Engineering Mechanics

Visual Quick-Reference Cheat Sheet

Structures • Fluids • Gas Dynamics • Rockets • Heat Transfer • Orbits • Vibrations • Math

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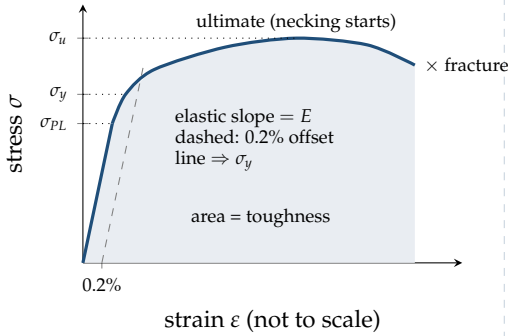
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How to read this sheet. Each topic is a card: the **sketch** on the left shows the geometry and loads, the **equations** on the right are what you plug into. boxed equations are the ones you will use most. *Italic gray* lines define the symbols. **Tip:** lines give rules of thumb and common pitfalls.

Units: SI unless noted. In US customary with lbm, divide mass by $g_c = 386.1$ in/s^2 (or 32.17 ft/s^2) to get consistent units.

Structural Mechanics

Stress–Strain Basics



$\sigma = E\varepsilon$ $\quad \varepsilon = \dfrac{\Delta L}{L}$ $\quad \nu = -\dfrac{\varepsilon_{lat}}{\varepsilon_{long}}$ $\quad G = \dfrac{E}{2(1+\nu)}$

E = Young's modulus, ν = Poisson's ratio (≈ 0.3 metals), G = shear modulus

True vs. engineering (valid up to necking)
$\sigma_{true} = \sigma_{eng}(1+\varepsilon_{eng}) \qquad \varepsilon_{true} = \ln(1+\varepsilon_{eng})$

Thermal expansion: $\varepsilon = \alpha\Delta T, \quad \Delta L = L\alpha\Delta T$

Toughness (energy/volume to fracture): $\displaystyle\int_0^{\varepsilon_f} \sigma\, d\varepsilon$

Resilience (elastic energy/vol.): $\dfrac{\sigma_y^2}{2E}$

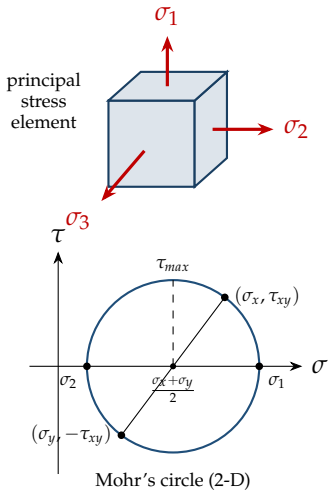
Tip: Yield strength σ_y is defined by the 0.2% offset line, drawn parallel to the elastic slope starting at $\varepsilon = 0.002$.

Typical Material Properties (room temperature)

Material	E [GPa]	σ_y [MPa]	σ_u [MPa]	ρ [kg/m^3]	ν	α [μ/K]	k [W/m·K]
Al 6061-T6	68.9	276	310	2700	0.33	23.6	167
Al 7075-T6	71.7	503	572	2810	0.33	23.4	130
Ti-6Al-4V (annealed)	114	880	950	4430	0.34	8.6	6.7
4130 steel (normalized)	205	435	670	7850	0.29	12.2	42.7
304 stainless (annealed)	193	215	505	8000	0.29	17.3	16.2
Inconel 718 (aged)	200	1030	1240	8190	0.29	13.0	11.4
Copper C101 (annealed)	115	69	220	8940	0.34	17.0	391

1 ksi = 6.895 MPa 1 Msi = 6.895 GPa Values are typical; always use an approved source (e.g. MMPDS) for design.

Stress State, Principal Stresses & Von Mises



Principal stresses (2-D):

$$\sigma_{1,2} = \frac{\sigma_x+\sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + \tau_{xy}^2} \qquad \tau_{max} = \frac{\sigma_1-\sigma_2}{2}$$

Von Mises (ductile yield):

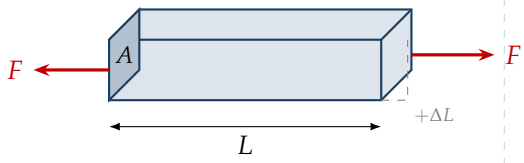
$$\boxed{\sigma_{VM} = \sqrt{\frac{(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2}{2}}}$$

Plane stress: $\sigma_{VM} = \sqrt{\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2}$

Tresca (conservative): $\tau_{max} = \frac{1}{2}(\sigma_{max} - \sigma_{min}) \le \frac{1}{2}\sigma_y$

Tip: Yield when $\sigma_{VM} \ge \sigma_y$. For brittle materials (elongation $< 3\%$) use max principal stress instead.

Axial Load (Bar in Tension)



$$\sigma = \frac{F}{A} \qquad \Delta L = \frac{FL}{EA} \qquad k = \frac{EA}{L}$$

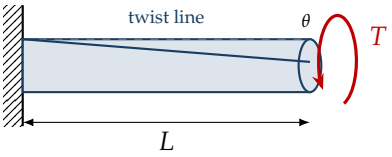
Lateral contraction: $\Delta S = -\nu \frac{F\,S}{EA}$

S = lateral dimension

Strain energy: $U = \frac{F^2 L}{2EA}$

Tip: Bars in series add like springs: $1/k_{tot} = \sum 1/k_i$; in parallel $k_{tot} = \sum k_i$.

Torsion of Shafts & Tubes



$$\tau = \frac{T\,r}{J} \qquad \theta = \frac{TL}{GJ} \qquad k_t = \frac{GJ}{L} \qquad \text{Power } P = T\omega$$

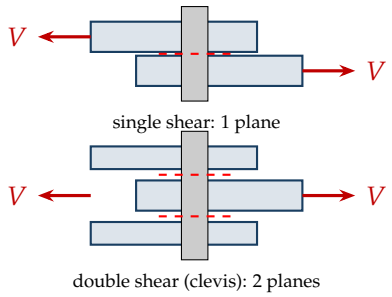
J = polar moment of inertia (see Area Moments card); r = radius to point of interest; θ in radians

Thin-walled closed tube (Bredt): $\tau = \frac{T}{2A_m t}$

A_m = area enclosed by wall mid-line

Tip: Max shear stress is at the outer surface. Open sections (slit tubes, I-beams) are dramatically weaker in torsion.

Pins in Shear (Single & Double)



$$\tau_{avg} = \frac{V}{A_{pin}} \qquad A_{pin} = \frac{\pi d^2}{4}$$

V = shear force carried by one shear plane

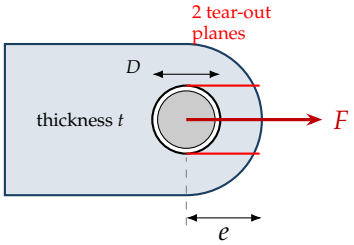
Double shear: each plane carries $V/2 \Rightarrow \tau = \frac{V}{2A_{pin}}$

Peak transverse shear (beam-type cross section):

Solid circle	$\tau_{max} = \frac{4}{3} V/A$
Thin hollow circle	$\tau_{max} = 2\,V/A$
Rectangle / square	$\tau_{max} = \frac{3}{2} V/A$

Tip: Ultimate shear strength $\tau_u \approx 0.6\,\sigma_u$ for most metals (von Mises gives $\sigma/\sqrt{3} = 0.577\,\sigma$).

Lug Bearing Stress & Shear Tear-Out



Bearing (contact on hole): $\sigma_{brg} = \frac{F}{Dt}$

single lug; divide F by 2 for each ear of a dual clevis

Shear tear-out: $\tau = \frac{F}{2et}$

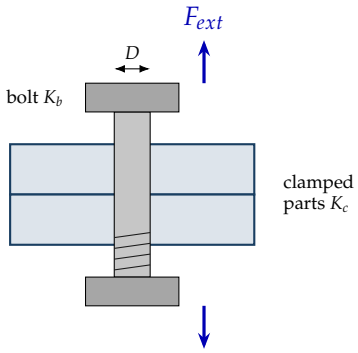
2et = total shear plane area; e = hole center to edge

Net-section tension: $\sigma = \frac{F}{(W - D)\,t}$

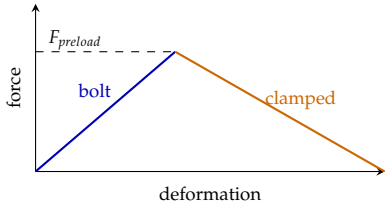
W = lug width

Tip: Allowable bearing σ_{bru} is a function of e/D. Keep $e/D \geq 1.5$ (2.0 preferred) to avoid tear-out governing.

