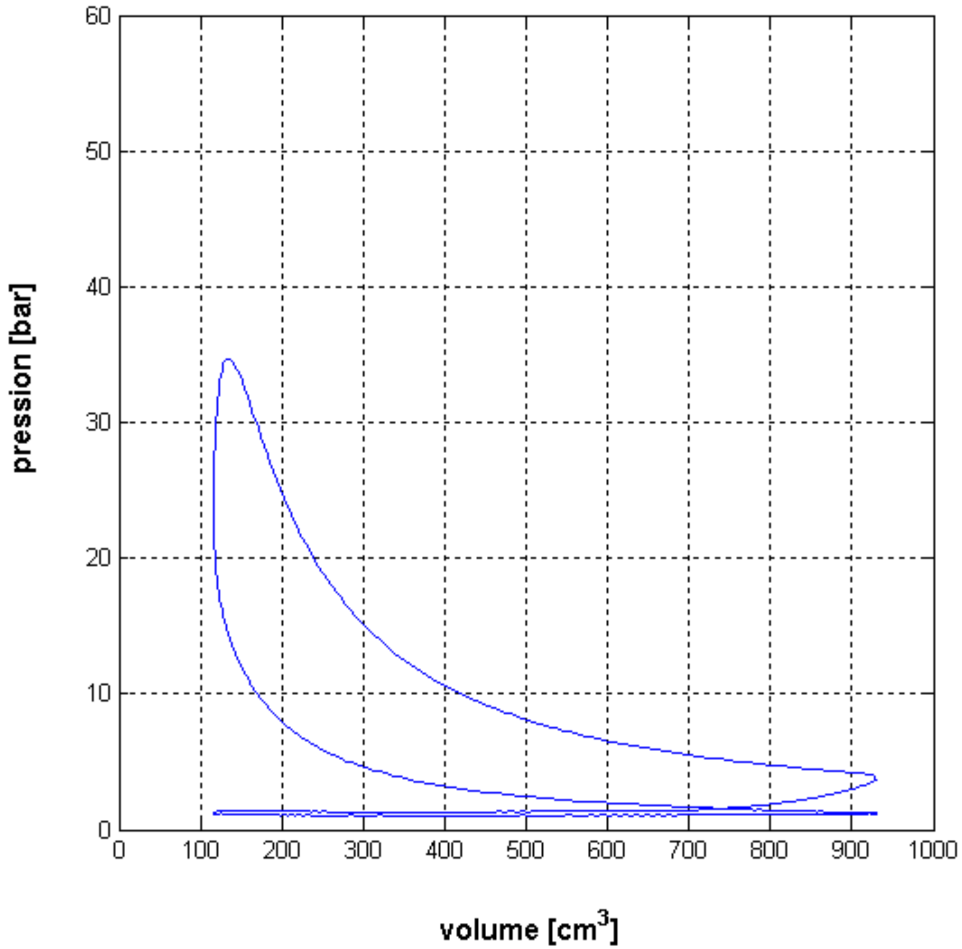
Chapter 2: Thermodynamic cycles



Pressure P

speed c

Power(W/m^2) = P x c

Pressure is indicative of power at a given speed

'indicated' power = indicated by a pressure sensor *inside* the cylinder = crankshaft power + losses (**friction**, oil/water pumps, other auxiliaries)

Learning objectives in Chapter 2

- ⇨ represent the thermodynamic transformations in **P-V** and T-s diagrams
- ⇨ know different **ideal** cycles and their representations
- ⇨ know how a **real** cycle can be measured and what are the differences between an **ideal** and a **real** cycle

Content Chapter 2

- **Thermodynamic basics**
  - ***P-v*** and *T-s* diagrams
  - Thermodynamic cycles
- <u>Ideal</u> cycles
  1. Carnot cycle
  2. Stirling cycle
  3. **Otto** cycle
  4. **Diesel** cycle
  5. Sabaté “combined” cycle
  6. “Wrapped” cycle
  7. Efficiency of ideal cycles
- <u>Real</u> cycles
  - Measurement method of the thermodynamic cycle on an engine
  - Difference between real cycle ⇔ ideal cycle

P-v diagram (Clapeyron diagram)

- *Pressure* on the Y axis (bar, (Pa))
- *(specific) volume v* on the X axis (m^3/kg)
- characteristic curves for an IDEAL GAS:
  - isothermal & isenthalpic curves

    $\Rightarrow$ equilateral hyperbola: $Pv = cte$
  - isentropic curve (steeper)

    $\Rightarrow$ $Pv^{\gamma} = cte$

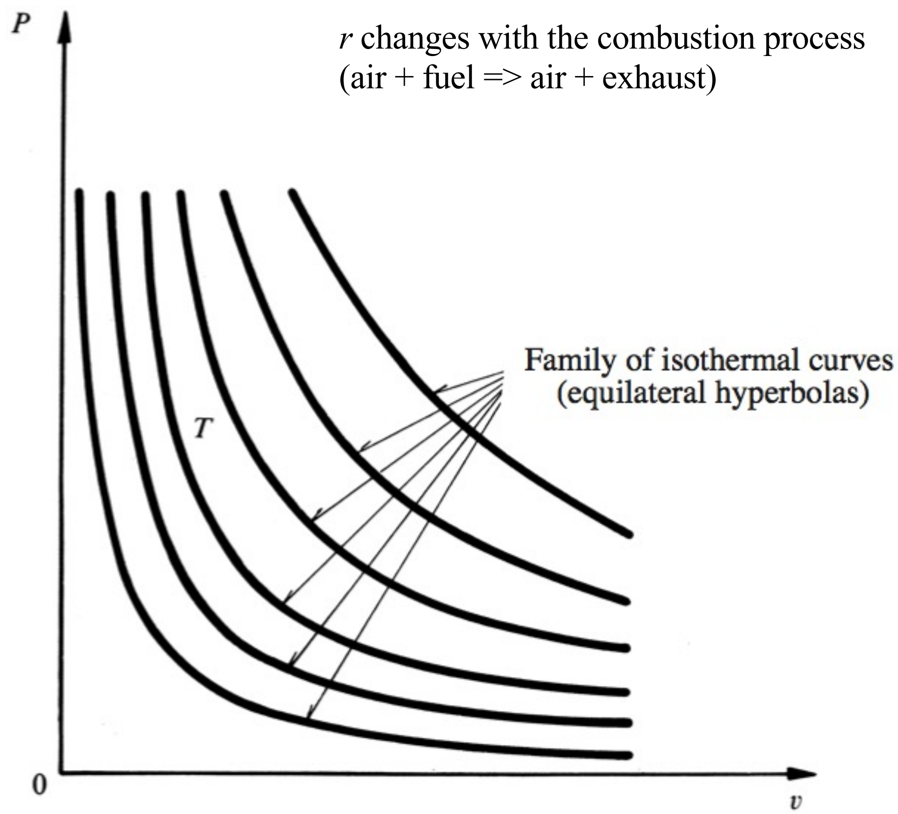
    with $\gamma = C_p / C_v$

- for a biphasic system: liquid-gas

  $\Rightarrow$ saturation curve

$$
\begin{cases}
P \cdot v = r \cdot T \quad with \quad r = \dfrac{\bar{r}}{\tilde{m}} = \dfrac{\Re}{\tilde{m}} \\
r_{air} = \dfrac{\bar{r}}{\tilde{m}_{air}} = \dfrac{8'314}{28.97} = 287 \ {}^{J}\!/\!_{kgK}
\end{cases}
$$

r changes with the combustion process (air + fuel => air + exhaust)



