EFRC Training Workshop
Design and operation of reciprocating compressors

Thermodynamics

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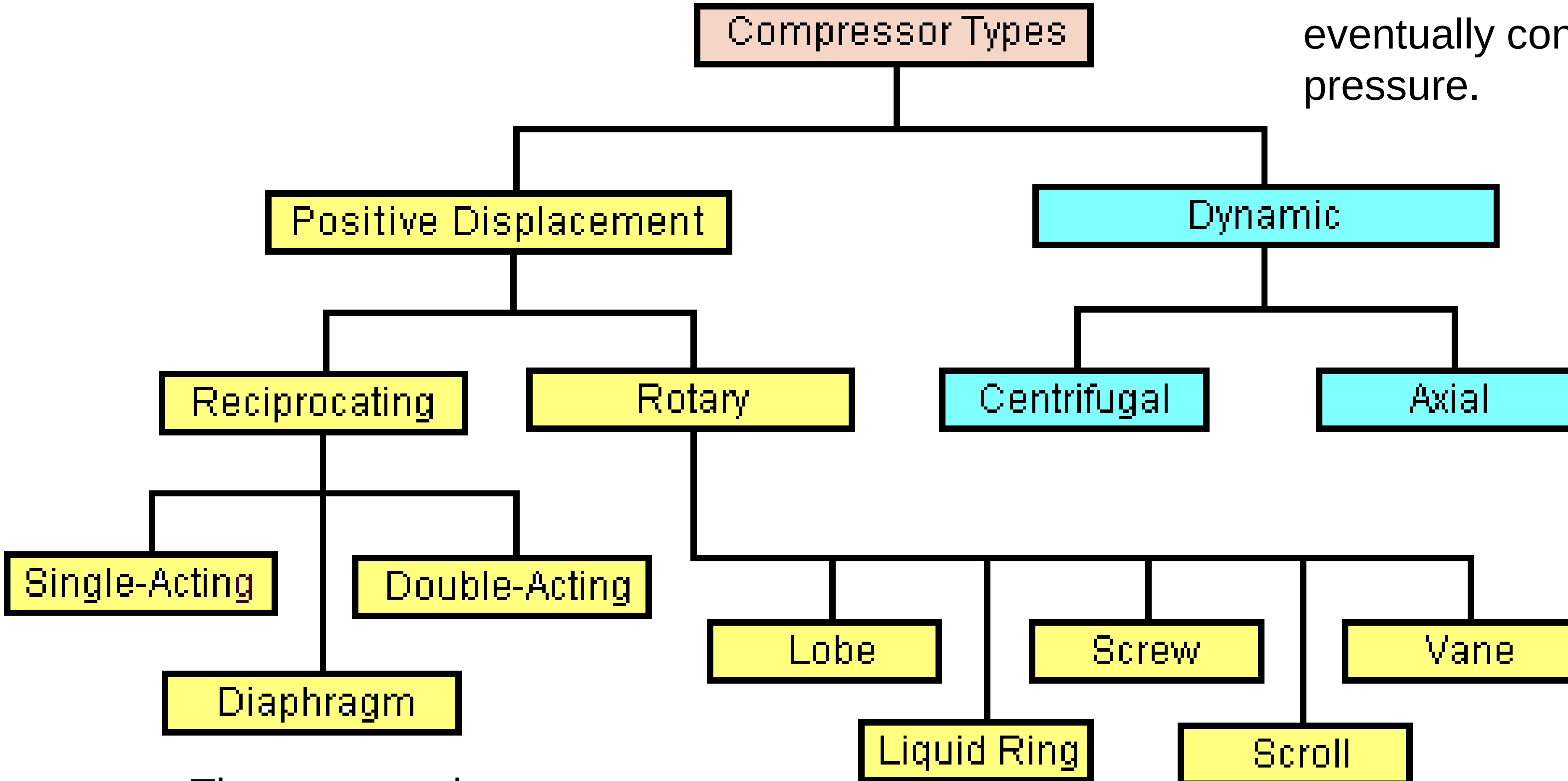
Types of compressors

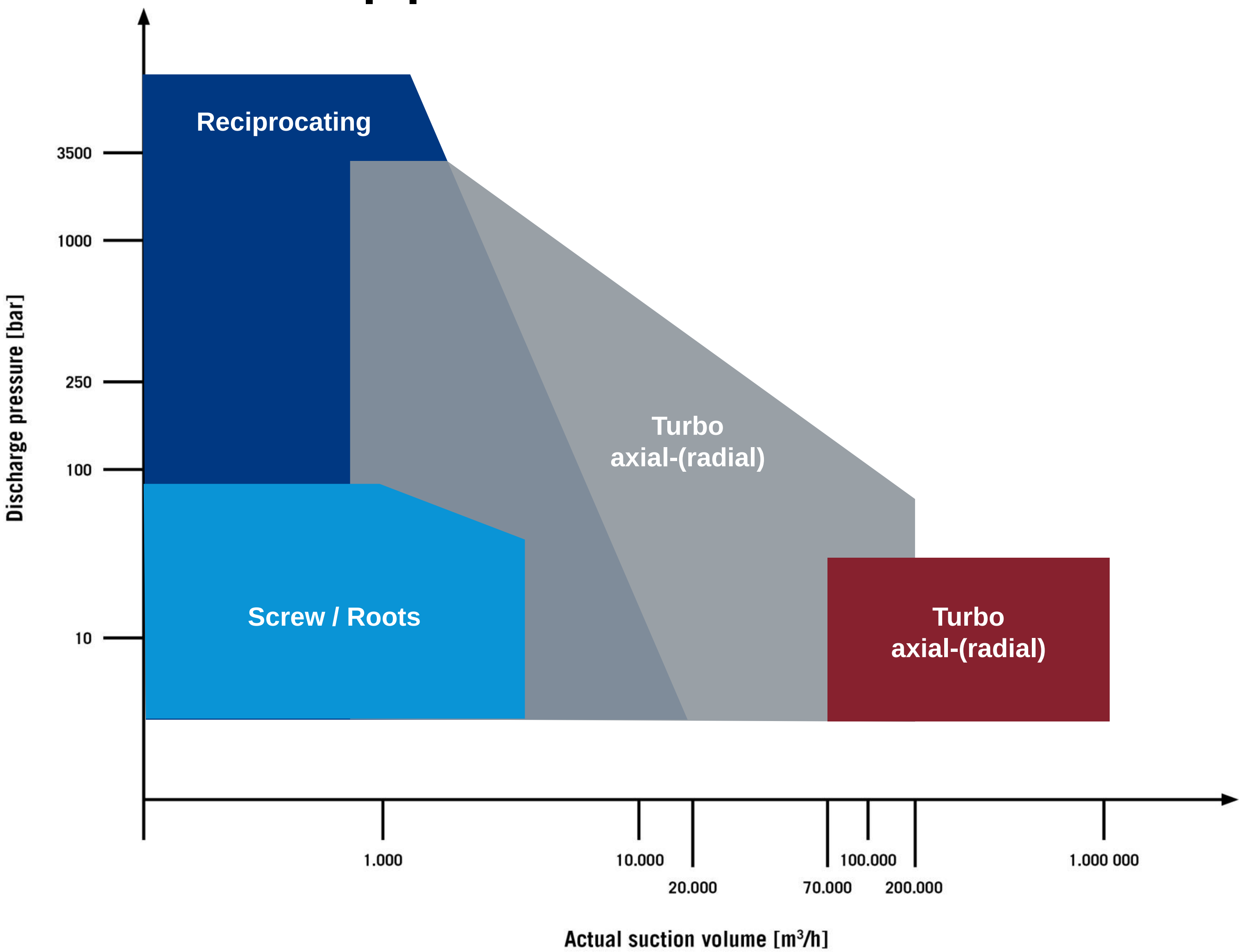
A basic overview

- Firstly the kinetic energy is increased and eventually converted into pressure.



- The pressure is increased by reduction of working volume

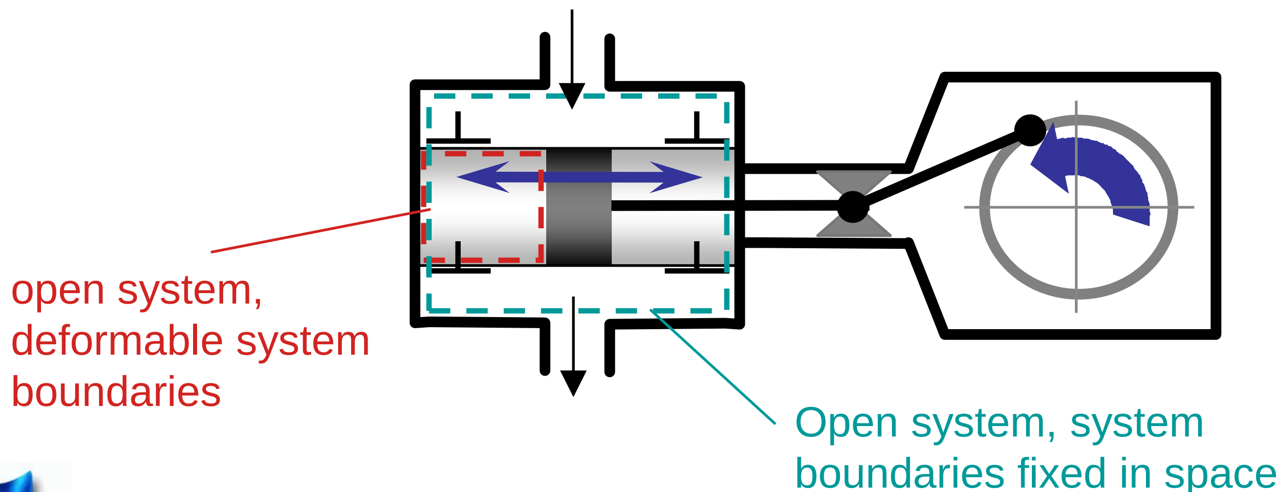
Application area



Thermodynamic systems

Separated by system boundaries (material or virtual) from surroundings (=everything except the system)

- Closed system....no transfer of matter across system boundaries
- Open system....transfer of matter across system boundaries
- Isolated system....no interaction with surroundings at all



Ideal Gases

Ideal gases …

- are defined by ideal gas law (p*V=R*T=const.)
- may be used to define the *absolute thermodynamic temperature*
- all real gases behave approximately as ideal gases for not too high pressures and temperatures $\Rightarrow$ ideal gas law is asymptotic limiting law for real gases

p ....absolute pressure [Pa]

V ....Volume

$\mathcal{R} = 8314 \ J/(kmol * K)$....universal gas constant

$R = \mathcal{R}/M$ ....special gas constant

M ....molar mass [kg/kmol]

T....absolute Temperatur [K]

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Pressure

- Definitions:

absolute pressure	bara
absolute pressure	psia
gauge pressure	barg
gauge pressure	psig

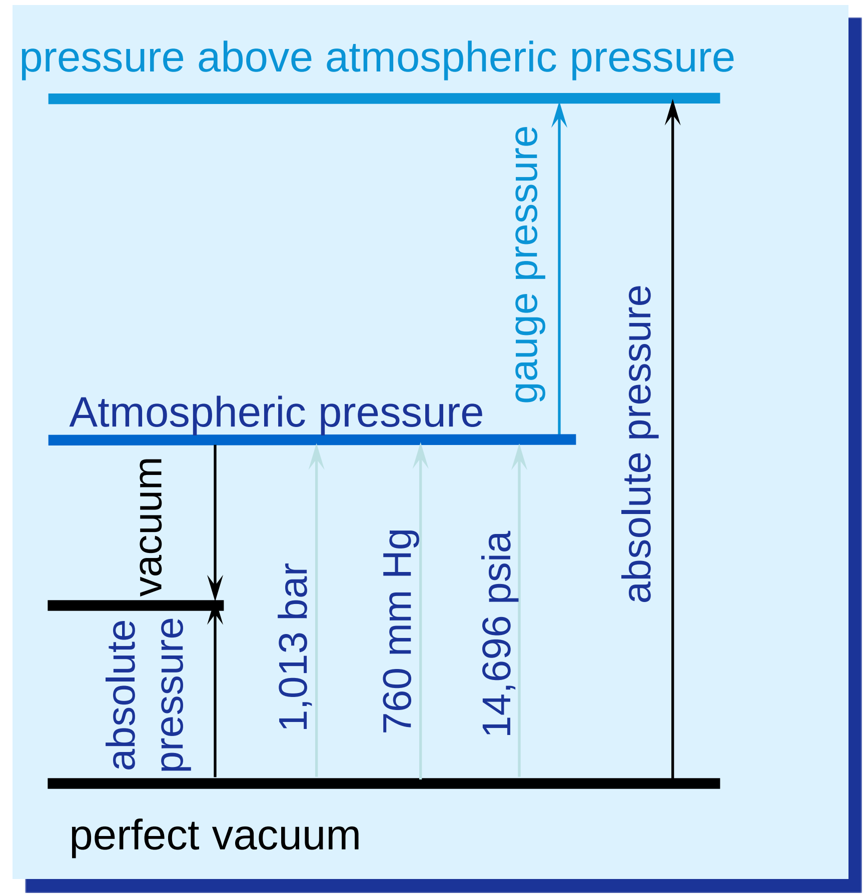
- Units:

1 Pa	= 1 Pascal	= 1 N/m²
1 bar	= 100.000	Pa
1 psi	= 6894	Pa

- Standard conditions:

atmospheric pressure

p_0 = 1,0133 bara



In thermodynamic calculations only the absolute pressure may be used!

