Calculations

Ariel Calculation Method

The Ariel Performance Software includes calculations for predicting the flow, power, temperatures, pressures, gas rod loads, crosshead pin load reversals as well as other values for the predictions and limitations of the compressor applications. The following outlines the general methods and some of the equations applied for Ariel reciprocating compressors.

Pressures:

Operating pressures are calculated allowing user input pressure losses prior to, after and between stages of compression. Ariel includes a typical pressure loss for the interstage and final discharge pressure losses. Typical values are:

Flange Pressure	Pressure Loss	Not To Exceed
35 psia and below	5%	1 psi
36 psia to 250 psia	3%	5 psi
251 psia to 1000 psia	2%	10 psi
1001 psia and above	1%	

Operating pressures at each stage are expressed in the term Compression Ratio. Depending upon the use of this term, it may be defined at the cylinder flange pressures or the internal pressures. Compression ratio is the discharge pressure divided by the suction pressure (in absolute pressure units).

Interstage flange pressures are determined by balancing the flow of gas through each stage (expressed in standard, not actual volume). Gas added through side streams or removed through side streams or condensate from each stage is considered.

Pressures inside the cylinder consider pressure losses through the cylinder gas passages and valves. Many of the calculations apply the use if internal pressures rather than flange to flange pressures. Cylinder and valve pressure losses are a function of the gas density and gas velocity through the cylinder gas passages and valves.

Temperatures:

Discharge temperature after each stage of compression is a function of the gas properties (k-value, or N-value), internal compression ratio and suction temperature. Suction temperature at each stage is user defined. The internal suction temperature may include a preheat value, depending upon the inlet gas temperature and discharge gas temperature.

Discharge Temperature (initial estimate using flange ratio): $Tdi = Tsi\,(Ri)^{\frac{k-1}{k}}$

Suction Temperature Preheat: Tsph = Ts + [(0.02 + (0.002 x Cyl bore)) x (Tdi - Ts)]

Discharge Temperature: Td = Tsph x (Rinternal)^((k-1)/k)

Flow:

Flow is a function of the piston displacement and the volumetric efficiency. The piston displacement is the piston area times the length of stroke times the rpm. Both ends of a double acting cylinder are included in this calculations, with the crank end considering the loss of piston area due to the piston rod. Flow is calculated on a per end basis.

$$Q = VE \text{ x } PD \text{ x } \left(\frac{Ps}{Zs} \right) \text{ x number of cylinders x } 10^{-6}$$

As Q is actual flow, it is often converted to be expressed in standard flow units.

Volumetric Efficiency:

Volumetric efficiency includes many factors that help explain the differences between ideal gas behavior and real gas behavior. In general, volumetric efficiency depends upon compression ratio, cylinder clearances, gas compressibility values and the ratio of specific heats (k or N value). A common representation of volumetric efficiency is:

$$VE = \frac{98 - R}{100} - CL\left[\left(\frac{Z_s}{Z_d}\right) R^{\frac{1}{n}} - 1\right]$$

Power:

Power is calculated through a power per unit flow equation, multiplied by the flow. Calculated power will include the power of compression, plus mechanical inefficiencies, plus frame friction power. Internal power of compression equations include:

$$IHP / MM = 43.6\left(\frac{k}{k-1}\right)\left(Rint^{\frac{k-1}{k}} - 1\right) x\, Za$$

CHP = Q x (IHP/MM)/ Mech Eff + Friction power.

Mechanical Efficiency is near 0.95, offering a 5% loss for mechanical inefficiencies of the cylinders. Friction power loss is dependent upon the frame size.

Gas Rod Load:

Ariel calculates gas rod load based upon internal cylinder pressures. The equations below are based upon pressures in gauge units. If absolute units are applied, then additional terms for Patm being applied on the piston rod diameter must be included.

