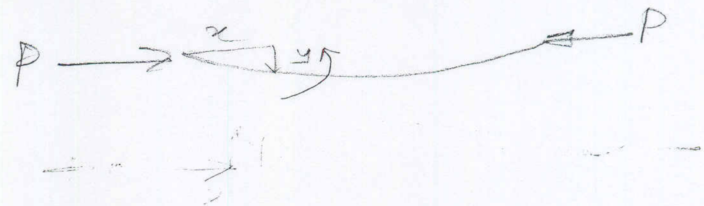
Solution (1)

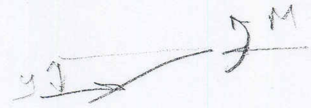
1 (a)



$\therefore \; EI \dfrac{d^2y}{dx^2} = -Py$

$EI \dfrac{d^2y}{dx^2} = -M$

$\therefore \; \dfrac{d^2y}{dx^2} + \dfrac{P}{EI} y = 0$



$M = Py$

ie $\dfrac{d^2y}{dx^2} + \alpha^2 y = 0$ — (1)

where $\alpha = \sqrt{\dfrac{P}{EI}}$

Solution to equation (1) is

$y = A \sin(\alpha x) + B \cos(\alpha x)$

Boundary conditions:– when $x = 0, \; y = 0$

and when $x = L, \; y = 0$

$\therefore \; 0 = B$ and $0 = A \sin \alpha L$

$\therefore \; \sin(\alpha L) = 0$ because $A = 0$ is a trivial solution

Hence $\alpha L = 0, \pi, 2\pi, \ldots n\pi \ldots$

Lowest buckling load is given by

$\alpha L = \pi$

ie $\sqrt{\dfrac{P_c}{EI}}\, l = \pi$ OR $P_c = \dfrac{\pi^2 EI}{l^2}$

(b) (i) $\hat{y} = e\left[\sec\left(\frac{\alpha l}{2}\right) - 1\right]$

$\alpha = \sqrt{\frac{P}{EI}}$

$l = 2\,m,$ $b = 36\,mm$ and $d = 36\,mm$

$I = \frac{bd^3}{12} = \frac{36^4}{12}\,mm^4$

$\hat{y} = 20\,mm$

$E = 200 \times 10^9 \frac{N}{m^2}$

$P = 20 \times 10^3\,N$

$e = ?$

$$\alpha = \sqrt{\frac{P}{EI}} = \sqrt{\frac{20000\,N \quad 12}{200 \times 10^9\,N/m^2 \times 36^4\,mm^4}}$$

$$= \sqrt{\frac{12}{10^7 \times 36^4}\,\frac{m^2}{mm^4}\left[\frac{10^6\,mm^2}{m^2}\right]}$$

$$= \sqrt{\frac{12}{36^4}}\,\frac{1}{mm}$$

ie $\alpha = 7.1445 \times 10^{-7}\,mm^{-1}$

$\therefore \hat{y} = e\left[\sec\left(\frac{\alpha l}{2}\right) - 1\right]$

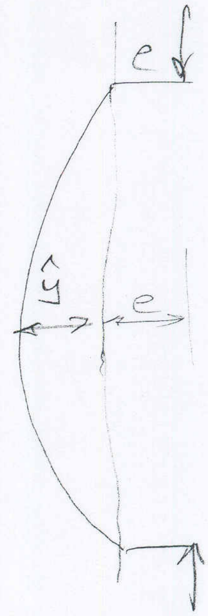
ie $20\,mm = e\left[\sec\left(\frac{8.4525 \times 10^{-4}}{2} \times 2000\right) - 1\right]$

25.102°

$20\,mm = e\left[\sec(0.438778) - 1\right] = 0.50687$

ie $e = 25 \div 0.5068 = \cancel{239}$ 49.32 mm

(ii) Maximum tensile and compressive stresses occur at the centre of the strut, (ie $x = \frac{L}{2}$).



$$\sigma = -\frac{P}{A} \pm \frac{P(y+e)}{I}\frac{d}{2}$$

$$\hat{\sigma}, \check{\sigma} = -\frac{P}{A} \pm \frac{P(\hat{y}+e)}{I} \times \frac{d}{2}$$

$$= -\frac{20 \times 10^3 \text{ N}}{36^2 \text{ mm}^2} \pm \frac{20 \times 10^3 \text{ N}(20 \text{ mm} + 49.3 \text{ mm}) \times 18 \text{ m}}{\left(\frac{36^4}{12}\right) \text{ mm}^4}$$

$$= \left(\frac{-20 \times 10^3}{36^2} \pm \frac{20 \times 10^3 \times 69.3 \times 18 \times 12}{36^4}\right) \frac{\text{N}}{\text{mm}^2}$$

$$= (-15.432 \pm 178.241) \frac{\text{N}}{\text{mm}^2}$$

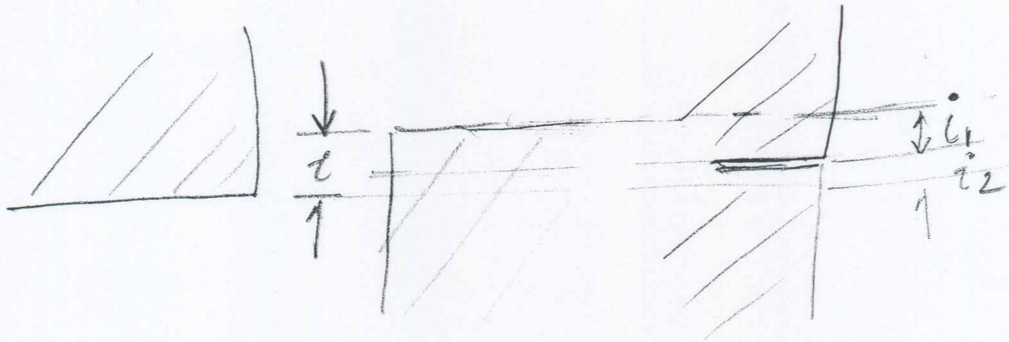
$\therefore$ Maximum tensile stress $= (178.241 - 15.432) \frac{\text{N}}{\text{mm}^2}$

$$= 162.8 \frac{\text{N}}{\text{mm}^2}$$

and maximum compressive stress $= (-178.241 - 15.432)$ "

$$= -193.7 \frac{\text{N}}{\text{mm}^2}$$

Solution (2)



Before shrink Fitting

After shrink Fitting

For cylinder 1

$$\sigma_r = A_1 - \frac{B_1}{r^2} \qquad \sigma_\theta = A_1 + \frac{B_1}{r^2}$$

At $r = 15\,mm$, $\sigma_r = 0$ $\therefore B_1 = 225 A_1$

At $r = 35\,mm$, $\sigma_r = -p$

$$\therefore -p = A_1 - \frac{225 A_1}{1225}$$

$$\therefore \underline{A_1 = -1.225p}$$

Thus $$(\sigma_r)_1 = -1.225p\left(1 - \frac{225}{r^2}\right)$$

and $$(\sigma_\theta)_1 = -1.225p\left(1 + \frac{225}{r^2}\right)$$

$$\varepsilon_\theta = \frac{u}{r} = \frac{1}{E}\left(\sigma_\theta - \nu(\sigma_r + \sigma_z)\right) = \frac{1}{E}\left(\sigma_\theta - \nu\sigma_r\right)$$