Temperature

- Definitions:

  absolute temperature T K

  Celsius temperature $\vartheta_{°C}$ °C

  Fahrenheit scale $\vartheta_{°F}$ °F

- Units:

  $1\ K = \text{Kelvin}$ K

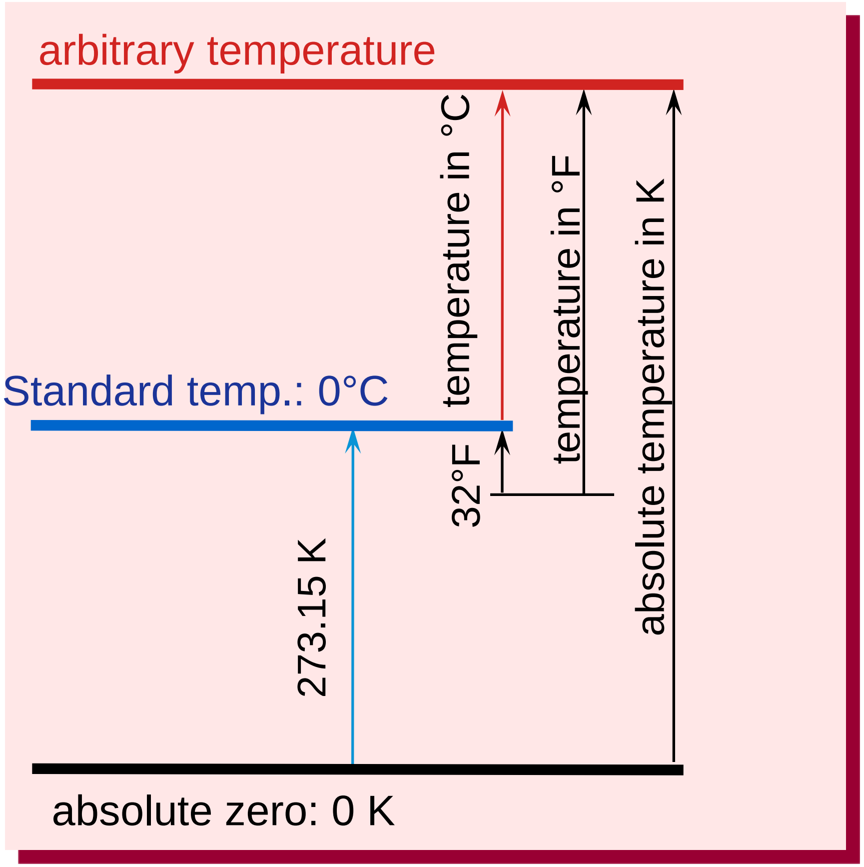
  $\vartheta_{°C} = T - 273.15$ °C

  $= (\vartheta_{°F} - 32°F) / 1{,}8$

  $\vartheta_{°F} = 1{,}8\ \vartheta_{°C} + 32$ °F

- Standard conditions:

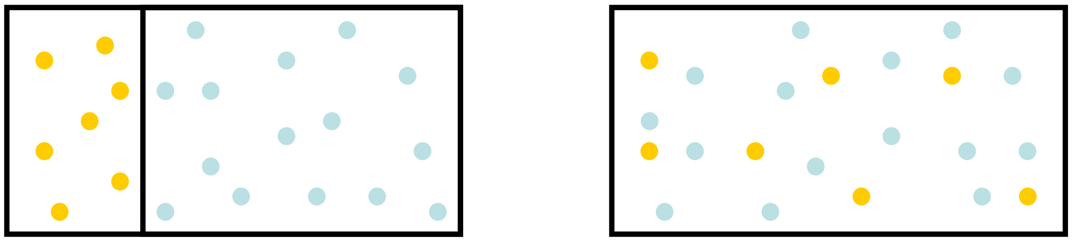
  T = 273.15 K = 0°C = 32°F



In thermodynamic calculations only the absolute temperature may be used!

Gas mixtures

Under consideration: Mixture of n gases i=1,2,...n



Definitions:

- Partial volume V_i....volume that gas i would occupy at temperature of mixture
- Partial pressure p_i.... pressure gas i would have if it alone occupied the volume V at the temperature of the mixture

Mixture of ideal gases:

- Pressure of mixture = sum of partial pressures ....„Dalton´s law“
- Volume of mixture = sum of partial volumes

Ideal gas mixture

Ideal gas law for components $p_i * V = n_i * \mathcal{R} * T$

summed over all components $\sum_{i=1}^{n} p_i * V = \sum_{i=1}^{n} n_i * \mathcal{R} * T$

with Dalton´s law and $m = \sum_{i=1}^{n} n_i * M_i$ it follows that $p * v = \frac{\mathcal{R}}{M} * T$

with the molar mass of the mixture $M = \sum_{i=1}^{n} \frac{n_i}{n} * M_i$

Measures for concentration:

mole fraction $y_i := n_i / n$

volumetric fraction $\alpha_i := V_i / \sum_{i=1}^{n} V_i$

$\alpha_i = y_i$ for ideal gas mixtures

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Ideal gas mixture

Example: Molar mass of dry air

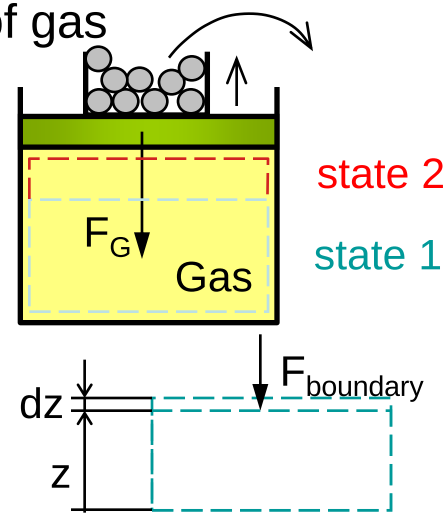
Dry air consists of approx. 78% N_2, 21%O_2 und 1%Ar (volume fraction). What is the molar mass of the air?

$$M = 0.78 * 2 * 14 kg/kmol + 0.21 * 2 * 16 kg/kmol$$
$$+ 0.01 * 40 kg/kmol = 29.0 kg/kmol$$

Energy balance….special cases of interest

Closed system at rest: $dU = d_e W + d_e Q \Leftrightarrow U_2 - U_1 = W_{12} + Q_{12}$

<u>Example:</u> Quasistatic, adiabatic (no heat transfer) expansion of gas



Work done on system....$d_e W = - F_{boundary} * dz$

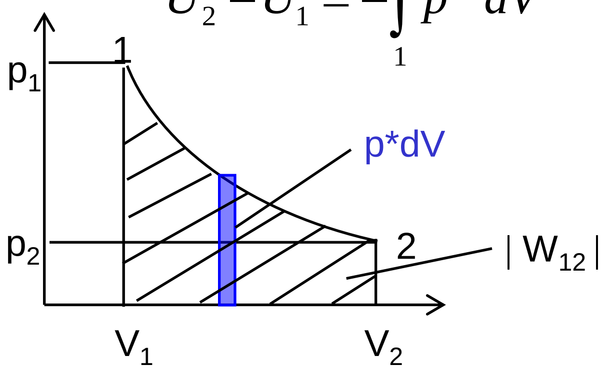
Quasistatic change of state:

- mechanic equilibrium piston....$F_{boundary} = F_G$
- pressure inside cylinder homogeneous.... $F_{boundary} = p*A$

$\Rightarrow$ quasistatic work $d_e W = - p*A*dz = - p*dV$

1st law

$$U_2 - U_1 = -\int_1^2 p * dV$$



Note:

- the work depends on the special path 1→2
- according to the 1st law, the work equals the change in the internal energy where the latter just depends on the states 1 and 2
- <u>solution:</u> further restriction of no heat transfer leaves only one quasistatic path $\Rightarrow$ *isentropic* change of state

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Summary of gas laws

$$p \cdot V = m \cdot R \cdot T$$

For any isentropic or adiabatic (no heat transfer in the process) change of conditions the relation of pressure and volume is following the equation:

$$p \cdot v^{\kappa} = const$$

Transformed to calculate the discharge temperature:

$$T_2 = T_1 \cdot \left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{\kappa}}$$

P … absolute pressure [Pa]

V … *volume [m³]*

m … *mass [kg]*

v … *spec. Volume [m³/kg]*
($v=1/\rho$ .. density)

R … *spec. gas constant [kJ / kgK]*

T … *absolute Temperature [K]*

κ ... *isentropic exponent [/]*

Cp …*spec. heat at const. pressure [kJ/kgK]*

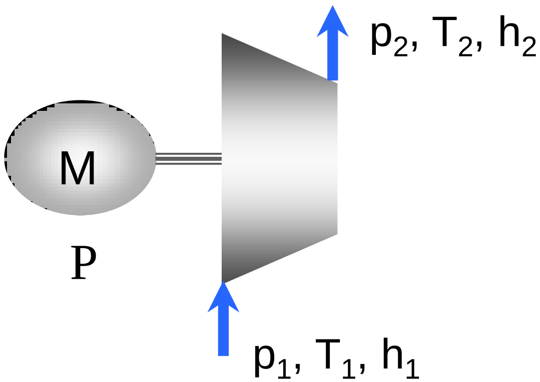
Cv … *spec. heat at const. volume [kJ/kgK]*

κ = Cp / Cv

1, 2 … *index for thermodynamic conditions at point 1 (inlet) and point 2(outlet)*

