vibrationdata_pressure: Equations

vibrationdata_pressure.mlapp by Tom Irvine

Twenty tabs cover thirty-one equations for pressure vessels, boilers and piping: membrane and thick-wall stresses, ASME Section VIII shells and heads, external pressure, test pressure and stored energy, shell vibration, thermal stress, corrosion life, ASME Section I tubes and drums, saturated steam, boiler output, blowdown, safety valves, B31.3 and B31.1 wall thickness, pipeline MAOP, pressure drop, water hammer, thermal expansion and pipe spans. Each tab has English or metric units, its equations in a strip at the top of the tab, and a results panel. This document gives the equations, assumptions and references for each tab. It is a study and checking aid: design to the current edition of the governing code, which this app does not replace.

The thirty-one equations

1. Thin-wall cylinder stresses
2. Thin-wall sphere, von Mises
3. Lame thick-wall cylinder
4. UG-27 cylinder thickness and MAWP
5. UG-27 sphere
6. Formed heads, Appendix 1-4
7. UG-34 flat heads
8. Cylinder collapse, external pressure
9. Sphere buckling
10. Hydrostatic and pneumatic test
11. Stored energy, TNT equivalent
12. Shell ring frequency
13. Shell ovaling modes
14. Thermal stress
15. Corrosion rate
16. Remaining life
17. PG-27 boiler tubes
18. PG-27 piping, drums, headers
19. IAPWS-IF97 saturation line
20. Boiler output, boiler horsepower
21. Blowdown
22. Napier steam relief
23. B31.3 / B31.1 pipe wall
24. Barlow design pressure, MAOP
25. Darcy-Weisbach pressure drop
26. Colebrook friction factor
27. Korteweg wave speed
28. Joukowsky surge
29. Thermal expansion
30. Guided cantilever leg
31. Pipe span natural frequency

Units

Quantity	*English*	*metric*
Pressure, stress	psi	MPa
Pressure drop, surge	psi	kPa
Elastic and bulk modulus	psi	GPa
Thickness, diameter	in	mm
Length, span	ft	m
Temperature, difference	F	C
Expansion coefficient	1/F	1/C
Density, volume	lbm/ft^3, ft^3	kg/m^3, m^3
Flow rate, viscosity	gal/min, cP	L/s, cP
Steam rate, enthalpy	lbm/h, Btu/lbm	kg/h, kJ/kg
Heat rate, energy	Btu/h, ft-lbf	kW, kJ
Corrosion rate	mil/yr	mm/yr

Changing the units converts the entered values. Calculations are done in SI with $g = 9.80665$ m/s^2. Gauge pressure unless marked absolute. The Napier formula is evaluated in its English units in both systems.

1. Thin-Wall Vessel (Refs. 6, 7)

Eq. 1: $\sigma_\theta = \dfrac{P\,D_m}{2t}$ $\sigma_z = \dfrac{P\,D_m}{4t}$ $\sigma_r = -P$ (bore) $D_m = D + t$

Eq. 2: $\sigma_{sphere} = \dfrac{P\,D_m}{4t}$ $\sigma_{vm} = \sqrt{\sigma_\theta^2 - \sigma_\theta\sigma_z + \sigma_z^2}$ $MS = \dfrac{S_a}{\sigma_{vm}} - 1$

Membrane stresses of a thin cylinder with closed ends and of a sphere under internal pressure P (Refs. 6, 7). D = inside diameter, t = wall thickness, D_m = mean diameter. The mean-diameter form is close to the average Lame hoop stress. Valid for $t/D < 0.1$; thicker walls: Thick-Wall Cylinder tab.

Inputs: P internal pressure, D inside diameter, t wall thickness, Allowable, opt.