At the outside of cylinder 1, $r = 35\,mm$

$$\therefore \frac{-i_1}{} = \frac{1}{207{,}000}\left(\sigma_\theta - \nu\sigma_r\right)$$

ie $$\frac{-i_1}{35} = \frac{1}{207000} \times (-1.225p)\left(1 + \frac{225}{35^2} - \nu\left(1 - \frac{225}{35^2}\right)\right)$$

ie $$\frac{i_1}{35} = \frac{p}{207000}(1.45 - \nu)$$

For cylinder 2

$$\sigma_r = A_2 - \frac{B_2}{r^2} \qquad \sigma_\theta = A_2 + \frac{B_2}{r^2}$$

At $r = 55$ mm, $\sigma_r = 0$, $\therefore B_2 = 3025 A_2$

At $r = 35$ mm, $\sigma_r = -p$,

$$\therefore -p = A_2 - \frac{3025 \cdot A_2}{1225}$$

i.e. $$\underline{A_2 = \frac{1225}{1800} p}$$

and $$\underline{B_2 = 3025 \times \frac{1225}{1800} p}$$

Therefore, $$(\sigma_r)_2 = \frac{1225}{1800} p \left(1 - \frac{3025}{r^2}\right)$$

and $$(\sigma_\theta)_2 = \frac{1225}{1800} p \left(1 + \frac{3025}{r^2}\right)$$

$$\varepsilon_\theta = \frac{u}{r} = \frac{1}{E}(\sigma_\theta - \nu\sigma_r)$$

At the inside of cylinder 2

$$\frac{i_2}{35} = \frac{1}{207000}(\sigma_\theta - \nu\sigma_r)$$

$$= \frac{1}{207000}\left(\frac{1225}{1800} p\right)\left[1 + \frac{3025}{1225} - \nu\left(1 - \frac{3025}{1225}\right)\right]$$

$$= \frac{p}{207000}(3.633 - 1.469\nu)$$

But $i_1 + i_2 = i = 0.05$ mm

$$\therefore \frac{35p}{207000}(1.45-\nu) + \frac{35p}{207000}(3.633 - 1.469\nu) = 0.05 \text{ mm}$$

Now, $\nu = 0.3$

$$\therefore \left[(1.45-0.3) + (3.633 - 0.441)\right]p = \frac{207000 \times 0.05}{35}$$

$$4.342\, p = \frac{207000 \times 0.05}{35}$$

$$\therefore p = 68.11 \ \frac{N}{mm^2}$$

For cylinder 1 $\quad \sigma_r = \underbrace{-1.225 \times 68.11}_{-83.43}\left(1 - \frac{225}{r^2}\right)$

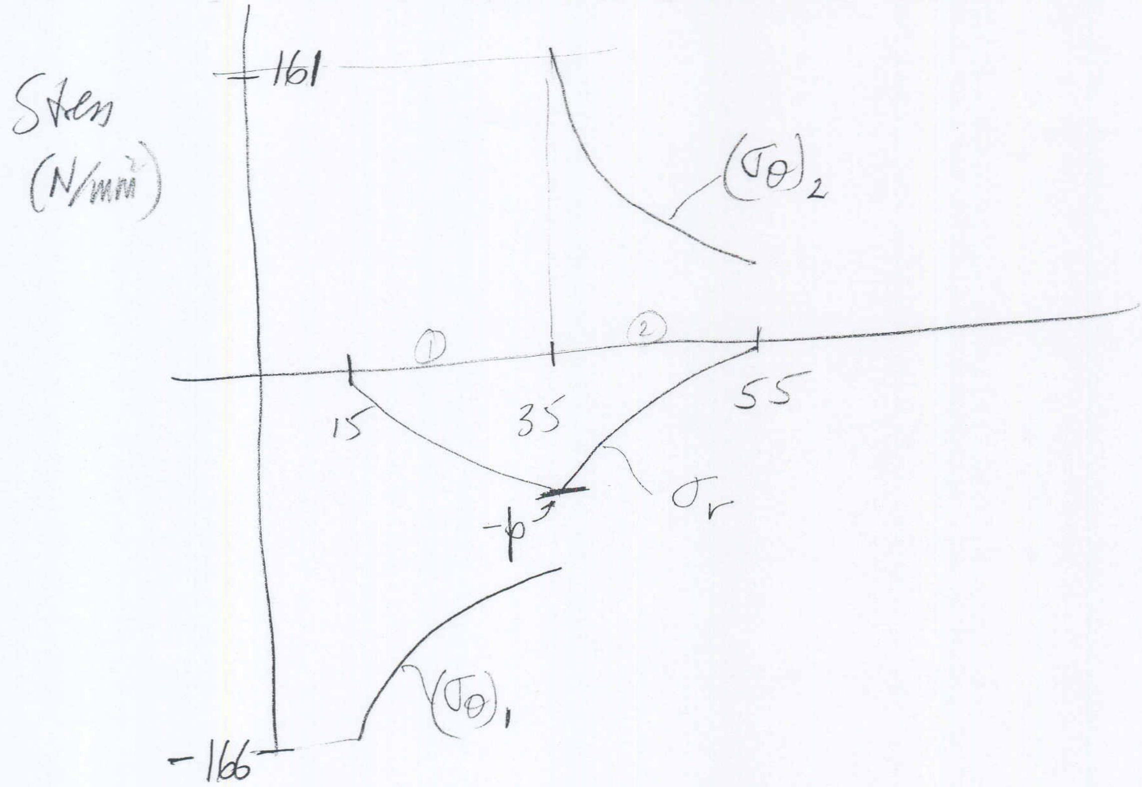
$$\sigma_\theta = -1.225 \times 68.11 \left(1 + \frac{225}{r^2}\right)$$

For cylinder 2 $\quad \sigma_r = \frac{1225}{1800} \times 68.11 \left(1 - \frac{3025}{r^2}\right)$

$$\sigma_\theta = \underbrace{\frac{1225}{1800} \times 68.11}_{46.35}\left(1 + \frac{3025}{r^2}\right)$$

At $r = 35$, $(\sigma_\theta)_2 = 160.8 \text{ N/mm}^2$

At $r = 15$, $(\sigma_\theta)_1 = -166 \text{ N/mm}^2$

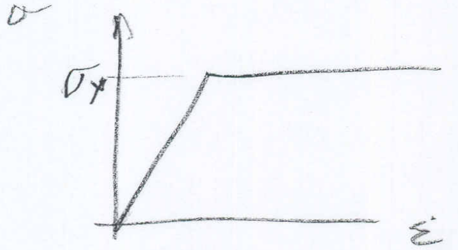


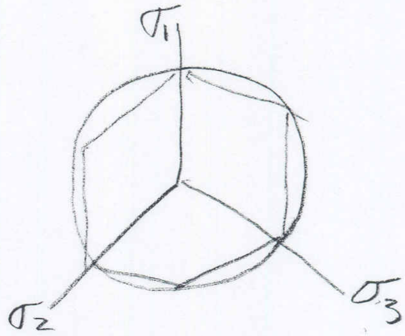
Stress (N/mm²)
161
−166
15
35
55
1
2
$(\sigma_\theta)_2$
σ_r
$-p$
$(\sigma_\theta)_1$