Adiabatic power consumption

Adiabatic compression of ideal gas:
(kinetic and pot. energy not considered)



$$0 = P + \dot{m} * (h_1 - h_2) \Rightarrow P = \dot{m} * c_p * (T_2 - T_1)$$

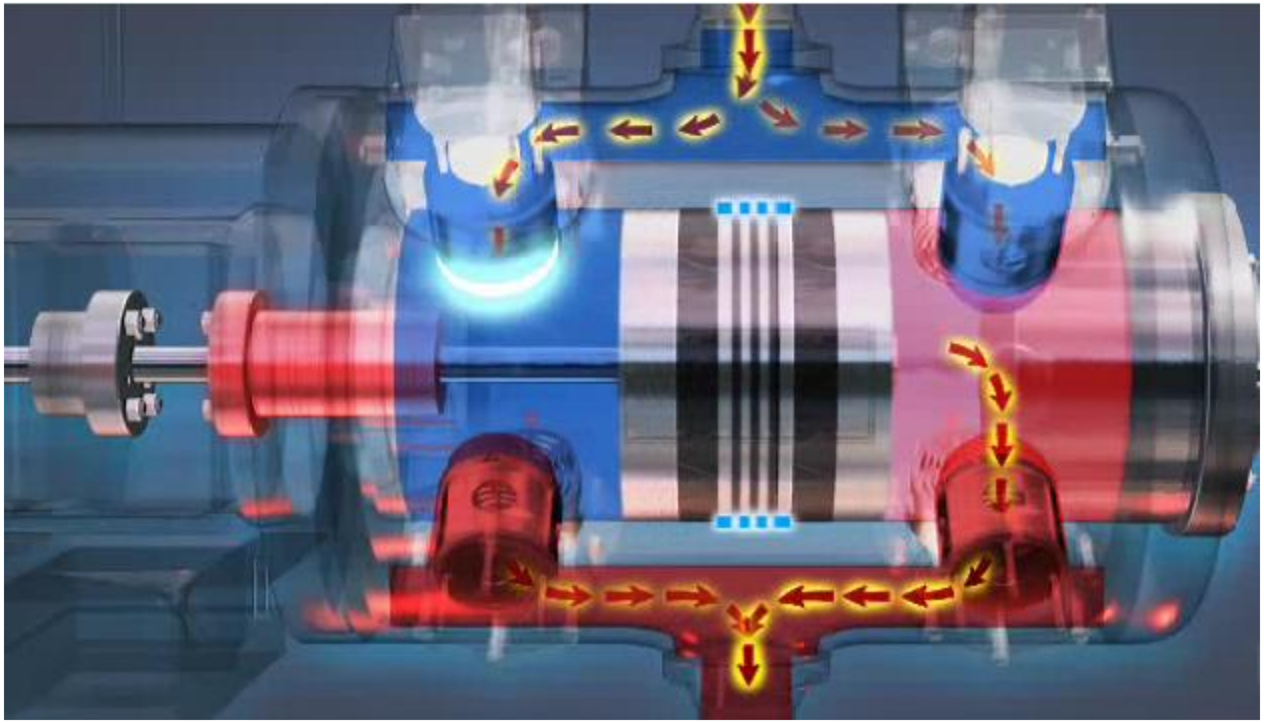
Reversible compression:

$$T_d = T_s * \left(\frac{p_d}{p_s} \right)^{\frac{\kappa - 1}{\kappa}}$$

$$\Rightarrow P = \dot{m} * c_p * T_s * \left[\left(\frac{p_d}{p_s} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right]$$

Note: losses increase discharge temperature and thereby power consumption!

Piston Compressors – Working principle



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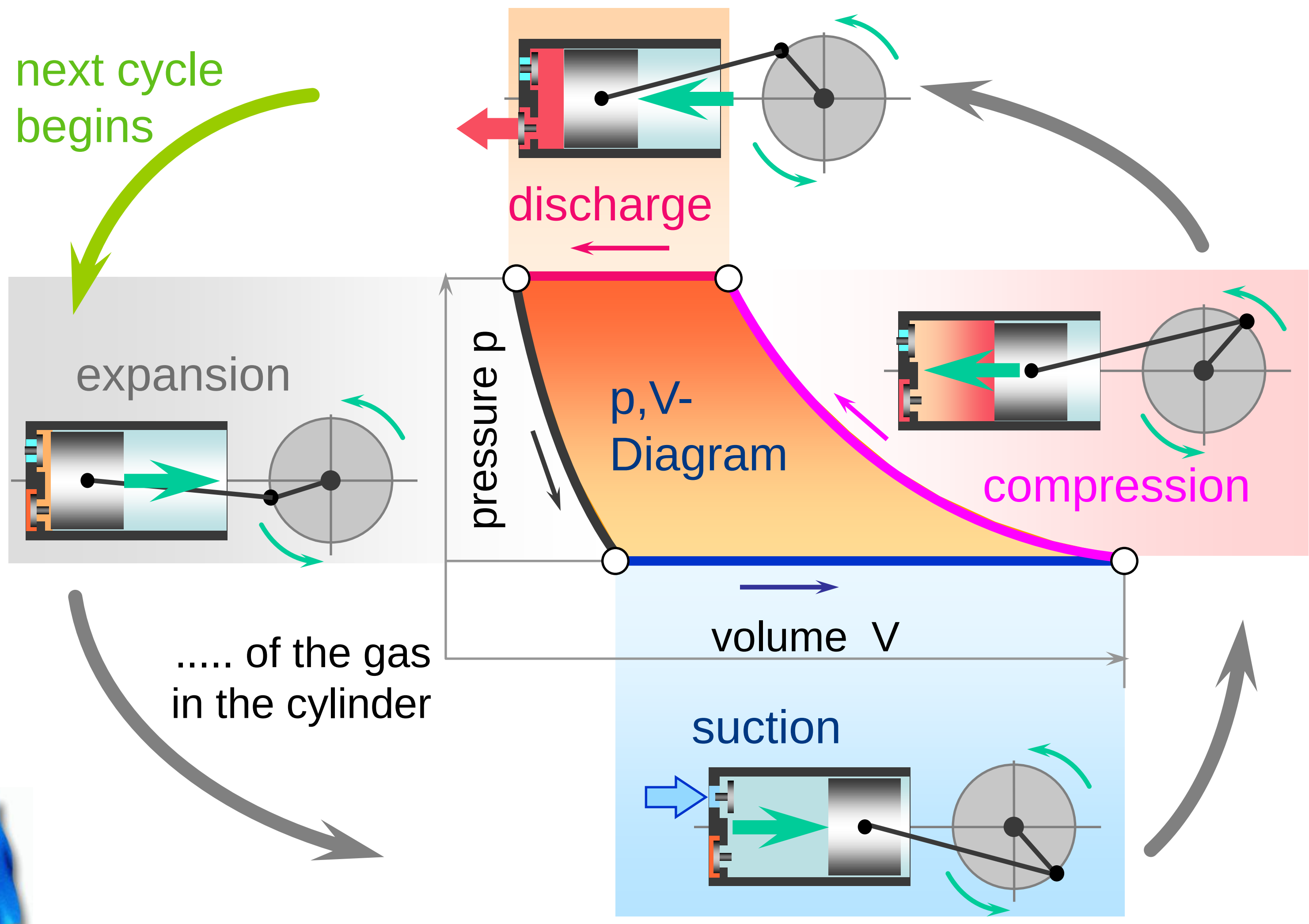
Single acting piston compressor pressure vs. volume diagram theory



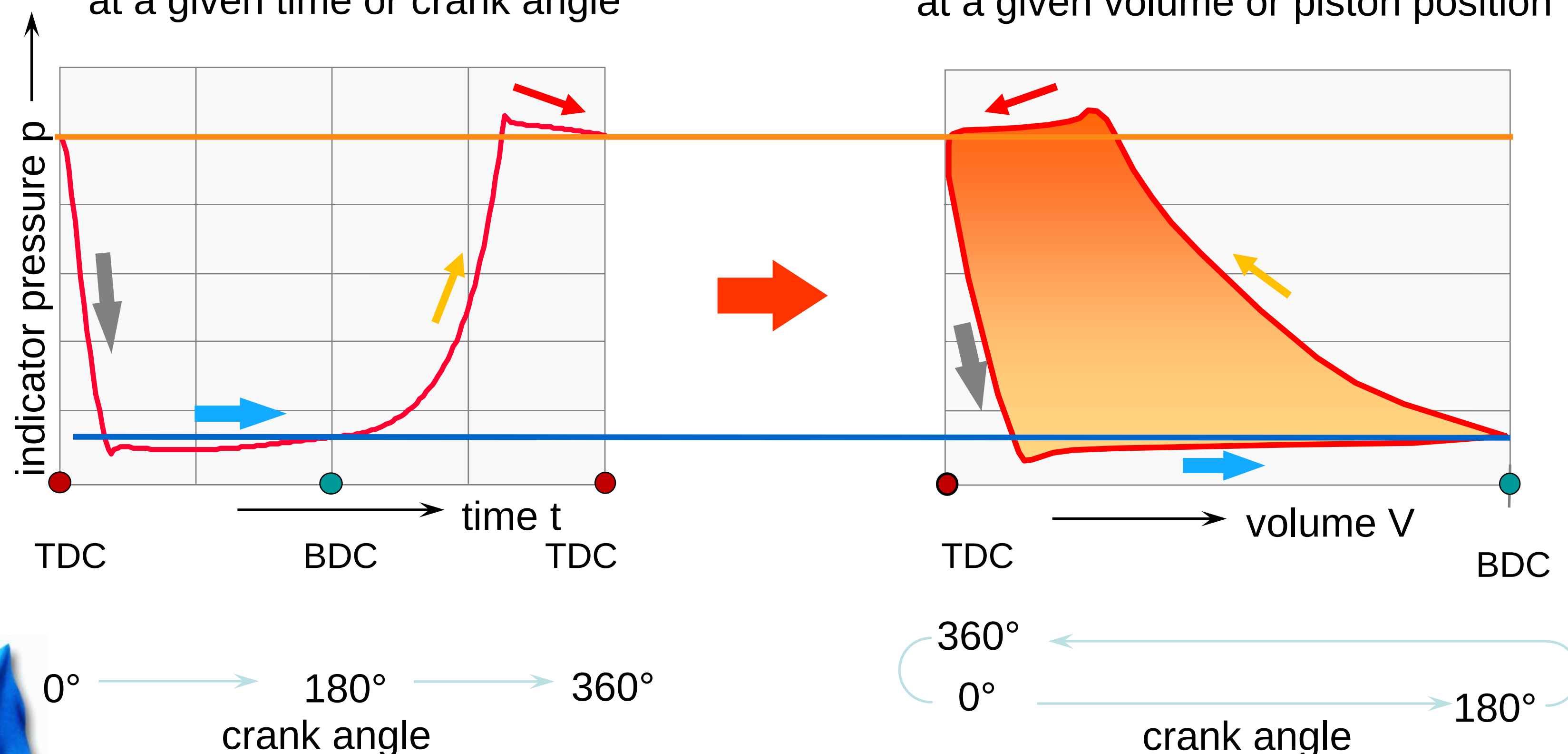
Single acting piston compressor pressure vs. volume diagram theory

Pressure - Time Diagram p,t

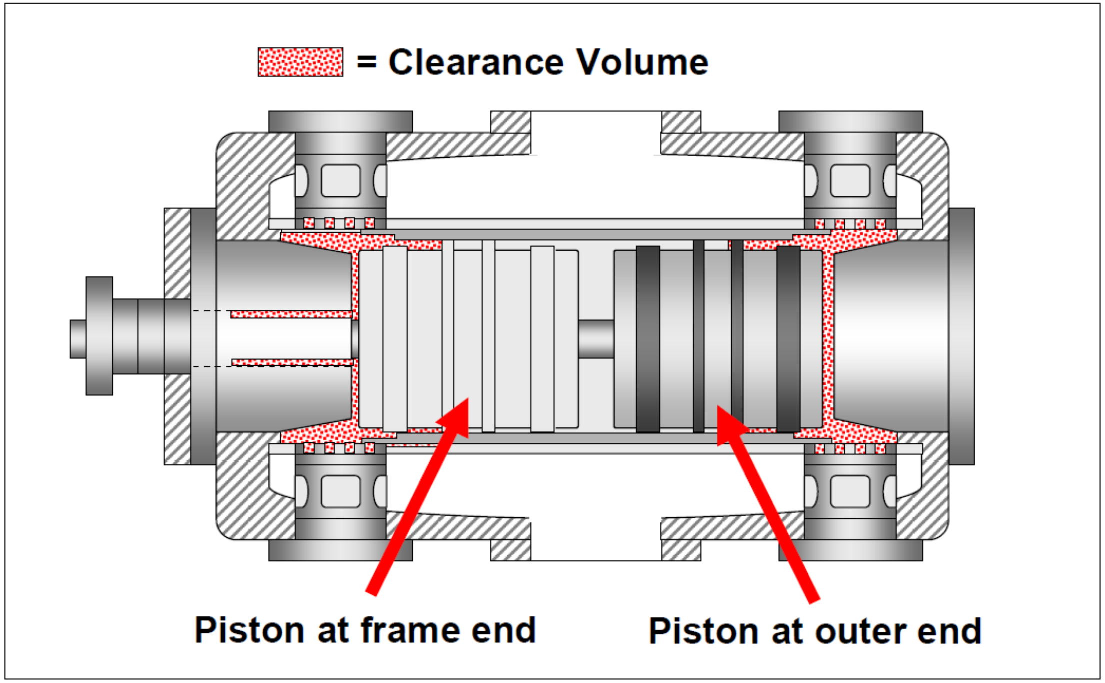
shows the pressure in the cylinder at a given time or crank angle

