# 1. STRENGHT OF MATERIAL

## Stress and Strain

Stress (σ)

$$\text{Nominal / Engg. stress} = \frac{\text{Load}}{\text{Original Area}}$$

$$\text{Actual /True stress} = \frac{\text{Load}}{\text{Actual Area}}$$

$$\text{Ultimate Stress} = \frac{\text{Maximum load}}{\text{Original cross sectional area}}$$

$$\text{Working stress} = \frac{\text{Actual axial load}}{\text{Original cross sectional area}}$$

**Strain (ε)**

$$\text{Strain}(\epsilon) = \frac{\text{change in dimension}}{\text{original dimension}}$$

$$\text{Linear strain}(\varepsilon_{long}) = \frac{\text{change in longitudinal dmension}}{\text{original longitudinal dimension}} = \frac{\delta L}{L}$$

$$\frac{\text{Change in lateral dimension}}{\text{Original lateral dimension}} = \frac{\delta d}{d} \text{ or } \frac{\delta t}{t}$$

## Deformation of Bodies:

Axial Elongation (Δ) Of Prismatic Bar Due To External Load

$$\Delta = \frac{PL}{AE}$$

Elongation Of Bar Due To Its Own Weight

Reactangular Bar : $\delta L = \frac{WL}{2AE} = \frac{\gamma L^2}{2E}$

Conical Bar : $\delta L = \frac{\gamma L^2}{6E}$

Deformation of a body of stepped c/s due to an axial load: $\delta L = \frac{P}{E}\left(\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3}\right)$

## Uniformly Varying Section :

Circular Tapering Bar : $\delta L = \frac{4PL}{\pi E d_1 d_2}$

Rectangular Tapering Bar:

$$\delta L = \frac{PL}{Et(a-b)}\log_e\frac{a}{b}$$

$$\text{Poisson's ratio}(\mu) = \frac{-\text{Lateral strain}}{\text{Linear strain}}$$

$$\text{Young Modulus (E)} = \frac{\text{Stress}}{\text{Strain}} = \frac{\sigma}{\epsilon}$$

$$\text{Modulus of rigidity}(G) = \frac{\text{Shear stress}}{\text{Shear strain}} = \frac{q}{\phi}$$

$$\text{Bulk modulus} = \frac{\text{Direct stress}}{\text{Volumetric strain}} = \frac{\sigma}{\frac{\delta V}{V}}$$

Volumetric Strain of Rectangular bar

$$e_v = \frac{\delta V}{V} = e_x + e_y + e_z$$

Concept Of Uni-Axial Loading:

$$\frac{\delta V}{V} = e_x + e_y + e_z$$

$$e_v = \frac{\delta V}{V} = e(1-2\mu)$$

Concept of Bi-Axial Loading:

$$e_x = \frac{\sigma_x}{E} - \mu.\frac{\sigma_y}{E}, e_y = \frac{\sigma_y}{E} - \mu.\frac{\sigma_x}{E}$$

Concept of Tri-Axial Loading:

$$e_x = \frac{\sigma_x}{E} - \mu\frac{\sigma_y}{E} - \mu\frac{\sigma_z}{E}$$

$$e_y = \frac{\sigma_y}{E} - \mu\frac{\sigma_x}{E} - \mu\frac{\sigma_z}{E}$$

$$e_z = \frac{\sigma_z}{E} - \mu\frac{\sigma_x}{E} - \mu\frac{\sigma_y}{E}$$

$$\frac{\delta V}{V} = e_x + e_y + e_z$$

$$e_v = \frac{\delta V}{V} = \left(\frac{\sigma_x + \sigma_y + \sigma_z}{E}\right)(1-2\mu)$$

Volumetric Strain of Cylindrical bar

$e_v$ = longitudinal strain + 2 ( diametric strain)

Volumetric Strain of sphere

$e_v$ = 3x diametric strain

Relation Between Elastic Constants E, G & K:

$$E = 2G(1+\mu) = 3K(1-2\mu)$$

$$E = \frac{9KG}{3K+G}, \quad \mu = \frac{3K-2G}{6K+2G}$$

## Temperature / Thermal stresses:

Thermal strain, $e_T = \alpha.\Delta T$

Thermal deformation, $\delta(T_L) = (\alpha\Delta T)L$

Thermal stress, $\sigma_T = E\alpha\Delta T$

## strain energy (U)

$$U = \frac{1}{2} \times P \times \delta L = \frac{1}{2} \times T \times \theta$$