Bolted Joints



joint diagram



Preload from torque: $F_{preload} = \dfrac{T}{KD}$

T = torque, D = nominal (shank) diameter, K = nut factor: 0.10–0.15 *lubed,* 0.15–0.20 *dry*

Bolt tensile stress (valid while $F_{ext} < F_{preload}$, joint not gapped):

$$\sigma = \frac{F_{preload} + n\,\phi\,F_{ext}}{A_t} \qquad \phi = \frac{K_b}{K_b + K_c} \approx 0.2$$

n = load introduction factor (≈ 0.5); ϕ = joint stiffness factor

Tensile stress area (UN threads, n_t threads/in):

$$A_t = \frac{\pi}{4}\left(D - \frac{0.9743}{n_t}\right)^2 \quad \text{or} \quad A_t = \frac{F_{bolt,ult}}{\sigma_{ult}}$$

Thread pull-out (shear of internal threads):

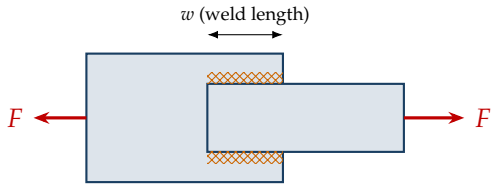
$$F_{shear,ult} = 0.4\,\pi D\,L_{eng}\,\tau_u$$

0.4: 40% of threads are effectively engaged (conservative rule of thumb; detailed methods use the thread shear area, roughly $0.5\text{–}0.75\,\pi D L_{eng}$*); L_{eng} = thread engagement length; τ_u from MMPDS, else $\sigma_u/\sqrt{3}$*

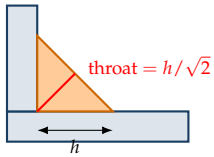
Gapping load: $F_{gap} = \dfrac{F_{preload}}{1 - n\phi}$

Tip: Typical preload target: 65–75% of bolt proof load. Use $L_{eng} \geq 1.5D$ in aluminum, $\geq 1D$ in steel.

Fillet Welds



top view: 2 parallel fillets



cross section

Shear on throat (two parallel fillets):

$$\tau \approx \frac{F/2}{\dfrac{h}{\sqrt{2}}\,w} = \frac{F}{\sqrt{2}\,h\,w}$$

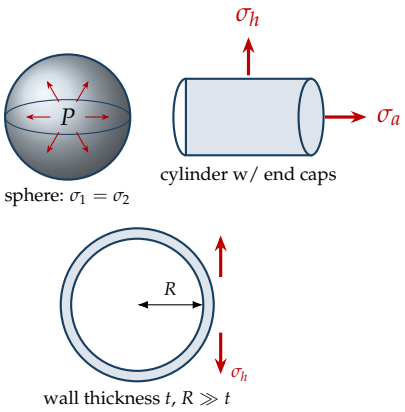
h = weld leg size, w = weld length; $h/\sqrt{2}$ = minimum (throat) shear thickness

General: $\tau = \dfrac{F}{0.707\,h\,L_{total}}$

Compare τ against τ_u of the weld/base metal **with a weld knock-down factor** (steel ≈ 0.8; aluminum heat-affected zone often loses 40–50% of temper strength).

Tip: Assumes $t_{weld} \ll L_{geom}$ and uniform shear along the weld. Long welds and eccentric loads need a moment/torsion analysis of the weld group.

Thin-Walled Pressure Vessels



Sphere $\sigma_{hoop} = \sigma_1 = \sigma_2 = \dfrac{PR}{2t}$ $\quad \sigma_{radial} = -P\ (\approx 0)$

Radial growth $\quad \Delta R = \dfrac{PR^2(1-\nu)}{2Et}$

Hoop strain $\quad \varepsilon_{hoop} = \dfrac{PR(1-\nu)}{2Et}$

Thermal growth $\quad \Delta R = R\,\alpha\,\Delta T$

Stored strain energy $\quad U = \dfrac{\pi P^2 R^4 (1-\nu)}{Et}$

Cylinder with end caps (thin wall, $R/t > 10$):

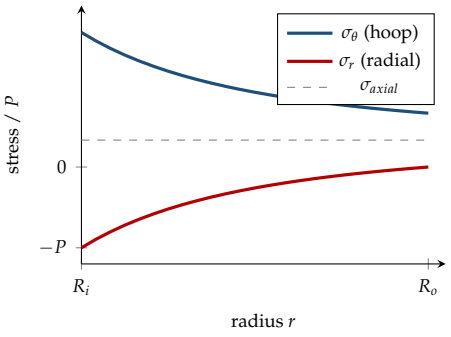
$\sigma_{hoop} = \sigma_1 = \dfrac{PR}{t}$ $\quad \sigma_{axial} = \sigma_2 = \dfrac{PR}{2t}$ $\quad \sigma_3 = -P$

$\Delta L = \dfrac{PRL}{Et}\left(\frac{1}{2} - \nu\right)$ $\quad \Delta R = \dfrac{PR^2}{Et}\left(1 - \frac{\nu}{2}\right)$

The ν terms are Poisson contraction from the other in-plane stress.

Tip: Hoop stress in a cylinder is twice the axial stress, which is why pipes split lengthwise.

Thick-Walled Cylinders (Lamé)



Shown for $R_o = 2R_i$, internal pressure only

Internal pressure P, no external pressure:

$$\sigma_\theta(r) = \frac{PR_i^2}{R_o^2 - R_i^2}\left(1 + \frac{R_o^2}{r^2}\right) \qquad \sigma_r(r) = \frac{PR_i^2}{R_o^2 - R_i^2}\left(1 - \frac{R_o^2}{r^2}\right)$$

Maximum hoop stress (at inner wall):

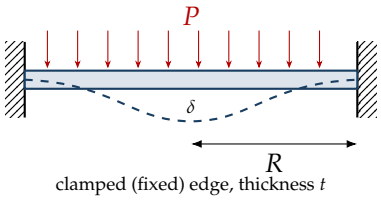
$$\boxed{\sigma_{\theta,max} = P\,\frac{R_o^2 + R_i^2}{R_o^2 - R_i^2}}$$

With end caps: $\sigma_{axial} = \dfrac{PR_i^2}{R_o^2 - R_i^2}$

Thin-wall limit ($R_{mean}/t > 10$): $\sigma_\theta \approx \dfrac{PR_i}{t}$

Tip: Peak stress is always at the bore. Adding wall thickness gives diminishing returns: hoop stress never drops below P.

Pressurized Circular Plates

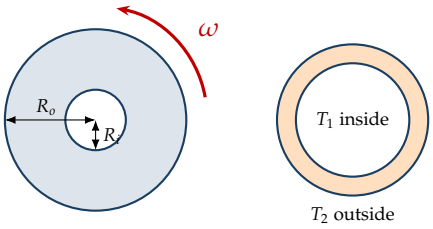


	Center deflection	Max stress
Clamped edge	$\delta = \dfrac{3}{16}\dfrac{PR^4(1-\nu^2)}{Et^3}$	$\sigma = \dfrac{3}{4}P\left(\dfrac{R}{t}\right)^2$
Simply supported	$\delta = \dfrac{3(5+\nu)(1-\nu)PR^4}{16Et^3}$	$\sigma = \dfrac{3(3+\nu)}{8}P\left(\dfrac{R}{t}\right)^2$

Max stress is at the edge for clamped plates and at the center for simply supported plates.

Tip: Valid for small deflection ($\delta < t/2$). Larger deflections stiffen the plate through membrane action.

Rotating Disks & Thermal Stress



Rotating hollow shaft/disk – outer radial growth:

$$\Delta R_o = \frac{\rho\,\omega^2 R_o}{4E}\left[(1-\nu)R_o^2 + (3+\nu)R_i^2\right]$$

Thin-walled cylinder, wall gradient $\Delta T = T_1 - T_2$ (unconstrained):

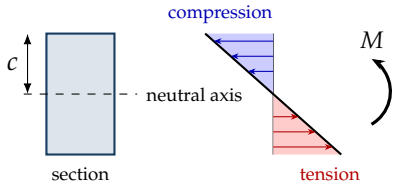
$$\sigma_\theta = \frac{\alpha E \Delta T}{2(1-\nu)}$$

Fully constrained thermal stress:

Bar (1-D): $\sigma = -E\alpha\Delta T$ Plate (2-D, biaxial): $\sigma = -\dfrac{E\alpha\Delta T}{1-\nu}$

Tip: Free thermal expansion causes no stress; stress only appears when expansion is restrained or there is a temperature gradient.