Double Acting Cylinders

RLc = Ahe x Pdi - Ace x Psi

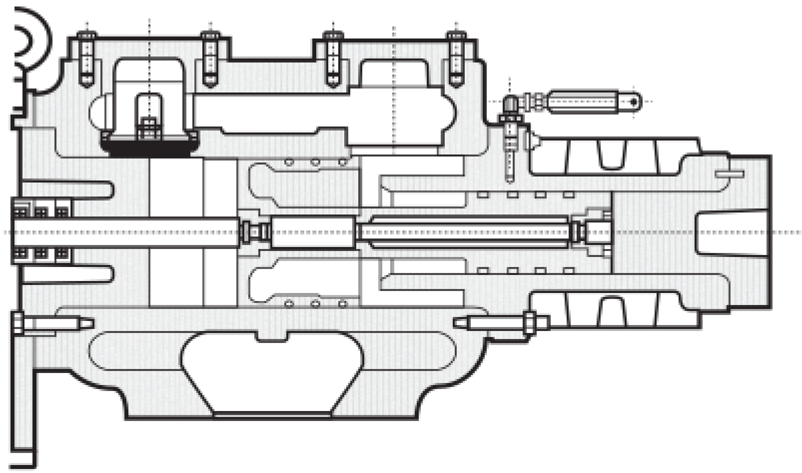
RLt = Ace x Pdi - Ahe x Psi

Single Acting Crank-End Cylinders

RLc = Ahe x Pdihe - Ace x Psi

RLt = Ace x Pdi - Ahe x Psihe

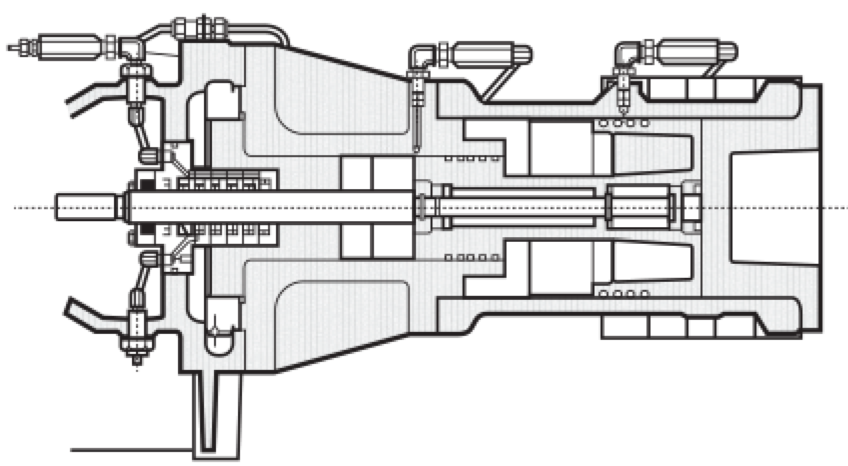
Tandem Cylinders - (High Pressure Cylinder Outboard)



RLc = Ahe(HP) x Pdi(HP) +[Ahe((LP) - Ahe(HP)] x Psflg(LP) - Ace(LP) x Psi(LP)

RLt = Ace(LP) x Pdi(LP) - [Ahe(LP) - Ahe(HP)] x Psflg(LP) - Ahe(HP) x Psi(HP)

Tandem Cylinders - (High Pressure Cylinder Inboard)



RLc = Ahe(LP) x Pdi(LP) - [Ahe(LP) - Ahe(HP)] x Psflg(LP) - Ace(HP) x Psi(HP)

RLt = Ace(HP) x Pdi(HP) + [Ahe(LP) - Ahe(HP)] x Psflg(LP) - Ahe(LP) x Psi(LP)

Since Gas Rod Load is calculated using internal pressures, the pressure losses through the cylinder gas passages and valves must be applied to the cylinder flange pressures. Pressure losses are calculated using the gas velocity at the suction and discharge. Gas velocities are calculated based upon the flow areas of the gas passages and valves. Flow areas are available through the Performance Software cylinder data sheets.

Psi = Psflg x ((100 - PLs)/100)

Pdi = Pdflg x ((100 + PLd)/100)

Rint = Pdi / Psi

$$PLs = \left[\frac{(GVs)^2}{6.55 \times 10^6}\right] \times \frac{SG}{Zsi} \times \frac{520}{Tsi}$$

$$PLd = \left[\frac{(GVd)^2}{4.55 \times 10^6}\right] \times \frac{SG}{Zd} \times \frac{520}{Td}$$

$$Avc = \left[\frac{1}{(AV)^2} + \frac{1}{(AC)^2}\right]^{-\frac{1}{2}} \quad \text{(HE or CE Suction or Discharge)}$$

$$GVs \text{ or } GVd = \frac{\text{Piston Area, in}^2 \text{ (HE or CE)} \times \text{Piston Speed, fpm}}{\text{Avc, in}^2 \text{ (HE or CE) Suction or Discharge / Corner}}$$

Definitions / Units

Area units are in square inches.

Diameter and stroke units are in inches.

Gas velocity units are in feet per minute

Pressure units are in psia for ratio calculations and psig for rod load calculations.