Pressure - Volume Diagram p,V

shows the pressure in the cylinder at a given volume or piston position



Clearance volume



•Clearance refers to the volumes in each end of the cylinder that retain gas after the piston has stopped

•Gas cannot be displaced; cylinder is not 100% efficient

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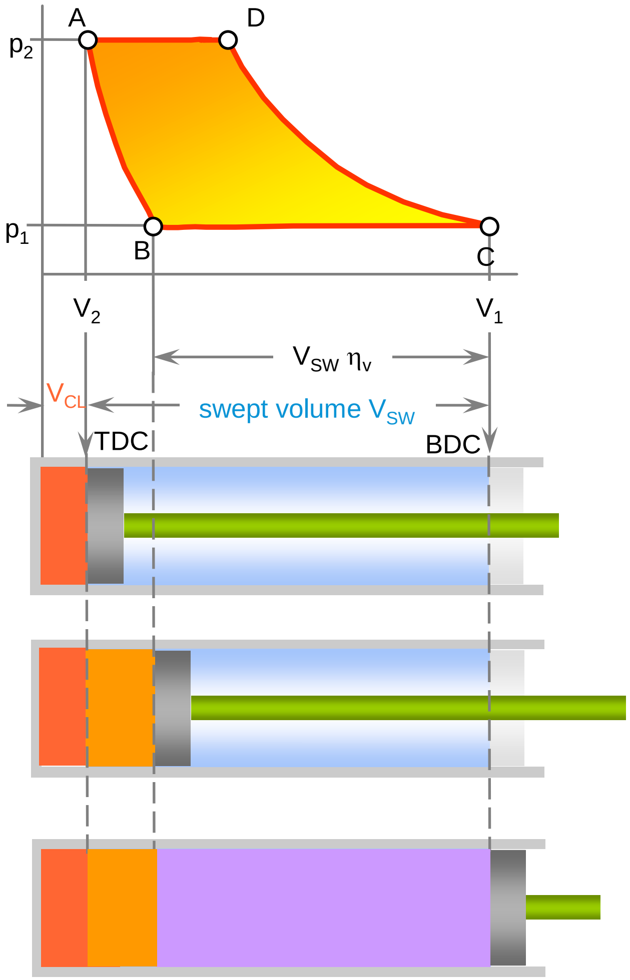
Volumetric efficiency



$$V_A = V_{CL}$$

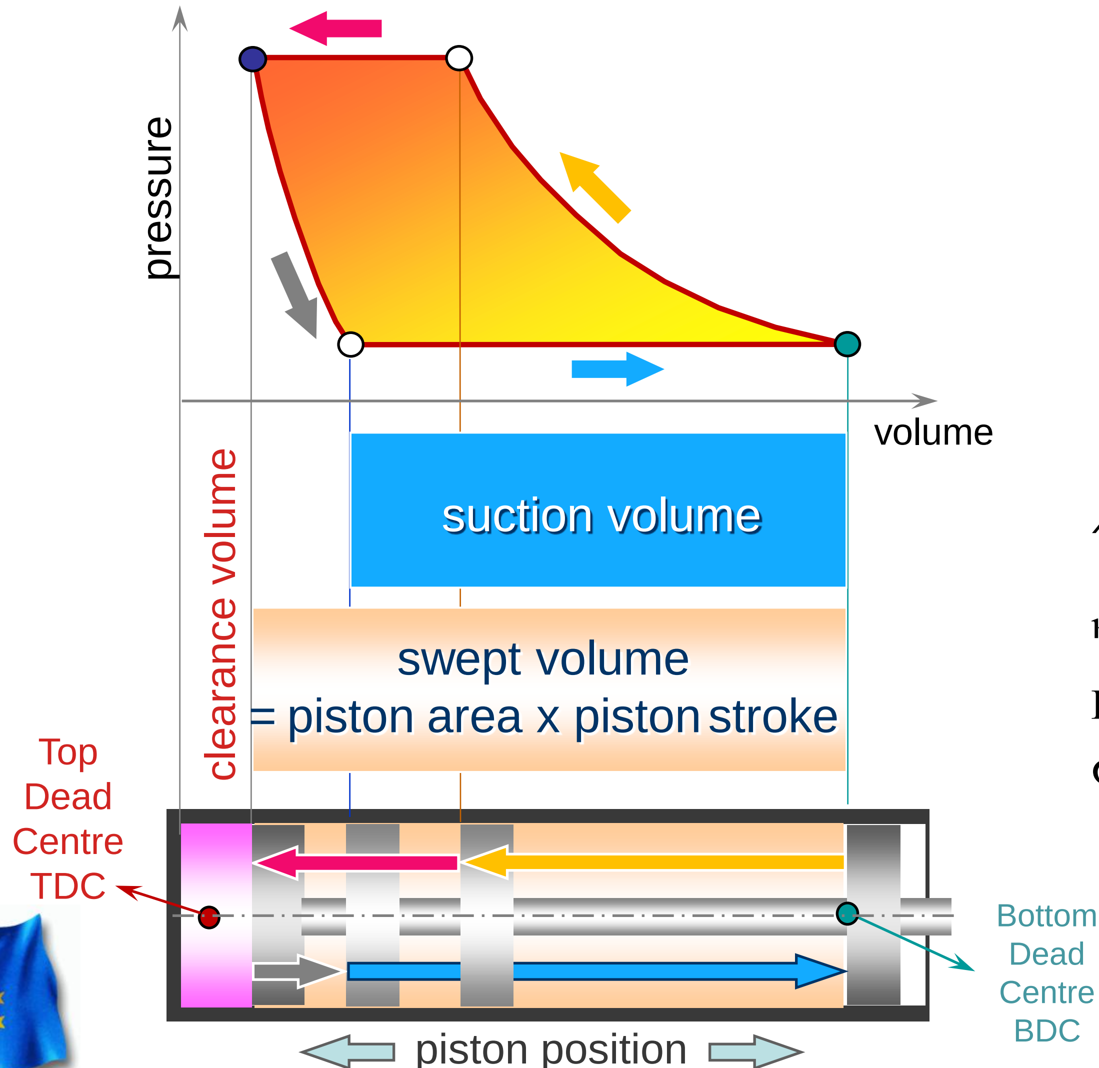
$$V_B = V_{CL}\left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}}$$

$$V_{SW}\eta_V = V_{SW} - (V_B - V_A)$$

$$V_{SW}\eta_V = V_{SW} - V_{CL}\left(\Pi^{\frac{1}{\kappa}} - 1\right)$$

$$\eta_V = 1 - \sigma_0\left(\Pi^{\frac{1}{\kappa}} - 1\right)$$

Single acting piston compressor pressure vs. volume diagram theory



Volumetric efficiency

$$\eta_v = \frac{\text{suction volume}}{\text{swept volume}}$$

$$\lambda = 1 - \sigma_0 (\Pi^{\frac{1}{\kappa}} - 1) - L$$

λ filling efficiency= volumetric efficiency η_V minus losses L

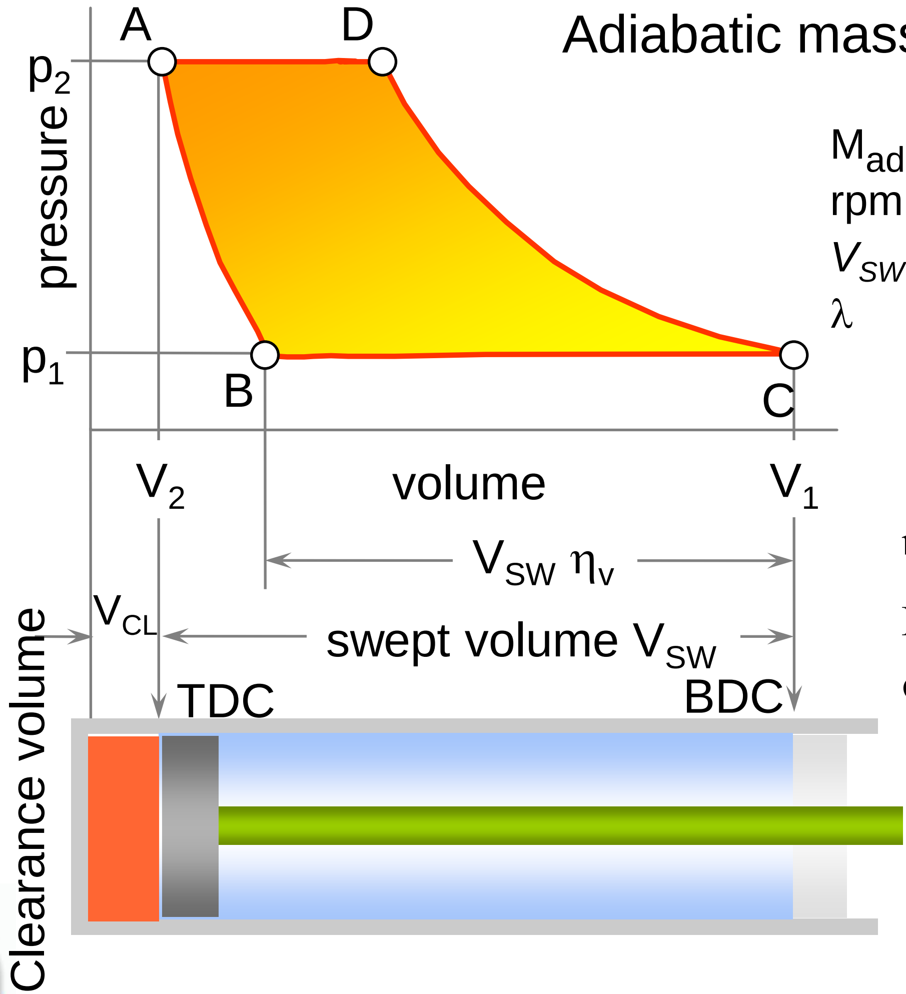
κ polytropic exponent

Π compression ratio p_2/p_1

σ_0 clearance volume referred to swept volume

L loss factor accounting for losses associated with leakage, heating up of gas during suction,….