Beam Bending Stress



$$\sigma_B = \frac{Mc}{I} = \frac{M}{S} \qquad S = \frac{I}{c}\ \text{(section modulus)}$$

M = bending moment, c = distance from neutral axis to outer fiber (radius or half-thickness), I = area moment of inertia

Transverse shear stress: $\tau = \dfrac{VQ}{It}$

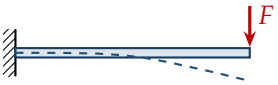
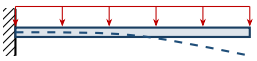
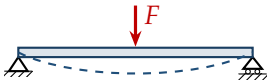
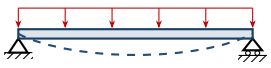
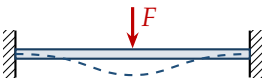
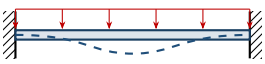
Q = first moment of area above the cut, t = width at the cut

Curvature: $\dfrac{1}{\rho} = \dfrac{M}{EI} = \dfrac{d^2y}{dx^2}$ **Parallel axis:** $I = I_c + Ad^2$

Tip: Stress is zero at the neutral axis and maximum at the outer fibers. Doubling depth h gives 8× stiffness and 4× strength for a rectangle.

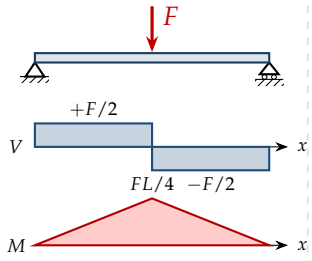
Beam Deflection & Moment Formulas

F = total load. For distributed cases, w = load per length and F = wL. δ = max deflection, M_{max} = max bending moment. Stiffness $K = F/\delta$.

Sketch	Case	δ_{max}	M_{max}	Notes
	Cantilever, end load	$\dfrac{FL^3}{3EI}$	FL	at wall; $K = \dfrac{3EI}{L^3}$
	Cantilever, uniform load	$\dfrac{FL^3}{8EI}$	$\dfrac{FL}{2}$	at wall
	Simply supported, center load	$\dfrac{FL^3}{48EI}$	$\dfrac{FL}{4}$	at center
	Simply supported, uniform load	$\dfrac{5}{384}\dfrac{FL^3}{EI}$	$\dfrac{FL}{8}$	at center
	Fixed–fixed, center load	$\dfrac{FL^3}{192EI}$	$\dfrac{FL}{8}$	at center & ends
	Fixed–fixed, uniform load	$\dfrac{FL^3}{384EI}$	$\dfrac{FL}{12}$	at ends ($FL/24$ center)
	Cantilever, end moment M_0	$\dfrac{M_0L^2}{2EI}$	M_0	constant along beam

Tip: Superposition: deflections from separate loads simply add (linear elastic). Fixing both ends cuts deflection 4× vs. simply supported (center load).

Shear & Moment Diagrams (example)



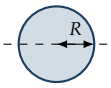
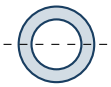
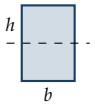
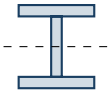
Rules for drawing them:

$$\frac{dV}{dx} = -w(x) \qquad \frac{dM}{dx} = V(x)$$

- Point load ⇒ jump in V; M has a kink.
- Uniform load ⇒ V linear, M parabolic.
- M is maximum where V crosses zero.
- Free and pinned ends have $M = 0$ (unless a moment is applied there); an unloaded free end also has $V = 0$.
- Area under the V diagram between two points = change in M.

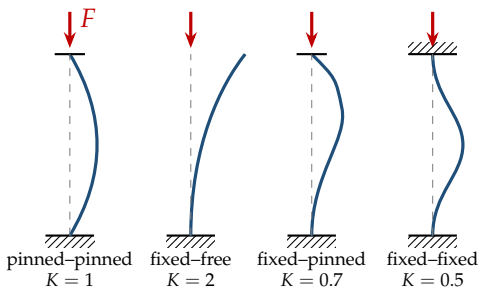
Tip: Always start with a free body diagram and solve the support reactions ($\Sigma F = 0$, $\Sigma M = 0$) before drawing.

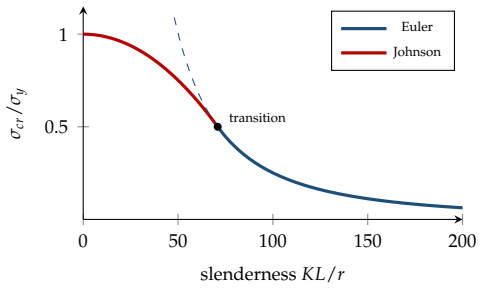
Area & Polar Moments of Inertia

Shape		I (bending)	J (torsion)	Area
	Solid circle	$\frac{\pi}{4}R^4 = \frac{\pi}{64}D^4$	$\frac{\pi}{2}R^4 = \frac{\pi}{32}D^4$	πR^2
	Hollow circle	$\frac{\pi}{4}(R_o^4 - R_i^4)$	$\frac{\pi}{2}(R_o^4 - R_i^4)$	$\pi(R_o^2 - R_i^2)$
	Thin-walled tube (radius R, wall t)	$\approx \pi R^3 t$	$\approx 2\pi R^3 t$	$\approx 2\pi R t$
	Rectangle	$\frac{bh^3}{12}$	$\frac{bh(b^2+h^2)}{12}$ (polar; true torsion constant is lower)	bh
	Triangle (about centroid)	$\frac{bh^3}{36}$	–	$\frac{1}{2}bh$
	I-beam (flanges $b \times t_f$, height h)	$\approx \frac{b\,t_f\,h^2}{2} + \frac{t_w h^3}{12}$	weak in torsion	$2bt_f + t_w h$

Radius of gyration $r = \sqrt{I/A}$ (used in buckling slenderness L/r). Parallel axis theorem: $I = I_c + Ad^2$.

Column Buckling (Euler & Johnson)





Euler buckling (long, slender columns):

$$F_{cr} = \frac{n\,\pi^2 EI}{L^2} = \frac{\pi^2 EI}{(KL)^2} \qquad n = \frac{1}{K^2}$$

End condition	K (eff. length)	n
pinned–pinned	1.0	1
fixed–free (flagpole)	2.0	0.25
fixed–pinned	0.7	2 (2.05)
fixed–fixed	0.5	4

Design practice: real "fixed" ends are never perfectly rigid, so use $n = 1$ for fixed–pinned and fixed–fixed unless rotation is truly prevented.

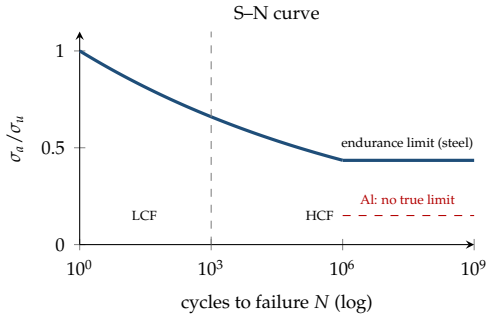
Johnson parabola (short/intermediate columns, yield + buckling):

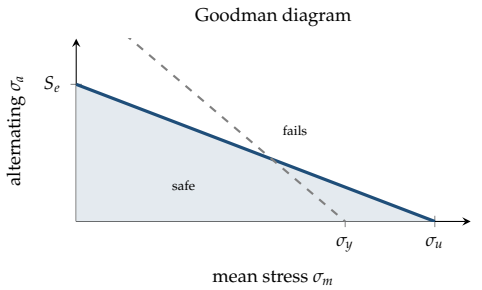
$$\sigma_{cr} = \sigma_y - \frac{\sigma_y^2}{4\pi^2 E}\left(\frac{KL}{r}\right)^2 \quad \text{for} \quad \frac{KL}{r} < \sqrt{\frac{2\pi^2 E}{\sigma_y}}$$

Alternative: Engesser (tangent modulus) method, replace E with E_t from the stress–strain curve.

Tip: Euler applies for $L/R > 10$ and slenderness above the transition point. Buckling depends on E and I, not strength, so a stronger alloy doesn't help a slender column.