Solution (3)

(a) **Elastic-perfectly-plastic**

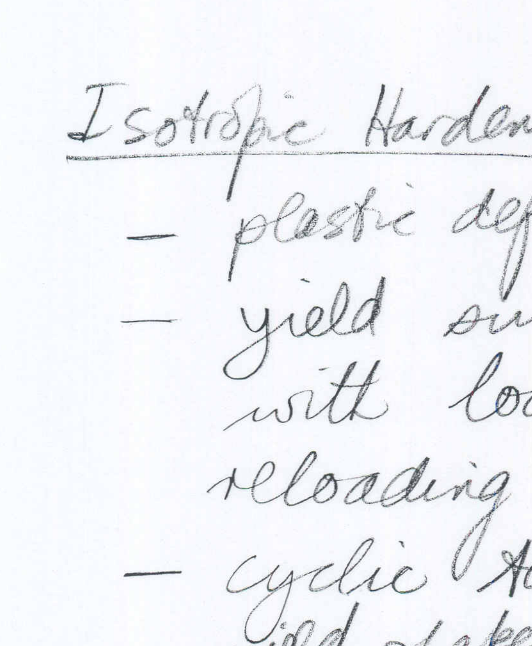
- no hardening
- yield locus does not change position size or shape
- gives hysteresis loops with alternating plasticity

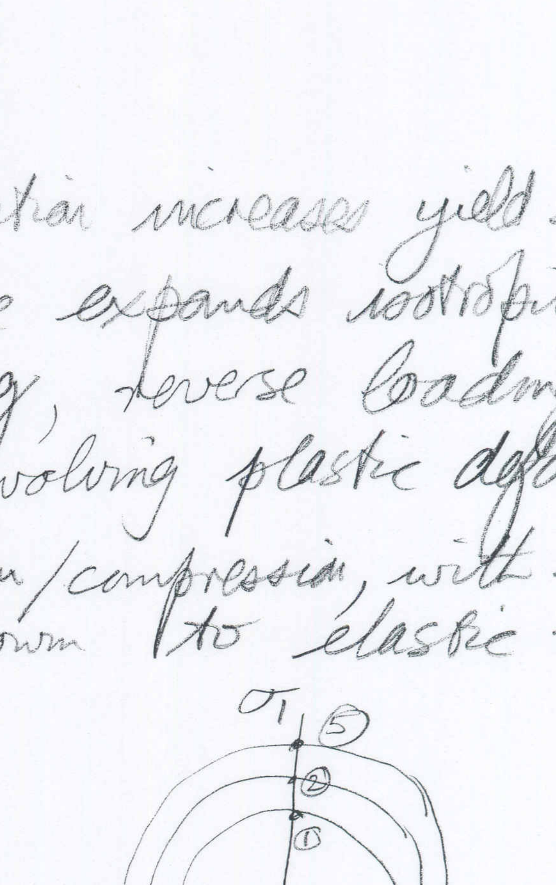




Isotropic Hardening

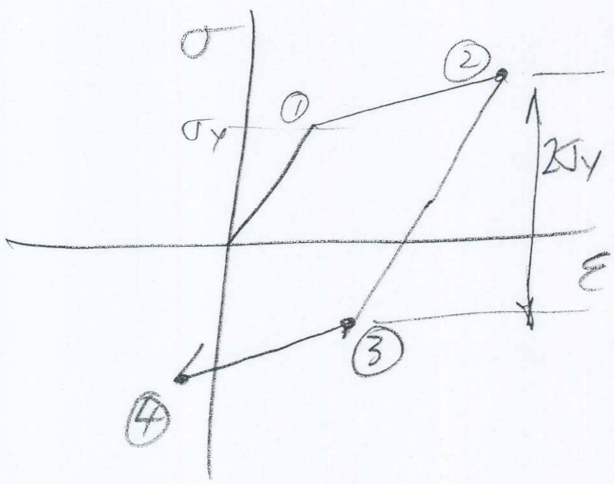
- plastic deformation increases yield stress
- yield surface expands isotropically with loading, reverse loading and reloading involving plastic deformation
- cyclic tension/compression, with initial yield shakes down to elastic behaviour

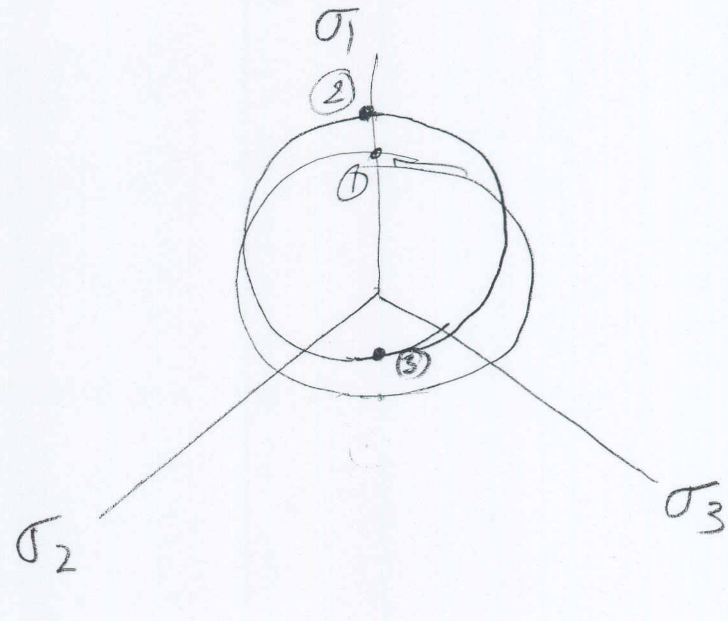




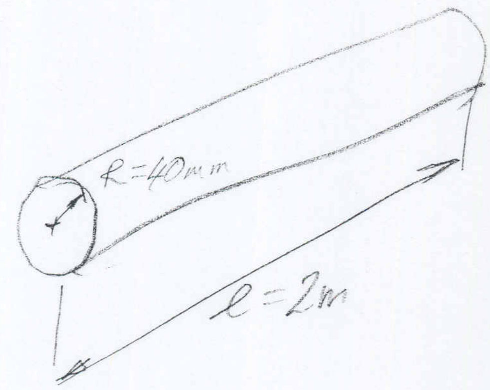
Kinematic Hardening

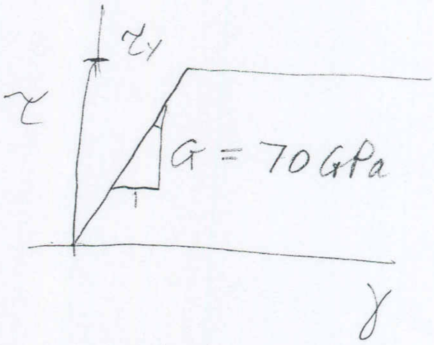
- assumes yield range remains constant at $2\sigma_Y$
- yield surface translates in stress space with no change in shape or size
- tensile/compressive cyclic loading which initially gives yield, also gives yield for subsequent cycles.
- steady-state of alternating plastic strain sets in (hysteresis loops)



