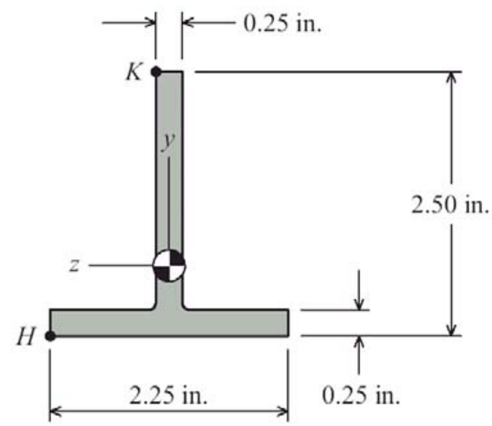
 Name: ___________________________

1. A cross section of a beam is shown. For any moment M about the *z*-axis, which point will have the greater bending stress magnitude (either tension or compression)?

   ______ Point *K*

   ______ Point *H*



2. If the beam in Problem 1 is subjected to a negative moment about the z-axis, which bending stress will have the greater magnitude?

   ______ Compression bending stress

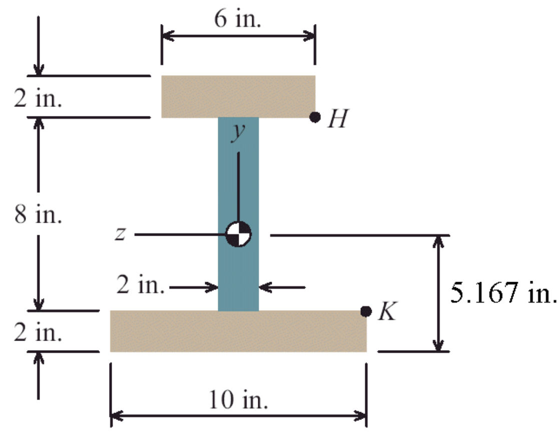
   ______ Tension bending stress

3. Determine the distance from the bottom of the cross section in Problem 1 to the centroid.

   $y_{\text{from bottom}}$ = ___________ in.

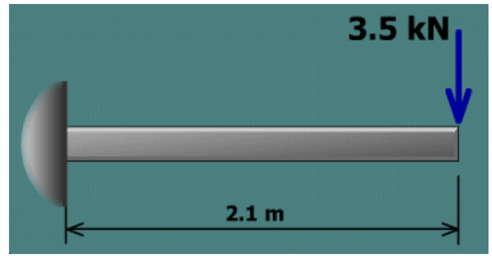
4. Determine the moment of inertia about the *z*-axis for the cross section shown. Note that the centroid is 5.167 in. from the bottom.