2. Thick-Wall Cylinder (Refs. 6, 7)

Eq. 3: $\sigma_r = A - \dfrac{B}{r^2}$ $\sigma_\theta = A + \dfrac{B}{r^2}$ $A = \dfrac{p_i a^2 - p_o b^2}{b^2 - a^2}$ $B = \dfrac{(p_i - p_o)\,a^2 b^2}{b^2 - a^2}$

$\sigma_z = A$ (closed), 0 (open), $2\nu A$ (plane strain) $u = \dfrac{r}{E}\left[\sigma_\theta - \nu(\sigma_r + \sigma_z)\right]$

Lame solution for a thick-walled cylinder, elastic, a = inside radius, b = outside radius, p_i, p_o = internal and external pressure (Refs. 6, 7). The largest hoop and von Mises stresses are at the bore. u = radial displacement. The plot shows the stresses through the wall.

Inputs: pi internal pressure, po external pressure, Di inside diameter, Do outside diameter, E elastic modulus, nu Poisson ratio.

3. ASME VIII Shell (Ref. 1)

Eq. 4: $t = \dfrac{PR}{SE - 0.6P}$ $\quad P = \dfrac{SEt}{R + 0.6t}$ $\quad t_L = \dfrac{PR}{2SE + 0.4P}$ $\quad P_L = \dfrac{2SEt}{R - 0.4t}$

Eq. 5: $t = \dfrac{PR}{2SE - 0.2P}$ $\quad P = \dfrac{2SEt}{R + 0.2t}$ $\quad$ (sphere) $\quad t_{nom} = t + CA$

ASME BPVC Section VIII Div. 1, UG-27 (Ref. 1): cylinder circumferential and longitudinal stress, and sphere. R = inside radius in the corroded condition, S = allowable stress (Section II Part D), E = joint efficiency (UW-12), CA = corrosion allowance. Limits: cylinder $t \le R/2$ or $P \le 0.385\,SE$; sphere $t \le 0.356\,R$ or $P \le 0.665\,SE$; beyond them use Appendix 1-2. UG-16(b) minimum thickness 1/16 in (1.5 mm), excluding corrosion allowance.

Inputs: P design pressure, R inside radius, S allowable stress, E joint efficiency, CA corrosion allow., t nominal (MAWP).

4. ASME VIII Heads (Ref. 1)

Eq. 6: $t = \dfrac{PDK}{2SE - 0.2P}$ $\quad K = \dfrac{2 + (D/2h)^2}{6}$ $\quad t = \dfrac{PLM}{2SE - 0.2P}$

$M = \dfrac{3 + \sqrt{L/r}}{4}$ $\quad t_{hemi} = \dfrac{PL}{2SE - 0.2P}$ $\quad t_{cone} = \dfrac{PD}{2\cos\alpha\,(SE - 0.6P)}$

Eq. 7: $t_{flat} = d\sqrt{\dfrac{CP}{SE}}$

Formed heads, UG-32 and Appendix 1-4 (Ref. 1). 2:1 ellipsoidal ($K = 1$, $h = D/4$): $t = PD/(2SE - 0.2P)$. Torispherical with $L = D$, $r = 0.06L$: $t = 0.885PL/(SE - 0.1P)$. D = inside diameter, h = inside depth, L = inside crown radius, r = knuckle radius, α = cone half-angle ($\le 30^\circ$). Flat heads UG-34: C from the attachment sketch (e.g. 0.17, 0.20, 0.30, 0.33); bolted covers add $1.9\,Wh_G/(SEd^3)$ under the root. The MAWP forms invert the same equations, e.g. $P = 2SEt/(KD + 0.2t)$. Corroded dimensions are used.

Inputs: P design pressure, D inside diameter, S allowable stress, E joint efficiency, CA corrosion allow., t nominal (MAWP), h ellipse depth, L crown radius, r knuckle radius, alpha half-angle (deg), C factor (flat).