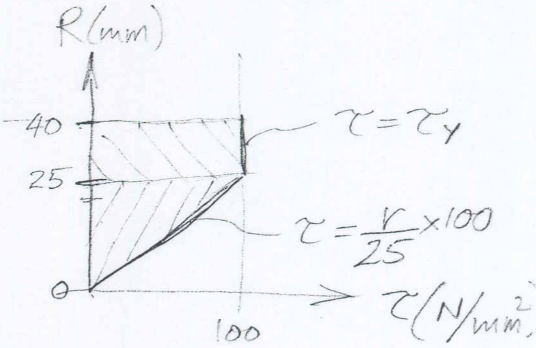


(b)





(i)



Equilibrium

$$T = \int_0^{40} r \times \tau \times 2\pi r \, dr$$

$$= \int_0^{40} 2\pi \tau r^2 dr$$

$$= \int_0^{25} 2\pi \times \frac{r}{\cancel{25}} \times \cancel{100}^{\,4} \times r^2 dr + \int_{25}^{40} 2\pi \times \tau_Y r^2 dr$$

$$= 8\pi \int_0^{25} r^3 dr + 200\pi \int_{25}^{40} r^2 dr$$

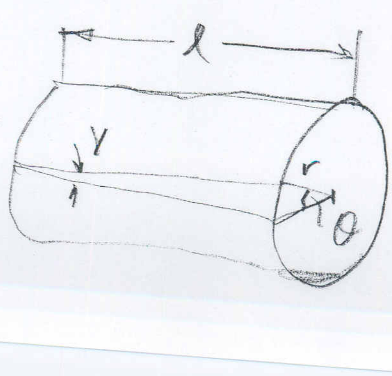
$$= 8\pi \left[\frac{r^4}{4}\right]_0^{25} + 200\pi \left[\frac{r^3}{3}\right]_{25}^{40}$$

$$= 2\pi \times 25^4 + \frac{200\pi}{3}\left[40^3 - 25^3\right]$$

$$= 2.4547 \times 10^6 + 10.1330 \times 10^6 \text{ Nmm}$$

ie $T = 12.588$ kNm

Compatability



$$r\theta = \gamma \ell$$

At $r = 25\,mm$, $\tau = \tau_Y$, $\gamma = \gamma_Y$

and $G = \dfrac{\tau_Y}{\gamma_Y}$, ie, $\gamma_Y = \dfrac{\tau_Y}{G}$

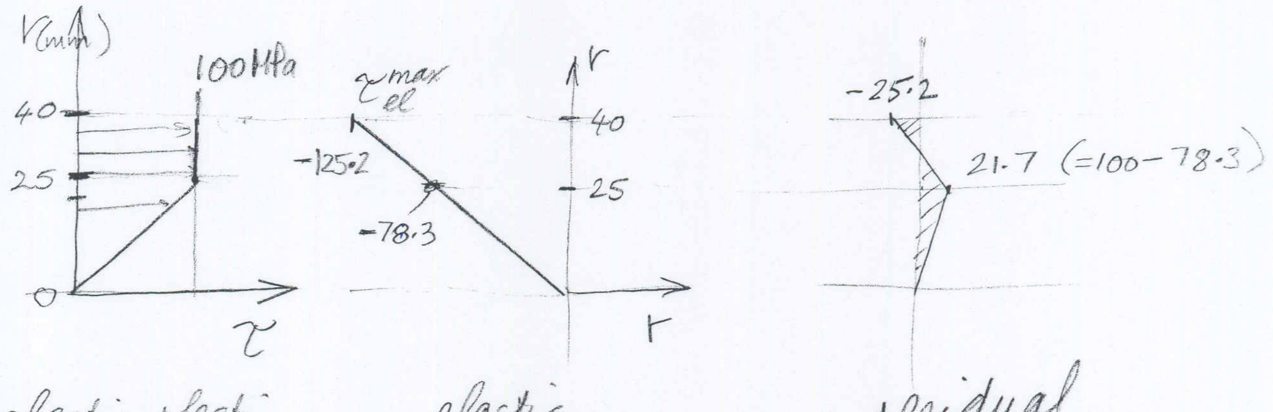
$$\therefore \gamma_Y = \frac{100\,N/mm^2}{70 \times 10^3\,N/mm^2} = 1.4286 \times 10^{-3}$$

$$\therefore \theta = \frac{\gamma l}{r} = 1.4286 \times 10^{-6} \times \frac{2000\,mm}{25\,mm}$$

$$\text{ie } \theta = 0.11429\,rad \times \frac{360}{2\pi}\frac{{}^\circ}{rad}$$

$$\therefore \underline{\theta = 6.547^\circ}$$

(ii)



elastic-plastic loading

elastic unloading

residual

Elastic unloading, $\tau_{el}^{max} = \dfrac{Tr}{J}$, $J = \dfrac{\pi d^4}{32} = \dfrac{\pi \times 80^4\,mm^4}{32}$

$$= 4.02176 \times 10^6\,mm^4$$

$$\tau_{el}^{max} = -\frac{12.588 \times 10^6\,Nmm \times 40\,mm}{4.02176 \times 10^6\,mm^4}$$

$$\underline{\tau_{el}^{max} = -125.2\,N/mm^2}$$

$$\therefore \left(\tau_{el}\right)_{r=250} = \frac{25}{40} \times (-125.2) \frac{N}{mm^2}$$

$$= \underline{-78.3 \ N/mm^2}$$

Residual angle of twist is (again) determined using the condition at interface between purely elastic and elastic-plastic behaviour, ie $r = 25m$

$$\gamma_{r=25} = \frac{\tau_{residual}}{G} = \frac{21.7}{70 \times 10^3} = 3.1 \times 10^{-4} \text{ rad}$$

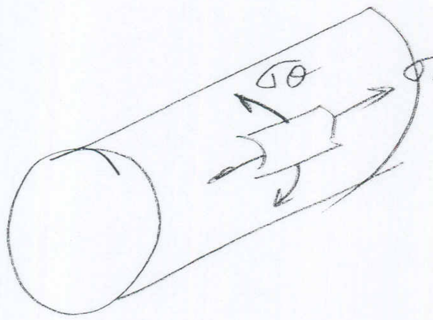
$$\text{i.e. } \theta_{residual} = \frac{3.1 \times 10^{-4} \times 2000}{25} \times \frac{360}{2\pi} \left(\theta = \frac{\gamma l}{r}\right)$$

$$= \underline{1.42^\circ}$$

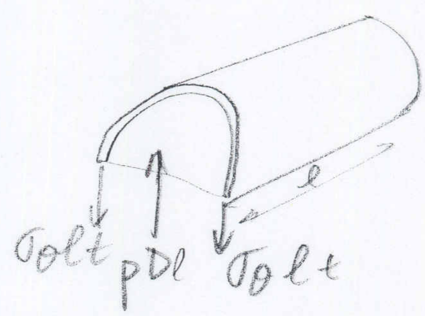
Solution (4)

(a) $D = 50\,mm$ (inside)