Basic principles – mass flow



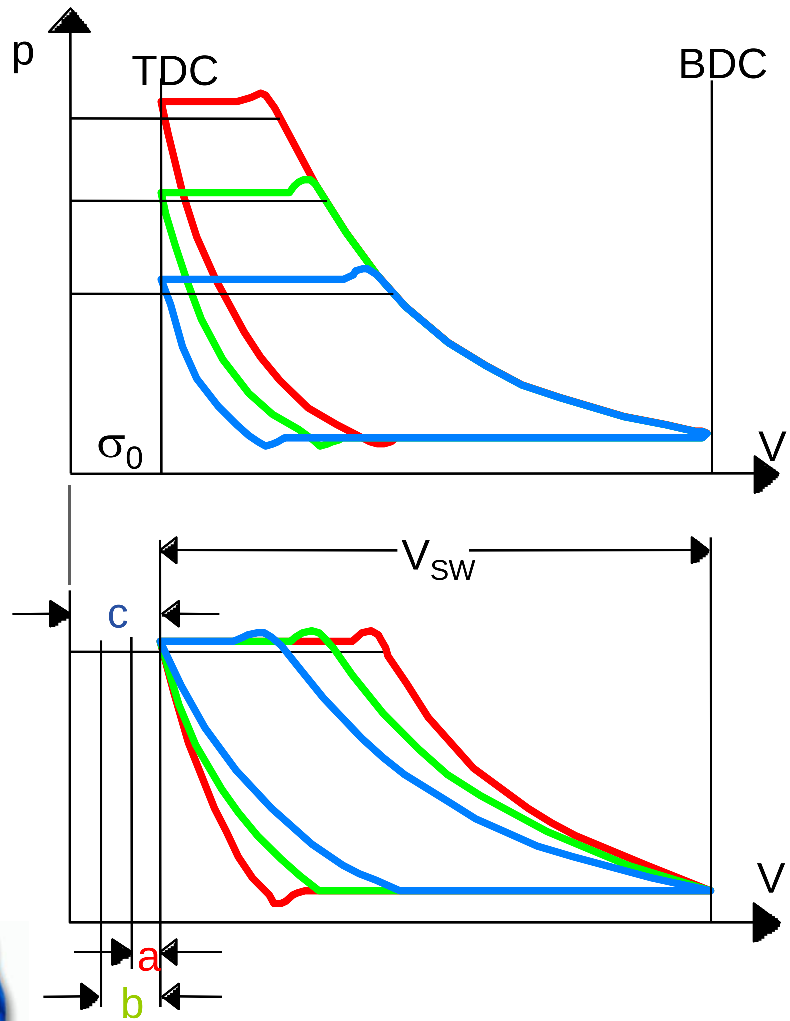
Adiabatic mass flow $M_{ad}=\rho_s{*}V_{SW}{*}\lambda{*}rpm/60$

M_{ad}	adiabatic mass flow (kg/s)
rpm	speed (min^{-1})
V_{SW}	swept volume
λ	filling efficiency= volumetric efficiency η_V minus losses L

$$\lambda = 1 - \sigma_0 (\Pi^{\frac{1}{\kappa}} - 1) - L$$

κ	polytropic exponent
Π	compression ratio p_2/p_1
σ_0	clearance volume referred to swept volume
L	loss factor accounting for losses associated with leakage, heating up of gas during suction,….

p-V Diagram - Influence of pressure ratio p2/p1 and clearance volume σ_0



Influence of pressure ratio p2/p1 on volumetric efficiency:

4, 6, 8

clearance σ_0 constant 10%V_{SW}

Influence of clearance σ_0 on vol. efficiency:

a: 6%,

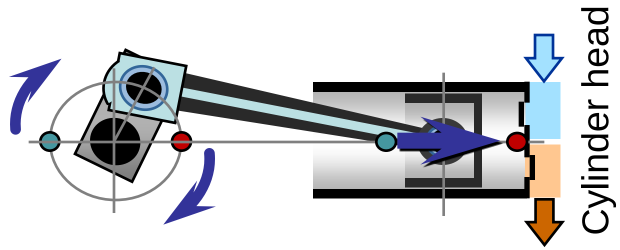
b: 10%,

c: 15 % of V_{SW}

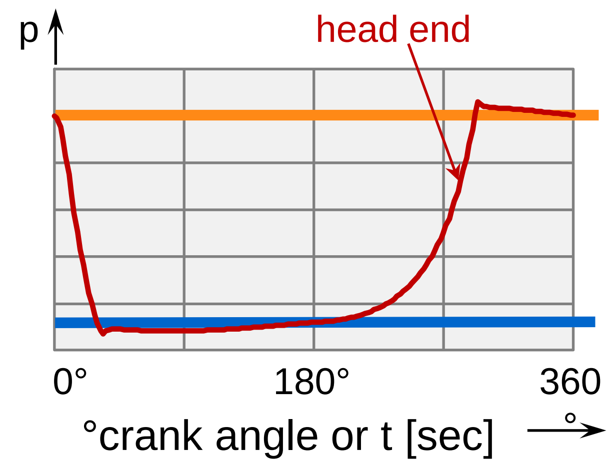
Pressure ratio p_2/p_1 constant

Single versus double acting compressor

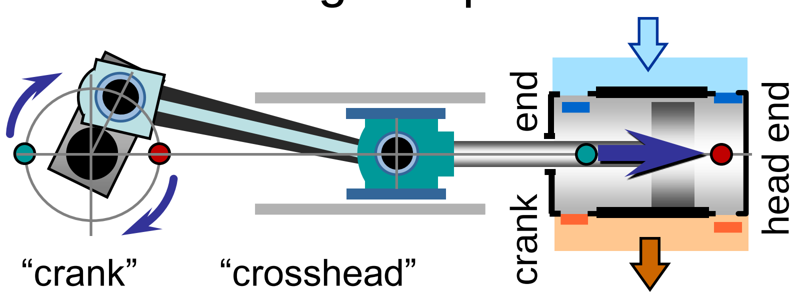
Single-acting compressor



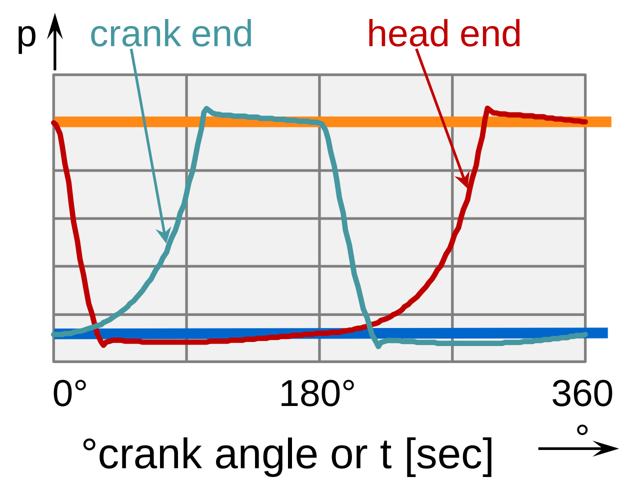
drive cylinder



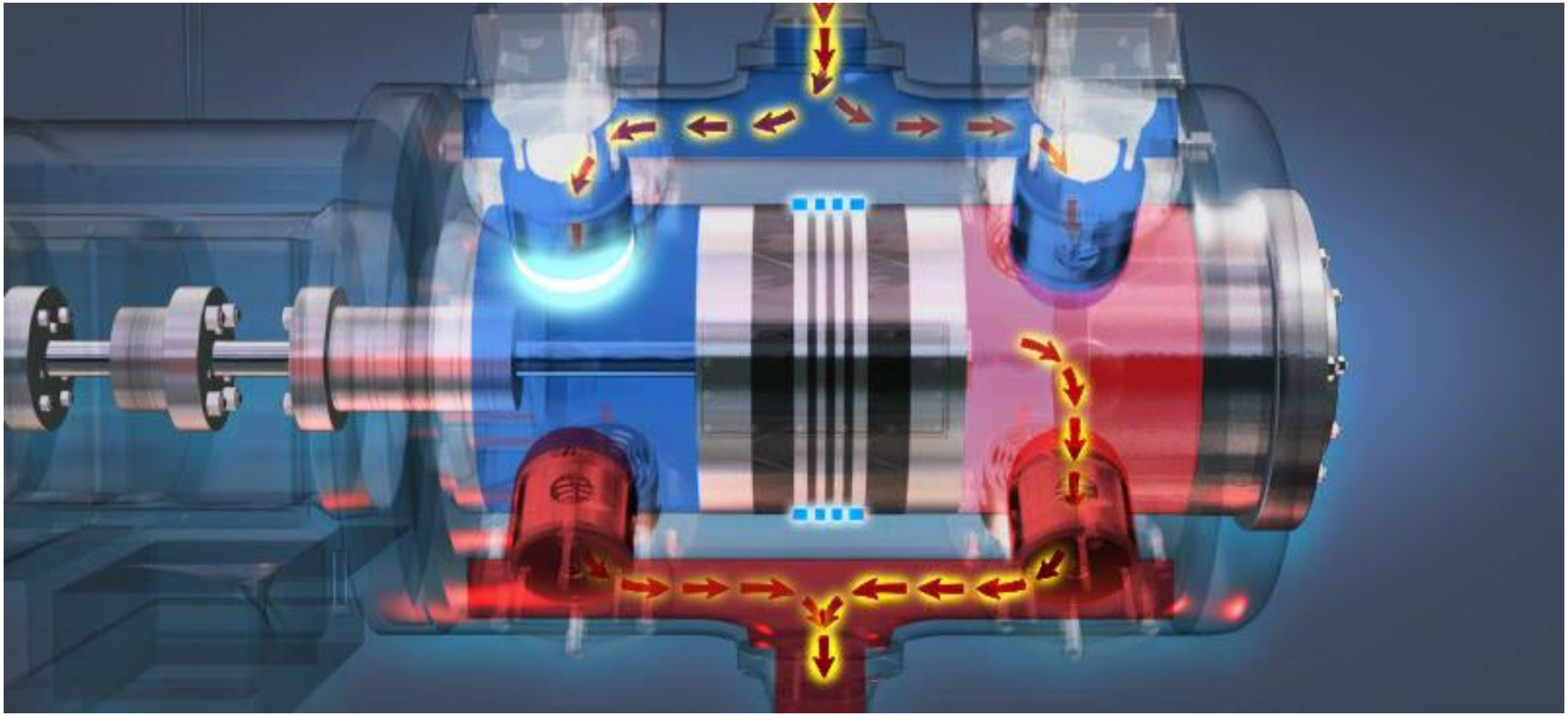
Double-acting compressor



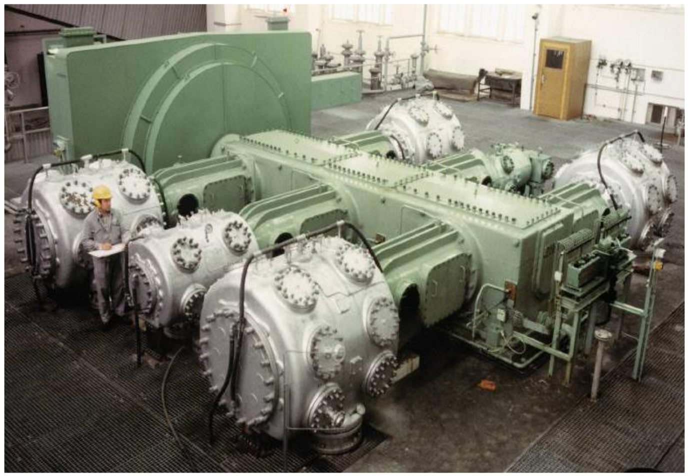
drive cylinder

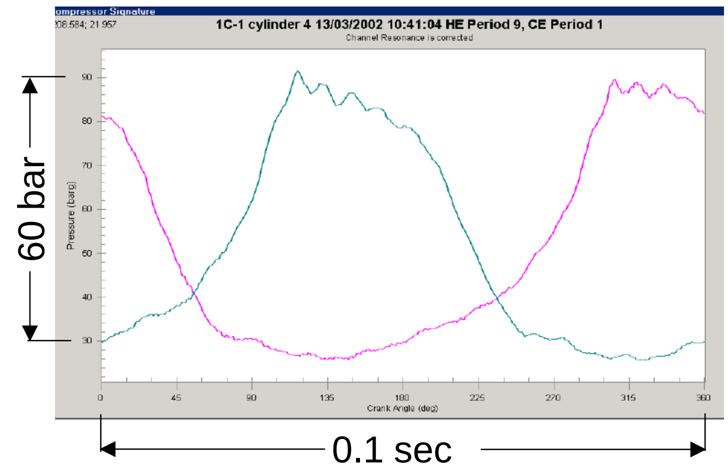


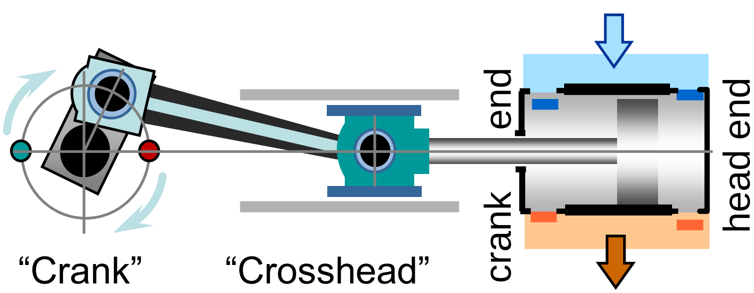
Double acting piston compressor



Double acting piston compressor pressure vs. volume diagram theory

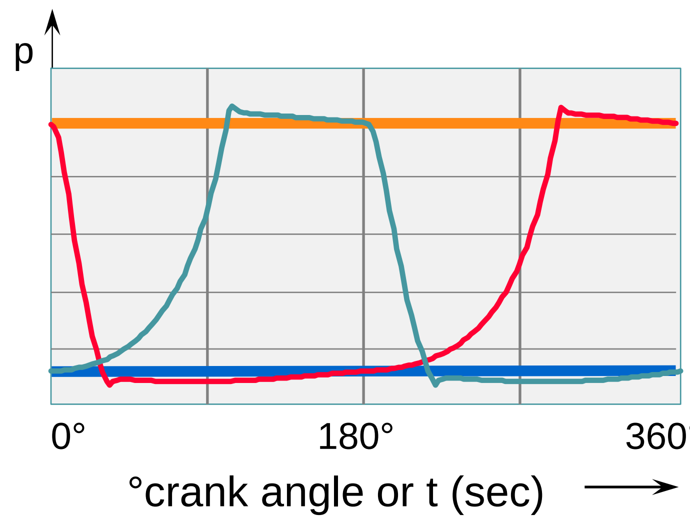






drive

cylinder



Compressor Limitations
Compression ratio

The equation $T_2 = T_1 \cdot \left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{\kappa}}$ limits the pressure ratio in one stage as neither material nor gas may stand such a high discharge temperature value.

