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CAE & Stress Analysis M5X02064

7/5/2020

13:00

Page 1/6

Up Force: P_{dL}
Resisting: $2t\sigma_{HL}$

$$P_{dL} = 2t\sigma_{HL}$$

$$\sigma_{HL} = \frac{P_{dL}}{2tL}$$

$$\sigma_{HL} = \frac{P_{dL}}{2t}$$

b $\sigma_L = \frac{P_{dL}}{4t}$ Which is half the hoop stress $\sigma_L = \frac{1}{2}\sigma_{HL}$
Hence material will break with hoop stress before longitudinal

c Density = ~~1.0258~~
 $1.0260 \times \frac{1 \times 10^{-3}}{1 \times 10^{-6}} = 1.026 \times 10^3 \text{ kg/m}^3 = 1026 \text{ kg/m}^3$

d $d = 5.6 \text{ m}$
 $t = 5.6 \times 10^{-3} \text{ m}$

Safety 2.1

$$\sigma_{HL} = 200 \text{ MPa}$$

$$\sigma_{HL} = \frac{P_{dL}}{2t}$$

$$P = \frac{200 \times 10^6 \times 2 \times 5.6 \times 10^{-3}}{5.6}$$

$$P = 400 \times 10^3 \text{ Pa} \quad P_{max}$$

$$P = \rho gh$$

$$h = \frac{400 \times 10^3}{1.026 \times 10^3 \times 9.81}$$

$$h = \frac{400 \times 10^3}{1026 \times 9.81}$$

$$h = 39.74 \text{ m}$$

with safety 2.1

$$h = 39.74 \times 10^6 \text{ m}$$

with safety $\div 2.1$

$$h = 19.87 \text{ m}$$

$$h = 18.92 \times 10^6 \text{ m}$$

$$h = 18.92 \text{ m}$$

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7/5/2020

13:00

1e

with internal pressure will go deeper

Page 2/
6

$$P = 0.4 \text{ MPa} + 0.1 \text{ MPa} = 0.5 \text{ MPa}$$

$$P = \rho g h$$

$$h = \frac{0.5 \times 10^6}{1026 \times 9.81}$$

$$h = 49.677 \text{ m}$$

with safety

$$h = \frac{24.84 \text{ m}}{23.66}$$

$$f \quad P = 0.4 + 0.2 = 0.6 \text{ MPa}$$

$$h = \frac{0.6 \times 10^6}{1026 \times 9.81}$$

$$h = 56.612 \text{ m}$$

with safety

$$h = \frac{24.81 \text{ m}}{26.96 \text{ m}}$$

g Pressure from the diving ~~sub~~ submarine is compressing the walls in. with internal pressure it will push walls out. with more internal pressure the sub can go deeper as seen in previous answers

H ~~A~~ In real results the density and temperature of the water would change with depth. The water would be more dense and colder meaning external pressure would be greater so the sub won't be able to safely dive as deep as ~~calculated~~ calculated