$t = 1\ mm$ $\Big\} \therefore \dfrac{R}{t} = 25$



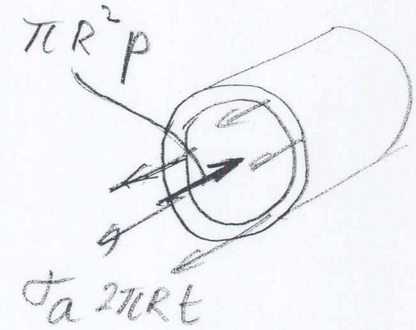
Assuming thin wall theory :-



$$2\sigma_\theta l t = pDl$$

$$\sigma_\theta = \frac{pD}{2t} = \frac{pR}{t}$$

$$\underline{\sigma_\theta = 25p}$$



$$\sigma_a \times 2\pi R t = \pi R^2 p$$

$$\sigma_a = \frac{pR}{2t} = \underline{12{\cdot}5p}$$

$\sigma_r \simeq 0$ (thin cylinder theory)

Tresca $\quad \hat{\sigma} - \check{\sigma} \leq 250\,MPa$

ie $25p \leq 250\,MPa$

ie $\underline{p \leq 10\,MPa}$

von Mises $\quad \frac{1}{\sqrt{2}}\sqrt{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2} \leq 250\,MPa$

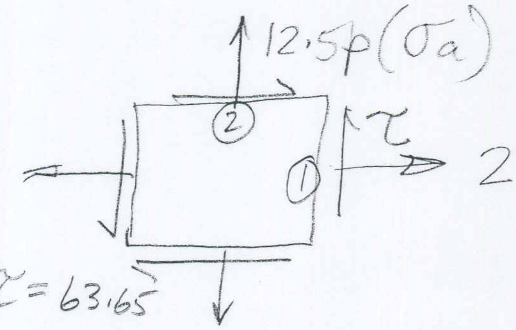
$$\frac{p}{\sqrt{2}}\sqrt{(25-12.5)^2+(12.5-0)^2+(0-25)^2} \leq 250\,MPa$$

$$\frac{p}{\sqrt{2}}\sqrt{12.5^2 + 12.5^2 + 25^2} \leq 250 \text{ MPa}$$

$$21.6506\, p \leq 250 \text{MPa}$$

$$\therefore p \leq 11.55 \text{ MPa}$$

(b)



Thin cylinder:-

$$J \simeq 2\pi R t \times R^3$$

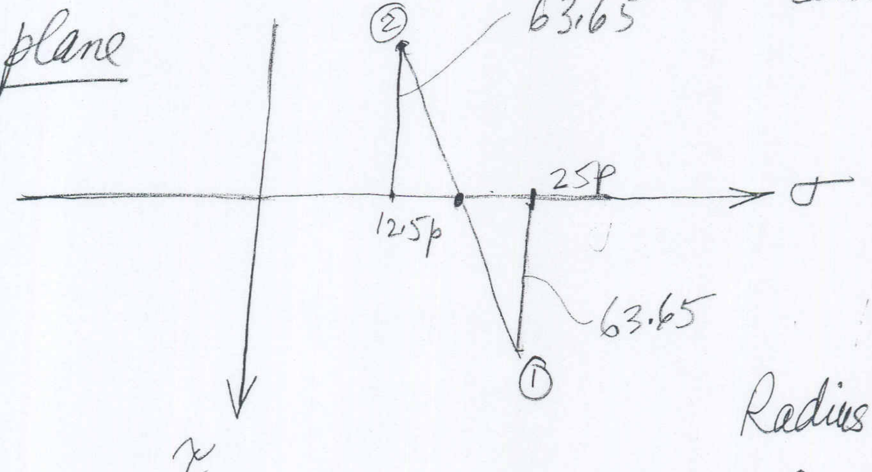
$$\text{ie } J \simeq 2\pi R^3 t$$

$$\frac{T}{J} = \frac{\tau}{r} \qquad \therefore \tau = \frac{Tr}{J} = \frac{0.25\times 10^6 \text{Nmm} \times 25\text{mm}}{2\pi \times 25^3 \times 1 \text{ mm}^4}$$

$$\tau = \frac{10^4}{50\pi} = 63.65 \frac{\text{N}}{\text{mm}^2}$$

Mohrs circle:-

σ_θ/σ_a plane



centre at $\sigma = \frac{(25+12.5)p}{2}$
$= 18.75p$

$$\text{Radius} = \sqrt{63.65^2 + (25-18.75)^2 p^2}$$

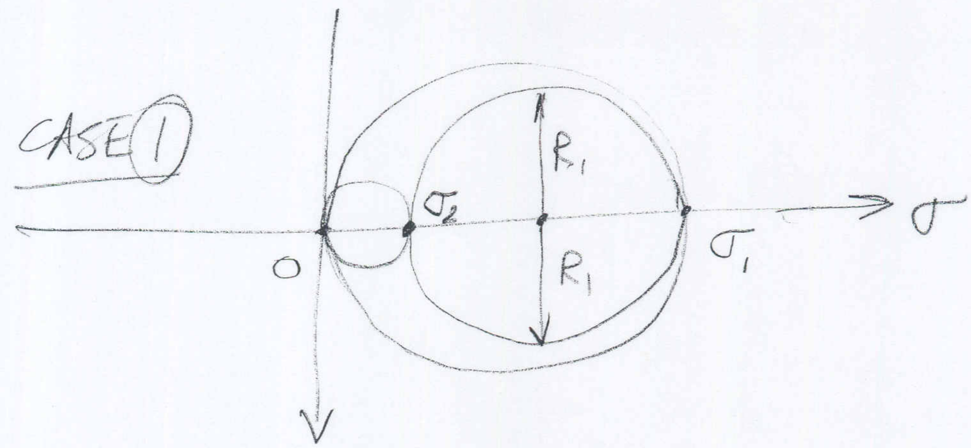
$$R_1 = \sqrt{4051 + 39.06 p^2}$$

$$\therefore \sigma_1 = 18.75\, p + R_1$$

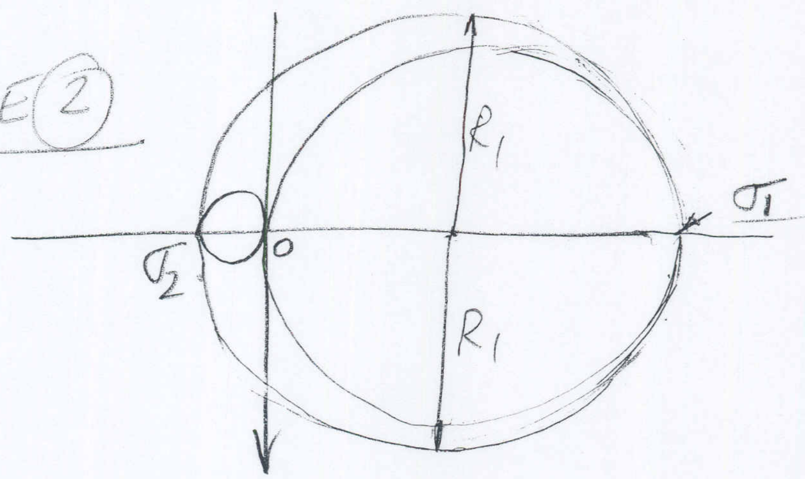
$$\text{and } \sigma_2 = 18.75p - R_1$$

Two possible solutions

Either CASE (1)



