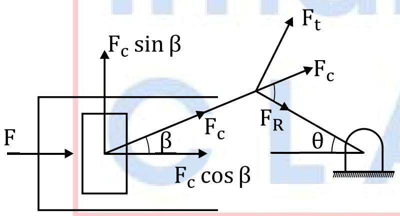
$A_1 = \left(\frac{\pi}{4}\right) D^2$, where D = diameter of piston

$P_2 =$ pressure of gas at crank end side of piston

$A_2 = \frac{\pi}{4}(D^2 - d^2)$

d = diameter of piston rod





1. $F_{CR} = \frac{F}{\cos\beta}$

2. $F_n = F_c \sin\beta =$ Normal thrust to cylinder walls

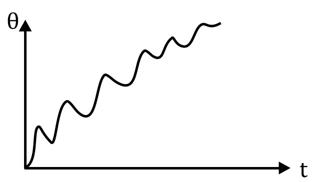
3. $F_r = F_c \cos(\theta + \beta)$

   $F_r =$ Radial thrust to crank shaft bearing

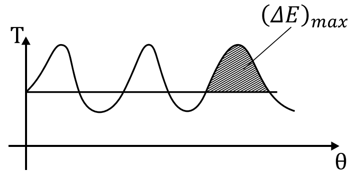
4. Crank effort $(F_t) = F_c \sin(\theta + \beta)$

$$= \frac{F}{\cos\beta}[\sin(\theta + \beta)]$$

5. Torque on crank $(T) = F_t \times$ radius(r)



Flywheel:



Change in energy of flywheel

$= (\Delta E)_{max} = \frac{1}{2} I (\omega_{max}^2 - \omega_{min}^2)$

$(\Delta E)_{max} = I\omega^2 C_s = E_{max} - E_{min}$

$$\omega = \frac{\omega_{max} + \omega_{min}}{2}$$

$$C_s = \frac{\omega_{max} - \omega_{min}}{\omega}$$

= Coefficient speed fluctutation

$C_E =$ Coefficient of energy fluctuation

$$= \frac{E_{max} - E_{min}}{(WD)_{cycle}}$$

$(WD)_{cycle} = T_{mean} \times \theta_{cycle}$

WD = work done

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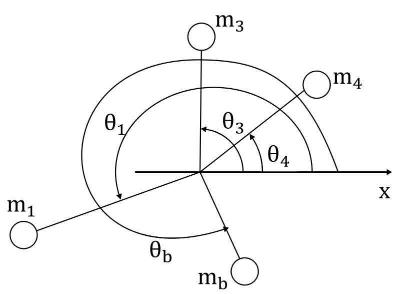
Chapter 8: BALANCING

- Balancing
  - Static Balancing (Balancing of forces) $\sum \vec{F} = 0$
  - Dynamic Balancing

    $\sum \vec{F} = 0$

    $\sum \vec{M}_R = 0$

Balancing of several masses rotating in Same plane:



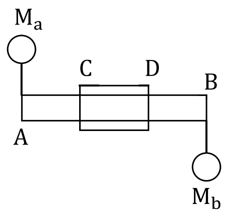
m_b = Balancing mass

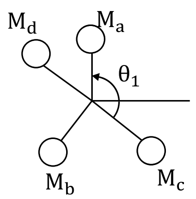
$\sum F_H = 0, \ \sum F_V = 0$

$\sum mr\cos\theta = 0$

$\sum mr\sin\theta = 0$

Balancing of several masses rotating in different plane:





$\sum F = 0; \ \ \sum M = 0$

Table can be made and get the answer

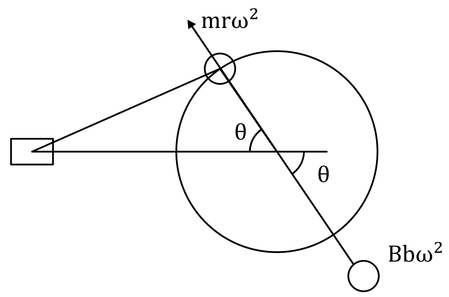
Balancing of Reciprocating Masses:

$$F_{Inertia} \text{ on Piston } = mr\omega^2\left[\cos\theta + \frac{\cos 2\theta}{n}\right]$$

Only primary force is balanced ⇒ which is maximum at $\theta = 0°$ and $180°$

2nd force is maximum 4 times in one rotation and negligible at moderate speeds.