Cp, Cv = *f* (T) (semi-ideal gas)

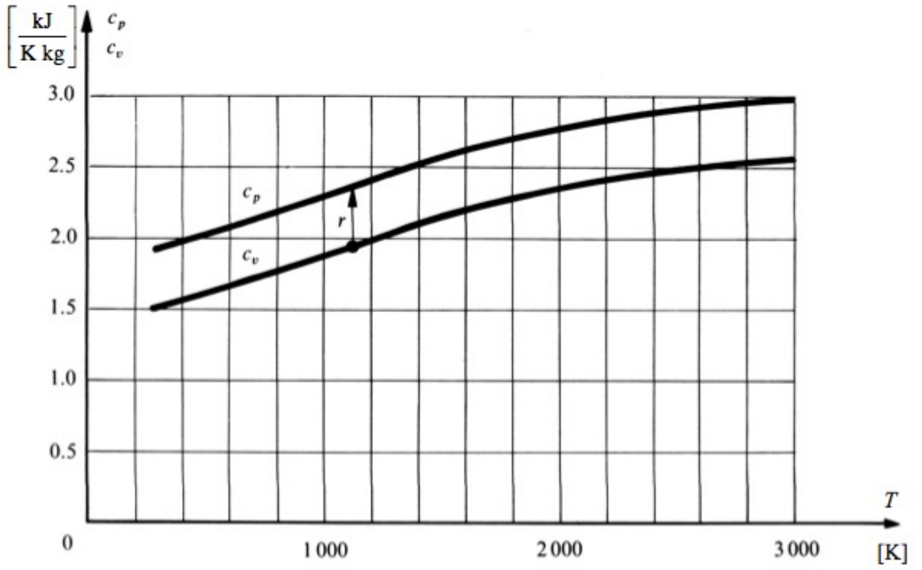


Fig. 5.9 Variation of c_v and c_p as a function of the temperature T for steam, in the extreme case where $P \to 0$.

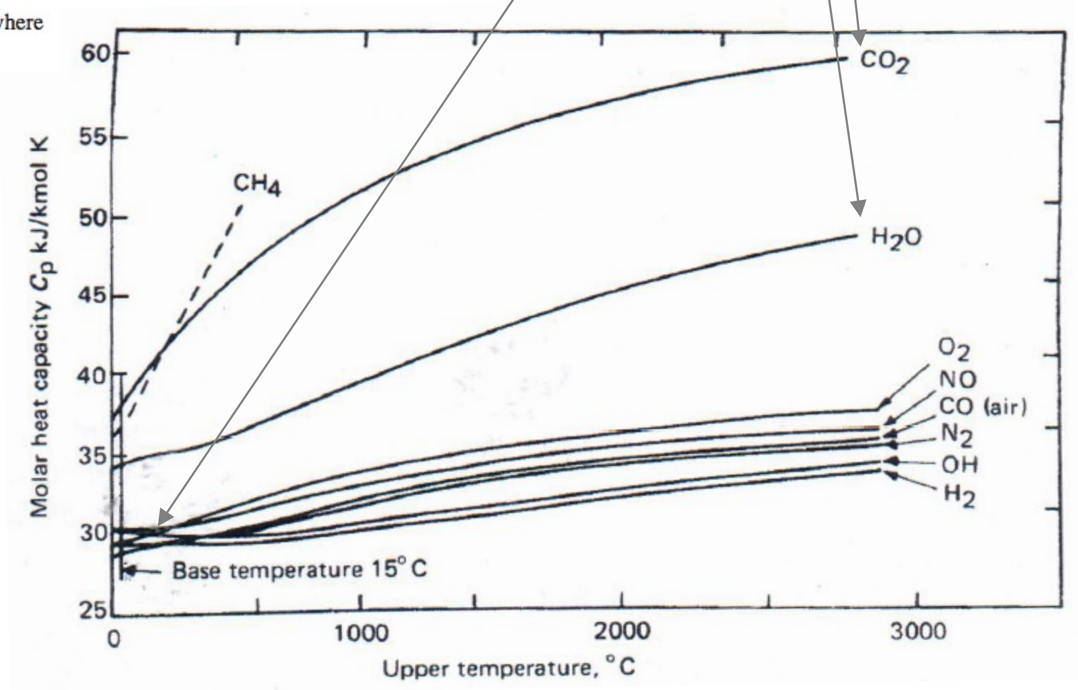
Borel/Favrat Fig 5.9

=> Cp, Cv ≠ const!

=> $\gamma \neq$ const!

(between intake – exhaust)

R.Stone Fig 2.9



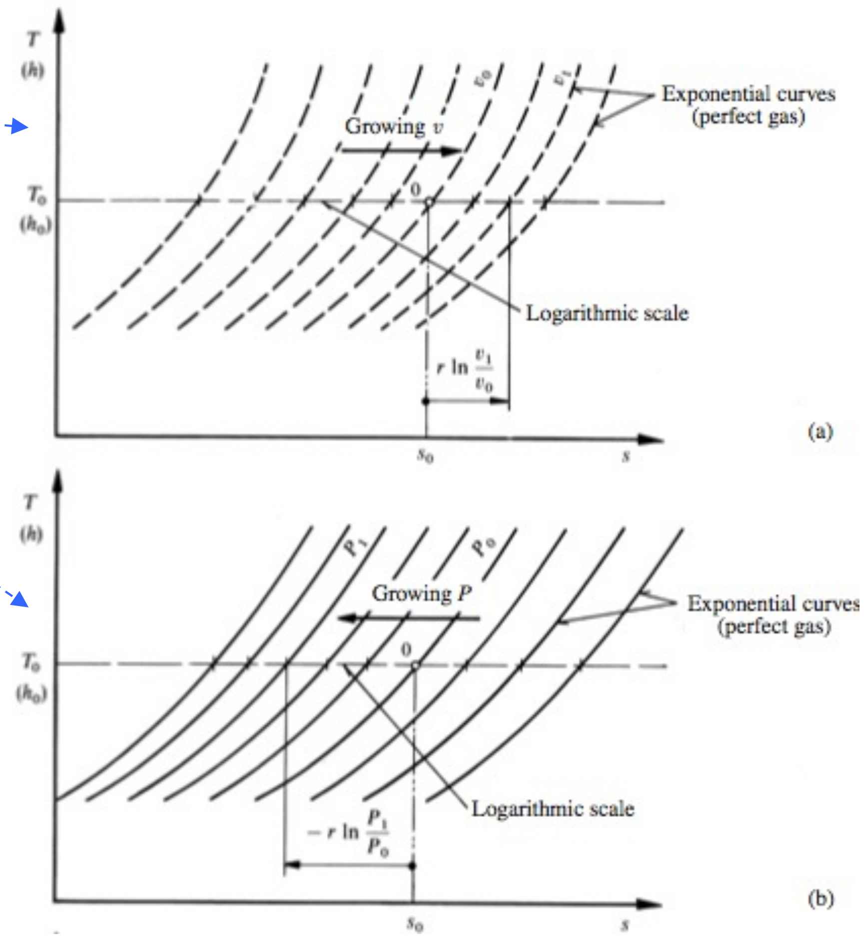
T-s diagram (entropy diagram)