Practice shows, that reasonable maximal pressure ratios per stage are:

$p_2/p_1 = 5$ for polyatomic gases with $\kappa = 1{,}3$ (natural gas, CO_2 etc. …..)
$p_2/p_1 = 4$ for diatomic gases with $\kappa = 1{,}4$ (air, N_2, H_2, CO etc. …..)
$p_2/p_1 = 3$ for monoatomic gases $\kappa = 1{,}67$ (He, Ne, Xe, Ar etc. …..)

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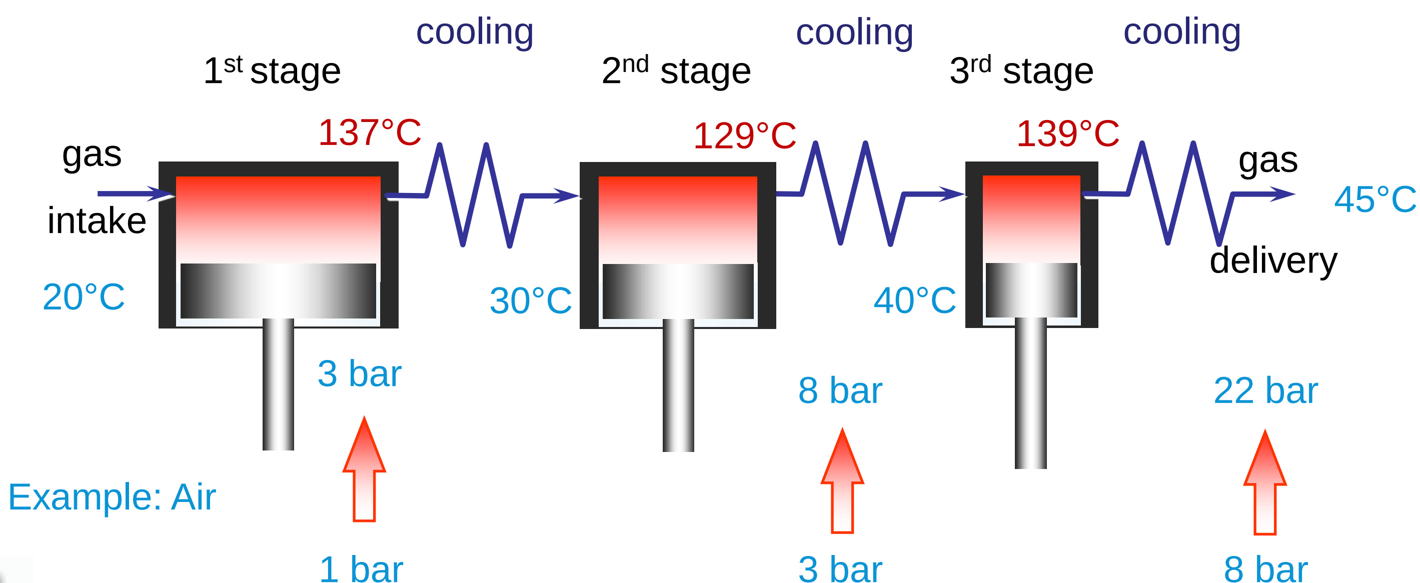
Multistage compressor

Function:

- Achieve higher pressures - up to several hundred bars in high pressure and ultra-high pressure compressors.
- Needs to be cooled down between stages in order to avoid exceeding permissible temperature for compressor materials and lubricating oil. Cooling significantly saves energy.

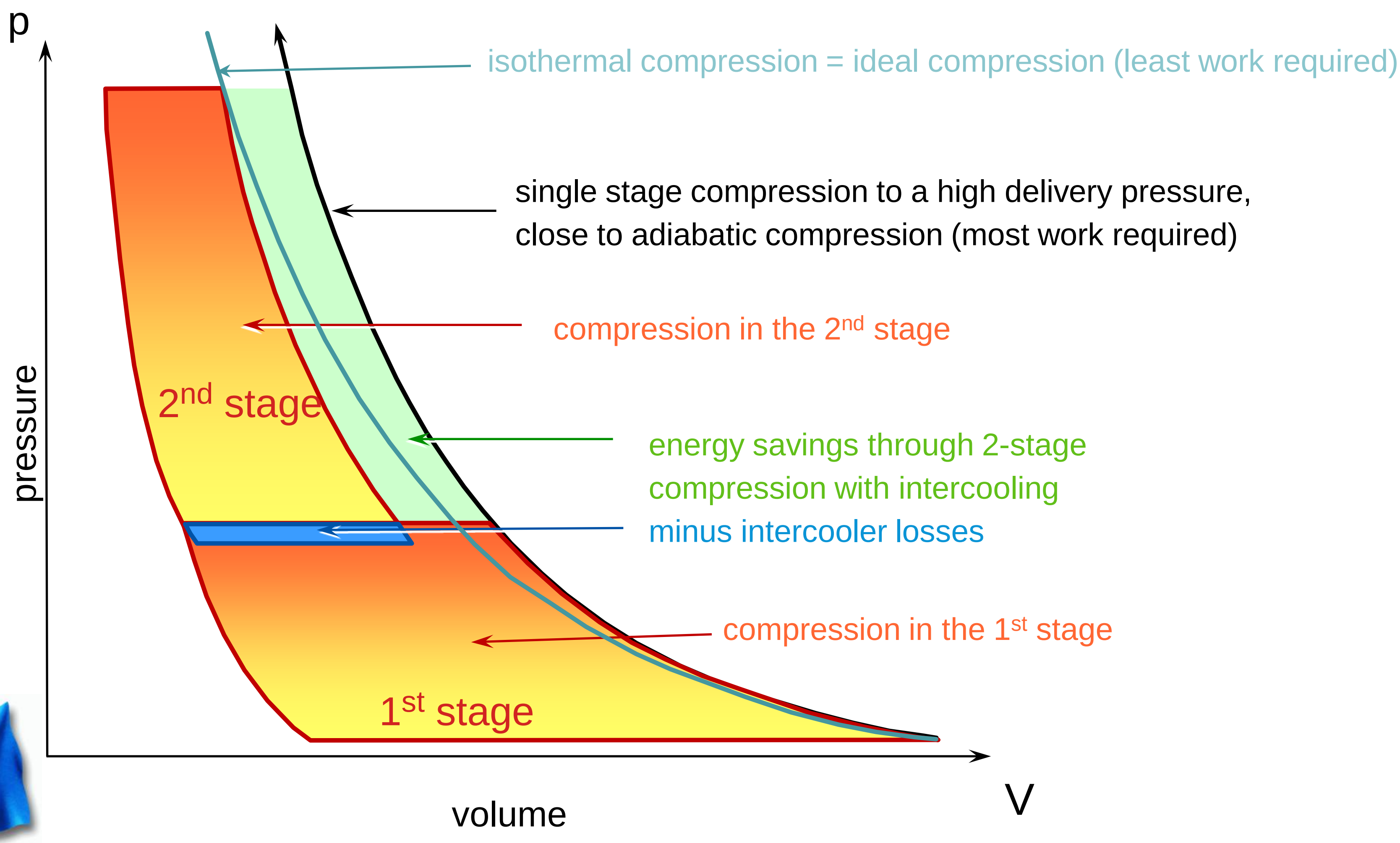


Discharge temperature: $T_d = T_s * \left(\frac{p_d}{p_s}\right)^{\frac{\kappa-1}{\kappa}}$

Pd … Discharge pressure
Ts … Suction temperature
PS … Suction pressure
k … Isentropic coeff.

Multistage compressor

Energy saving explanation:



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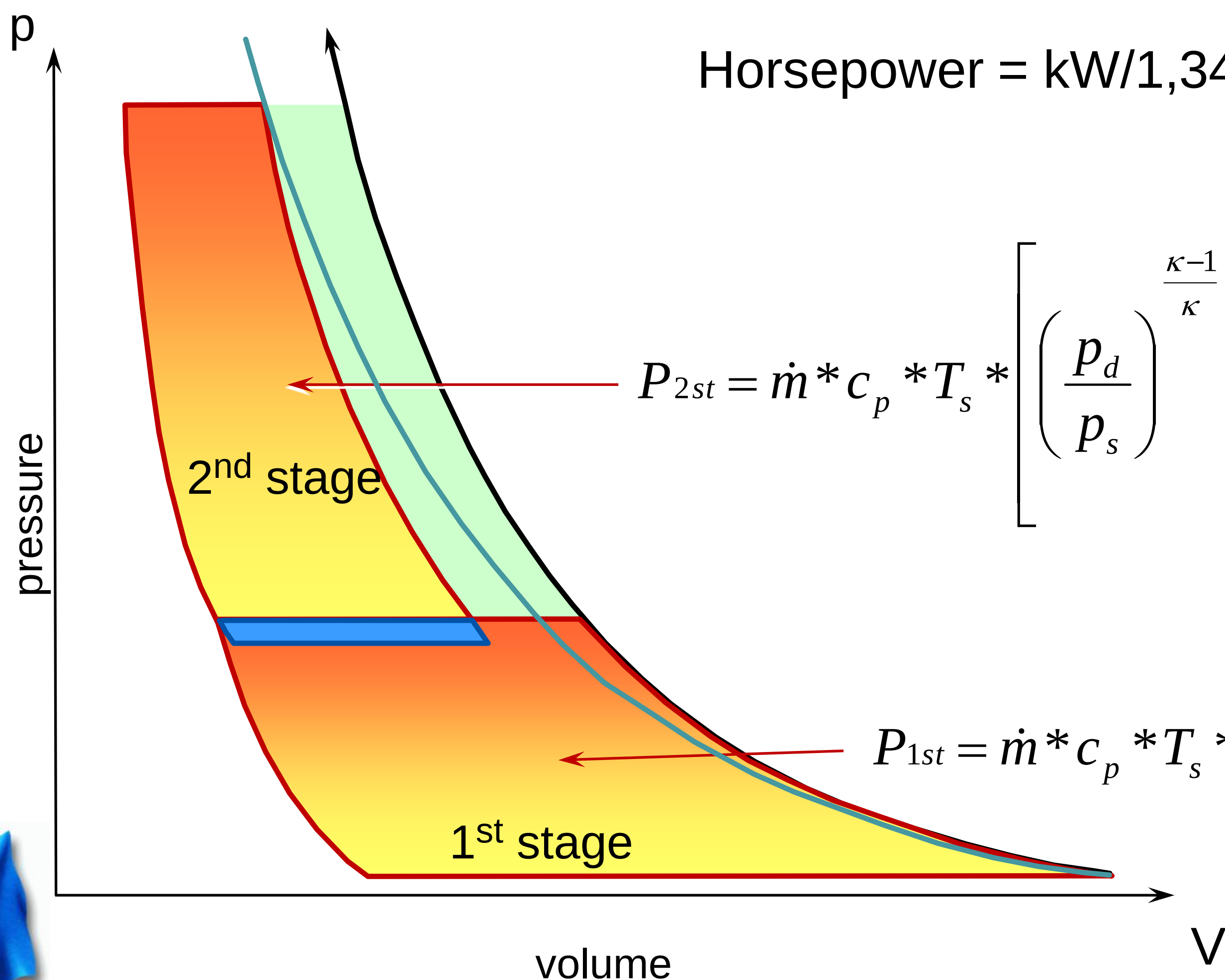
Multistage compressor