Fatigue





LCF = low-cycle fatigue ($N \lesssim 10^3$, part yields) **HCF** = high-cycle fatigue (elastic only)

Coffin–Manson / Manson universal slopes (total strain range):

$$\Delta\varepsilon \approx 3.5\,\frac{\sigma_u}{E}N^{-0.12} + \left(\frac{\ln\left(\frac{1}{1-RA}\right)}{N}\right)^{0.6}$$

RA = reduction in area (fraction); 1st term = elastic, 2nd = plastic

Basquin (HCF): $\sigma_a = \sigma_f'\,(2N)^b$, $b \approx -0.05$ to -0.12

Stress terms: $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$, $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$, $R = \dfrac{\sigma_{min}}{\sigma_{max}}$

Goodman (mean stress): $\dfrac{\sigma_a}{S_e} + \dfrac{\sigma_m}{\sigma_u} = \dfrac{1}{FS}$

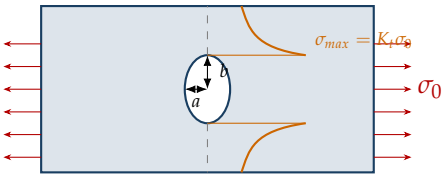
Miner's rule (damage): $D = \sum \dfrac{n_i}{N_i} < 1$

n_i = cycles applied at level i, N_i = cycles to failure at that level

Endurance limit (steel, rough): $S_e \approx 0.5\,\sigma_u$ (polished specimen)

Tip: Correct S–N data for surface finish, size, temperature and stress concentration. Include vibration cycles; they dominate the count.

Stress Concentrations



$$\sigma_{max} = K_t\,\sigma_0$$

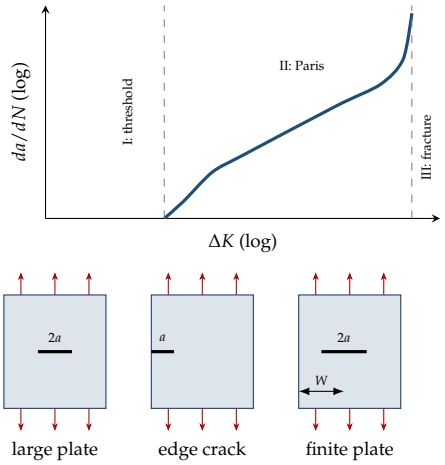
Elliptical hole, infinite plate: $K_t = 1 + \dfrac{2b}{a}$

b = semi-axis perpendicular to load, a = semi-axis along load

Circular hole, infinite plate	$K_t = 3$
Shoulder fillet, $r/d \approx 0.1$	$K_t \approx 1.7$ (bending)
Sharp crack ($a \to 0$)	$K_t \to \infty$ (use fracture mechanics)

Tip: Ductile parts under static load usually redistribute around K_t; for fatigue and brittle materials it must always be included.

Fracture Mechanics



Stress intensity factor (Mode I): $K_I = Y\sigma\sqrt{\pi a}$ Fracture if $K_I \geq K_{IC}$

Center crack $2a$, large plate	$K_I = \sigma\sqrt{\pi a}$
Edge crack a, large plate	$K_I = 1.12\,\sigma\sqrt{\pi a}$
Center crack, finite plate (half-width W)	$K_I = \sigma\sqrt{\dfrac{\pi a}{\cos\left(\frac{\pi a}{2W}\right)}}$

Critical crack length: $a_c = \dfrac{1}{\pi}\left(\dfrac{K_{IC}}{Y\sigma}\right)^2$

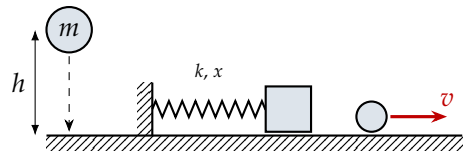
Paris law (region II): $\dfrac{da}{dN} = C\,\Delta K^m$

$$N = \int_{a_{init}}^{a_{crit}} \frac{da}{C\left(Y\Delta\sigma\sqrt{\pi a}\right)^m}$$

No crack growth if $\Delta K < \Delta K_{threshold}$. Typical $m \approx 3$–4 for metals.

Tip: Typical K_{IC}: Al 7075 ≈ 25, Ti-6Al-4V ≈ 55, steels 50–200 MPa$\sqrt{\text{m}}$.

Energy Relations



Work	$W = \int F\,dx$	Kinetic	$\frac{1}{2}mv^2$
Potential	mgh	Spring	$\frac{1}{2}kx^2$
Heat	$mc_p\Delta T$	Electrical	VIt
Rotational KE	$\frac{1}{2}I\omega^2$	Efficiency	$\eta = \dfrac{E_{out}}{E_{in}}$

Tip: Energy methods are the fastest way to estimate impact loads: set $\frac{1}{2}mv^2 = \frac{1}{2}k\delta^2$ to get δ, then $F = k\delta$.

Margin of Safety & Factors of Safety

$MS > 0$: positive margin
$MS = 0$: exactly meets FoS
$MS < 0$: fails requirement

$$MS = \frac{\text{Allowable Load}}{\text{Applied Load} \times FoS} - 1$$

Common spacecraft factors of safety:

Yield	1.1	Ultimate	1.25–1.4
Thermal	1.0	Joint gapping	1.2–1.4
Fittings	1.15	Castings	1.25
Fatigue	1.15	Service life factor	4

Tip: Human-rated hardware typically requires ultimate FoS ≥ 1.4. Always check yield and ultimate margins separately.

"Any altitude is attainable with the right attitude."

Fluid Mechanics

Fluid Properties & Hydrostatics

Fluid (20 °C, 1 atm)	ρ [kg/m^3]	μ [Pa·s]	$\nu = \mu/\rho$ [m^2/s]
Water	998	1.0×10^{-3}	1.0×10^{-6}
Air	1.20	1.8×10^{-5}	1.5×10^{-5}
Liquid oxygen (−183 °C)	1141	1.9×10^{-4}	1.7×10^{-7}
RP-1 / kerosene	810	1.6×10^{-3}	2.0×10^{-6}

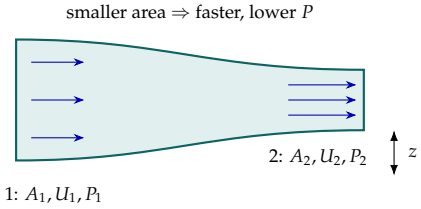
Hydrostatic pressure: $P = P_0 + \rho g h$

Buoyancy: $F_B = \rho_{fluid}\, g\, V_{displaced}$

μ = dynamic viscosity, ν = kinematic viscosity, U = velocity

1 atm = 101.325 kPa = 14.696 psi
1 bar = 100 kPa 1 psi = 6.895 kPa

Conservation Laws (Continuity, Bernoulli, Energy)



Continuity (mass): $\rho A U = \text{const} = \dot{m}$ incompressible: $A_1 U_1 = A_2 U_2$

Bernoulli (momentum along a streamline):

$$\frac{P}{\rho} + \frac{U^2}{2} + gz = \text{const}$$

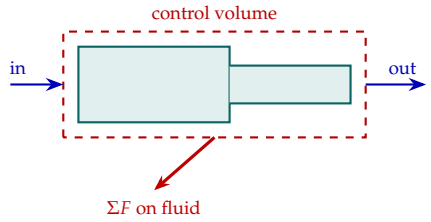
Inviscid, incompressible, steady, along a streamline. g = gravitational (or vehicle) acceleration, z = height.

Steady-flow energy (adiabatic, no shaft work):

$$c_p(T_2 - T_1) + \frac{U_2^2 - U_1^2}{2} + g(z_2 - z_1) = 0$$

Tip: Bernoulli is only valid between points with no pumps, turbines or friction losses between them. Add head-loss terms otherwise.

Control Volume Momentum (forces on pipes, nozzles, bends)



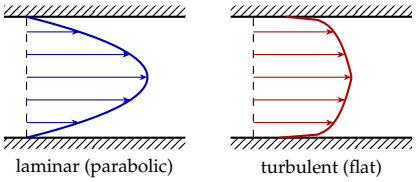
$$\boxed{\sum F = (\dot{m}_e V_e - \dot{m}_i V_i) + (P_e A_e - P_i A_i)}$$

ΣF = force exerted by the walls/hardware on the fluid. Vector equation: apply separately in each direction. Use gauge pressures if the outside is at ambient.

Hydraulic diameter (non-circular ducts): $D_h = \dfrac{4A}{P_{wetted}}$

Tip: The reaction force on the hardware is equal and opposite to ΣF on the fluid. This gives thrust, nozzle blow-off loads and elbow loads.