OR CASE (2)



Case (1) TRESCA $\sigma_1 - 0 = \sigma_Y$

ie $18.75p + R_1 - 0 = \sigma_Y$

ie. $18.75p + \sqrt{4051 + 39.06p^2} = 250$

$(250 - 18.75p)^2 = 4051 + 39.06p^2$

$62500 - 9375p + 351.5625p^2 = 4051 + 39.06p^2$

$312.5p^2 - 9375p + 58449 = 0$

$p^2 - 30p + 187 = 0$

$\therefore p = \dfrac{30 \pm \sqrt{30^2 - 4 \times 187}}{2} = \dfrac{30 \pm \sqrt{900 - 748}}{2}$

$= \dfrac{30 \pm 6.16}{2} = 8.84$ OR 21.16 MPa

21.16 is unrealistic since $p=10$ was obtained for p_{max} with pressure only. Therefore,

$$p = 8.84 \text{ MPa}$$

Case (2) Tresca:-

$$\sigma_1 - \sigma_2 = 250$$

ie $2R_1 = 250$

296.3

$\therefore \quad 4051 + 39.06p^2 = 125^2$

ie $p = 17.2$

Again, this is unrealistic because it is higher than the pressure which would cause yield on its own.

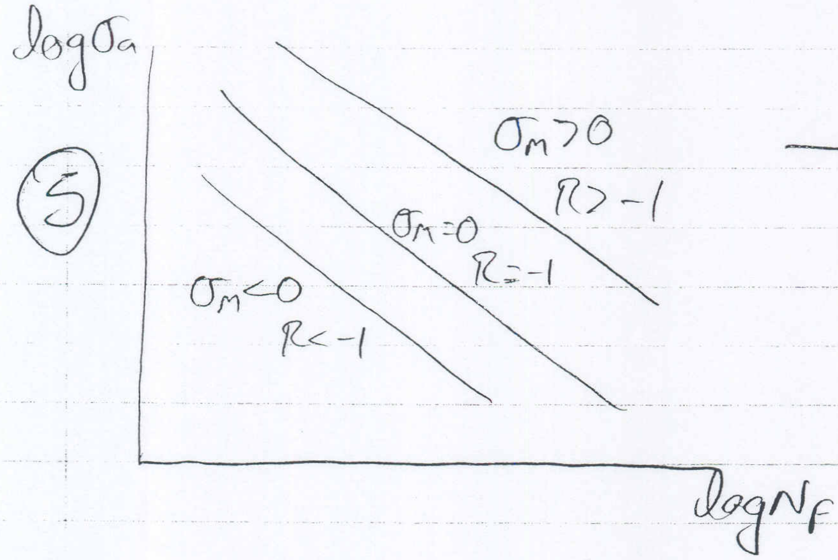
$$\therefore \quad p = 8.84 \text{ MPa}$$

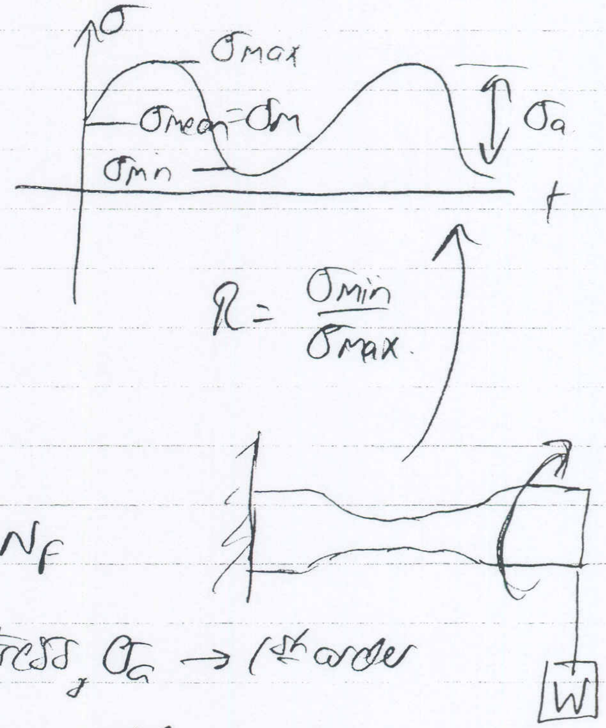
Solution (5)

(a)

Total life approach

(5) Does not distinguish between crack initiation and propagation
Smooth specimens under stress-controlled condition.



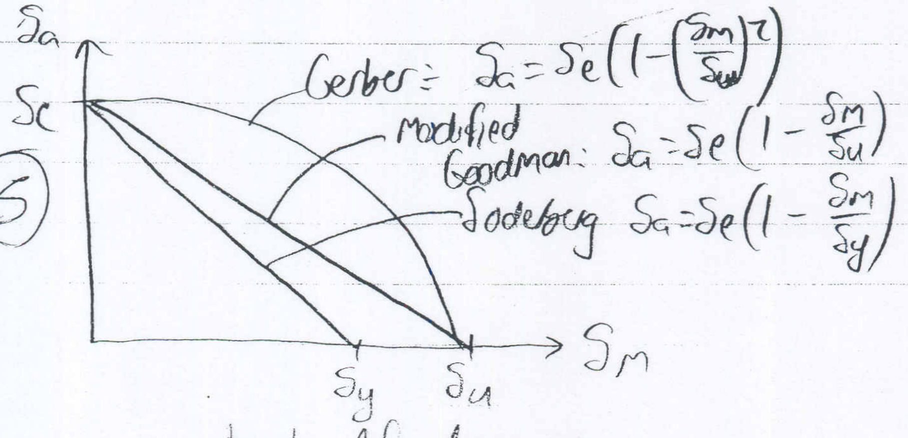


(5)

Effect of cyclic stress, σ_a → 1st order

effect of mean stress on life. — 2nd order
compressive → beneficial
tensile → detrimental

(5)

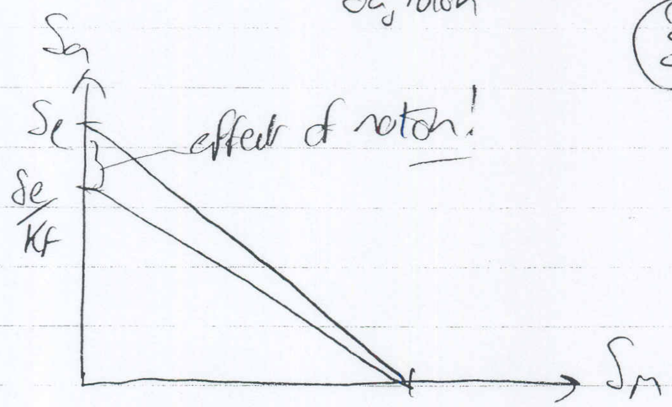


constant life lines

Effect of stress concentrations

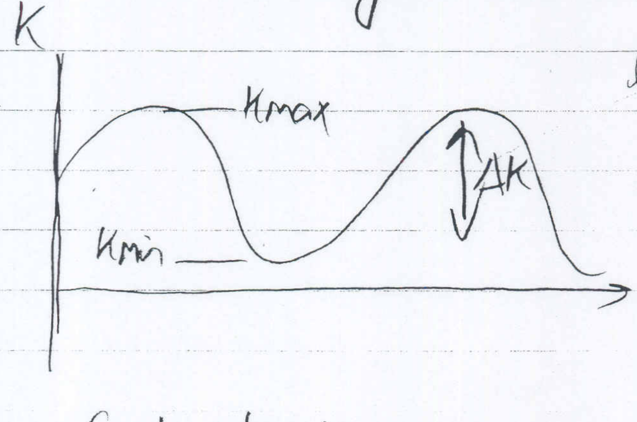
$$K_f = \frac{S_{a,\,smooth}}{S_{a,\,notch}}$$