   I_z = ___________ in.^4

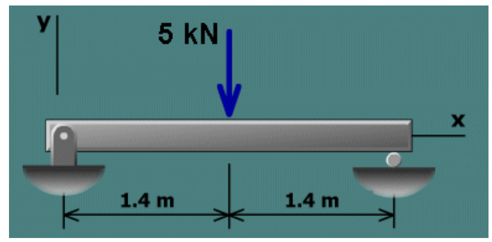


5. Determine the maximum bending moment in the following beams.

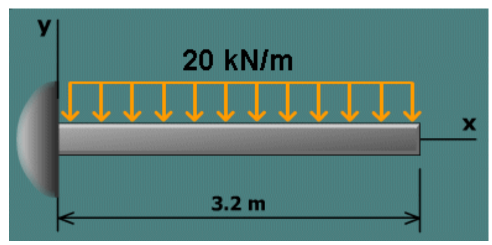
M_{max} = __________ kN-m



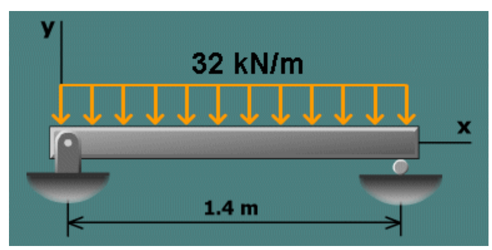
M_{max} = __________ kN-m



M_{max} = __________ kN-m

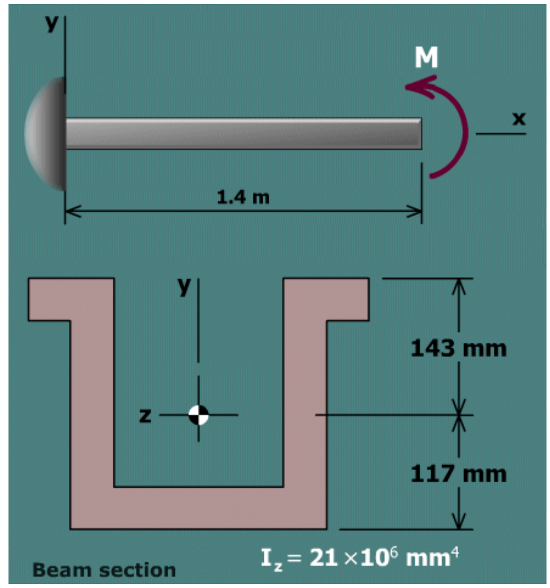


M_{max} = __________ kN-m



6. For the beam shown, the allowable compression bending stress is 60 MPa, and the allowable tension bending stress is 70 MPa. Determine the maximum value of *M* that can be applied as shown to the beam.

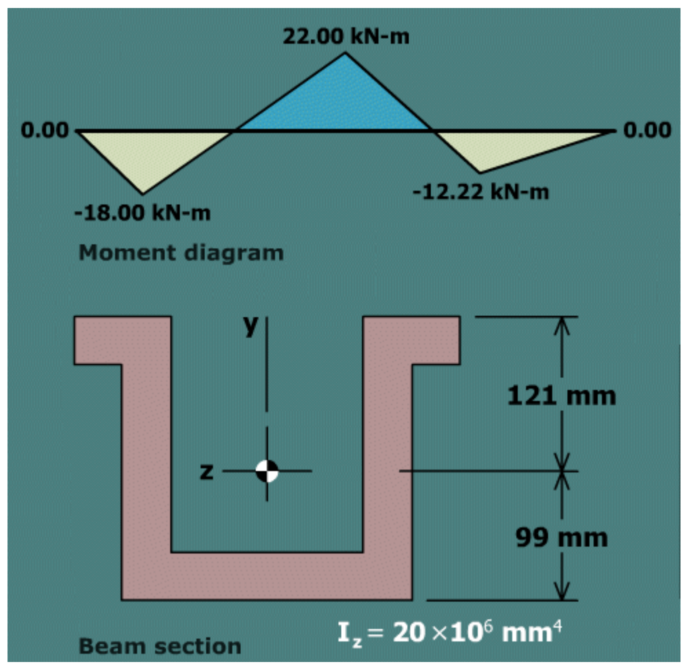
   M = __________ kN-m



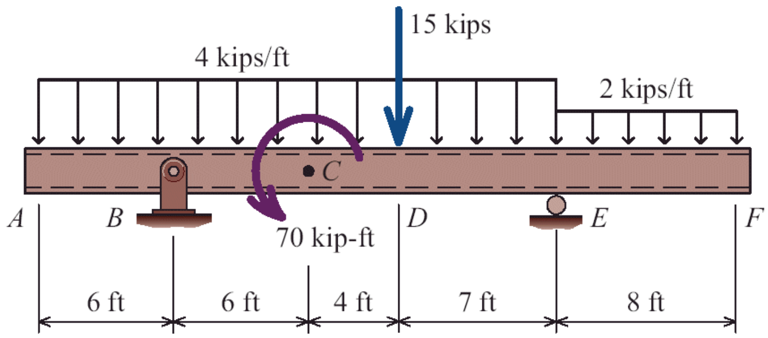
7. For the moment diagram and cross section shown, compute the maximum tension and compression bending stresses produced at any location along the beam span.