Reynolds Number & Pipe Velocity Profiles



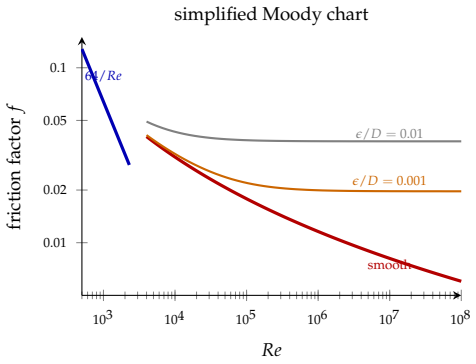
$$\boxed{Re = \frac{UD}{\nu} = \frac{\rho U D}{\mu} = \frac{4\dot{m}}{\pi D \mu}}$$

$Re < 2300$	laminar (smooth, orderly)
$2300 < Re < 4000$	transitional
$Re > 4000$	turbulent (mixing, higher losses)

Wall shear: $\tau(r) = \mu \dfrac{du}{dr}$ Laminar: $U_{max} = 2\,U_{avg}$

Tip: Re is the ratio of inertial to viscous forces. Use D_h for non-circular ducts.

Pressure Loss in Pipes (Darcy–Weisbach)



$$\boxed{\Delta P = f \frac{L}{D} \frac{\rho U^2}{2}}$$

f = Darcy friction loss factor, L/D = length over diameter ratio

Laminar: $f = \dfrac{64}{Re_D}$

Turbulent (Haaland, within 2% of Colebrook):

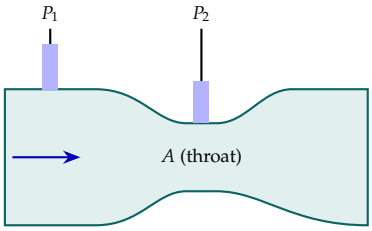
$$\frac{1}{\sqrt{f}} = -1.8 \log_{10}\left[\frac{6.9}{Re} + \left(\frac{\epsilon/D}{3.7}\right)^{1.11}\right]$$

ϵ = wall roughness: drawn tubing $\approx 1.5\,\mu m$, commercial steel $\approx 45\,\mu m$

Minor losses (fittings): $\Delta P = K \dfrac{\rho U^2}{2}$

90° elbow $K \approx 0.3$–0.9; open ball valve ≈ 0.05; sharp entrance 0.5; exit 1.0

Orifice & Venturi Flow Meters



Incompressible flow through a restriction:

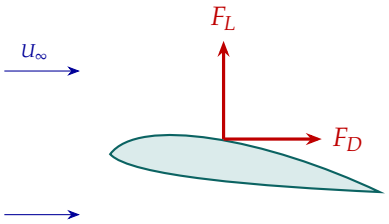
$$\boxed{\dot{m} = C_D\, A \sqrt{2\rho\, \Delta P}}$$

$\Delta P = P_1 - P_2$, A = throat or orifice area, C_D = discharge coefficient. If the throat is not small vs. the pipe, divide by $\sqrt{1-\beta^4}$ with $\beta = d_{throat}/D_{pipe}$ (approach-velocity correction).

Sharp orifice $C_D \approx 0.6$–0.65; venturi $C_D \approx 0.95$–0.99; rounded nozzle ≈ 0.97

Tip: For gases with $\Delta P/P_1 >\sim 10\%$, use the compressible (choked) relations in Gas Dynamics.

Lift & Drag



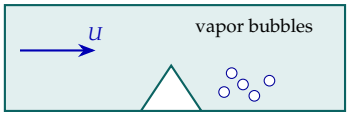
$$F_L = \tfrac{1}{2} C_L \rho A U^2 \qquad F_D = \tfrac{1}{2} C_D \rho A U^2 \qquad q = \tfrac{1}{2}\rho U^2$$

q = dynamic pressure; A = reference area (frontal for bluff bodies, planform for wings)

Typical C_D (high Re):

Flat plate (normal)	1.28	Sphere	0.47
Long cylinder	1.2	Car	0.25–0.35
Cube	1.05	Streamlined body	0.04

Cavitation Check



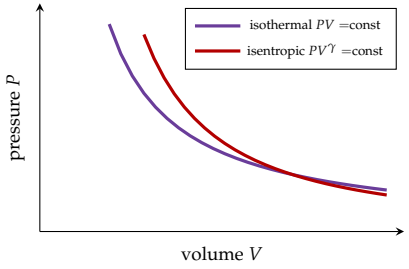
$$K = \frac{2\,\Delta P}{\rho U^2}$$ cavitation is initiated for $K \in [0.2, 1.5]$

$\Delta P = P_{local} - P_{vapor}$. Lower $K \Rightarrow$ more likely to cavitate.

Tip: Cavitation happens where local pressure drops below vapor pressure: pump inlets, valve seats, sharp restrictions, venturi throats.

Gas Dynamics

Ideal Gas & Process Approximations



$$P = \rho R T \qquad R = \frac{R_u}{MW} \qquad R_u = 8.314\ \frac{\text{J}}{\text{mol}\cdot\text{K}}$$

$R(air)$ = 287 J/kg·K; MW = molecular weight [kg/mol]

Ideal gas Good for a first cut. Use a compressibility factor Z ($P = Z\rho RT$) when $T < -100\,^\circ$F and $P > 1000$ psia.

Adiabatic No heat flow (insulated).

Isentropic Adiabatic + reversible (insulated, no viscous losses).

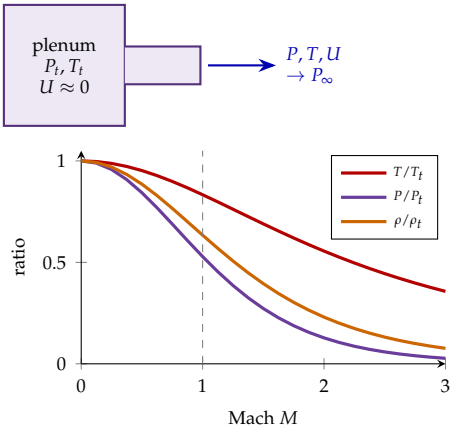
Isentropic relations:

$$\frac{P_2}{P_1} = \left(\frac{\rho_2}{\rho_1}\right)^\gamma = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} \qquad \gamma = \frac{c_p}{c_v}, \quad c_p - c_v = R$$

Ratio of Specific Heats γ

Gas type	γ	Examples
Monatomic	$5/3 = 1.67$	He, Ne, Ar, Xe
Diatomic	$7/5 = 1.40$	N_2, O_2, H_2, air
Combustion gases	1.1–1.25	rocket and jet exhaust

Mach Number & Stagnation (Total) Properties



Plotted for $\gamma = 1.4$

Mach number: $M = \dfrac{U}{a} = \dfrac{U}{\sqrt{\gamma R T}}$ a = speed of sound

$M < 0.3$: incompressible OK • $M < 1$ subsonic • $M = 1$ sonic • $M > 1$ supersonic

Total pressure (incompressible): $P_t = P + \frac{1}{2}\rho U^2$

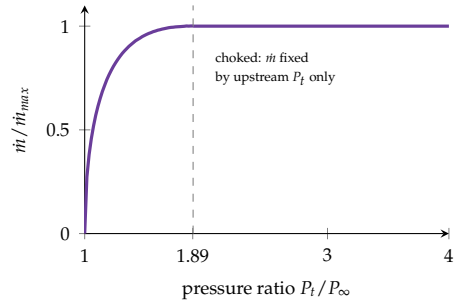
Stagnation relations (compressible, isentropic):

$$\frac{T_t}{T} = 1 + \frac{\gamma - 1}{2} M^2$$

$$\frac{P_t}{P} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{\gamma}{\gamma - 1}} \qquad \frac{\rho_t}{\rho} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{1}{\gamma - 1}}$$

Tip: Total (stagnation) conditions are what a gas reaches if brought to rest isentropically. A pitot probe reads P_t.

Choked Flow & Mass Flow Rate



Air, $\gamma = 1.4$

Flow is choked ($M = 1$ at the throat) when:

$$\frac{P_t}{P_\infty} \geq \left(\frac{\gamma + 1}{2}\right)^{\frac{\gamma}{\gamma - 1}}$$

≈ 1.89 air, ≈ 1.7 combustion gas

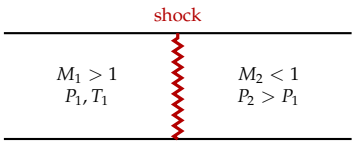
Choked mass flow:

$$\dot{m} = \frac{A^* P_t}{\sqrt{T_t}} \sqrt{\frac{\gamma}{R}} \left(\frac{\gamma + 1}{2}\right)^{-\frac{\gamma + 1}{2(\gamma - 1)}}$$

A^* = throat area; for air: $\dot{m} = 0.0404\, A^* P_t / \sqrt{T_t}$ (SI units)

Tip: Once choked, lowering the downstream pressure no longer increases flow. Only raising P_t or A^* does.