$$\text{Resilience} = \frac{1}{2} \times P \times \delta L = \frac{\sigma^2}{2E} \times V$$

$$\text{Proof Resilience } (U_P) := \frac{\sigma_{max}^2}{2E} \times V$$

**Modulus of resilience:** $\frac{\text{Proof Resilience}}{\text{Volume}} = \frac{\sigma^2}{2E}$

For strain energy stored in a body when the load is applied with impact load

$$\sigma = \frac{P}{A}\left(1 + \sqrt{1 + \frac{2AEh}{PL}}\right)$$

For strain energy stored in a body due to shear stress

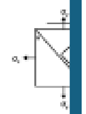
$$\text{Strain energy stored} = \frac{\tau^2}{2C} \times V$$

## Principal Stress And Principal Strain

**Normal and Shear stresses on oblique/ inclined plane**

Normal stresses ($\sigma_\theta$) on a plane :

$$\sigma_\theta = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \left(\frac{\sigma_x - \sigma_y}{2}\right)\cos 2\theta + \tau_{xy}\sin 2\theta$$



Shear / Ta...

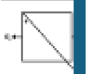
$\tau_\theta = \left(\frac{\sigma_x ...}{}\right)$ ...

**Resultant** ...

Angle of ...

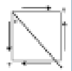
$\varnothing = \tan^{-1}$ ...

**Dire...**



Normal st...

Tangential...



Normal st...

Shear stres...

Maximum...

$\sigma_1 = \left(\frac{\sigma_x ...}{}\right)$ ...

Minimum...

$\sigma_2 = \left(\frac{\sigma_x ...}{}\right)$ ...

**Position o...**

Magnitude...

$\tau_{max} = \pm\sqrt{\left(\frac{...}{2}\right)^2 ...}$

Normal stress on maximum shear stress plane

$$\sigma_{avg} = \left(\frac{\sigma_1 + \sigma_2}{2}\right)$$

Resultant stress on the plane of maximum shear stress plane is

$$\sigma_R = \sqrt{(\sigma_{...})^2 + (\tau_{max})^2}$$

## PRESSURE VESSELS

Hoop or circumferential stress, $\varepsilon_{avg} = \frac{(\varepsilon_x + \varepsilon_y)}{2}$

Longitudinal stress, $\sigma_L = \frac{pd}{4t}$

Maximum Shear Stress, $\tau_{max} = \frac{pd}{8t}$

## Hollow circular section:

Polar moment of inertia, $J = \frac{\pi}{32}D^4$

Polar section modulus, $Z_p = \frac{\pi}{16}\left(\frac{D^4 - d^4}{D}\right)$

$\frac{16T}{...}\left(\frac{D}{...}\right)$

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| Operational | A/F ratio |
|---|---|
| Idling | 12 – 12.5 |
| Cruising / normal | 16 – 16.5 |
| Maximum power range | 12 – 13 |
| Transient operation/starting | 3 – 5 |

**Engine performance parameters –**

Indicated power (IP) – Inside cylinder by buying fuel

$IP = \frac{P_m LANK}{60}$ [Per 2 – stroke engine]

$IP = \frac{P_m LANK}{120}$ [Per 4 – stroke engine]

Brake power (BP) – Available at engine crank shaft for doing useful work $BP = \frac{2\pi NT}{60}$

Here, N = no. of rotation (rpm)

Frictional Power (FP) –

F.P = Indicated power (IP) – Brake power (BP)

Mechanical Efficiency = $\eta_{mech} = \frac{BP}{IP}$

Brake Thermal Efficiency ($\eta_{bth}$) – $\eta_{bth} = \frac{BP}{mf \times C.V}$

Indicated specific fuel consumption (ISFC)

$IFSC = \frac{mf}{IP}\left(\frac{kg}{kw} hr\right)$

Brake specific fuel consumption (BSFC)

$BSFC = \frac{mf}{LP}(kg/kw - hr)$ $\eta_{th} = \frac{IP}{m_f \times C.V}$

$\eta_{th} = \frac{IP}{m_f \times C.V}$

$m_f$ = mass flow rate (kg / sec)

C.V. = calorific value (j / kg)

### Knocking in SI engine

(I) knocking or detonation is due to auto ignition of end charge before reaching the flame front in that part of combustion chamber

(II) In S.I engine knocking takes place at the end of combustion process

(III) Pressure rise is very high during knocking due to homogeneous mixture

### Knocking in C.I engine

(I) Due to combustion of accumulated fuel during large delay period creates very high pressure