Temperature units are in Rankine.

Power units are in hp.

Flow units are in MMCFD, actual for Q and standard for Flow.

Clearance values are in percent.

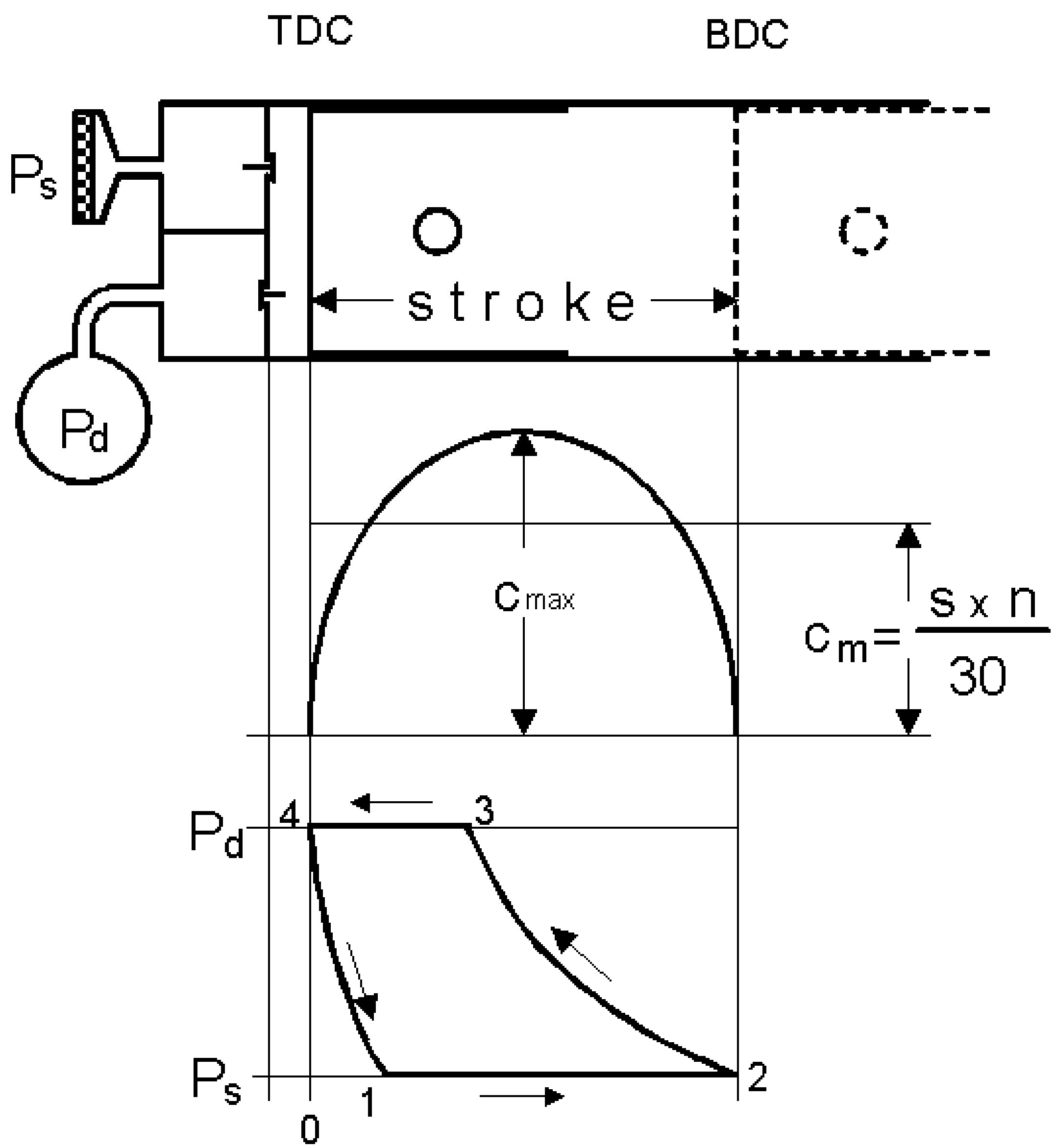
Pressure loss is expressed in percent.

Compressor Theory

Ideal P-V Diagram

Given the laws governing and definitions describing gas behavior, let us look at how they are applied to a typical reciprocating compressor cycle. One of the preferred tools for analyzing compressor performance is the Pressure-Volume (P-V) diagram. The P-V diagram which is to be discussed here depicts the relationship of the pressure and volume of the gas within one end of a cylinder of a reciprocating compressor to the displacement of the piston. To start with we will look at an ideal P-V diagram in which there are no valve losses and the compression is adiabatic.