- *Temperature* on the Y axis (K)
- *(specific) entropy s* on the X axis (J/K/kg)
- characteristic curves for an IDEAL GAS:
  - iso**baric** & iso**choric** (steeper) curves

    ⇒ exponential curves with X axis

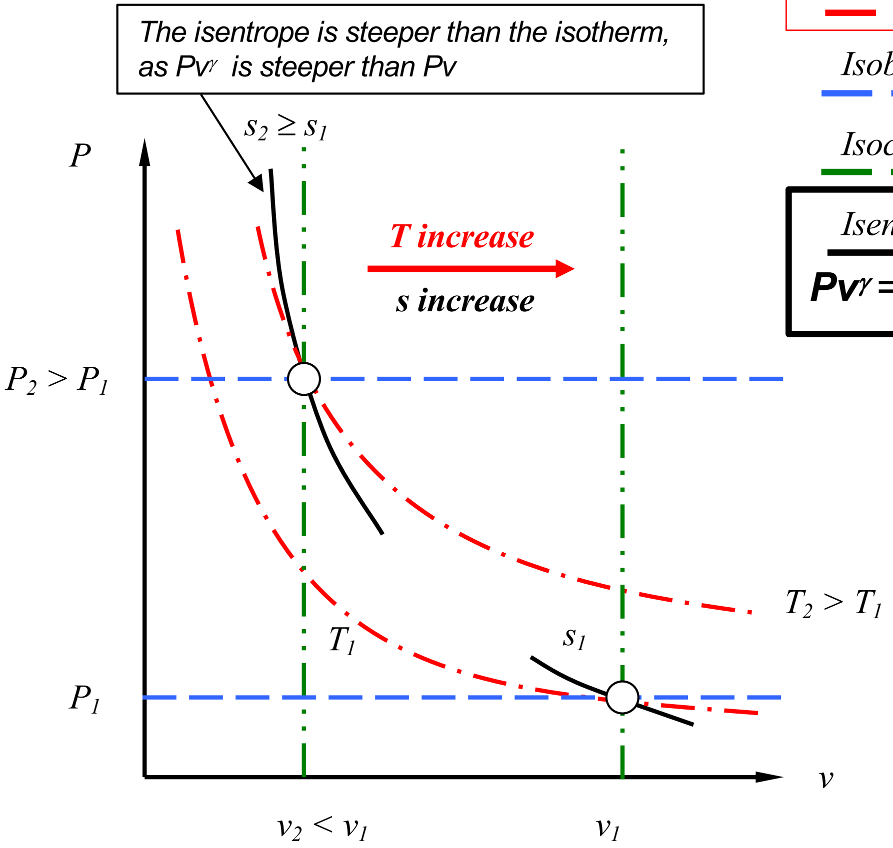
    as asymptote
- for a biphasic system: liquid-gas

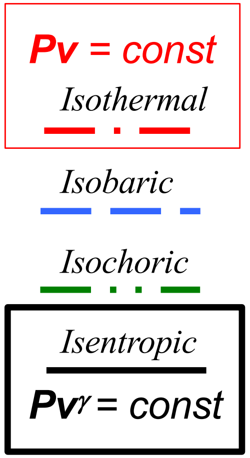
  ⇒ saturation curve



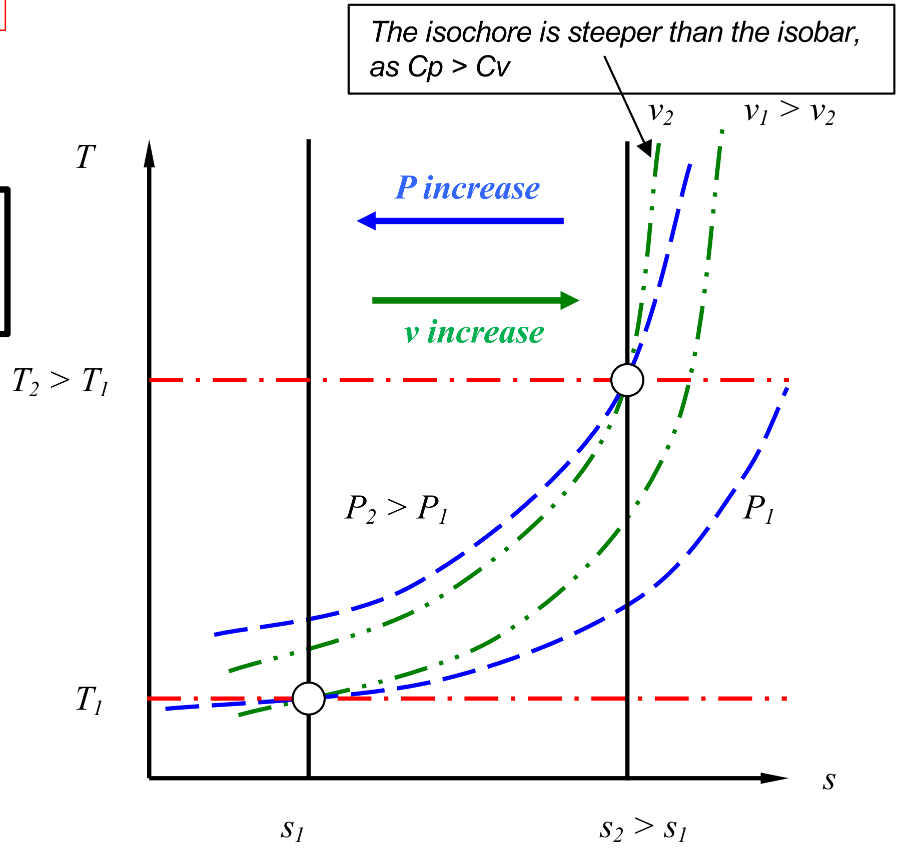
Isocurves in **P-v** and T-s (ideal gas)

P-v diagram





T-s diagram



For p – log v plot, there would be straight lines, with slope 1 for pv = const (isothermal), and slope 1.4 for air isentropic transformation

$$c_v \equiv T\left(\frac{\partial s}{\partial T}\right)_v \longleftrightarrow du = -Pdv + Tds \longleftrightarrow c_v = \left(\frac{\partial u}{\partial T}\right)_v$$

$$c_p \equiv T\left(\frac{\partial s}{\partial T}\right)_P \longleftrightarrow dh = vdP + Tds \longleftrightarrow c_p = \left(\frac{\partial h}{\partial T}\right)_P$$