To Balance primary force a balancing mass is attached opposite to crank.



B = Balance mass

b = Radial distance of balance mass

$mr\omega^2\cos\theta = Bb\omega^2\cos\theta$

$mr = Bb$

We will go for partial Balancing

$cmr = Bb$

$F_{unbalance}$ (along horizontal direction)

$= (1-c)mr\omega^2\cos\theta$

$F_{unbalance}$ ($\perp^r$ to line of stroke)

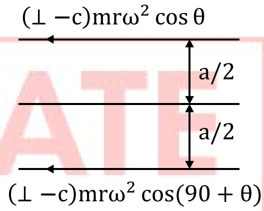
$= cmr\omega^2\sin\theta$

$$F_R = \sqrt{((F_{unbalance})_H)^2 + ((F_{unbalance})_{\perp^r})^2}$$

$$\text{For minimum } F_R = \frac{dF_R}{dc} = 0$$

Resultant unbalance is minimum when $c = 1/2$.

Balancing of Locomotive:



1. **Tractive Force (F_t):**

   $(1-c)mr\omega^2\cos\theta + (1-c)mr\omega^2\cos(90+\theta)$

   F_t is max at $\theta = 135°$ and $315°$

2. **Swaying Couple (S):**

   $$S = (1-c)mr\omega^2\cos\theta\left(\frac{a}{2}\right) - (1-c)mr\omega^2\cos(90+\theta) \times \frac{a}{2}$$

   $$S = (1-c)mr\omega^2\left(\frac{a}{2}\right)(\cos\theta + \sin\theta)$$

   S_{max} at $\theta = 45°$ and $225°$

3. **Hammer Blow:**

   It is unbalanced vertical force due to balancing mass (B) at radius (b).

   $P = Bb\omega^2\sin\theta = cmr\omega^2\sin\theta$

   P_{max} at $\theta = 90°$ and $270°$

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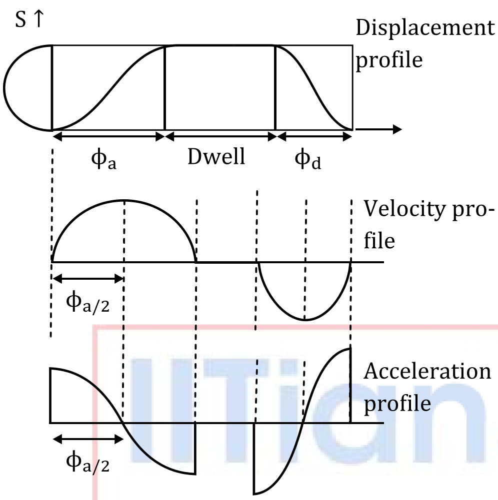
Chapter 9: CAMS & FOLLOWER

Types of motion of Follower

1. **SHM (Simple Harmonic Motion):**

ϕ_a = Angle of ascent

ϕ_d = Angle of descent

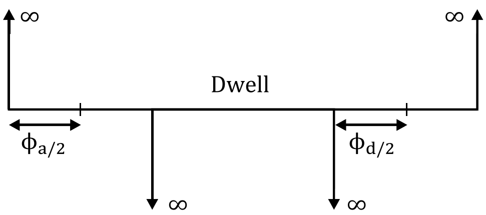


$$(\text{Displacement})y = \frac{h}{2}\left(1 - \cos\left(\frac{\pi\theta}{\phi_a}\right)\right)$$