1. Suction valve opens and gas is drawn into the cylinder (1 - 2).
2. Suction valve closes and gas compression begins (2 - 3).
3. Discharge valve opens and the compressed gas is discharged from the cylinder (3 - 4).
4. Discharge valve closes. Note that a gap is shown in this diagram between zero volume and the volume at position 4. This represents the clearance volume in the cylinder. As the piston begins its return stroke, the gas which remains in this space re-expands (4 - 1).



Effect of Clearance Volume on Capacity

Gas is trapped in the clearance volume after each stroke. This volume of gas must be re-expanded before gas at suction conditions is admitted into the cylinder to be compressed.

Effect of Clearance Volume

The re-expansion of the gas trapped at the end of the discharge cycle does not influence the horsepower losses but has a direct effect on the volumetric efficiency of the cylinder. Figure: Effect of Clearance Volume shows two P-V curves with different clearance volumes superimposed. The opening positions of the valves change dramatically. With the higher clearance volume the volumetric efficiency is considerably lower but so is the horsepower consumption.

$$VE_s = 100\text{-}CL\%\left[\left(\frac{P_d}{P_s}\right)^{\frac{1}{k}}\text{-}1\right]$$

$$VEd = \frac{VEs}{\left[\frac{Z_s}{Z_d}\left(\frac{P_d}{P_s}\right)^{\frac{1}{k}}\right]}$$

Pd = discharge pressure [psia]

Ps = suction pressure [psia]

CL% = clearance volume in %

k = effective cylinder isentropic exponent

Figure: PV vs Clearance

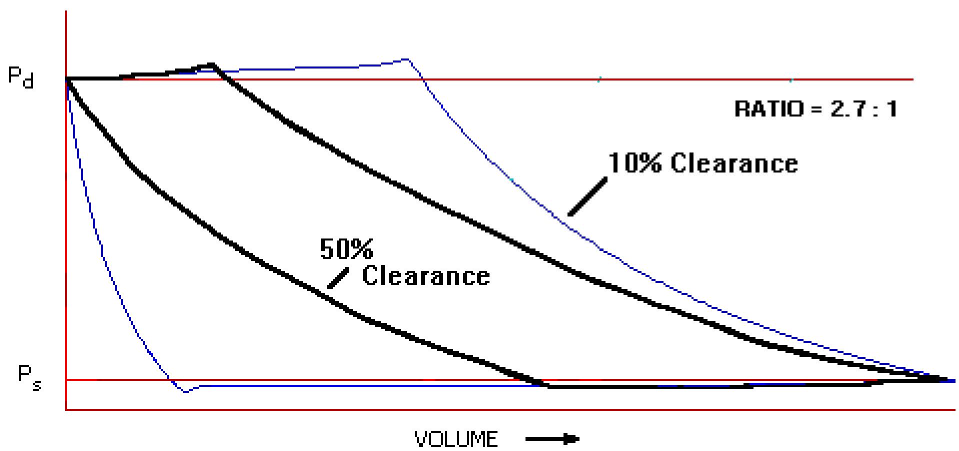
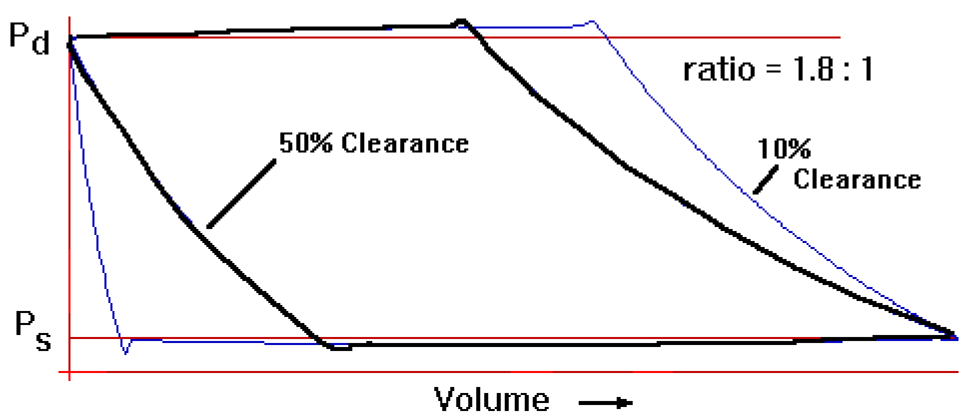