Reminder: adiabatic compression (ideal gas)

(1)

$$dU = \delta Q + \delta W$$

$$\delta Q = 0$$

$$du = C_v dT = 0 - Pdv$$

(2)

$$Pv = rT$$

$$\Rightarrow d(Pv) = Pdv + vdP = rdT$$

Eliminate dT from (1) and (2):

$$\frac{C_v}{r}(Pdv + vdP) = -Pdv$$

$$\frac{C_v}{r} vdP = -\left(\frac{C_v}{r} + 1\right) Pdv$$

$$\frac{dP}{P} = -\frac{r}{C_v}\left(\frac{C_v + r}{r}\right)\frac{dv}{v} = -\left(\frac{C_P}{C_v}\right)\frac{dv}{v} = -\gamma \frac{dv}{v}$$

Ideal gas => U is only kinetic energy => depends only on T.
Work δW done at constant pressure for ideal gas = PdV = rdT.
Hence, adding heat to an ideal gas at constant pressure = adding extra heat rdT for each kg of gas, beyond the heat added at constant volume. Hence the specific heat of an ideal gas at constant pressure is $c_P = c_v + r$

Or, in similar terms:
at constant volume, all the added heat increases the internal energy and thus the temperature T;
at constant pressure, to reach the same T we need to add to the previous amount an extra amount of heat equal to the work done due to the gas undergoing expansion against atmosphere.

s-P-v-T relations (state equations)

$$ds = c_p \frac{dv}{v} + c_v \frac{dP}{P} \qquad ds = r\frac{dv}{v} + c_v \frac{dT}{T} \qquad ds = -r\frac{dP}{P} + c_p \frac{dT}{T}$$

Isentropic process:

$$C_p \frac{dv}{v} = -C_v \frac{dP}{P}$$

$$C_P d\ln v = -C_v d\ln P$$

$$C_P \ln\frac{v_2}{v_1} = -C_v \ln\frac{P_2}{P_1}$$

$$\frac{C_P}{C_v}\ln\frac{v_2}{v_1} = -\ln\frac{P_2}{P_1}$$

$$\ln\left(\frac{v_2}{v_1}\right)^{C_P/C_v} = \ln\frac{P_1}{P_2}$$

$$\left(\frac{v_2}{v_1}\right)^{\gamma} = \frac{P_1}{P_2}$$

$$\boxed{P_2 v_2^{\gamma} = P_1 v_1^{\gamma} = P v^{\gamma} = const}$$

$$C_p \frac{dT}{T} = r\frac{dP}{P}$$

$$C_P d\ln T = r d\ln P$$

$$C_P \ln\frac{T_2}{T_1} = (C_P - C_v)\ln\frac{P_2}{P_1}$$

$$\ln\frac{T_2}{T_1} = \frac{C_P - C_v}{C_P}\ln\frac{P_2}{P_1}$$

$$\ln\left(\frac{T_2}{T_1}\right) = \frac{\gamma - 1}{\gamma}\ln\frac{P_2}{P_1}$$

$$\ln\left(\frac{T_2}{T_1}\right) = \ln\left(\frac{P_2}{P_1}\right)^{\Gamma}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\Gamma}$$

$$\boxed{\frac{P_2^{\Gamma}}{T_{2,s}} = \frac{P_1^{\Gamma}}{T_{1,s}} = \frac{P^{\Gamma}}{T_s} = const}$$

Consequences of C_p, C_v, γ

Gas	γ
Air	1.40
Ammonia	1.32
Argon	1.66
Benzene	1.12
n- ou iso-butane	1.18
Iso-butane	1.19
Carbon Dioxide	1.28
Carbon Monoxide	1.40
Ethane	1.18
Ethyl alcohol	1.13
Helium	1.66
n-heptane	1.05
Hexane	1.06
Hydrogen	1.41
Methyl alcohol	1.20
Natural Gas (Methane)	1.32
Nitric oxide	1.40
Nitrogen	1.40
Nitrous oxide	1.31
n-octane	1.05
Oxygen	1.40
n- ou Iso-pentane	1.08
Propane	1.13
R-134a	1.20
Steam (water)	1.33
Sulphur dioxide	1.26

Gas	Example	c_P	c_v	γ	Γ
Monoatomic	He, Ar	2.5r	1.5r	1.67	0.4
Diatomic	N_2, O_2,.. (air)	3.5r	2.5r	1.41	0.29
Triatomic etc.	H_2O, CO_2,..	4r	3r	1.33	0.25

$$C_P - C_v = r$$

$$\Gamma \equiv \frac{\gamma - 1}{\gamma}$$

Compressing air (**C.I.**) has $\gamma = 1.40$
Compressing air / fuel mixture (**S.I.**) has $\gamma \approx 1.35$

Hence for a <u>same</u> compression ratio ε, a higher final pressure is reached in a **C.I.** engine:

ε	P_2	P_2	η_I	η_I
for γ equal:	**γ=1.40**	**γ=1.35**	**γ=1.40**	**γ=1.35**
ε = 10	**25.1 bar**	**22.4 bar**	**60%**	**55%**
ε = 13	**36.3 bar**	**31.9 bar**	**64%**	**59%**

=indicated efficiency

$$\eta_I = 1 - \frac{1}{\varepsilon^{\gamma-1}}$$

5% points efficiency penalty C.I. → S.I. only due to γ

Some property values

https://www.ohio.edu/mechanical/thermo/property_tables/gas/idealGas.html

Gas (300K)	Chemical Formula	Molar mass	Gas constant	Specific Heat at const. P	Specific Heat at const. V	Specific Heat Ratio
		m [kg/kmol]	r [kJ/kg.K]	Cp [kJ/kg.K]	Cv [kJ/kg.K]	γ = Cp/Cv
Air	--	28.97	0.287	1.005	0.718	1.4
Argon	Ar	39.948	0.2081	0.5203	0.3122	1.667
Butane	C_4H_{10}	58.124	0.1433	1.7164	1.5734	1.091
Carbon Dioxide	CO_2	44.01	0.1889	0.846	0.657	1.289
Carbon Monoxide	CO	28.011	0.2968	1.04	0.744	1.4
Ethane	C_2H_6	30.07	0.2765	1.7662	1.4897	1.186
Ethylene	C_2H_4	28.054	0.2964	1.5482	1.2518	1.237
Helium	He	4.003	2.0769	5.1926	3.1156	1.667
Hydrogen	H_2	2.016	4.124	14.307	10.183	1.405
Methane	CH_4	16.043	0.5182	2.2537	1.7354	1.299
Neon	Ne	20.183	0.4119	1.0299	0.6179	1.667
Nitrogen	N_2	28.013	0.2968	1.039	0.743	1.4
Octane	C_8H_{18}	114.231	0.0729	1.7113	1.6385	1.044
Oxygen	O_2	31.999	0.2598	0.918	0.658	1.395
Propane	C_3H_8	44.097	0.1885	1.6794	1.4909	1.126
Steam	H_2O	18.015	0.4615	1.8723	1.4108	1.327

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- Ideal cycles
  1. Carnot cycle
  2. Stirling cycle
  3. **Otto** cycle
  4. **Diesel** cycle
  5. Combined cycle
  6. Wrapped cycle
  7. Efficiency of ideal cycles
- Real cycles
  - Measurement method of the thermodynamic cycle on an engine
  - Difference between real cycle ⇔ ideal cycle

Ideal cycles

- General assumptions:
  - Combustion process ⇒ assimilated to a heat transfer
  - Working fluid is not subjected to modification of composition (!!)