(5)

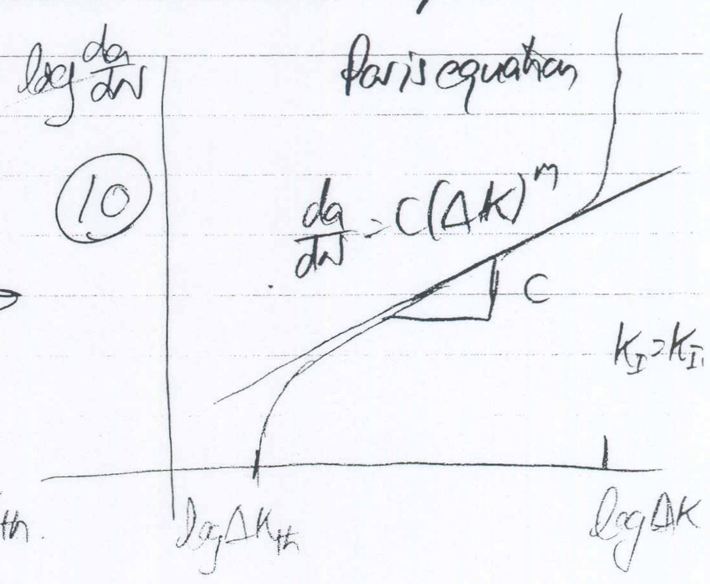


Damage tolerant approach

(5) Assumes that all structures have initial cracks and defines ~~the~~ fatigue life as number of cycles for crack to grow to a length which causes unstable fracture.

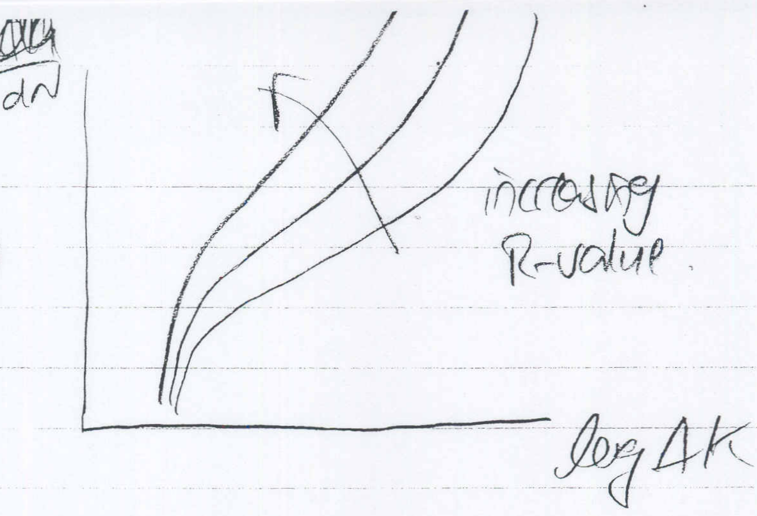


(10)



K_{Ic} = fracture toughness
no crack growth if $\Delta K < \Delta K_{th}$.

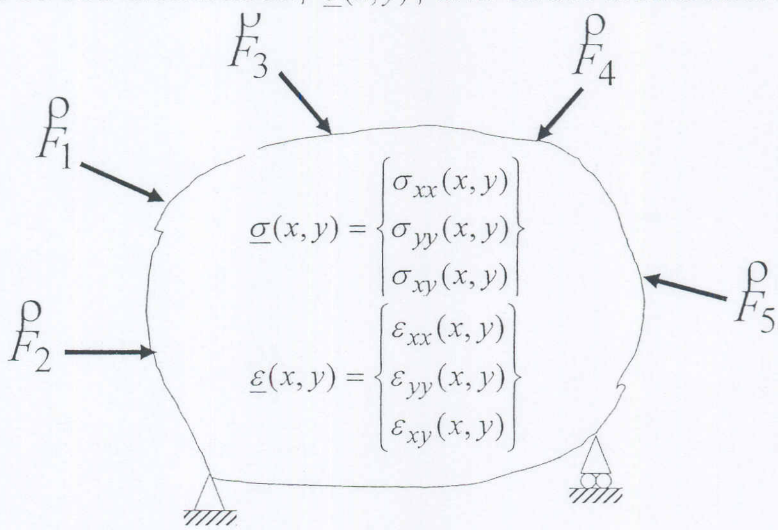
5



Solution (5)

~~**University of Nottingham**~~
~~**School of Mechanical, Materials & Manufacturing Engrng**~~
~~Solid Mechanics 3: The Finite Element (FE) Method~~

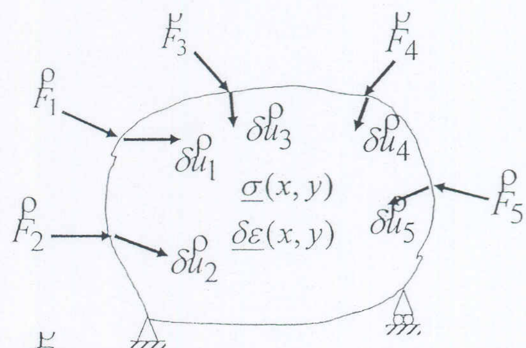
(b)(i) Consider a 2D body in equilibrium under the action of a set of vector forces, $\vec{F}_i$, giving rise to a set of vector displacements and thence to an internal stress distribution, $\underline{\sigma}(x,y)$, and strain distribution, $\underline{\varepsilon}(x,y)$.



The principal of virtual work (PVW) says that the work done by a set of forces in equilibrium moving through a set of small, compatible displacements is zero.

For deformable bodies, the total virtual work consists of the sum of external virtual work and internal virtual work:

$$\delta W = \delta W_{\text{int}} + \delta W_{ext}$$



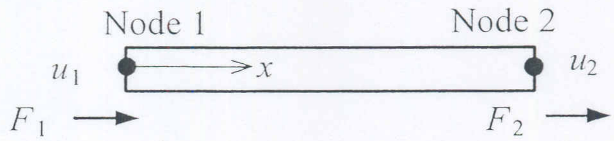
where

$$\delta W_{\text{int}} = -\int_V (\delta\varepsilon_{xx}\sigma_{xx} + \delta\varepsilon_{yy}\sigma_{yy} + \delta\varepsilon_{xy}\sigma_{xy})dV$$

$$\delta W_{ext} = \delta\vec{u}_1 \cdot \vec{F}_1 + \delta\vec{u}_2 \cdot \vec{F}_2 + \delta\vec{u}_3 \cdot \vec{F}_3 + \delta\vec{u}_4 \cdot \vec{F}_4 + \delta\vec{u}_5 \cdot \vec{F}_5$$

The principle of virtual work or displacement is the foundation of structural finite element (FE) analysis as shown below. It permits the development of expressions for the stiffness matrices of a range of different structural element types. In turn this permits computational modeling of complex geometries to give approximate solutions for displacement, stress and strain distributions.