Figure: PV vs Clearance Low Ratio



Clearance Volume - Low Ratio

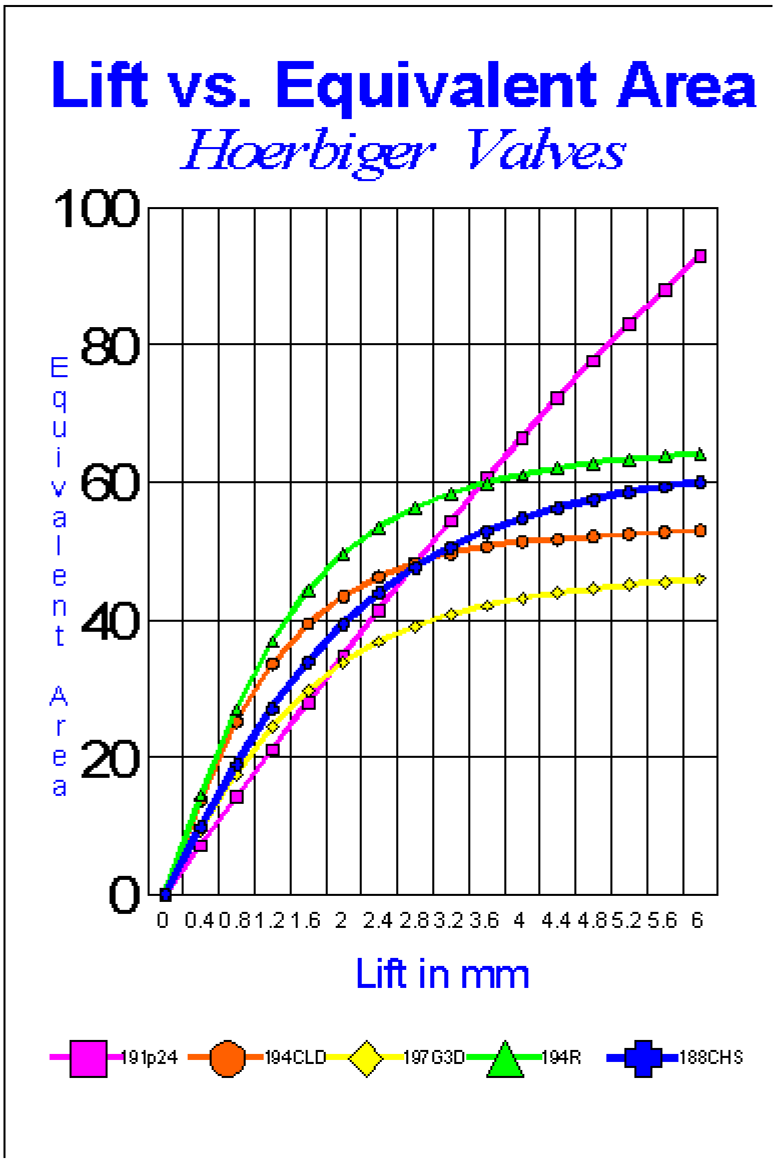
In applications where the compression ratio is relatively small, the extra clearance volume has a lesser effect on the volumetric efficiency. The reason for this is demonstrated in Figure: Effect of Clearance Volume and Figure PV vs Clearance Low Ratio. In Figure: PV vs Clearance Low Ratio an application with a 2.7:1 compression ratio is shown with clearance volumes of 10% and 50%. With more clearance volume, the piston must travel farther in its compression stroke before the cylinder pressure exceeds the discharge line pressure enough to open the discharge valve. The discharge volumetric efficiency (VEd) decreases and can be reduced to a point (especially in high-speed machines) where the compressed gas cannot be discharged quickly enough and the valve is forced to close late. A similar reduction in volumetric efficiency, and therefore incapacity, can be expected on the suction valve when the clearance volume is increased.

Adjusted Equivalent Valve Area

Adjusted equivalent valve area is a measure of the effective orifice area of the complete valve assembly. The equivalent area is a static measure and does not assure good dynamic behavior of the valve. This is a useful term to compare valve designs, as a valve with a higher adjusted equivalent area will generally have a lower pressure drop and better efficiency.

Equivalent area is a function of valve lift and port area. Increasing lift or port area increases equivalent area. However, higher lift valves generally have a shorter life than lower lift valves--thus the trade-off between efficiency and durability

Below is a comparison of lift vs. equivalent area for several valves.