(II) In CI engine it takes place at beginning of combustion

(III) Knocking is due to auto ignition of more fuel accumulated due to long delay period. Pressure rise is not so high Factors tending to reduce detonation & knocking in S.I & CI engine

| Factors | S.I engine | CI engine |
|---|---|---|
| compression | Low | High |
| Inlet temperature | Low | High |
| Inlet pressure | Low | High |
| Self Ignition Temp | High | low |

| Time lag or delay period | Long | Short |
|---|---|---|
| Load on engine | Low | High |
| Combustion wall temp | Low | High |
| Speed | High | low |

Firing order of IC engine → firing order is maintained for proper balancing of engine & adjustment of unbalanced forces and also for proper heat dissipation

1) 4 cylinder → 1-2-4-3 or 1-3-4-2

2) 5 cylinder → 1-2-4-5-3

3) 6 cylinder → (1) 1-5-3-6-2-4
(2) 1-3-5-6-2-4
(3) 1-3-2-6-4-5

| | |
|---|---|
| Constant Volume (Isochoric) | $W_{(1-2)} = 0$ |
| Constant Temperature (Isothermal) | $W_{1-2} = P_1V_1 \ln\left(\frac{P_1}{P_2}\right)$ <br> $= mRT \ln\left(\frac{P_1}{P_2}\right)$ <br> $= mRT \ln\left(\frac{V_2}{V_1}\right)$ |
| Adiabatic (Isentropic) | $W_{1-2} = \frac{(P_1V_1 - P_2V_2)}{(\gamma - 1)}$ <br> $= \frac{mR(T_1 - T_2)}{\gamma - 1}$ |

| | |
|---|---|
| Boiled evaporator | $Q_{boil} = h_2 - h_1$ |

### Unsteady State Transient –

From mass conservation –

$\left(\frac{dm}{dt}\right)_{cv} = m_i - m_e$

Here, $m_i$ = mass flow/sec at inlet, $m_e$ = mass flow/sec at exit From Energy Balance Equation

$\left(\frac{dv}{dt}\right)_{cv} = m_i h_i - m_e h_e + Q - W$

Here, V = Total internal energy
cv = Control volume  Q = Heat input in the cv
W = Work done by cv

### Heat Engine

Entropy change $ds \geq \frac{dQ}{T}$ $ds = \left[\frac{dQ}{T}\right]_{rev}$

$(ds)_{universe} \geq 0$

Here, ds = Entropy change  dQ = change in heat
T = Temperature.

### Inequality of clause's –

$\oint \frac{dQ}{T} = 0$, the cycle is reversible

$\oint \frac{dQ}{T} < 0$, the cycle is irreversible & possible.

$\oint \frac{dQ}{T} > 0$, the cycle is impossible

### Change in Entropy of system –

$= Q_1\left(1 - \frac{T_0}{T_H}\right)$

$W_{max} = Q_1 - Q_2 = Q_1 - (\Delta S)T_0$

Here, $(\Delta S)\ T_0$ = Unavailable energy

Note – The maximum work can be obtain when the lower temperature would be ambient temperature.

### Loss of Available Energy in cycle

Loss of available energy $= Q_1\left[\frac{T_0}{T_1'} - \frac{T_0}{T_1}\right]$



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| Self Ignition Temp | High | low |


| Process | K (index) |
|---|---|
| P = Const | K = 0 |
| V = Const | K = ∞ |
| T = Const | K = 1 |
| Adiabatic | K = γ |
| Polytropic | K = n |

## Reversible closed system process

W=P.dv

Closed system work in various process –

| Process | Work - done |
|---|---|
| Constant pressure (Isobaric) | $W_{1-2} = P(V_2 - V_1)$ <br> $= MR(T_2 - T_1)$ |

is less than zero, $\oint \frac{dQ}{T} < 0$

### Available Energy (cycle)

$T_0$ = Ambient temp. $W_{max} = Q_1\ \eta_{max}$

## PERT and CPM

EST= Earliest start time.
EFT=Earliest finish time.
LST=Latest start time.
LFT=Latest finish time.
Now,
(1) Total float=LST-EST = $L_j - E_i - T_{ij}$
(2) Free float=Total float-head slack event.
(3) Independent float =Free float – (Tail event slack)

The event with zero slack time are known as critical event. $t_e = \frac{t_0 + 4t_m + t_p}{6}$

where,

$t_e$ = expected time, $t_o$ = optimistic time
$t_m$ = most likely time, $t_p$ = pessimistic time

Standard deviation(σ) $\sigma = \frac{t_p - t_o}{6}$

## Quality control and Analysis

(1) X- chart :-

(a) central line= $\bar{\bar{x}} = \frac{\sum_{i=1}^{N} \bar{x}_i}{N}$

(b) Upper control limit : UCL= $\bar{\bar{x}} - 3\sigma_{\bar{x}}$

Where, $\sigma_{\bar{x}} = \frac{\sigma}{\sqrt{n}}$ n=sample size.