5. External Pressure (Refs. 1, 8, 9)

Eq. 8: $P_{cr} = \dfrac{2E}{1-\nu^2}\left(\dfrac{t}{D}\right)^3$ $\qquad P_{cr} = \dfrac{2.42\,E\,(t/D)^{5/2}}{(1-\nu^2)^{3/4}\left[L/D - 0.45\sqrt{t/D}\right]}$

$L_c = 1.11\,D\sqrt{D/t}$ $\qquad$ Eq. 9: $P_{cr,sphere} = \dfrac{2E}{\sqrt{3(1-\nu^2)}}\left(\dfrac{t}{R}\right)^2$ $\qquad P_y = \dfrac{2S_y t}{D}$

Elastic buckling of a long tube (Bresse) and of a short tube between stiffeners (Windenburg and Trilling, Ref. 8); D = mean diameter, L = unsupported length. The larger of the two is used (the short-tube form tends to zero for long tubes). Sphere: classical (Zoelly) value; tests fall far below it (Ref. 9). These are estimates for perfect shells. Design for external pressure follows UG-28 and the charts of Section II Part D.

Inputs: Do outside diameter, t wall thickness, L unsupported length, Sy yield strength, P external, opt., E elastic modulus, nu Poisson ratio, Design factor.

6. Test Pressure & Energy (Refs. 1, 10)

Eq. 10: $P_{hydro} = 1.3\,MAWP\,\dfrac{S_t}{S}$ $\qquad P_{pneu} = 1.1\,MAWP\,\dfrac{S_t}{S}$

Eq. 11: $E_{gas} = \dfrac{pV}{\gamma - 1}\left[1 - \left(\dfrac{p_a}{p}\right)^{(\gamma-1)/\gamma}\right]$ $\qquad E_{water} = \dfrac{P^2 V}{2K}$ $\qquad W_{TNT} = \dfrac{E}{4.267\ \text{MJ/kg}}$

Test pressures of UG-99 (hydrostatic) and UG-100 (pneumatic), Ref. 1. S_t/S = lowest ratio of the allowable stress at test and at design temperature. Stored energy of the gas for isentropic expansion to atmosphere, absolute pressures, and TNT equivalent (ASME PCC-2, Art. 5.1, Ref. 10). Water: compression energy only (K = bulk modulus, 2.2 GPa); the strain energy of the vessel is not included. A pneumatic test stores far more energy.

Inputs: MAWP, Stress ratio St/S, V vessel volume, gamma of test gas, pa atmospheric, K water bulk mod.

7. Shell Ring Frequencies (Ref. 11)

Eq. 12: $f_r = \dfrac{1}{2\pi R}\sqrt{\dfrac{E}{\rho\,(1-\nu^2)}}$ $\qquad R = D/2$

Eq. 13: $f_n = \dfrac{1}{2\pi}\,\dfrac{n(n^2-1)}{\sqrt{n^2+1}}\,\dfrac{t}{R^2}\sqrt{\dfrac{E}{12\,\rho\,(1-\nu^2)}}$ $\qquad n = 2, 3, \ldots$

Ring frequency (breathing mode, $n = 0$) and in-plane bending (ovaling) modes of a long thin cylindrical shell with no axial wavelength (Ref. 11). R = mean radius. Plane-strain modulus $E/(1-\nu^2)$ for the long shell. Contained liquid adds mass and lowers the bending frequencies.

Inputs: D mean diameter, t wall thickness, Number of modes, E elastic modulus, rho density, nu Poisson ratio.

8. Thermal Stress (Ref. 6)

Eq. 14: $\sigma = \pm\dfrac{E\,\alpha\,\Delta T}{2(1-\nu)}$ $\qquad$ linear through-wall gradient

$\sigma = \dfrac{E\,\alpha\,\Delta T}{1-\nu}$ (restrained plate) $\qquad \sigma = E\,\alpha\,\Delta T$ (restrained bar)

Elastic thermal stresses away from edges and discontinuities (Ref. 6). ΔT = through-wall temperature difference (gradient) or the change from the restrained state. The hotter surface is in compression. Thermal stresses are secondary: the codes limit their range (e.g. $3S$ for primary plus secondary).

Inputs: Delta T, Allowable, opt., E elastic modulus, nu Poisson ratio, alpha expansion.