$$V = \frac{\pi h\omega}{2\phi_a}\sin\left(\frac{\pi\theta}{\phi_a}\right)$$

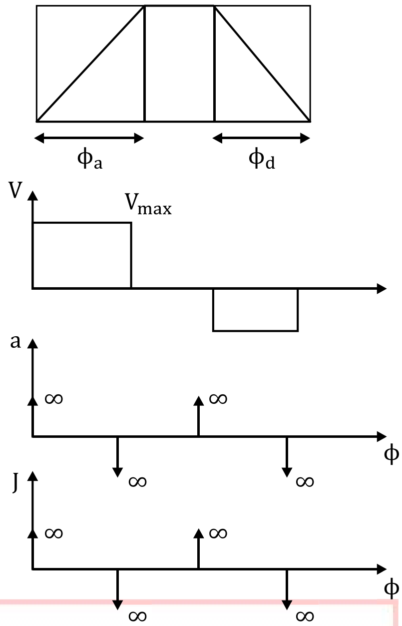
$$a = \frac{\pi^2 h\omega^2}{2\phi_a^2}\left[\cos\left(\frac{\pi\theta}{\phi_a}\right)\right]$$

Jerk profile



$$J = \frac{da}{dt}$$

2. **Uniform Velocity (Worst motion)**

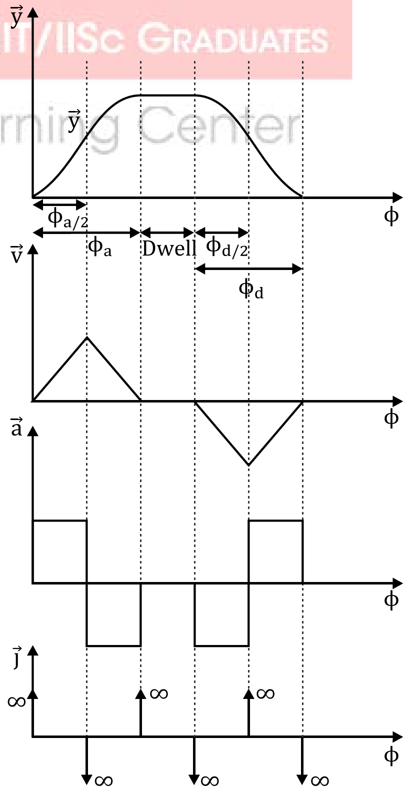


$$y = \frac{h\theta}{\phi_a}$$

$$V = \frac{h\omega}{\phi_a}$$

$a = 0, J = 0$

3. **Uniform Acceleration:**

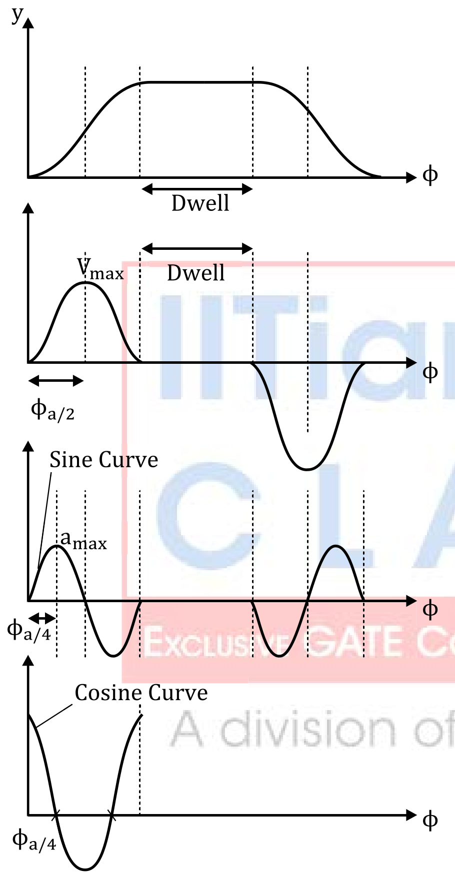


$$y = \frac{2h\theta^2}{\phi_a^2}$$

$$V = \frac{4h\theta\omega}{\phi_a^2}$$

$$a = \frac{4h\omega}{\phi_a^2}$$

4. **Cycloidal Motion (Best Motion):**



$$y = \frac{h}{\pi}\left(\frac{\pi\theta}{\phi_a} - \frac{1}{2}\sin\left(\frac{2\pi\theta}{\phi_a}\right)\right)$$

$$V = \frac{h\omega}{\phi_a}\left(1 - \cos\left(\frac{2\pi\phi}{\phi_a}\right)\right)$$

NOTE:

$V_{(max)_{cycloidal}} > V_{SHM} > V_{v=constant}$

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