   $\sigma_{\text{max tension}}$ = ___________ MPa

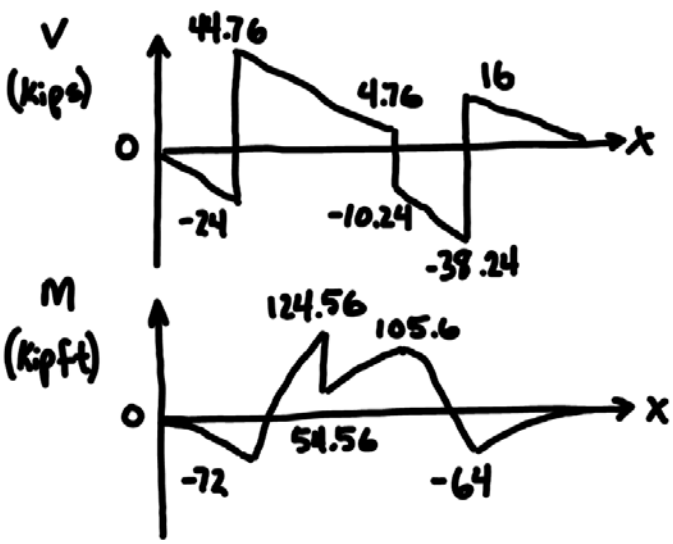
   $\sigma_{\text{max compression}}$ = ___________ MPa

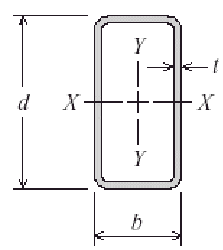


8. A HSS10×4×1/2 standard steel shape is used to support the loads shown on the beam. The shape is oriented so that bending occurs about the strong axis. Determine the magnitude of the maximum bending stress in the beam. Note that the shear-force and bending-moment diagrams have been provided in kips and kip-ft.



σ_{max} = ___________ ksi

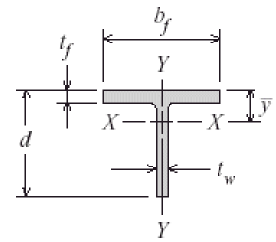




Hollow Structural Sections or HSS Shapes

Designation	Depth d	Width b	Wall thickness (nom.) t	Weight per foot	Area A	I_x	S_x	r_x	I_y	S_y	r_y
	in.	in.	in.	lb/ft	in.^{2}	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
HSS12×8×1/2	12	8	0.5	62.3	17.2	333	55.6	4.41	178	44.4	3.21
×8×3/8	12	8	0.375	47.8	13.2	262	43.7	4.47	140	35.1	3.27
×6×1/2	12	6	0.5	55.5	15.3	271	45.2	4.21	91.1	30.4	2.44
×6×3/8	12	6	0.375	42.7	11.8	215	35.9	4.28	72.9	24.3	2.49
HSS10×6×1/2	10	6	0.5	48.7	13.5	171	34.3	3.57	76.8	25.6	2.39
×6×3/8	10	6	0.375	37.6	10.4	137	27.4	3.63	61.8	20.6	2.44
×4×1/2	10	4	0.5	41.9	11.6	129	25.8	3.34	29.5	14.7	1.59
×4×3/8	10	4	0.375	32.5	8.97	104	20.8	3.41	24.3	12.1	1.64
HSS8×4×1/2	8	4	0.5	35.1	9.74	71.8	17.9	2.71	23.6	11.8	1.56
×4×3/8	8	4	0.375	27.4	7.58	58.7	14.7	2.78	19.6	9.80	1.61
×4×1/4	8	4	0.25	19.0	5.24	42.5	10.6	2.85	14.4	7.21	1.66
×4×1/8	8	4	0.125	9.85	2.70	22.9	5.73	2.92	7.90	3.95	1.71
HSS6×4×3/8	6	4	0.375	22.3	6.18	28.3	9.43	2.14	14.9	7.47	1.55
×4×1/4	6	4	0.25	15.6	4.30	20.9	6.96	2.20	11.1	5.56	1.61
×4×1/8	6	4	0.125	8.15	2.23	11.4	3.81	2.26	6.15	3.08	1.66
×3×3/8	6	3	0.375	19.7	5.48	22.7	7.57	2.04	7.48	4.99	1.17
×3×1/4	6	3	0.25	13.9	3.84	17.0	5.66	2.10	5.70	3.80	1.22
×3×1/8	6	3	0.125	7.30	2.00	9.43	3.14	2.17	3.23	2.15	1.27

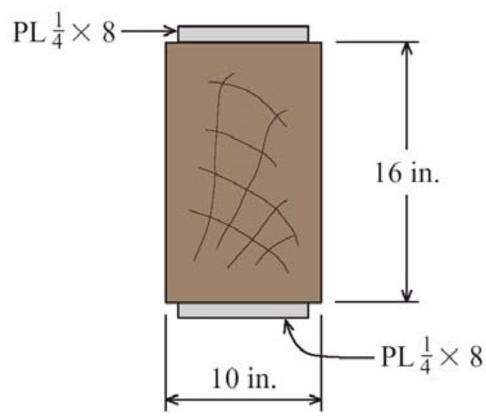
9. Circle the most economical WT beam in the following table if a section modulus $S_x \geq 8$ in.^{3} is required.