5(b)(ii) 1D example of the finite element method (FEM)

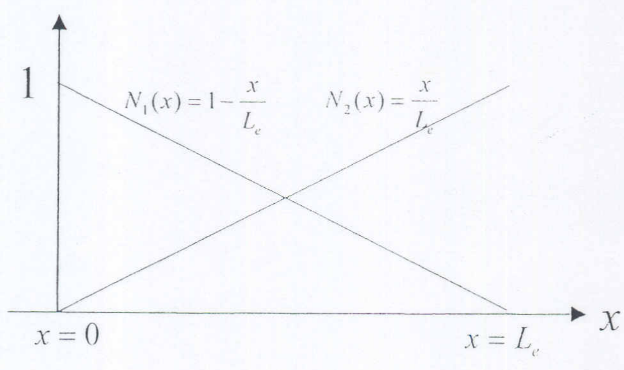


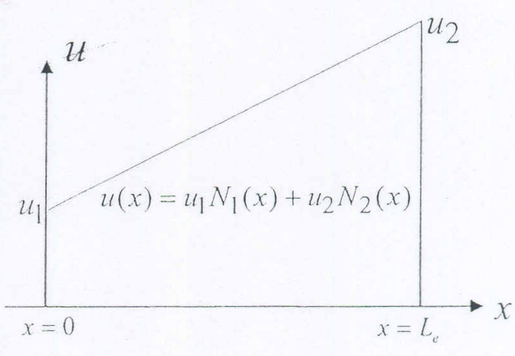
Length, L_e
Area, A
Young's modulus, E

Structure is divided up into elements, connected together at nodes. The basic approximation for this element in displacement-based FE analysis is:

$$
\begin{aligned}
u(x) &= u_1 N_1(x) + u_2 N_2(x) \\
&= \{N_1(x) \quad N_2(x)\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix}
\end{aligned}
$$

where u_1, u_2 are the nodal degrees of freedom (dofs) and $N_1(x)$, $N_2(x)$ are called the shape functions of the element.





Strain-displacement relation:

$$\varepsilon = \frac{du(x)}{dx}$$

$$\Rightarrow \varepsilon = \left\{ \frac{dN_1(x)}{dx} \quad \frac{dN_2(x)}{dx} \right\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \left\{ -\frac{1}{L_e} \quad \frac{1}{L_e} \right\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \{\underline{B}\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \{\underline{B}\}\{\underline{u}\}$$

Stress-strain (constitutive) relation:

$$\sigma = E\varepsilon = E\{\underline{B}\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \{\underline{D}\}\{\underline{B}\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix}$$

Internal virtual work:

$$\delta W_{\text{int}} = -\int_0^{L_e} \delta \underline{\varepsilon}^T \, \underline{\sigma} A dx$$

$$= -A \int_0^{L_e} \{\delta \underline{u}\}^T \{\underline{B}\}^T E\{\underline{B}\}\{\underline{u}\} dx$$

$$= -EA\{\delta \underline{u}\}^T \int_0^{L_e} \{\underline{B}\}^T \{\underline{B}\} dx \{\underline{u}\}$$

External virtual work:

$$\delta W_{ext} = \delta u_1 P_1 + \delta u_2 P_2 = \{\delta \underline{u}\}^T \{\underline{P}\}$$

Principal of virtual work (PVW)

$$\delta W_{ext} + \delta W_{\text{int}} = 0$$

$$EA\{\delta \underline{u}\}^T \int_0^{L_e} \{\underline{B}\}^T \{\underline{B}\} dx \{\underline{u}\} = \{\delta \underline{u}\}^T \{\underline{P}\}$$

$$\Rightarrow EA \int_0^{L_e} \{\underline{B}\}^T \{\underline{B}\} dx \{\underline{u}\} = \{\underline{P}\}$$

$$\Rightarrow \{K\}\{\underline{u}\} = \{\underline{P}\}$$

This could be provided with more words and less equations provided the process is clearer explained

This is the stiffness equation for the element of a 1D stress system.

$$\{K\} = EA \int_0^{L_e} \{\underline{B}\}^T \{\underline{B}\} dx$$

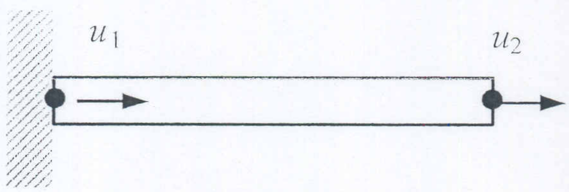
$$= EA \int_0^{L_e} \begin{Bmatrix} -\dfrac{1}{L_e} \\ \dfrac{1}{L_e} \end{Bmatrix} \begin{Bmatrix} -\dfrac{1}{L_e} & \dfrac{1}{L_e} \end{Bmatrix} dx$$

$$= \frac{EA}{L_e} \begin{Bmatrix} 1 & -1 \\ -1 & 1 \end{Bmatrix}$$

$$\Rightarrow \frac{EA}{L_e} \begin{Bmatrix} 1 & -1 \\ -1 & 1 \end{Bmatrix} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \begin{Bmatrix} P_1 \\ P_2 \end{Bmatrix}$$

Solution of stiffness equation:

To solve the stiffness equation, need to apply boundary conditions – otherwise, elements can move as rigid bodies and we cannot solve the equations for unique values of displacement.



Suppose $u_1 = 0$, then $u_2 = \dfrac{P_2 L_e}{EA}$ which is the exact solution for this problem. This is because the assumed linear variation of displacement is the exact variation in this case.

Once the primary variables have been solved for, by solving the stiffness matrix, then back-substitution into the strain-displacement and stress-strain relations yield the strains and stresses in each element and consequently in the complete FE model for a multi-element analysis.

Strain:

$$\Rightarrow \varepsilon = \left\{ -\frac{1}{L_e} \quad \frac{1}{L_e} \right\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \{\underline{B}\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = B_2 u_2 = \frac{1}{L_e} \frac{P_2 L_e}{EA} = \frac{P_2}{EA}$$

Stress:

$$\sigma = E\{\underline{B}\} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = E\varepsilon = E\frac{P_2}{EA} = \frac{P_2}{A}$$

Reactions:

$$\frac{EA}{L_e} \begin{Bmatrix} 1 & -1 \\ -1 & 1 \end{Bmatrix} \begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} = \begin{Bmatrix} P_1 \\ P_2 \end{Bmatrix} = \begin{Bmatrix} R \\ P \end{Bmatrix}$$

$$u_1 = 0$$

$$\Rightarrow -\frac{EA}{L_e} u_2 = R$$

$$\Rightarrow R = -\frac{EA}{L_e} \frac{PL_e}{EA} = -P$$

Solution requires description of the process to this point including a clear statement of the various stages. Another element type (or approach) could be used instead to illustrate the process.