    ⇒ working fluid = air

    ⇒ c_p , c_v = constant (!!)

$$\Delta U_{cz} = \sum_k \delta E_k^+ + \sum_i \delta Q_i^+ + \sum_j h_{cz,j} \cdot dM_j^+$$

  - Thermal losses supposed to be zero (!!)
  - 1st Law: $\Delta U_{cz} = 0$ (cycle for a state function), $dM = 0$ (mass conservation) ⇒ $\sum_k E_k^+ + \sum_i Q_i^+ = 0$; $\delta S^i = \delta S^r = \dfrac{\delta R}{T}$
  - 2nd Law: $TdS = \delta Q + \delta R$ ⇒ $\oint dS = \oint \dfrac{\delta Q_a^+}{T_a} + \oint \dfrac{\delta Q_b^+}{T_b} + \ldots + \underbrace{\oint \delta S^i}_{\geq 0}$ (=0 for a cycle on a state function)

$$\frac{Q_a^+}{T_a} + \frac{Q_b^+}{T_b} + \ldots + S^i = 0$$

$$dS = \underbrace{\delta S^e}_{=0} + \delta S^i \quad and \quad \delta S^i \geq 0$$

⇒ The condition $E^+ < 0$ (i.e. $E^- > 0$, work generated) requires to have at least 2 thermal sources Q_a and Q_b. Ideality means $S^i = 0$ (no friction, no dissipation, δR=0)

State function cycle integral = 0
P-v cycle integral = work

Fundamental equations: Borel / Favrat book, § 13.2.2

Closed system

$$\oint \delta a^- + \oint d\frac{\overline{C}^2}{2} + g\oint d\overline{Z} + \oint \delta r = \oint P dv = -\oint du + \oint \delta q^+ + \oint \delta r = -\oint du + \oint T ds$$

Open system

$$\oint \delta e^- + \oint d\frac{C^2}{2} + g\oint dZ + \oint \delta r = -\oint v dP = -\oint dh + \oint \delta q^+ + \oint \delta r = -\oint dh + \oint T ds$$

State functions:

$$\oint d\frac{\overline{C}^2}{2} = 0$$

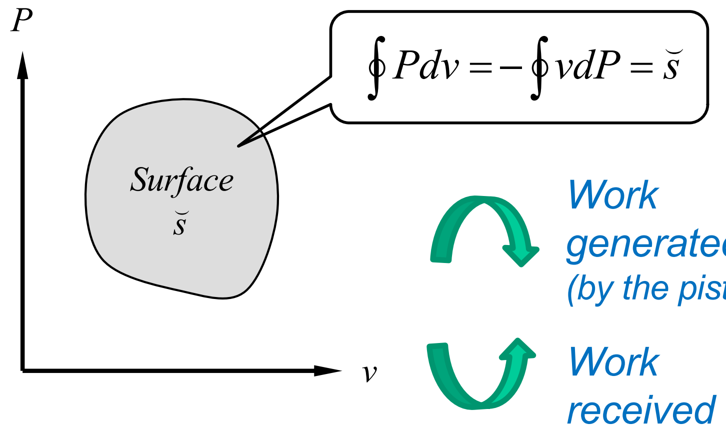
$$\oint d\overline{Z} = 0$$

$$\oint du = 0$$

$$\oint dh = 0$$

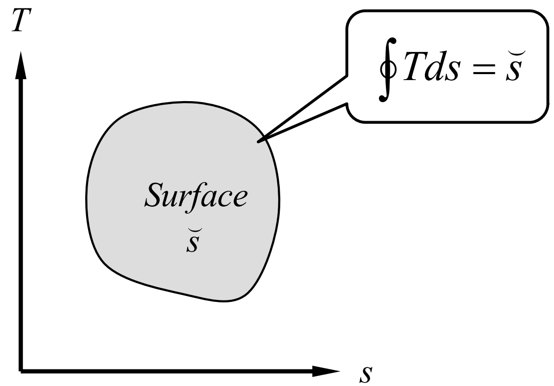
$$a^- = q^+ = \underbrace{\oint P dv}_{\breve{s}} - r = \underbrace{\oint T ds}_{\breve{s}} - r \qquad e^- = q^+ = -\underbrace{\oint v dP}_{\breve{s}} - r = \underbrace{\oint T ds}_{\breve{s}} - r$$

r : friction losses



Work generated
(by the piston)

Work received
(by the piston)



Surface = work done by the ideal cycle (no friction loss). Indicated work

Thermodynamic cycles

- always <u>bi-thermal</u> :

  Heat transfer with 2 thermal sources

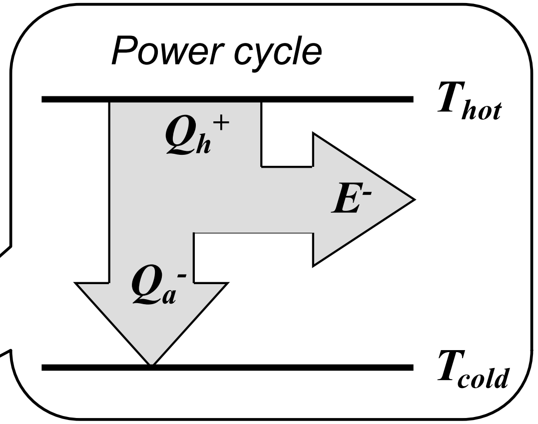
  - If $Q^{+} > 0$ and $E^{+} < 0 \Rightarrow$ **Power cycles**

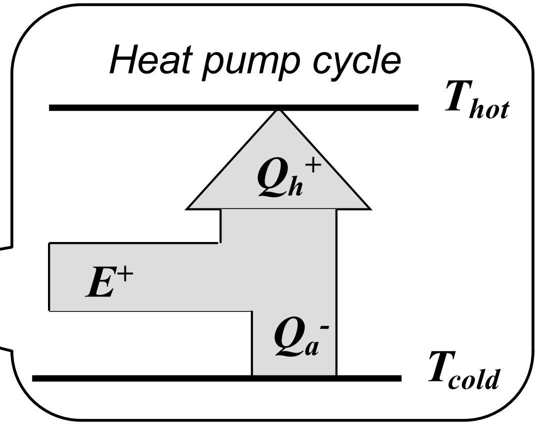
    example: Internal Combustion engines