9. Corrosion & Remaining Life (Refs. 12, 13)

Eq. 15: $CR = \dfrac{t_{prev} - t_{act}}{\Delta \text{ years}}$ Eq. 16: $RL = \dfrac{t_{act} - t_{req}}{CR}$ $\text{interval} = \min\left(\dfrac{RL}{2},\ 10 \text{ yr}\right)$

Corrosion rate and remaining life of API 510 (vessels) and API 570 (piping), Refs. 12, 13. Long-term rate: initial to actual; short-term: previous to actual; use the rate that gives the shorter life. Internal or on-stream inspection interval for vessels: the lesser of half the remaining life and 10 years (if RL is under 4 years the interval may be the full remaining life, up to 2 years). API 570 uses the piping class maximum instead of 10 years.

Inputs: t previous (or initial), t actual (latest), Years between, t required.

10. ASME I Tubes & Drums (Ref. 2)

Eq. 17: $t = \dfrac{PD}{2S + P} + 0.005D + e$ $P = S\,\dfrac{2t - 0.01D - 2e}{D - (t - 0.005D - e)}$

Eq. 18: $t = \dfrac{PD}{2SE + 2yP} + C$ $P = \dfrac{2SE\,(t - C)}{D - 2y\,(t - C)}$

ASME BPVC Section I, PG-27 (Ref. 2): PG-27.2.1 for tubes up to 5 in (125 mm) outside diameter, PG-27.2.2 for piping, drums and headers. D = outside diameter, e = thickness factor for expanded tubes (0 for strength-welded tubes), y = temperature coefficient (0.4 for ferritic steels below 900 F), C = allowance for threading, stability and corrosion, E = ligament or weld efficiency.

Inputs: P MAWP, D outside diameter, S allowable stress, t thickness (MAWP), E efficiency (drums), y coefficient, e (tubes), C allowance (pipe).

11. Saturated Steam (Ref. 14)

Eq. 19: $p_s = \left[\dfrac{2C}{-B + \sqrt{B^2 - 4AC}}\right]^4$ $\quad \vartheta = T + \dfrac{n_9}{T - n_{10}}$ $\quad A = \vartheta^2 + n_1\vartheta + n_2$

$B = n_3\vartheta^2 + n_4\vartheta + n_5$ $\quad C = n_6\vartheta^2 + n_7\vartheta + n_8$ $\quad p_s$ in MPa, T in K

IAPWS-IF97 region 4 saturation equation and its explicit inverse $T_s(p)$ (Ref. 14), 273.15 K to 647.096 K (0.000611 to 22.064 MPa). Gauge pressure plus atmospheric pressure gives the absolute pressure. Check: 212 F gives 14.709 psia (101.42 kPa).

Inputs: p, gauge, pa atmospheric, T saturation.

12. Boiler Output & Efficiency (Ref. 15)

Eq. 20: $Q = \dot{m}_s\,(h_s - h_f)$ $\quad \text{BHP} = \dfrac{Q}{33{,}475\ \text{Btu/h}}$ $\quad \dot{m}_e = \dfrac{Q}{970.3\ \text{Btu/lbm}}$

$Q_{fuel} = \dfrac{Q}{\eta}$ $\quad \dot{m}_{fuel} = \dfrac{Q_{fuel}}{HV}$ $\quad FE = \dfrac{h_s - h_f}{970.3\ \text{Btu/lbm}}$

Boiler heat balance. One boiler horsepower = 33,475 Btu/h = 34.5 lbm/h evaporated from and at 212 F (Ref. 15); 970.3 Btu/lbm = 2257 kJ/kg. h_s, h_f = enthalpy of the steam and the feedwater (steam tables), η = boiler efficiency on the same heating-value basis as HV, FE = factor of evaporation.

Inputs: Steam rate, h steam, h feedwater, Efficiency (%), Fuel heating value.