(2) R-chart

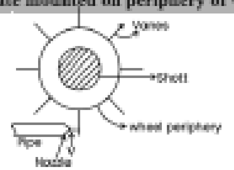
$\bar{R} = \frac{\sum_{i=1}^{N} R_i}{N}$ UCL=$d_4\bar{R}$ LCL = $d_3\bar{R}$

(3) u-chart

$\bar{u} = \frac{\sum_{i=1}^{N} u_i}{N}$ UCL=$\bar{u} + 3\sqrt{\frac{\bar{u}}{n}}$ LCL=$\bar{u} - 3\sqrt{\frac{\bar{u}}{n}}$

## 11. FLUID MACHINERY

### (1)Plate mounted on periphery of wheel

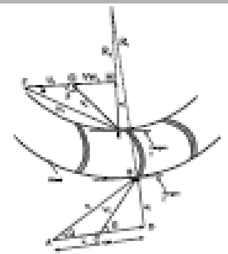



Mass of water per second striking the series of plates = ρAV
$F_x$=ρAV u (V-u)
Work done by jet (W) = ρAVu(V–u)
Efficiency of work done on wheel = η = $\frac{2u(V-u)}{V^2}$
When blade velocity (u) = $\frac{\text{Jet velocity}(V)}{2}$
$\eta_{max}$ = 50%

### (2)Force exerted on series of radial curved vanes u1≠u2)

| Parameters | Inlet | outlet |
|---|---|---|
| 1)velocity if Jet | $V_1$ | $V_2$ |
| 2)angle of Jet | α | β |
| 3)Relative velocity of Jet | $Vr_1$ | $Vr_2$ |
| 4)velocity of whirl | $Vw_1$ | $Vw_2$ |
| 5)velocity of flow | $Vf_1$ | $Vf_2$ |
| 6)velocity of blade | $u_1$ | $u_2$ |
| 7)Angle of blade | θ | ϕ |

### Force exerted on series of radial curved vanes



Work done per second on the wheel

WD/sec = $\rho AV[V_{(w1)} u_1 \pm V_{(w2)} u_2]$ kW

β < 90° (+ve sign)
β > 90° (-ve sign)

$\frac{WD}{weight} = \frac{[V_{w_1} u_1 \pm V_{w_2} u_2]}{g}$ in meters

### Turbine

(1) Water power= $\frac{energy}{second} = \rho gH = \frac{\rho gQH}{1000}$ kw

(2) Runner power=

$\frac{\text{work done}}{\text{second}} = \frac{\rho AV[V_{w_1} u_1 \pm V_{w_2} u_2]}{1000}$ k W

(3) Volumetric efficiency = $\frac{\text{volume of water actually striking the runner}}{\text{volume of water supply to turbine}}$

(4) Shaft power $= \frac{2\pi NT}{60 \times 1000}$ kw

(5) $\eta_{hydraulic} = \frac{\text{Runner power}}{\text{Water power}} = \frac{V_{w_1} u_1 \pm V_{w_2} u_2}{gHnet}$

(6) $\eta_{mechanical} = \frac{\text{shaft power}}{\text{runner power}}$

(7) $\eta_{overall} = \frac{\text{shaft power}}{\text{water power}} = \eta_{mechanical} \times \eta_{hydraulic}$

(8) Speed Ratio ($C_u$ or $K_u$)= $\frac{\text{Blade velocity}}{\text{Jet velocity}} = \frac{u}{\sqrt{2gH}}$

(9) Flow ratio ($K_f$) = $\frac{\text{Flow velocity}}{\text{Jet velocity}} = \frac{V_f}{\sqrt{2gH}}$

Degree of reaction=

$\frac{\text{change in pressure energy inside runner}}{\text{change in total energy inside runner}}$