Normal Shocks

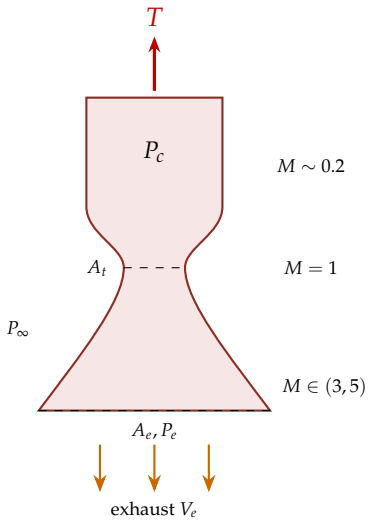


$$M_2^2 = \frac{1 + \frac{\gamma - 1}{2} M_1^2}{\gamma M_1^2 - \frac{\gamma - 1}{2}} \qquad \frac{P_2}{P_1} = 1 + \frac{2\gamma}{\gamma + 1}\left(M_1^2 - 1\right)$$

Tip: Total temperature is conserved across a shock, but total pressure drops (entropy rises).

Rocket Mechanics

Nozzle Flow & Thrust



Thrust:

$$\boxed{T = \dot{m} V_e + (P_e - P_\infty) A_e} = C_F \, P_c \, A_t = I_{sp} \, \dot{m} \, g_0$$

C_F = thrust coefficient, I_{sp} = specific impulse [s], $g_0 = 9.81$ m/s^2

Vacuum thrust coefficient:

$$C_{F,vac} = \sqrt{\frac{2\gamma^2}{\gamma - 1} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}} \left[1 - \left(\frac{P_e}{P_c} \right)^{\frac{\gamma-1}{\gamma}} \right]} + \frac{P_e}{P_c} \frac{A_e}{A_t}$$

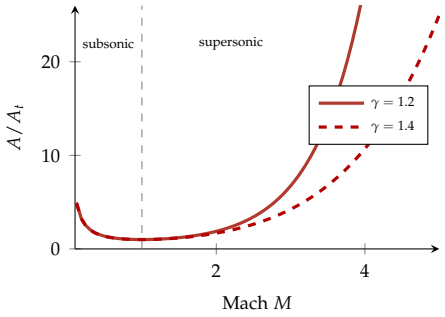
The square root alone is the momentum term (sea-level C_F subtracts $P_\infty A_e / P_c A_t$).

Characteristic velocity: $c^* = \dfrac{P_c A_t}{\dot{m}}$ $\qquad I_{sp} = \dfrac{c^* C_F}{g_0}$

Exhaust velocity (ideal): $V_e = \sqrt{\dfrac{2\gamma}{\gamma - 1} \dfrac{R_u T_c}{MW} \left[1 - \left(\dfrac{P_e}{P_c} \right)^{\frac{\gamma-1}{\gamma}} \right]}$

Tip: For given chamber conditions, thrust is maximized by the expansion ratio that gives $P_e = P_\infty$ (ideally expanded). Hot, light exhaust (high T_c, low MW) gives high I_{sp}.

Area–Mach Relation



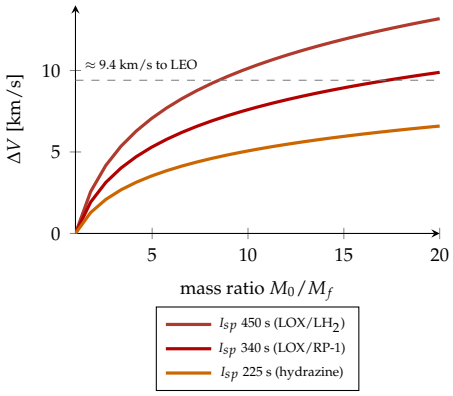
Use to estimate pressure as a function of axial distance:

$$\boxed{\frac{A_t}{A} = \left(\frac{\gamma + 1}{2} \right)^{\frac{\gamma+1}{2(\gamma-1)}} \frac{M}{\left(1 + \frac{\gamma-1}{2} M^2 \right)^{\frac{\gamma+1}{2(\gamma-1)}}}}$$

Procedure: local area $A(x) \rightarrow$ solve for M (two roots: pick subsonic before throat, supersonic after) $\rightarrow P/P_c$ from stagnation relations (Gas Dynamics).

Tip: Converging section accelerates subsonic flow; diverging section accelerates supersonic flow. Expansion ratio A_e/A_t sets exit Mach.

Rocket Equation

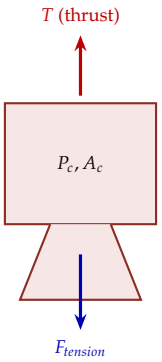


$$\Delta V = V_e \ln\left(\frac{M_{initial}}{M_{final}}\right) \qquad V_e = I_{sp}\, g_0$$

Propellant mass needed: $M_{prop} = M_f\left(e^{\Delta V/V_e} - 1\right)$

Propulsion type	Vacuum I_{sp} [s]
Cold gas (N_2)	60–75
Monopropellant (hydrazine)	220–235
Solid motor	260–300
LOX / RP-1	330–350
LOX / CH_4	350–380
LOX / LH_2	440–465
Electric (Hall / ion)	1500–4000

Chamber Tension (thrust structure FBD)



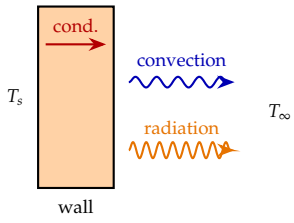
$$F_{tension} = P_c A_c - T$$

Axial load carried by the chamber wall/nozzle joint. A_c = chamber cross-section area.

Tip: The pressure on the injector face pushes forward with $P_c A_c$; the nozzle walls pull back. The difference is the joint tension that flanges and bolts must carry.

Heat Transfer

Three Modes of Heat Transfer



Conduction (Fourier) $\dot{Q} = -kA\dfrac{\Delta T}{\Delta x}$

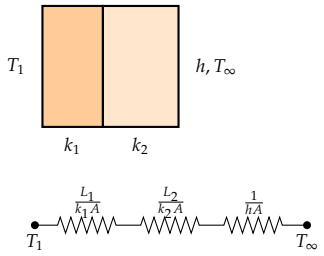
Convection (Newton) $\dot{Q} = hA\,(T_s - T_\infty)$

Radiation (Stefan–Boltzmann) $\dot{Q} = F_{12}\,\varepsilon\,\sigma A\left(T_s^4 - T_\infty^4\right)$

$\sigma = 5.67 \times 10^{-8}$ W/m^{2}K^4; temperatures in Kelvin for radiation; F_{12} = view factor; ε = emissivity

Energy balance: $\dot{Q}_{in} + \dot{Q}_{generated} - \dot{Q}_{out} = \dot{Q}_{stored} = mc_p\dfrac{dT}{dt}$

Thermal Resistance Networks

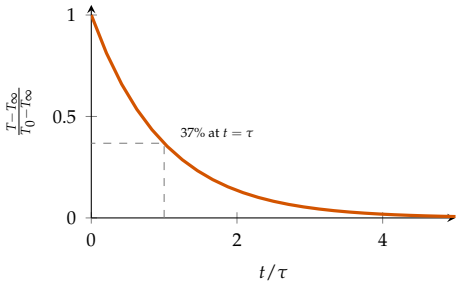


$$\dot{Q} = \frac{\Delta T}{\sum R_{thermal}} \qquad \text{(like Ohm's law: } I = V/R\text{)}$$

Plane wall	$R = \dfrac{L}{kA}$
Cylinder wall	$R = \dfrac{\ln(r_o/r_i)}{2\pi k L}$
Convection	$R = \dfrac{1}{hA}$
Contact / joint	$R = \dfrac{1}{h_c A}$

Tip: Series resistances add; parallel paths add as $1/R_{tot} = \sum 1/R_i$. The largest resistance controls the heat flow.