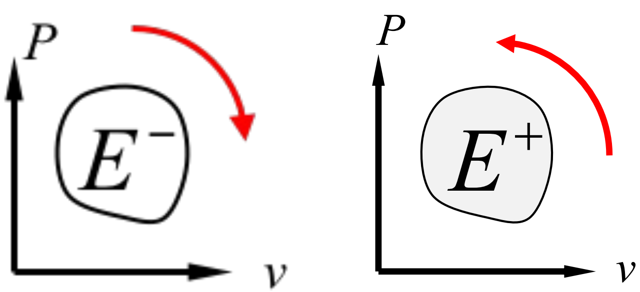
    External Combustion engines (gas turbines)

  - If $E^{+} > 0$ and $Q^{+} < 0 \Rightarrow$ (Heat) pump cycles

    example: Heat pump and refrigeration cycles







Ideal cycles

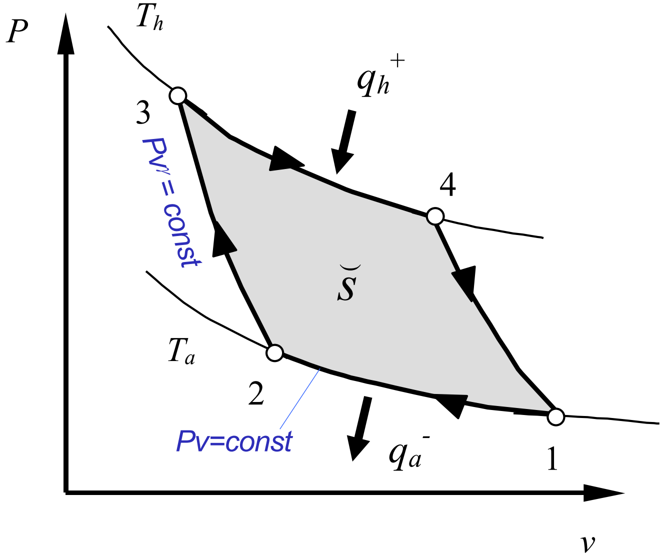
1. Carnot cycle

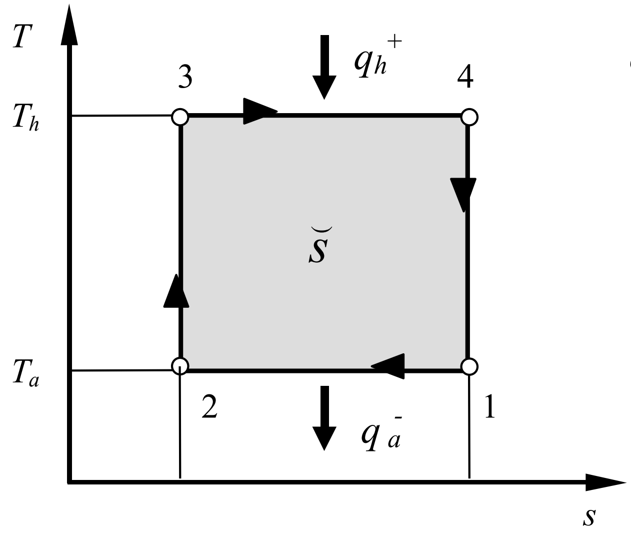
$\delta q = Tds - \delta r \quad \rightarrow \quad \delta q = Tds$

$$e^{-} = \underbrace{\oint Tds}_{\breve{s}} - \underbrace{r}_{=0}$$

⇒ bi-thermal reversible power cycle (δr = 0)

⇒ **2 isothermal + 2 adiabatic (isentropic** (δr = 0)**) processes**





$$e^{-} = (T_h - T_a)\cdot(s_4 - s_3) = \breve{s}$$

$$q_h^{+} = T_h \cdot (s_4 - s_3)$$

$$q_a^{-} = T_a \cdot (s_1 - s_2)$$

$$\varepsilon^* = \eta_I^* = \Theta = 1 - \frac{T_a}{T_h}$$

$$\eta^* = 1$$

I : « indicated »
(=thdyn cycle eff.)