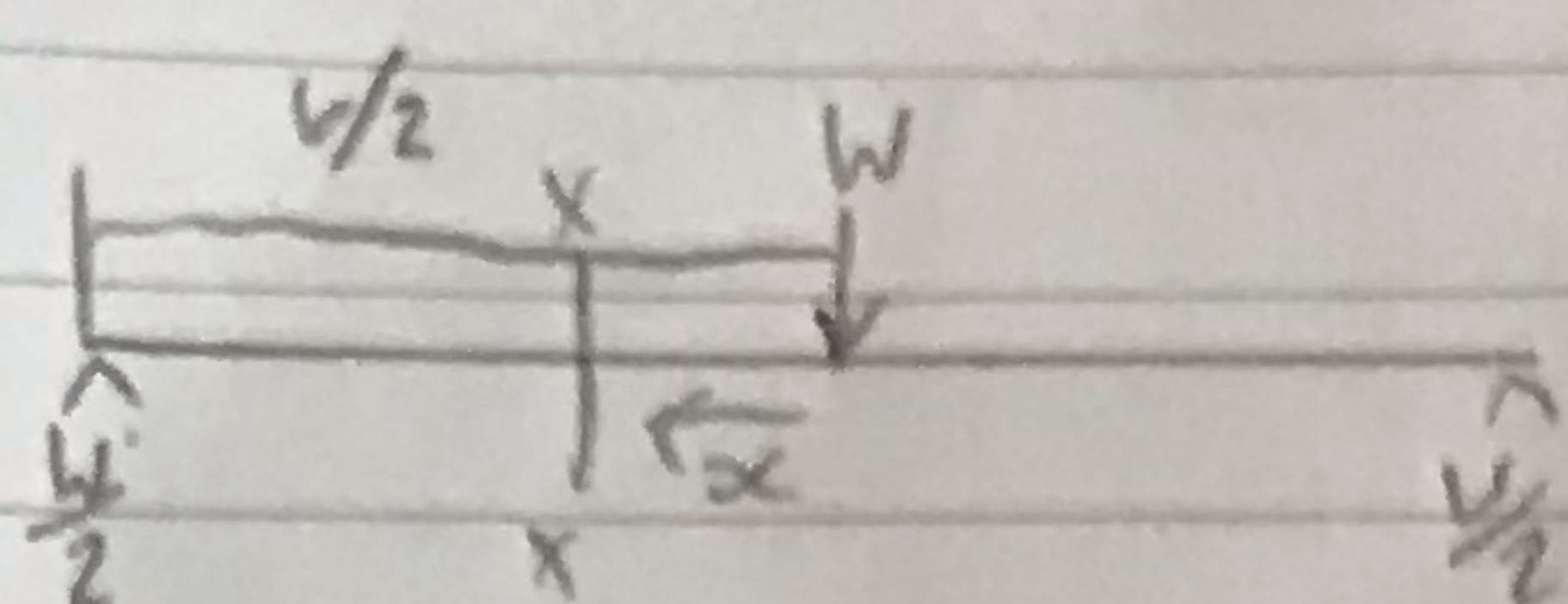
Daniel Wilcox
1802007

CAE & Stress Analysis MSX02064

7/5/2020 3

13:00 a

Page 3/6



$$y=0 \text{ when } x = \frac{L}{2} \text{ \& } -\frac{L}{2}$$

Origin in centre at W

$$EI \frac{d^2 y}{dx^2} = \frac{W}{2} \left(\frac{L}{2} - x \right) \\ = \frac{WL}{4} - \frac{Wx}{2}$$

$$EI \frac{dy}{dx} = \frac{WLx}{4} - \frac{Wx^2}{4} + a$$

$$EI y = \frac{WLx^2}{8} - \frac{Wx^3}{12} + ax + b$$

$$\frac{dy}{dx} = 0 \text{ when } x=0 \\ a=0$$

$$y=0 \text{ at } x = \pm \frac{L}{2}$$

$$0 = \frac{WL^3}{32} - \frac{WL^3}{96} + b$$

$$b = -\frac{WL^3}{48}$$

$$EI y = \frac{WLx^2}{8} - \frac{Wx^3}{12} - \frac{WL^3}{48}$$

$$y = \frac{1}{EI} \left(\frac{WLx^2}{8} - \frac{Wx^3}{12} - \frac{WL^3}{48} \right)$$

$$b \frac{dy}{dx} \text{ max at } x = \frac{L}{2}$$

$$EI \frac{dy}{dx} = \frac{WL^2}{8} - \frac{WL^2}{16}$$

Daniel Wilcox

1802007

7/5/2020

13:00

3b

Max deflection $x=0$

Page 4/6

$$EIy = 0 - 0 - \frac{WL^3}{48}$$

$$y_{\text{max}} = -\frac{WL^3}{EI48}$$

$$C \quad y_{\text{max}} = 11\text{mm} = 0.011\text{m}$$

$$W = 46\text{kN}$$

$$E = 235 \times 10^9 \text{ Pa}$$

$$I = 80 \times 10^{-6} \text{ m}^4$$

Find L

$$-0.011 = -\frac{46 \times 10^3 \times L^3}{235 \times 10^9 \times 80 \times 10^{-6} \times 48}$$

$$L = \sqrt[3]{\frac{-0.011 \times 235 \times 10^9 \times 80 \times 10^{-6} \times 48}{-46 \times 10^3}}$$

$$L = 5.998\text{m}$$

d Results may be different firstly with change in load W with deflection as the train loads other wheels. However the main factor is the Mass of the line and sleepers which creates a v.d.l and point loads across the span meaning a far reduced length is possible