13. Boiler Blowdown (Ref. 15)

Eq. 21: $B = \dfrac{\dot{m}_s\, C_f}{C_b - C_f}$ $\qquad F = \dot{m}_s + B$ $\qquad \dfrac{B}{F} = \dfrac{C_f}{C_b}$ $\qquad \text{cycles} = \dfrac{C_b}{C_f}$

$Q_B = B\,(h_b - h_m)$

Dissolved solids balance: the steam carries no solids, so the feedwater solids leave in the blowdown (Ref. 15). C_f, C_b = feedwater and boiler water concentrations (TDS), h_b = enthalpy of the blowdown (saturated liquid at drum pressure), h_m = makeup water enthalpy. Q_B: heat that a flash tank and heat recovery can reclaim.

Inputs: Steam rate, Feedwater TDS (ppm), Boiler TDS limit (ppm), h blowdown water, h makeup water.

14. Steam Safety Valve (Ref. 16)

Eq. 22: $W = 51.5\, A\, P\, K\, K_{sh}$ $\qquad$ (lbm/h, in^2, psia) $\qquad W = 5.25\, A\, P\, K$

$P = p_{set} + \Delta p_{over} + p_a$ $\qquad K = 0.9\, K_d$ $\qquad F_N = \dfrac{0.1906P - 1000}{0.2292P - 1061},\ P > 1500$ psia

Napier formula for dry saturated steam (Section I PG-69, Section VIII UG-131; now ASME Section XIII), Ref. 16. Metric form: kg/h, mm^2, MPa absolute. Overpressure: Section I 3% or 2 psi, Section VIII 10% or 3 psi, whichever is greater. K_d = certified coefficient of discharge, K_{sh} = superheat correction factor.

Inputs: Set pressure, gauge, Overpressure (%), pa atmospheric, K coefficient, Ksh superheat, W capacity, A flow area.

15. Pipe Wall Thickness (Refs. 3, 4, 5)

Eq. 23: $t = \dfrac{PD}{2\,(SEW + PY)}$ $\qquad t_m = t + c$ $\qquad t_{nom} = \dfrac{t_m}{1 - \text{tol}}$ $\qquad P = \dfrac{2\,SEW\,t}{D - 2Yt}$

$t = t_{nom}(1 - \text{tol}) - c \qquad t < D/6$

ASME B31.3, 304.1.2 Eq. (3a), Ref. 3, and ASME B31.1, 104.1.2 Eq. (7), Ref. 4 (same form; B31.1 calls the allowance A). D = outside diameter, S = allowable stress, E = quality factor, W = weld joint strength reduction factor, Y = coefficient (0.4 for ferritic steels below 900 F), c = corrosion, erosion and thread allowances, tol = mill under-tolerance (pipe specification, e.g. 12.5%). NPS dimensions and schedules: ASME B36.10M (Ref. 5).

Inputs: P design pressure, NPS, opt., D outside diameter, S allowable stress, E quality factor, W weld strength factor, Y coefficient, c allowances, Mill tolerance (%), t nominal (MAWP).

16. Pipeline MAOP (Refs. 17, 18)

Eq. 24: $P = \dfrac{2St}{D}\,F\,E\,T$ $\qquad$ (B31.8) $\qquad P = \dfrac{2St}{D}\,0.72\,E$ $\qquad$ (B31.4) $\qquad \sigma_h = \dfrac{PD}{2t}$

$\%\,SMYS = 100\,\dfrac{\sigma_h}{S}$

Barlow design pressure of ASME B31.8 (gas, 841.1.1) and B31.4 (liquid), Refs. 17, 18. S = specified minimum yield strength, D = outside diameter, t = nominal wall, F = design factor (Class 1 Div 1 0.80, Div 2 0.72, Class 2 0.60, Class 3 0.50, Class 4 0.40), E = joint factor, T = temperature derating.

Inputs: D outside diameter, t nominal wall, SMYS, E joint factor, F design factor, T temperature factor, P operating, opt.