Dimensionless Numbers & Transient Cooling



Lumped capacitance, valid for $Bi < 0.1$

Nusselt	$Nu = \dfrac{hL}{k_{fluid}}$	convection vs. conduction in fluid
Biot	$Bi = \dfrac{hL}{k_{solid}}$	surface vs. internal resistance
Prandtl	$Pr = \dfrac{\nu}{\alpha}$	momentum vs. thermal diffusion
Thermal diffusivity	$\alpha = \dfrac{k}{\rho c_p}$	how fast heat spreads

Lumped capacitance: $\dfrac{T - T_\infty}{T_0 - T_\infty} = e^{-t/\tau}, \qquad \tau = \dfrac{\rho c_p V}{hA}$

For the $Bi < 0.1$ check, use characteristic length $L = V/A$.

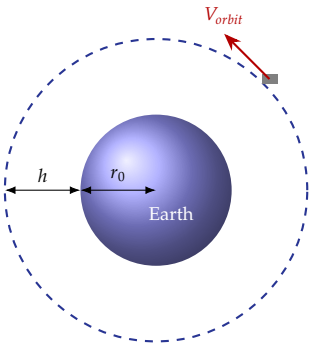
Typical Values

Convection	h [W/m^{2}K]
Free convection, air	2–25
Forced convection, air	25–250
Forced convection, water	100–20,000
Boiling / condensing	2,500–100,000

Surface	Emissivity ε
Polished aluminum / gold	0.03–0.05
Anodized aluminum	0.8–0.9
White / black paint	0.85–0.95
Oxidized steel	0.6–0.8

Orbital Mechanics

Circular Orbits, Escape & Period



Orbit speed $$V_{orbit} = r_0\sqrt{\frac{g_0}{r_0+h}} = \sqrt{\frac{\mu}{r}} \approx \sqrt{g_0 r_0}$$

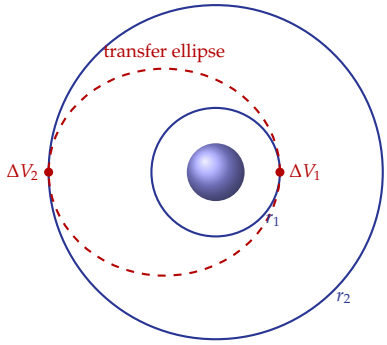
Escape speed $$V_{escape} = \sqrt{\frac{2\mu}{r}} \approx \sqrt{2g_0 r_0} = \sqrt{2}\,V_{orbit}$$

Orbital period $$\tau = \frac{2\pi(r_0+h)^{3/2}}{r_0\sqrt{g_0}} = 2\pi\sqrt{\frac{a^3}{\mu}} \approx 2\pi\sqrt{\frac{r_0}{g_0}}$$

$\mu = g_0 r_0^2 = GM$; r = distance from planet center; a = semi-major axis. Analogous to a pendulum: $\tau_{pend} \approx 2\pi\sqrt{l/g}$.

Tip: LEO: $V \approx 7.8$ km/s, period ≈ 90 min. Escape from Earth's surface: 11.2 km/s.

Elliptical Orbits & Hohmann Transfer



Vis-viva (speed anywhere on an orbit):

$$\boxed{V^2 = \mu\left(\frac{2}{r} - \frac{1}{a}\right)} \qquad a = \frac{r_p + r_a}{2}, \quad e = \frac{r_a - r_p}{r_a + r_p}$$

Hohmann transfer (circular $r_1 \to r_2$):

$$\Delta V_1 = \sqrt{\frac{\mu}{r_1}}\left(\sqrt{\frac{2r_2}{r_1+r_2}} - 1\right) \qquad \Delta V_2 = \sqrt{\frac{\mu}{r_2}}\left(1 - \sqrt{\frac{2r_1}{r_1+r_2}}\right)$$

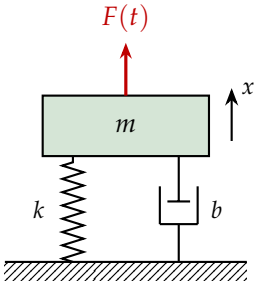
Transfer time $= \frac{1}{2}$ period of transfer ellipse $= \pi\sqrt{\frac{a_t^3}{\mu}}$

Constants

g_0 (Earth)	$9.81\ \text{m/s}^2$	Earth radius r_0	6.37×10^6 m
μ_{Earth}	$3.986 \times 10^{14}\ \text{m}^3/\text{s}^2$	g_{Mars}	$3.71\ \text{m/s}^2$
g_{Moon}	$1.62\ \text{m/s}^2$	G	$6.674 \times 10^{-11}\ \text{N·m}^2/\text{kg}^2$
h_{ISS}	$\sim 4 \times 10^5$ m (400 km)	$h_{geostationary}$	$\sim 3.6 \times 10^7$ m (35,786 km)
Earth–Mars (min)	$\sim 5.6 \times 10^{10}$ m	Earth–Moon	3.84×10^8 m

Dynamics & Vibration

Single-Degree-of-Freedom (SDOF) Oscillator



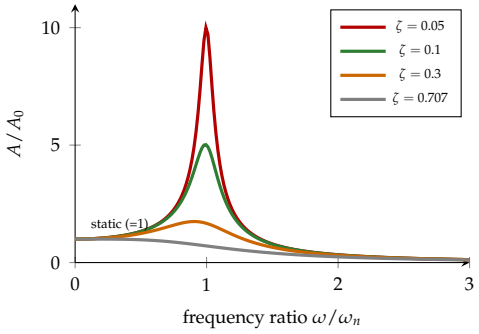
Equation of motion:

$$\boxed{F(t) = m\ddot{x} + b\dot{x} + kx} \quad \Longleftrightarrow \quad \frac{F(t)}{m} = \ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x$$

Natural frequency	$\omega_n = \sqrt{\frac{k}{m}}$ [rad/s]	(stiffness-to-mass ratio)
Hz conversion	$\omega\,[\text{rad/s}] = 2\pi f\,[\text{Hz}]$	
Damping ratio	$\zeta = \frac{b}{2\sqrt{km}} = \frac{b}{b_{crit}}$	
Damped frequency	$\omega_d = \omega_n\sqrt{1-\zeta^2}$	($\omega_d \approx \omega_n$ generally)

Tip: If using lbm, divide m by 386 in/s^2. Stiffer ⇒ higher frequency; heavier ⇒ lower.

Frequency Response (Resonance)



Amplification at resonance:

$$\boxed{Q = \frac{A_{max}}{A_0} \approx \frac{1}{2\zeta}}$$

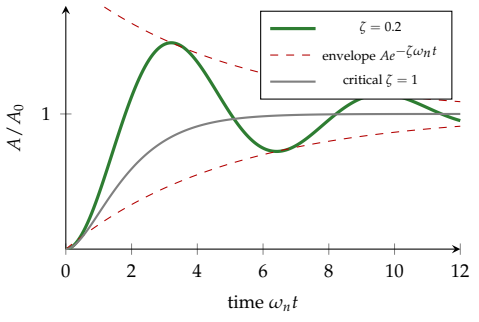
Thus doubling the damping ratio halves the resonant amplitude.

$\zeta \approx 1\%$	all metals (bare structure), $Q \approx 50$
$\zeta \approx 2$–3%	with silicone damping / bolted joints
$\zeta \approx 5$–10%	with fluid damping

Transfer function: $\dfrac{A}{A_0} = \dfrac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}, \quad r = \omega/\omega_n$

Tip: Well below resonance ($r \ll 1$) the part follows the input statically. Well above ($r > \sqrt{2}$) the response is attenuated.

Time Response to a Step Input



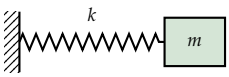
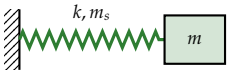
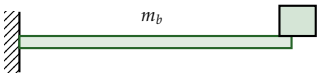
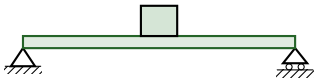
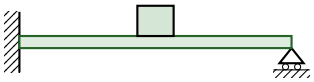
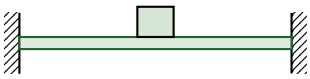
1% settling time	$t_s \approx \frac{4.6}{\zeta\omega_n}$	
Overshoot	$A_{overshoot} = e^{-\pi\zeta/\sqrt{1-\zeta^2}}$	
Log decrement	$\delta = \ln\frac{x_n}{x_{n+1}} \approx 2\pi\zeta$	

$\zeta < 1$ underdamped (oscillates) • $\zeta = 1$ critically damped (fastest, no overshoot) • $\zeta > 1$ overdamped

Tip: Damping ratio can be estimated by fitting $Ae^{-\zeta\omega_n t}$ to test data, or from two successive peaks via the log decrement.

Natural Frequencies [Hz] (fundamental mode)

m = attached mass, m_s = spring mass, m_b = beam mass, L = beam length. Rayleigh effective-mass approximations.