For Impulse turbine DOR = 0

For Reaction Turbine DOR = 1 - $\frac{\cot\alpha}{2(\cot\alpha - \cot\theta)}$

For pure reaction turbine DOR = 1

### Pelton Turbine

(1) Based on Newton's second law of motion.
(2) Pressure through the turbine is atmospheric.
(1) Runner power
$\frac{(V_{w_1} + V_{w_2})}{g}$ in m
(2) Speed Ratio (
(3) Flow ratio (K
(4) Jet Ratio= Dia
(5) Number of bu
(6) Discharge (Q
(7) Maximum nu
(8) Minimum dia
(9) DOR = 0
(10) Hydraulic eff
$\eta_{hyd} = \frac{2(V_1 - u)[1}{V_1}$
(11)For max effic
$\frac{\text{Jet velocity}}{2}$
(12) $\eta_{vol} = \frac{Q}{Q - \Delta Q}$

### Francis Turbine

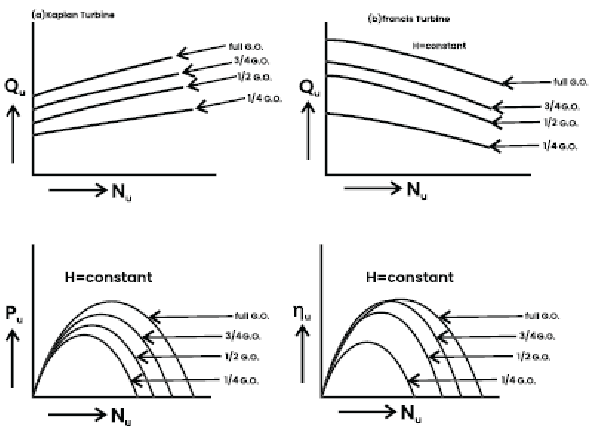
(1) Runner power
(2) Hydraulic effi
(3) Speed ratio =
(4) Flow ratio =
(5)Width Ratio
$\frac{B_1}{D_1} = \frac{\text{width of blade}}{\text{dia. of blade}}$
(6) Number of bla
(7) Discharge Q =
(8)Degree of reac

### K

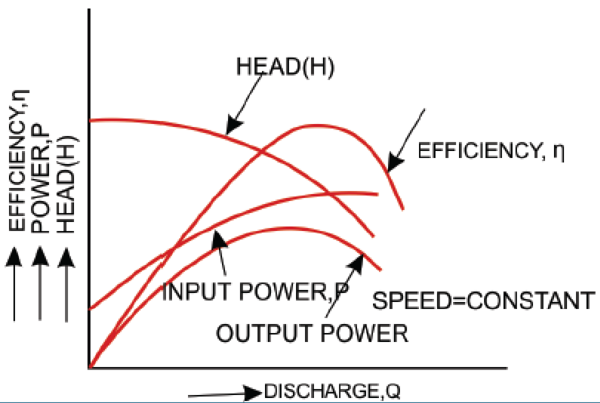
Axial flow reactio
Runner power depends upon inlet conditions
(1) Runner power =( m) $V_{(w1)} u_1$

(2) Discharge (Q) = $\pi/4(D_o^2 - D_b^2)V_f$
(3) Speed ratio = 1.4 – 2
(4) Flow ratio = 0.7 ($VF_1$ or $VF_2$)
(5) Number of blades = 4–8
(6) Width ratio = 0.1–0.3
(7) Degree of reaction = 0.5-1
(8) peripheral velocity at inlet &outlet
$u_1 = u_2 = \frac{\pi DN}{60}$

## FOR Reaction turbine




## Operating characteristics curve or constant speed




(V) Manometric efficiency =

$\frac{\text{monometric head}}{\text{Head imparted by impellor}} = \frac{gHm}{Vw_2 u_2}$

(VI) Mechanical efficiency=

$\eta_{mech} = \frac{\text{Impeller power}}{\text{Shaft power}}$

(VII) Overall efficiency $\eta_0 = \eta_{mech} \times \eta_{hydralic}$

(VIII) Specific speed $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$

(IX)Maximum suction height =

$H_{max} = H_{atm} - H_V - \frac{V_s^2}{2g} - H_{Fx}$

(X)Net positive suction Head

NPSH = H_atm-$H_V$–$H_s$–$H_{Fx}$

## Reciprocating pump-

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

d = diameter of cylinder
L = 2r
$H_s$ = Suction head
$H_d$ = Delivery head
N = Speed or revolution of crank per min

(I) Discharge

(IV) Power of pump = P = $\frac{\rho QgH_m}{\eta_m \times 1000}$