Shapes Cut from Wide-Flange Sections or WT Shapes

Designation	Area A	Depth d	Web thickness t_w	Flange width b_f	Flange thickness t_f	Centroid $\bar{y}$	I_x	S_x	r_x	I_y	S_y	r_y
	in.^{2}	in.	in.	in.	in.	in.	in.^{4}	in.^{3}	in.	in.^{4}	in.^{3}	in.
WT12×47	13.8	12.2	0.515	9.07	0.875	2.99	186	20.3	3.67	54.5	12.0	1.98
WT12×38	11.2	12.0	0.440	8.99	0.680	3.00	151	16.9	3.68	41.3	9.18	1.92
WT12×34	10.0	11.9	0.415	8.97	0.585	3.06	137	15.6	3.70	35.2	7.85	1.87
WT12×27.5	8.10	11.8	0.395	7.01	0.505	3.50	117	14.1	3.80	14.5	4.15	1.34
WT10.5×34	10.0	10.6	0.430	8.27	0.685	2.59	103	12.9	3.20	32.4	7.83	1.80
WT10.5×31	9.13	10.5	0.400	8.24	0.615	2.58	93.8	11.9	3.21	28.7	6.97	1.77
WT10.5×25	7.36	10.4	0.380	6.53	0.535	2.93	80.3	10.7	3.30	12.5	3.82	1.30
WT10.5×22	6.49	10.3	0.350	6.50	0.450	2.98	71.1	9.68	3.31	10.3	3.18	1.26
WT9×27.5	8.10	9.06	0.390	7.53	0.630	2.16	59.5	8.63	2.71	22.5	5.97	1.67
WT9×25	7.33	9.00	0.355	7.50	0.570	2.12	53.5	7.79	2.70	20.0	5.35	1.65
WT9×20	5.88	8.95	0.315	6.02	0.525	2.29	44.8	6.73	2.76	9.55	3.17	1.27
WT9×17.5	5.15	8.85	0.300	6.00	0.425	2.39	40.1	6.21	2.79	7.67	2.56	1.22
WT8×28.5	8.39	8.22	0.430	7.12	0.715	1.94	48.7	7.77	2.41	21.6	6.06	1.60
WT8×25	7.37	8.13	0.380	7.07	0.630	1.89	42.3	6.78	2.40	18.6	5.26	1.59
WT8×20	5.89	8.01	0.305	7.00	0.505	1.81	33.1	5.35	2.37	14.4	4.12	1.56
WT8×15.5	4.56	7.94	0.275	5.53	0.440	2.02	27.5	4.64	2.45	6.2	2.24	1.17

10. Two ¼ in.×8 in. steel [E = 30,000 ksi] plates are securely attached to a pine [E = 2,000 ksi] timber to form a composite beam. Determine the maximum bending stress magnitude in the steel if a moment of 100 kip-ft is applied about the horizontal axis of the beam.



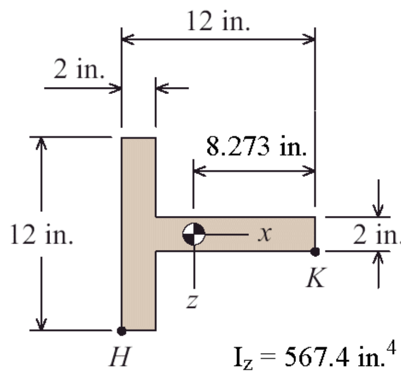
$\sigma_{\text{max steel}}$ = ___________ ksi