Description

1-2 : isothermal compression: $Q^+ < 0$; $E^+ > 0$

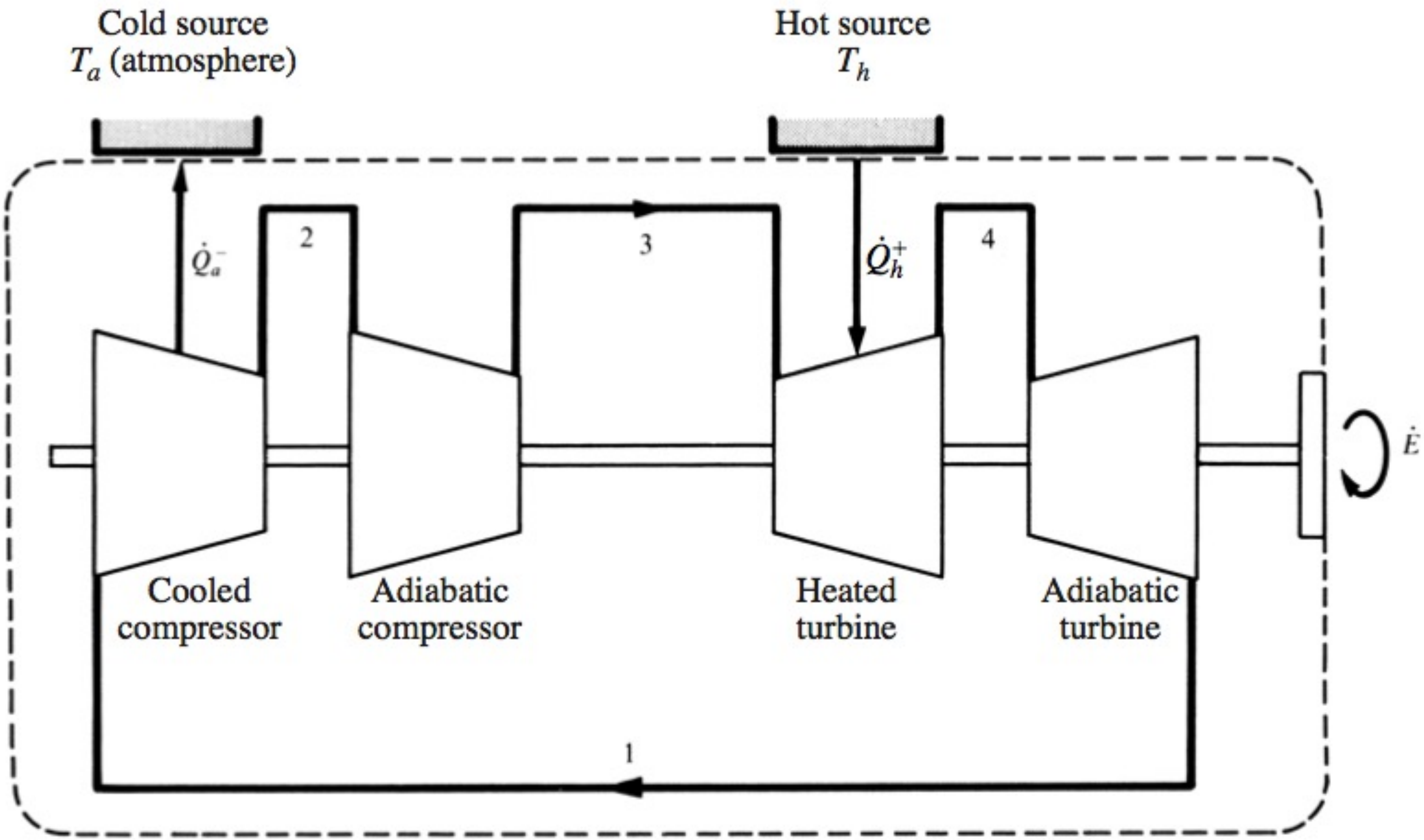
2-3 : isentropic compression: $E^+ > 0$

3-4 : isothermal expansion: $E^+ < 0$; $Q^+ > 0$

4-1 : isentropic expansion: $E^+ < 0$

Adiabatic compression: the system receives work without heat loss (all heat stays inside) => no entropy increase (if dissipation (friction) is neglected, which is the case for an ideal cycle)

Carnot machine/engine

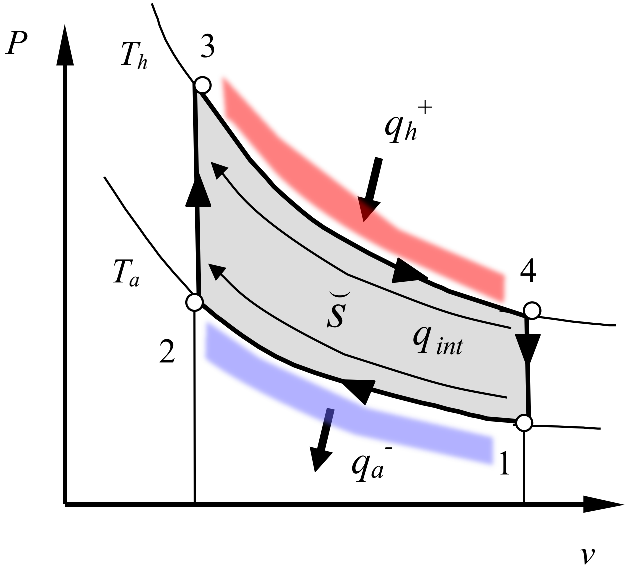


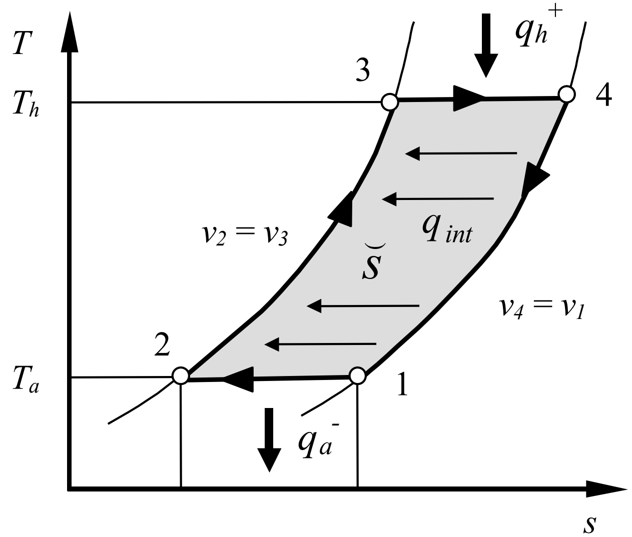
Ideal cycles

2. **Stirling** cycle

⇒ bi-thermal reversible power cycle

⇒ composed of **2 isothermal + 2 isochoric processes**





$$\varepsilon^* = \eta_I^* = \Theta = 1 - \frac{T_a}{T_h}$$

$$\eta^* = 1$$

Description

1-2 : isothermal compression : $Q^+ < 0$; $E^+ > 0$

2-3 : isochoric heat-addition (internal heat transfer) (= Otto)

3-4 : isothermal expansion: $E^+ < 0$; $Q^+ > 0$

4-1 : isochoric heat-removal (internal heat transfer)

2-3: The system receives <u>heat</u> at const. volume (ideal Otto cycle) => P, s increase

Stirling cycle

1-2 : isothermal compression ⇒ rise of the working piston *P*t

Gas : $e^+ > 0$ and $q^+_{cold} < 0$

2-3 : isochoric heat supply ⇒ descent of the displacement piston *P*d (gas receives heat from the internal heat recuperator A from bottom to top)

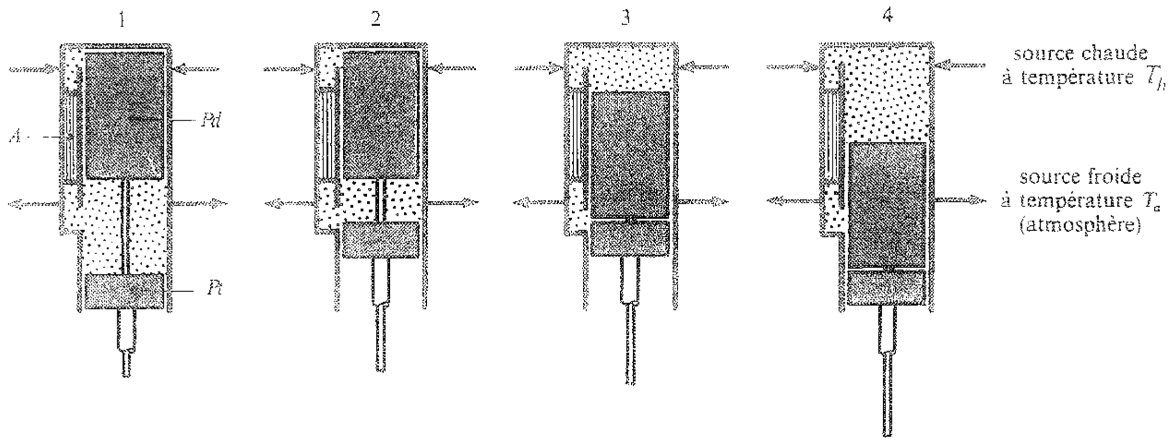
Gas : $q^+_{internal} > 0$

3-4 : isothermal expansion ⇒ descent of the 2 pistons *P*d and *P*t

Gas : $e^- > 0$ and $q^+_{hot} > 0$

4-1 : isochoric heat removal ⇒ rise of the displacement piston *P*d (gas gives heat to the internal heat recuperator A from top to bottom).