Piston Rod Root Stress

Piston rod root stress is used to design the appropriate piston rod attachment preload levels. Ariel piston rod to crosshead nut preload levels are 30,000 psi. This is well above 1-1/2 times the root stress potential from inertia and gas loading. Ariel does not publish a limit based upon piston rod root stress.

Root areas for Ariel frames are below:

Frame Model	Rod Diameter (in)	Crosshead Thread Pitch	Thread Root Area (in^2)
JGM:N:P:Q	1-1/8 inch	1"-12	0.625 in^2
JG / JGA	1-1/8 inch	1"-12	0.625 in^2
JGW:R:J	1-1/2 inch	1-3/8"-12	1.260 in^2
JGH:E:K:T KBK:T	2 inch	1-3/4"-12	2.120 in^2
JGC:D:F	2-1/2 inch	2-1/4"-8	3.420 in^2
KBZ:U	2-7/8 inch	2-5/8"-8	4.76 in^2
KBB:V	3-1/8 inch	2-7/8"-8	5.780 in^2

Unit Conversions -Flow

Multiply units in left column by proper factor below.

Units of Flow	MMSCFD	Sm^3/hr	Nm^3/hr
1 MMSCFD @14.7 psia & 60oF	1	1179.87	1116.28
1 Sm^3/hr @1.0135 BarA & 15oC	0.0008476	1	0.9461
1 Nm^3/hr @1.0135 BarA & 0oC	0.0008958	1.0569	1

Flow Units: CFM -Cubic feet per minute.

ACFM -Actual cubic feet per minute (pressure and temperature condition must also be stated)

ICFM -Inlet cubic feet per minute based upon the inlet pressure and temperature at the cylinder flange.

SCFM -Standard cubic feet per minute is gas flow measured at standard conditions (14.7 psia & 60 °F).

MMSCFD -Million standard cubic feet per day is million cubic feet of gas per day measured at standard conditions (14.7 psia & 60 °F). (not to be confused with MSCFD, 1000 SCFD)

Sm^3/hr -Standard cubic meters per hour, measured at standard metric conditions(1 Atm & 15 °C). Used primarily in Canada, South America, and New Zealand

Nm^3/hr -Normal cubic meters per hour, measured at normal metric conditions (1 Atm & 0 °C). Used primarily in Europe.

Unit Conversions - Volume

Multiply units in left column by proper factor below.

Units of Volume	cu.in.	cu.ft.	cu.meter	liter	US.gal	Imp. gal
1 cu.inch	1	0.00058	0.0000164	0.0164	0.0043	0.0036
1 cu.foot	1728	1	0.0283	28.32	7.481	6.229
1 cu.meter	61,023	35.31	1	1000	264.2	220
1 liter	61.025	0.0353	0.0010	1	0.2642	0.220
1 US. gal	231	0.1337	0.0038	3.785	1	0.8327
1 Imp. gal	277.4	0.1605	0.0045	4.546	1.201	1

Unit Conversions - Power

Multiply units in left column by proper factor below.

Units of Power	hp	kw
1 horsepower	1	0.7457
1 kilowatt	1.3410	1

Unit Conversions - Pressure

Multiply units in left column by proper factor below.

Units of Pressure	psi	atm	kg/sq.cm.	kPa	bar
1 pound/sq.in.	1	0.06805	0.0703	6.89476	0.06895
1 atmosphere	14.696	1	1.0332	101.325	1.01325
1 kilogram/sq. cm.	14.223	0.9678	1	98.066	0.98066
1 kilopascal	0.14504	0.009869	0.010197	1	0.01
1 bar	14.504	0.9869	1.0197	100	1

Unit Conversions - Length

Multiply units in left column by proper factor below.

Units of Length	in	ft	mm	cm	m
1 inch	1	0.0833	25.4	2.54	0.0254
1 foot	12	1	304.8	30.48	0.3048
1 millimeter	0.394	0.00328	1	0.1	0.001
1 centimeter	0.3937	0.03281	10	1	0.01
1 meter	39.37	3.281	1000	100	1

Unit Conversions - Weight

Multiply units in left column by proper factor below.

Units of Weight	lbs	ton	kg	metric ton
1 pound	1	0.0005	0.4535	0.00045
1 ton	2000	1	907	0.907
1 kilogram	2.205	0.0011	1	0.001
1 metric ton	2205	1.102	1000	1