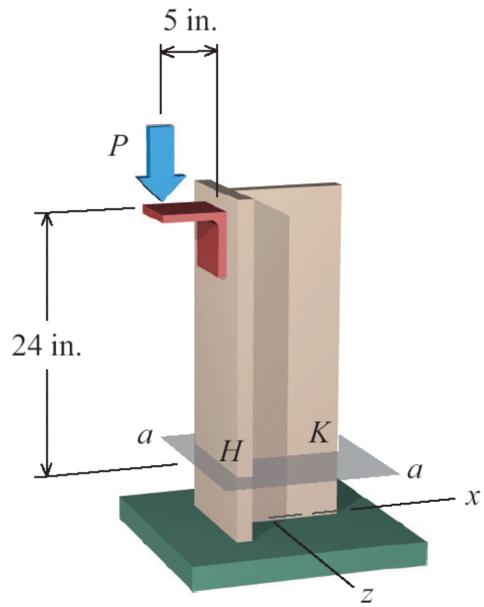
11. The tee shape is used as a short post to support a load of $P = 2{,}500$ lb. The load P is applied at a distance of 5 in. from the surface of the flange. Determine the normal force and bending moment located at section *a-a*. Also determine the magnitude of the bending stress at point *K*. Note that the centroid location and moment of inertia are provided.

$N =$ __________ lb

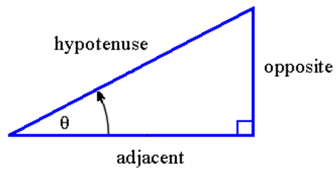
$M_z =$ __________ lb-in.

$\sigma_K =$ __________ psi





TRIGONOMETRY



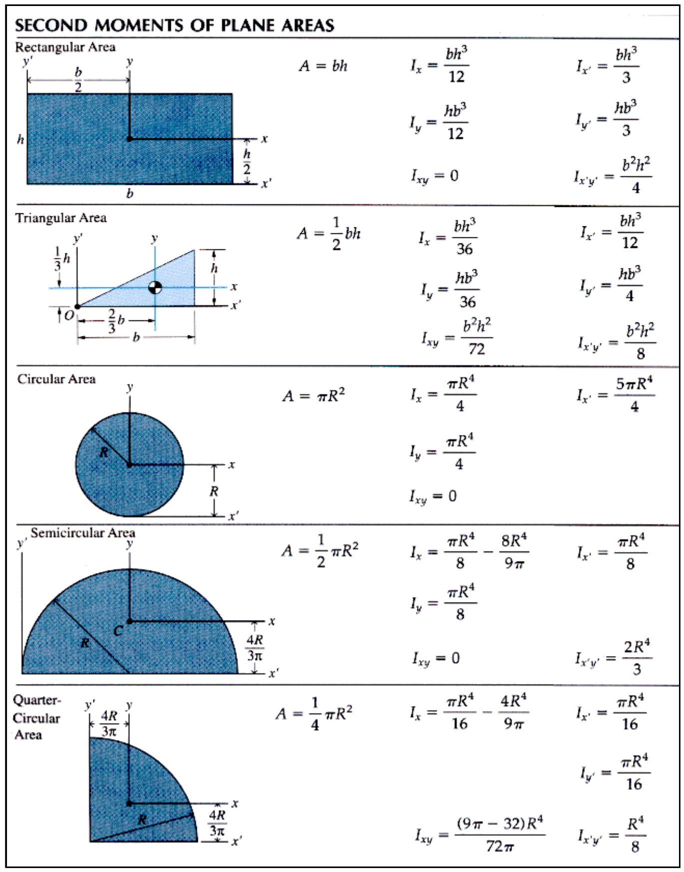
$\sin\theta = opp / hyp$

$\cos\theta = adj / hyp$

$\tan\theta = opp / adj$

STATICS

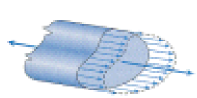
Symbol	Meaning	Equation	Units
$\bar{x}, \bar{y}, \bar{z}$	centroid position	$\bar{y} = \Sigma\bar{y}_i A_i / \Sigma A_i$	in, m
I	moment of inertia	$I = \Sigma(I_i + d_i^2 A_i)$	in^4, m^4
J	polar moment of inertia	$J_{solid\ circular\ shaft} = \pi d^4/32$ $J_{hollow\ circular\ shaft} = \pi(d_o^4 - d_i^4)/32$	in^4, m^4
N	normal force		lb, N
V	shear force	$V = \int -w(x)\,dx$	lb, N
M	bending moment	$M = \int V(x)\,dx$	in-lb, Nm
equilibrium		$\Sigma\mathbf{F} = \mathbf{0}$ $\Sigma\mathbf{M}_{(any\ point)} = \mathbf{0}$	lb, N in-lb, Nm

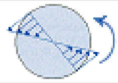
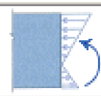
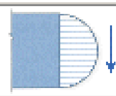