Power [kW] = $P_{1st} + P_{2st} + \dots$

Horsepower = kW/1,341

$$P_{2st} = \dot{m} * c_p * T_s * \left[\left(\frac{p_d}{p_s} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right]$$

$$P_{1st} = \dot{m} * c_p * T_s * \left[\left(\frac{p_d}{p_s} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right]$$



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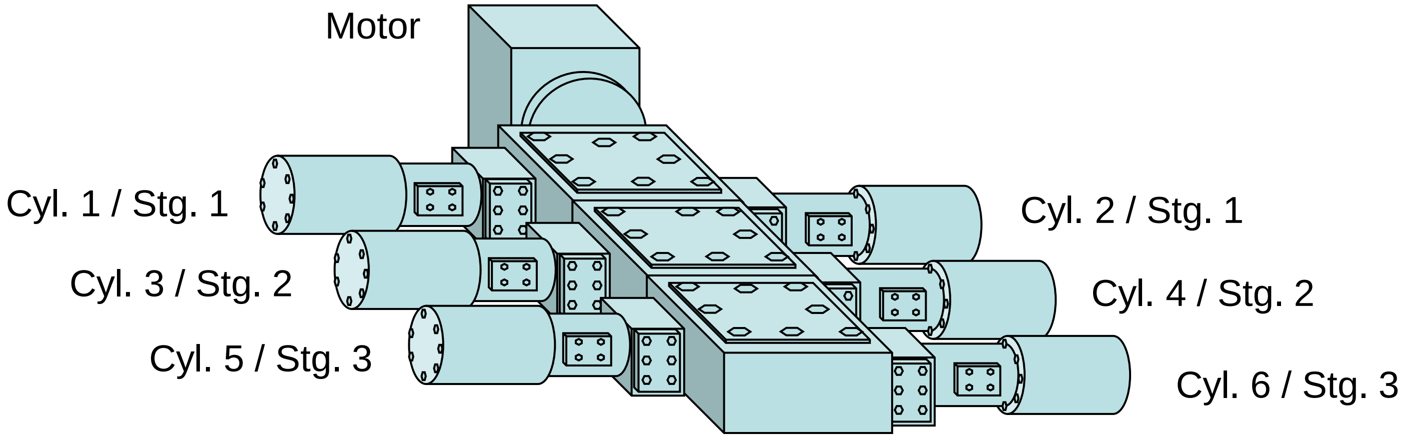
Example of multi-staging compressor

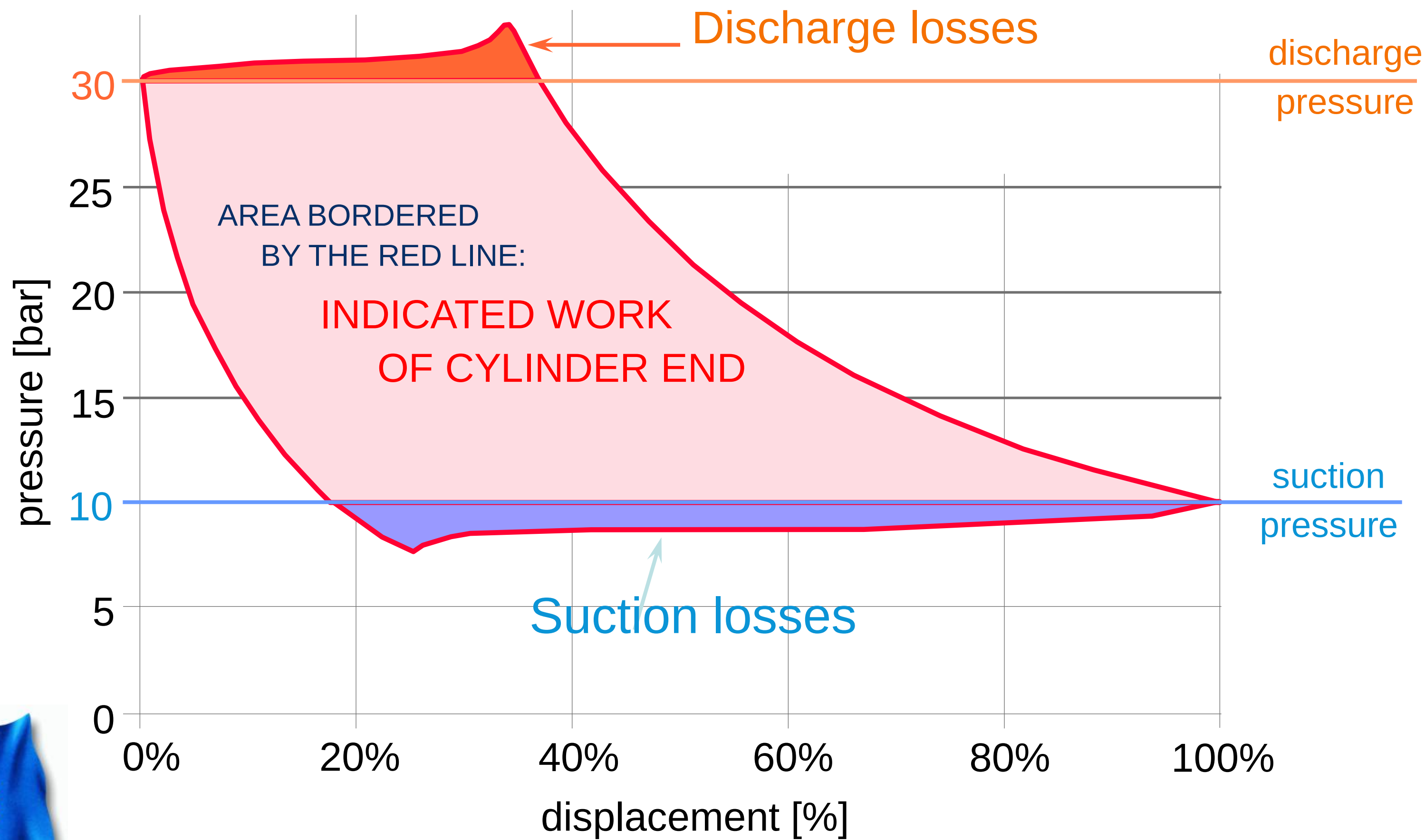
6 Cylinders - 3 stages compressor layout:

(Stg. … Stage, Cyl. … Cylinder)



Valve losses
pressure vs. volume diagram theory

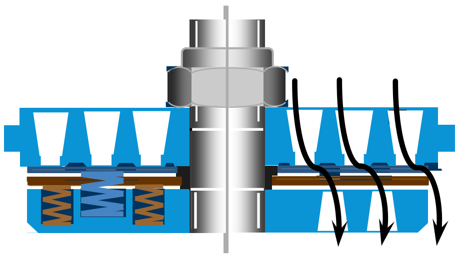
A certain % of compressor work is lost due to losses (suction and discharge



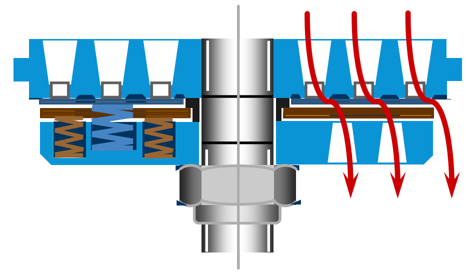
Valve loss

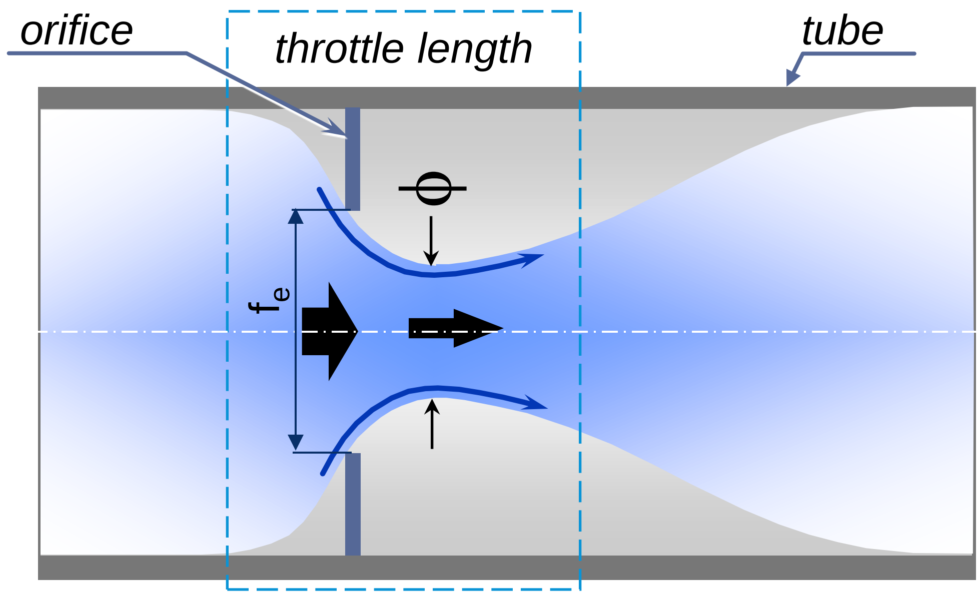
Valve loss:

suction valve = intake



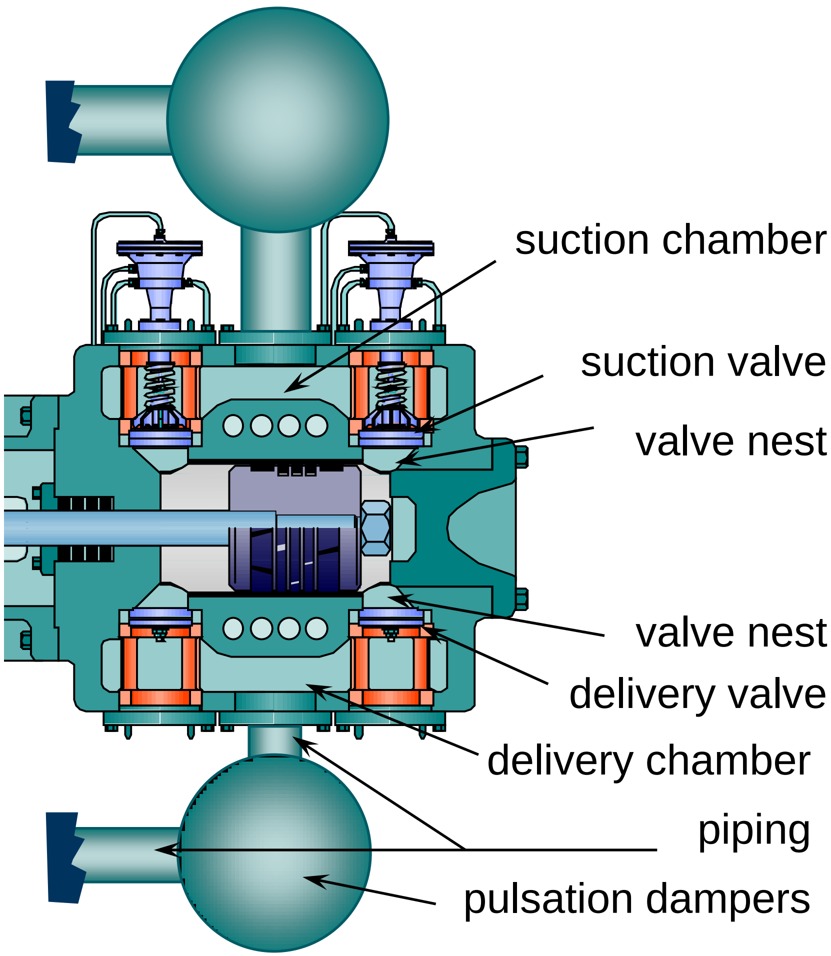
delivery valve = outlet





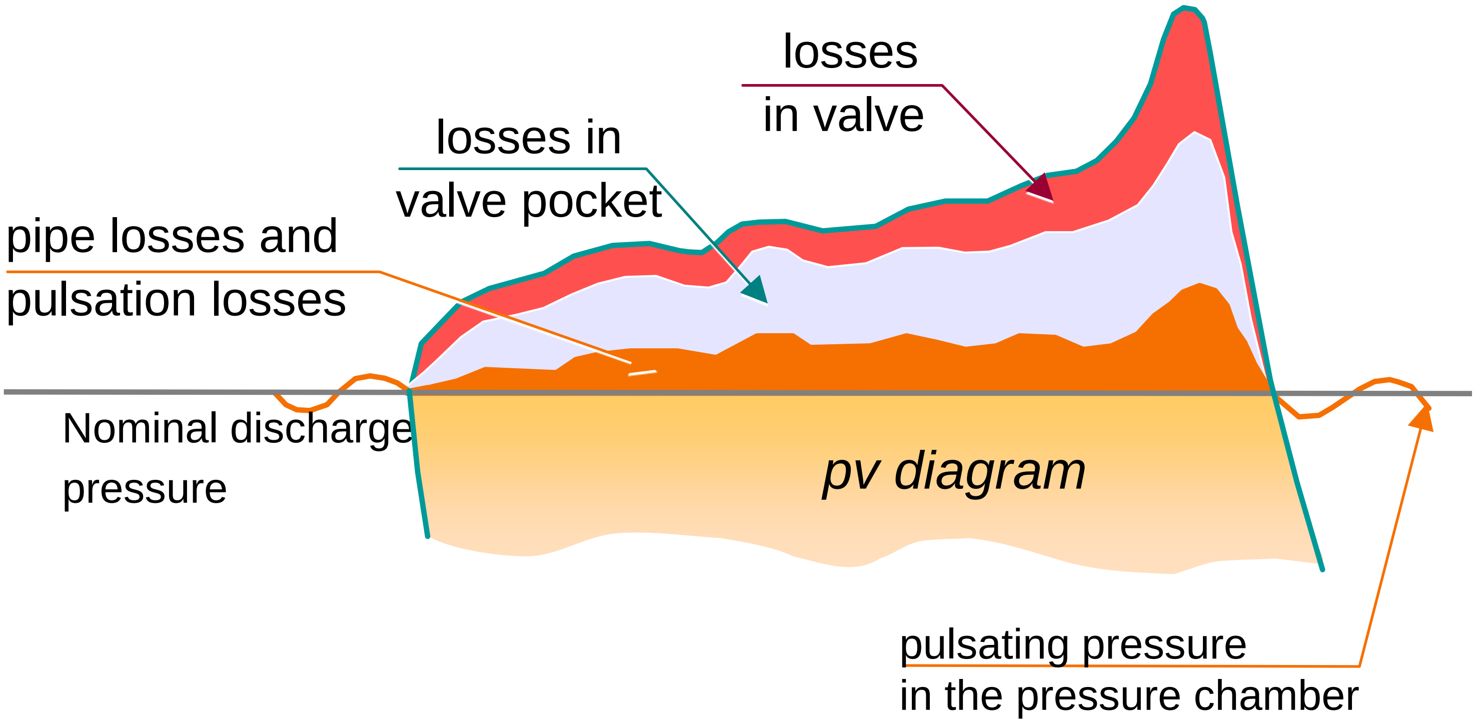
Ventilation loss

Ventilation losses are the overall losses incurred in:



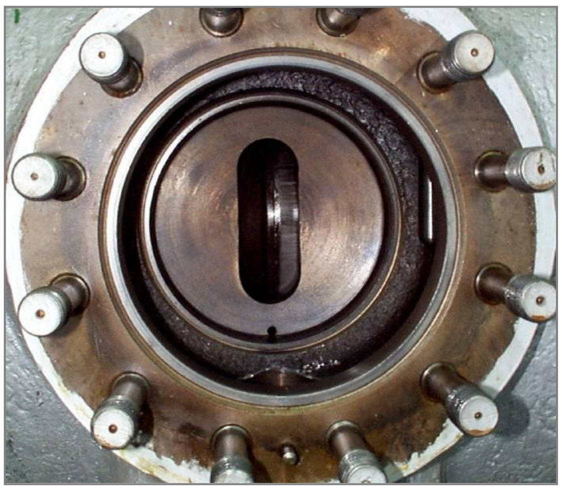
Valve & Ventilation losses

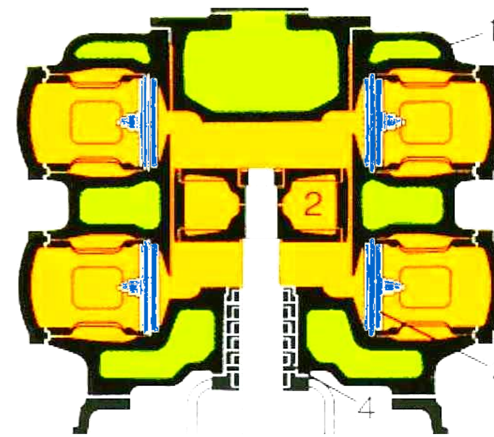
The areas exceeding nominal discharge pressure show the different losses at the delivery side.



Shape of valve pocket and valve losses

The shape of the valve pocket in the cylinder has a considerable influence on total gas flow losses. In the calculation of total flow losses, the valve nest shape can be taken into account with ‘Pocket Factors’.





Low losses in the cylinder pockets can be achieved by minimizing restriction of gas flow into cylinder,

- by wide passage areas
- by pockets or recesses in the cylinder head

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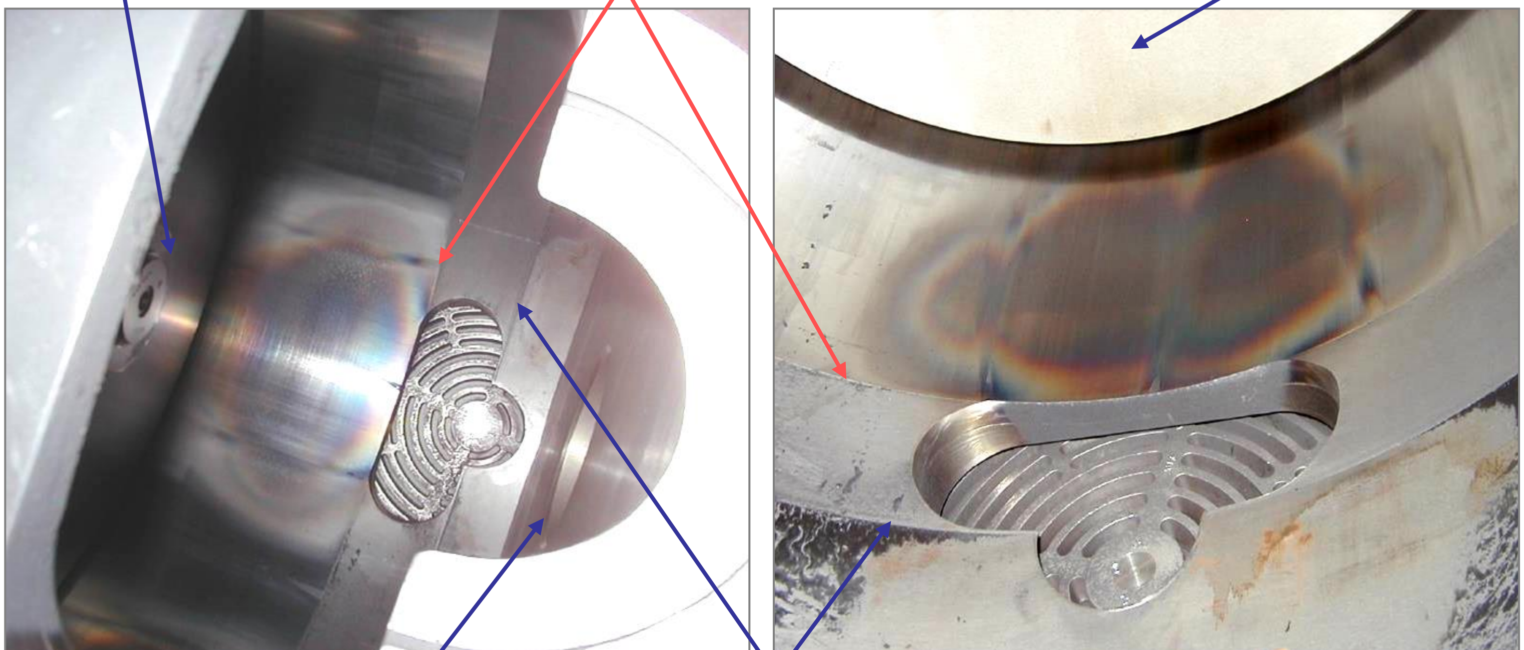
Shading of valve nest by cylinder head

Narrow ports and gaps are restricting the gas flow too much

retracted piston

end of contact area of piston rings

retracted piston



cylinder head, retracted

cylinder head protrudes until here into cylinder

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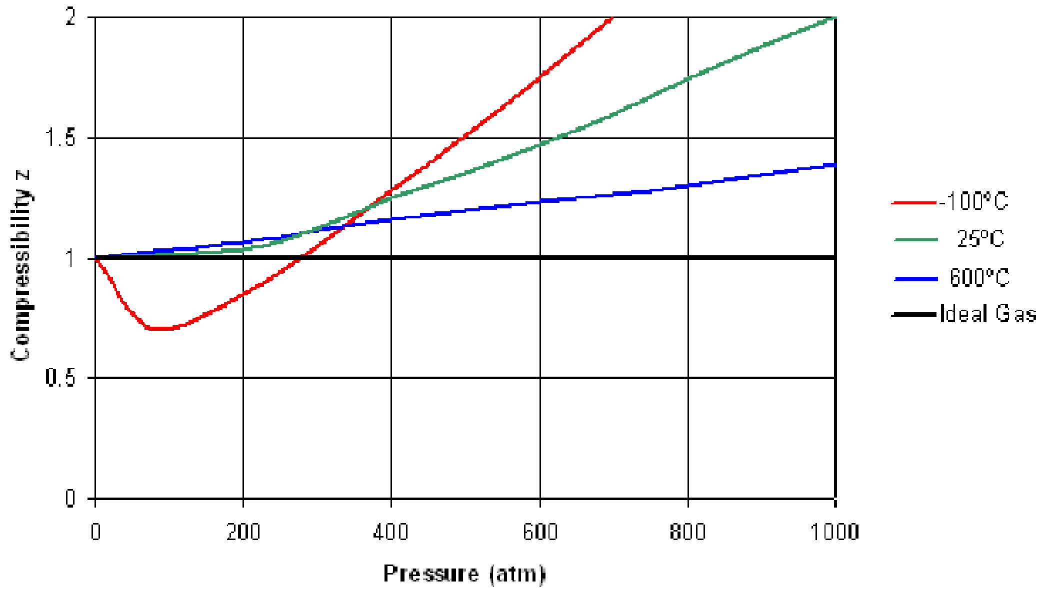
Real gas (compressibility) factor

Real gas (compressibility) factor Z defined by

$$p*v = Z*R*T$$

- describes deviations from ideal gas law
- Z=Z(p,T) has to be determined for each gas – detailed TD design needed!

Compressibility factor of Nitrogen



Summary

- A reciprocating compressor is the most efficient device to compress gas
- Although seemingly simple, a lot of fluid mechanics and thermodynamic knowledge is required
- Technology and R&D is constantly progressing to keep the reciprocating compressor a very attractive choice!

Have a look also at

http://recip.org/reciprocating/introduction-to-thermodynamics/