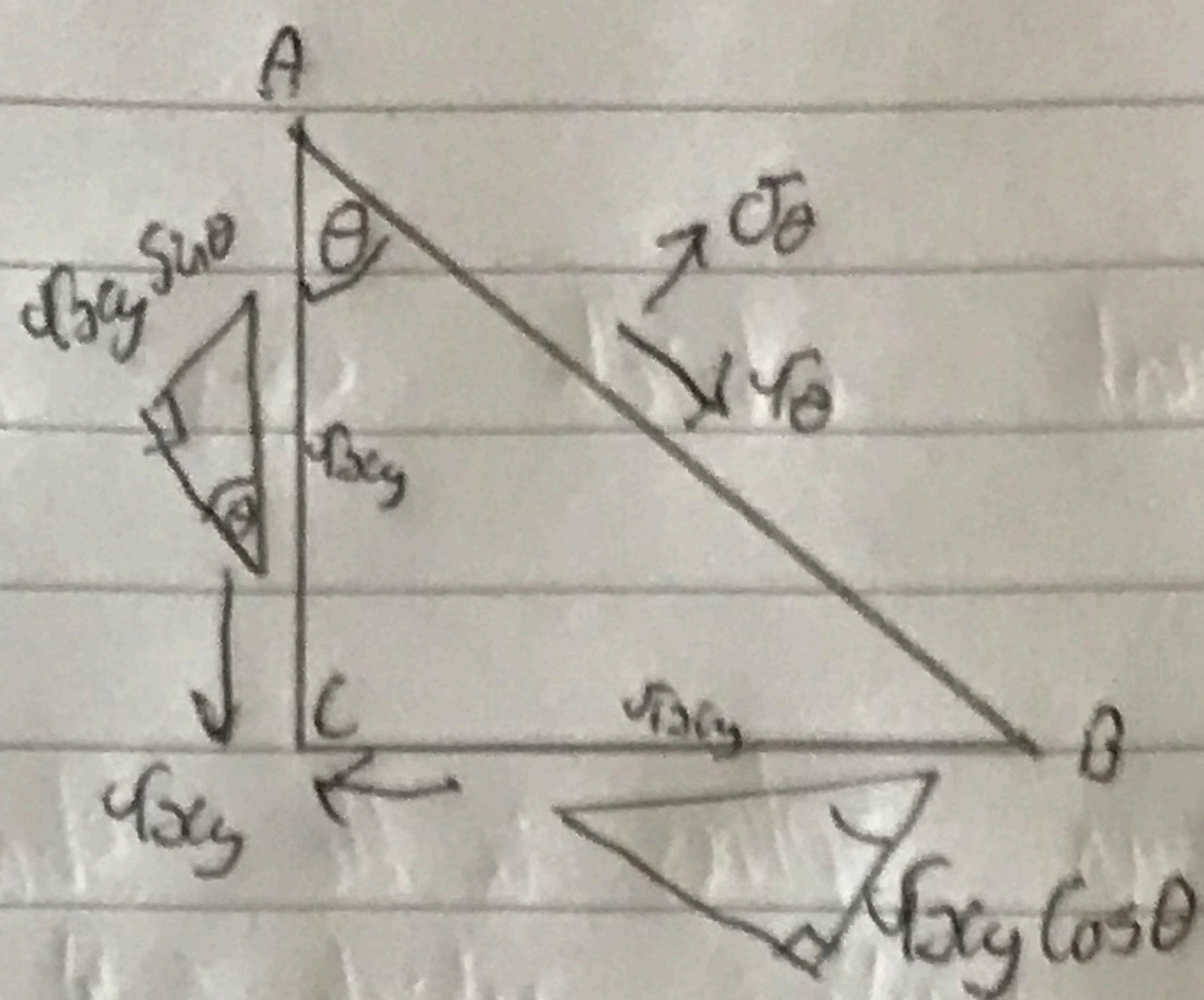
e ~~Permanent deformation with the train line being strained to yield point.~~

e Permanent deformation of the line should be considered as loading beyond yield will change the dimension of the track and reduce life of the line.

7/5/2020 4

13:00 a

Page 5/6



$$AB \sigma_{\theta} = AC \tau_{xy} \sin \theta + BC \tau_{xy} \cos \theta$$

$$\sigma_{\theta} = \frac{AC}{AB} \tau_{xy} \sin \theta + \frac{BC}{AB} \tau_{xy} \cos \theta$$

$$\sigma_{\theta} = \cos \theta \tau_{xy} \sin \theta + \sin \theta \tau_{xy} \cos \theta$$

$$\sigma_{\theta} = 2 \tau_{xy} \sin \theta \cos \theta$$

$$\cos \theta \sin \theta = \frac{1}{2} \sin 2\theta$$

$$\therefore \sigma_{\theta} = \tau_{xy} \sin 2\theta$$

b $\theta = 65^{\circ}$

$$\tau_{xy} = 90 \times 10^6 \text{ Pa}$$

$$\sigma_{\theta} = 90 \times 10^6 \times \sin(2 \times 65)$$

$$\sigma_{\theta} = 68.944 \text{ MPa}$$

c $\tau_{\theta} = -\tau_{xy} \cos 2\theta$

$$= -90 \times 10^6 \cos(2 \times 65)$$

$$\tau_{\theta} = 57.851 \text{ MPa}$$

d Under braking a brake pad material would see large shear stress to negligible compression force

e This is unlikely to happen as brake pressure will apply compressive force to the pad to keep it applied to the disk

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CAE & Stress Analysis MSX02064

7/5/2020

13:00 4 d
Page 6/6

Drive shaft is a component which sees largely
shear stress only

e Drive shaft angle and even splines in the shaft will
add compressive and tensile loadings which could be
measured. Therefore, the formula $\sigma_{\theta} = \frac{1}{2}(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$
would be used as σ_x and σ_y would affect
the final value.