SECOND MOMENTS OF PLANE AREAS

Area	A	Centroidal axes	Axes x', y'
Rectangular Area	$A = bh$	$I_x = \frac{bh^3}{12}$, $I_y = \frac{hb^3}{12}$, $I_{xy} = 0$	$I_{x'} = \frac{bh^3}{3}$, $I_{y'} = \frac{hb^3}{3}$, $I_{x'y'} = \frac{b^2h^2}{4}$
Triangular Area	$A = \frac{1}{2}bh$	$I_x = \frac{bh^3}{36}$, $I_y = \frac{hb^3}{36}$, $I_{xy} = \frac{b^2h^2}{72}$	$I_{x'} = \frac{bh^3}{12}$, $I_{y'} = \frac{hb^3}{4}$, $I_{x'y'} = \frac{b^2h^2}{8}$
Circular Area	$A = \pi R^2$	$I_x = \frac{\pi R^4}{4}$, $I_y = \frac{\pi R^4}{4}$, $I_{xy} = 0$	$I_{x'} = \frac{5\pi R^4}{4}$
Semicircular Area	$A = \frac{1}{2}\pi R^2$	$I_x = \frac{\pi R^4}{8} - \frac{8R^4}{9\pi}$, $I_y = \frac{\pi R^4}{8}$, $I_{xy} = 0$	$I_{x'} = \frac{\pi R^4}{8}$, $I_{x'y'} = \frac{2R^4}{3}$
Quarter-Circular Area	$A = \frac{1}{4}\pi R^2$	$I_x = \frac{\pi R^4}{16} - \frac{4R^4}{9\pi}$, $I_{xy} = \frac{(9\pi - 32)R^4}{72\pi}$	$I_{x'} = \frac{\pi R^4}{16}$, $I_{y'} = \frac{\pi R^4}{16}$, $I_{x'y'} = \frac{R^4}{8}$

MECHANICS OF MATERIALS

Topic	Symbol	Meaning	Equation	Units
axial	σ, sigma	normal stress 	$\sigma_{axial} = N/A$ $\tau_{cutting} = V/A$ $\sigma_{bearing} = F_b/A_b$	psi, Pa
	ε, epsilon	normal strain	$\varepsilon_{axial} = \Delta L/L_o = \delta/L_o$ $\varepsilon_{transverse} = \Delta d/d$	in/in, m/m
	γ, gamma	shear strain	γ = change in angle, $\gamma = c\theta$	rad
	E	Young's modulus, modulus of elasticity	$\sigma = E\varepsilon$ (one-dimensional only)	psi, Pa
	G	shear modulus, modulus of rigidity	$G = \tau/\gamma = E / 2(1+\nu)$	psi, Pa
	ν, nu	Poisson's ratio	$\nu = -\varepsilon'/\varepsilon$	
	δ, delta	deformation, elongation, deflection	$\delta = NL_o/EA + \alpha\Delta T L_o$	in, m
	α, alpha	coefficient of thermal expansion (CTE)		in/inF, m/mC
	F.S.	factor of safety	F.S. = actual strength / design strength	

Topic	Symbol	Meaning	Equation	Units
torsion	τ, tau	shear stress 	$\tau_{torsion} = Tc/J$	psi, Pa
	ϕ, phi	angle of twist	$\phi = TL/GJ$	rad, degrees
	θ, theta	angle of twist per unit length, rate of twist	$\theta = \phi / L$	rad/in, rad/m
	P	power	$P = T\omega$; $r_2T_1 = r_1T_2$, $r_1\omega_1 = r_2\omega_2$	watts = Nm/s hp=6600 in-lb/s
	ω, omega	angular speed, speed of rotation		rad/s
	f	frequency	$\omega = 2\pi f$	Hz = rev/s
	K	stress concentration factor	$\tau_{max} = KTc/J$	psi, Pa
flexure	σ, sigma	normal stress 	$\sigma_{beam} = -My/I$	psi, Pa
	σ, sigma	composite beams, $n = E_B/E_A$	$\sigma_A = -My / I^T$; $\sigma_B = -nMy / I^T$	psi, Pa
	τ, tau	shear stress 	$\tau_{beam} = VQ/Ib$ where $Q = \Sigma(y_{bar\ i} A_i)$	psi, Pa
	q	shear flow	$q = V_{beam}Q/I = nV_{fastener}/s$	
	v or y	beam deflection	$v = \iint M(x)\, dx^2 / EI$	in, m