Sketch	Case	f_n [Hz]
	Mass on massless spring	$\frac{1}{2\pi}\sqrt{\frac{k}{m}}$
	Mass on massive spring	$\frac{1}{2\pi}\sqrt{\frac{k}{m+m_s/3}}$
	Cantilever, tip mass	$\frac{\sqrt{3}}{2\pi}\sqrt{\frac{EI}{(m+0.24\,m_b)L^3}}$
	Pinned–pinned, center mass	$\frac{4\sqrt{3}}{2\pi}\sqrt{\frac{EI}{(m+m_b/2)L^3}}$
	Fixed–pinned, center mass	$\frac{1}{2\pi}\sqrt{\frac{768}{7}\frac{EI}{(m+0.45\,m_b)L^3}}$
	Fixed–fixed, center mass	$\frac{8\sqrt{3}}{2\pi}\sqrt{\frac{EI}{(m+0.37\,m_b)L^3}}$

Beam in tension: multiply f_n by $\sqrt{1+\frac{F_{tension}}{F_{buckling}}}$ (compression: use $-F$)

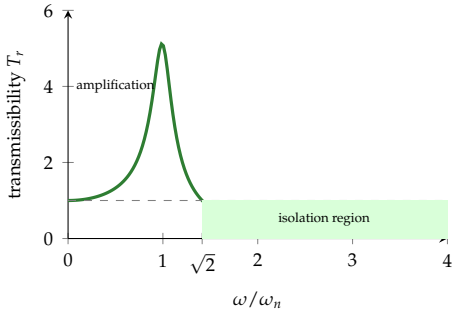
Uniform beam, no attached mass: $f_n = \frac{\lambda^2}{2\pi L^2}\sqrt{\frac{EI}{\rho_L}}$

$\lambda = 1.875$ cantilever $\lambda = \pi$ pinned–pinned $\lambda = 3.927$ fixed–pinned $\lambda = 4.730$ fixed–fixed

ρ_L = mass per unit length

Quick estimate from static sag: $f_n \approx \frac{1}{2\pi}\sqrt{\frac{g}{\delta_{static}}} \approx \frac{15.8}{\sqrt{\delta_{static}\,[\text{mm}]}}$ Hz

Vibration Isolation & Random Vibration



Base-excitation transmissibility:

$$T_r = \sqrt{\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2}}$$

Isolation only above $r > \sqrt{2}$: make the isolator soft so f_n is well below the forcing frequency.

Miles' equation (random vibration, SDOF):

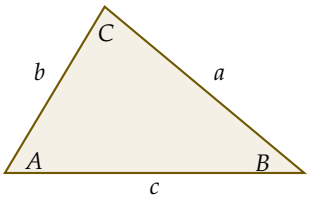
$$\boxed{G_{rms} = \sqrt{\frac{\pi}{2} f_n\, Q\, ASD(f_n)}}$$

ASD = input acceleration spectral density at f_n [g^2/Hz]; $Q \approx 1/(2\zeta)$. Design load $\approx 3\,G_{rms}$ (3-sigma).

Tip: Keep payload and structure modes separated by a factor of $\sqrt{2}$ or more (octave rule: 2×) to avoid coupled resonance.

Mathematics & Units

Trigonometry



Law of cosines: $\boxed{c^2 = a^2 + b^2 - 2ab\cos C}$

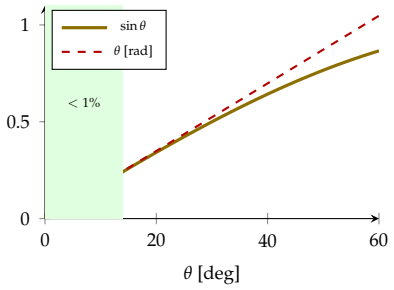
Law of sines: $\dfrac{a}{\sin A} = \dfrac{b}{\sin B} = \dfrac{c}{\sin C}$

Identities: $\sin^2\theta + \cos^2\theta = 1,\ \sin 2\theta = 2\sin\theta\cos\theta,\ \cos 2\theta = 1 - 2\sin^2\theta$

$\sin(a \pm b) = \sin a\cos b \pm \cos a\sin b \qquad \cos(a \pm b) = \cos a\cos b \mp \sin a\sin b$

Degrees → radians: $\times\pi/180 \quad (1\text{ rad} = 57.3°)$

Numerical Approximations



Small angle (θ in radians):

$$\sin\theta = \theta - \frac{\theta^3}{3!} + \cdots \approx \theta \qquad \cos\theta = 1 - \frac{\theta^2}{2!} + \cdots \approx 1$$

$$\tan\theta \approx \theta$$

Error $\leq$ 1%: $\sin\theta \approx \theta$ up to 14°; $\tan\theta \approx \theta$ up to 10°; $\cos\theta \approx 1$ only up to 8° (use $\cos\theta \approx 1 - \theta^2/2$ beyond that, good to $<$ 0.1% at 15°).

Binomial (small ϵ):

$$\boxed{(1+\epsilon)^n \approx 1 + n\epsilon}$$

e.g. $\dfrac{1}{1+\epsilon} \approx 1 - \epsilon, \qquad \dfrac{1}{1-\epsilon} \approx 1 + \epsilon, \qquad \sqrt{1+\epsilon} \approx 1 + \dfrac{\epsilon}{2}$

Other: $e^\epsilon \approx 1 + \epsilon, \quad \ln(1+\epsilon) \approx \epsilon$

Area & Volume

	Hollow circle area	$\frac{\pi}{4}(D_o^2 - D_i^2) \approx 2\pi Rt$		Sphere surface	$4\pi R^2$
	Tube surface	πDL		Spherical shell volume	$\frac{4}{3}\pi(R_o^3 - R_i^3)$
	Ellipse area	πab		Sphere volume	$\frac{4}{3}\pi R^3$
	Cylinder volume	$\pi R^2 L$		Cone volume	$\frac{1}{3}\pi R^2 h$

Unit Conversions

Length / Mass	
1 in	25.4 mm
1 ft	0.3048 m
1 mile	1.609 km
1 lbm	0.4536 kg
1 slug	14.59 kg

Force / Pressure	
1 lbf	4.448 N
1 psi	6.895 kPa
1 ksi	6.895 MPa
1 atm	101.3 kPa = 14.7 psi
1 bar	14.50 psi

Energy / Temp.	
1 BTU	1055 J
1 hp	745.7 W
1 ft·lbf	1.356 J
K	°C + 273.15
°R	°F + 459.67

$°F = 1.8\,°C + 32. \qquad g_c = 32.17\ lbm{\cdot}ft/(lbf{\cdot}s^2) = 386.1\ lbm{\cdot}in/(lbf{\cdot}s^2).$

Analysis Checklist

1. Set Up

- ☐ Define analysis goals and start with a free body diagram.
- ☐ Estimate results with hand calculations and double-check units at all steps.
- ☐ Obtain material properties and factors of safety from an approved source.
- ☐ Tabulate all loads, temperatures, accelerations, and number of cycles.

2. Loads & Conditions

- ☐ Evaluate design at worst-case load and thermal conditions (incl. transients).
- ☐ Sanity-check material cards and ensure they cover the full range of temperatures.
- ☐ Include relevant thermal, weld, and casting knockdown factors.
- ☐ Scale loads for symmetry/periodicity.
- ☐ Account for fluid/gas mass in lines.
- ☐ Ensure relevant blow-off loads are included (see free body diagram).
- ☐ Account for worst-case tolerances and fit-up loads.

3. Model Quality (FEA)

- ☐ Ensure blends/fillets are large enough to avoid fatigue and the mesh adequately captures stress concentrations (3 elements across radii).
- ☐ Perform mesh convergence study.
- ☐ Exaggerate deformations to verify boundary conditions and constraints.

4. Failure Modes

- ☐ Ensure parts in sliding are resistant to galling.
- ☐ Use principal stress for brittle materials (elongation $< 3\%$).
- ☐ Assess creep for sustained loads at high temperatures.
- ☐ Ensure fatigue calcs are corrected for temperature and include vibration.
- ☐ Verify human-rated factors of safety are used when personnel are present.
- ☐ Assess margins of safety on yield, ultimate, elongation, stability, fatigue, and gapping.

5. Verify & Document

- ☐ Check results with hand calcs ($\Sigma F = 0$, thermal strain, hoop stress, mass).
- ☐ Anchor model (strain gages, test data) and obtain peer review.
- ☐ Document inputs, assumptions, and results in a revision-controlled archive.