17. Pipe Pressure Drop (Refs. 19, 20)

Eq. 25: $\Delta p = \left(f \frac{L}{D} + \sum K \right) \frac{\rho V^2}{2} + \rho g \, \Delta z \qquad Re = \frac{\rho V D}{\mu}$

Eq. 26: $\frac{1}{\sqrt{f}} = -2 \log_{10} \left(\frac{e/D}{3.7} + \frac{2.51}{Re\sqrt{f}} \right) \qquad f = \frac{64}{Re}, \; Re < 2300$

Darcy-Weisbach with the Colebrook friction factor (solved by iteration from the Swamee-Jain value), Refs. 19, 20. Incompressible flow, full pipe. e = absolute roughness (commercial steel 0.0018 in, 0.045 mm), K = fitting loss coefficients, Δz = elevation rise. 2300 < Re < 4000: transitional, uncertain.

Inputs: Q flow rate, D inside diameter, L pipe length, e roughness, Sum of K (fittings), Elevation rise, rho density, mu viscosity (cP).

18. Water Hammer (Ref. 21)

Eq. 27: $a = \sqrt{\frac{K/\rho}{1 + c_1 \frac{KD}{Et}}} \qquad$ Eq. 28: $\Delta p = \rho \, a \, \Delta V \qquad \Delta H = \frac{a \, \Delta V}{g} \qquad t_{cr} = \frac{2L}{a}$

$\Delta p \approx \frac{2 \rho L \Delta V}{t_c}, \; t_c > \frac{2L}{a}$

Korteweg wave speed in an elastic pipe and the Joukowsky surge for a rapid velocity change (Ref. 21). K = fluid bulk modulus, E = pipe modulus, D = inside diameter, t = wall, c_1 = restraint factor (1 with expansion joints; $1 - \nu^2$ anchored against axial movement). Slow closure: Michaud estimate.

Inputs: D inside diameter, t wall thickness, L pipe length, Delta V velocity, Closure time (s), opt., E pipe modulus, K fluid bulk modulus, rho fluid, c1 restraint factor.

19. Thermal Expansion & Loop (Refs. 3, 22)

Eq. 29: $\Delta L = \alpha\, L\, \Delta T$ $\sigma = E\, \alpha\, \Delta T$ $F = \sigma A_m$ Eq. 30: $L_{leg} = \sqrt{\dfrac{3\, E\, D\, \Delta}{S_A}}$

$\Delta = \Delta L$

Free expansion of a straight run, the stress and anchor force when it is fully restrained, and the guided-cantilever leg that absorbs the expansion at the stress range S_A (Ref. 22). A simplified check: a flexibility analysis to B31.3 (319) or B31.1 is needed for design.

Inputs: L pipe run, Delta T, D outside diameter, t wall thickness, SA stress range, E elastic modulus, alpha expansion.

20. Pipe Span & Frequency (Refs. 6, 11)

Eq. 31: $f_n = \dfrac{\lambda^2}{2\pi L^2}\sqrt{\dfrac{EI}{m}}$ $\lambda^2 = \pi^2,\ 22.373,\ 3.516$ $m = m_{pipe} + m_{contents} + m_{ins}$

$M = c_M\, wL^2$ $c_M = \dfrac{1}{8},\ \dfrac{1}{12},\ \dfrac{1}{2}$ $\delta = c_\delta \dfrac{wL^4}{EI}$ $c_\delta = \dfrac{5}{384},\ \dfrac{1}{384},\ \dfrac{1}{8}$

Uniform beam under its own weight $w = mg$: pinned-pinned, fixed-fixed and cantilever (Refs. 6, 11). I = area moment of inertia, $Z = 2I/D$, bending stress M/Z. The fundamental frequency is often kept above about 4 Hz to avoid low-frequency vibration.

Inputs: D outside diameter, t wall thickness, L span, rho contents, Insulation weight, Allowable, opt., E elastic modulus, rho pipe.

References

1. ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, *Rules for Construction of Pressure Vessels* (UG-16, UG-27, UG-28, UG-32, UG-34, UG-99, UG-100, Appendices 1-2 to 1-5).
2. ASME Boiler and Pressure Vessel Code, Section I, *Rules for Construction of Power Boilers* (PG-27).
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Code paragraph numbers change between editions; check against the edition that governs the design. The example inputs (allowable stresses, material properties, enthalpies, water quality limits) are illustrative.