Topic	Equations		Units
stress trans-formation	*planar rotations* $\sigma_u = (\sigma_x+\sigma_y)/2 + (\sigma_x-\sigma_y)/2 \cos(2\theta) + \tau_{xy}\sin(2\theta)$ $\sigma_v = (\sigma_x+\sigma_y)/2 - (\sigma_x-\sigma_y)/2 \cos(2\theta) - \tau_{xy}\sin(2\theta)$ $\tau_{uv} = -(\sigma_x-\sigma_y)/2 \sin(2\theta) + \tau_{xy}\cos(2\theta)$	*principals and max in-plane shear* $\tan(2\theta_p) = 2\tau_{xy} / (\sigma_x-\sigma_y)$, $\theta_s = \theta_p \pm 45^o$ $\sigma_{1,2} = (\sigma_x+\sigma_y)/2 \pm \text{sqrt}\{ [(\sigma_x-\sigma_y)/2]^2 + \tau_{xy}^2 \}$ $\tau_{max} = \text{sqrt}\{ [(\sigma_x-\sigma_y)/2]^2 + \tau_{xy}^2 \} = (\sigma_1-\sigma_2)/2$ $\sigma_{avg} = (\sigma_x+\sigma_y)/2 = (\sigma_1+\sigma_2)/2$	psi, Pa
strain trans-formation	*planar rotations* $\varepsilon_u = (\varepsilon_x+\varepsilon_y)/2 + (\varepsilon_x-\varepsilon_y)/2 \cos(2\theta) + \gamma_{xy}/2 \sin(2\theta)$ $\varepsilon_v = (\varepsilon_x+\varepsilon_y)/2 - (\varepsilon_x-\varepsilon_y)/2 \cos(2\theta) - \gamma_{xy}/2 \sin(2\theta)$ $\gamma_{uv}/2 = -(\varepsilon_x-\varepsilon_y)/2 \sin(2\theta) + \gamma_{xy}/2 \cos(2\theta)$ $\varepsilon_z = -\nu (\varepsilon_x + \varepsilon_y) / (1-\nu)$	*principals and max in-plane shear* $\tan(2\theta_p) = \gamma_{xy} / (\varepsilon_x-\varepsilon_y)$, $\theta_s = \theta_p \pm 45^o$ $\varepsilon_{1,2} = (\varepsilon_x+\varepsilon_y)/2 \pm \text{sqrt}\{ [(\varepsilon_x-\varepsilon_y)/2]^2 + (\gamma_{xy}/2)^2 \}$ $\gamma_{max}/2 = \text{sqrt}\{ [(\varepsilon_x-\varepsilon_y)/2]^2 + (\gamma_{xy}/2)^2 \}$ $\varepsilon_{avg} = (\varepsilon_x+\varepsilon_y)/2$	psi, Pa in/in, m/m
Hooke's law	*1D strain to stress* $\sigma = E\varepsilon$ *2D strain to stress* $\sigma_x = E(\varepsilon_x+\nu\varepsilon_y) / (1-\nu^2)$ $\sigma_y = E(\varepsilon_y+\nu\varepsilon_x) / (1-\nu^2)$ $\tau_{xy} = G\gamma_{xy} = E\gamma_{xy} / 2(1+\nu)$	*2D stress to strain* $\varepsilon_x = (\sigma_x-\nu\sigma_y) / E$ $\varepsilon_y = (\sigma_y-\nu\sigma_x) / E$ $\varepsilon_z = -\nu(\varepsilon_x+\varepsilon_y) / (1-\nu)$ $\gamma_{xy} = \tau_{xy}/G = 2(1+\nu)\tau_{xy} / E$	psi, Pa in/in, m/m
pressure	$\sigma_{spherical} = pr/2t$ $\sigma_{cylindrical,\ hoop} = pr/t$ $\sigma_{cylindrical,\ axial} = pr/2t$	$\sigma_{radial,\ outside} = 0$ $\sigma_{radial,\ inside} = -p$	psi, Pa
failure theories	*maximum principal stress theory* $\sigma_{1,2} < \sigma_{yp}$	*maximum shear stress theory* $\tau_{max} < 0.5\ \sigma_{yp}$	psi, Pa