Gas: $q^+_{internal} < 0$



*P*d : <u>d</u>isplacement piston

A : Regenerator

*P*t : Working piston (<u>t</u>ravail)

- the one connected to the connecting rod

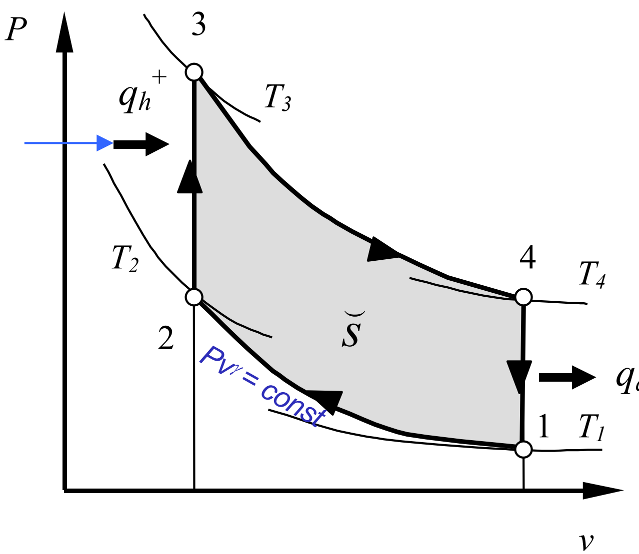
Ideal cycles

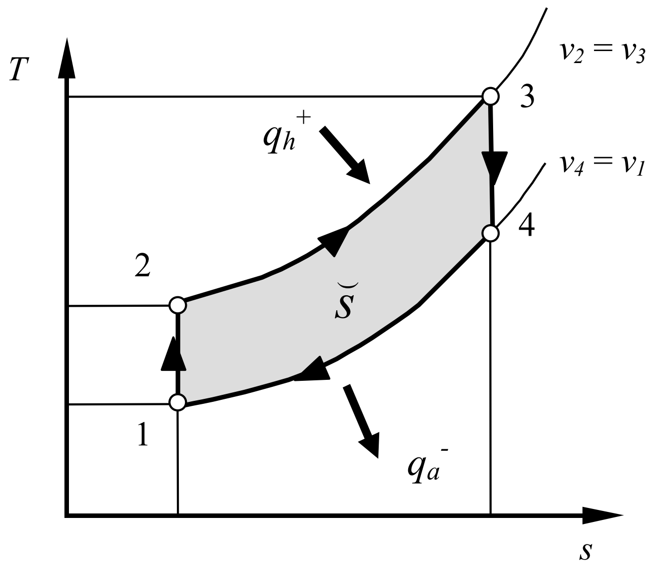
3. **Otto** cycle *or* constant-volume cycle (Beau de Rochas)

⇒ bi-thermal reversible power cycle

⇒ **2 isentropic + 2 isochoric processes**

ignition at TDC





cf. exercise today

$$\eta_I = 1 - \frac{1}{\chi^{\gamma-1}}$$

$$\eta_I = 1 - \frac{1}{\varepsilon^{\gamma-1}}$$

$$(\chi \equiv \varepsilon) = \frac{V_1}{V_2}$$

(Ideal) Efficiency only depends on the compression ratio ! (and gamma-value)

Description

1-2 : isentropic compression (hence adiabatic and reversible): $E^+ > 0$

2-3 : isochoric heat supply: $Q^+ > 0$ (spark at TDC + **combustion (instantaneous)**)

3-4 : isentropic expansion: $E^+ < 0$ ($E^- > 0$: work generated)

4-1 : isochoric heat-removal: $Q^+ < 0$ (=>exhaust) =open outlet valves at BDC **P**↓ **T**↓ = 'blowdown' loss

2-3: The system receives <u>heat</u> at const. volume => P, s increase

Residual combustion gases are always trapped in the clearance volume for the next cycle. As this is not air, this reduces volumetric efficiency (fresh air intake).

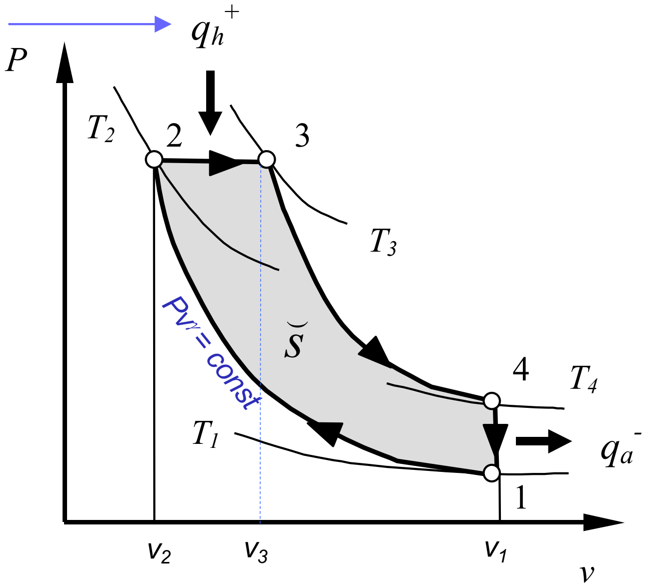
Ideal cycles

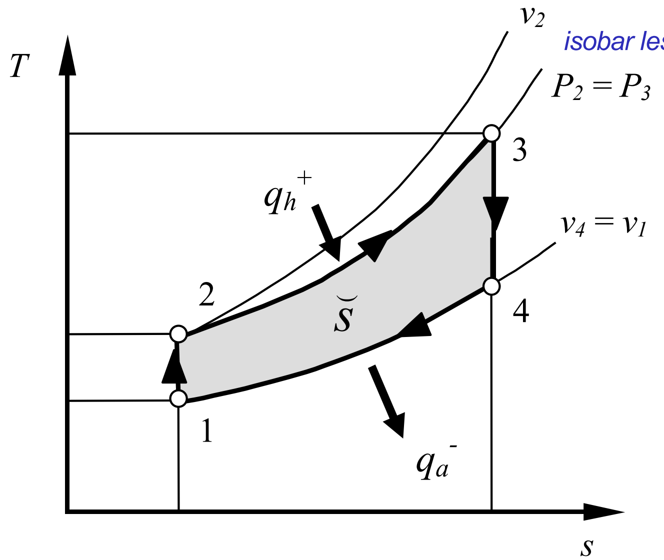
4. **Diesel** cycle *or* constant-pressure cycle

⇒ bi-thermal reversible power cycle

⇒ **2 isentropic + 1 isobaric + 1 isochoric processes**

combustion at TDC, starting the expansion, combustion at const. p takes time





isobar less steep than isochore

$$\eta_I = 1 - \frac{1}{\varepsilon^{\gamma-1}}\left[\frac{\delta^\gamma - 1}{\gamma\cdot(\delta - 1)}\right]$$

$$with \quad \delta = \frac{V_3}{V_2} = \frac{T_3}{T_2}$$

>1

≈ combustion delay

Description

Combustion takes time => this is worse for efficiency since this leaves more time for heat exchange with the environment

1-2 : isentropic compression (adiabatic and reversible): $E^+ > 0$

2-3 : isobaric heat supply (isobaric expansion) : $Q^+ > 0$

3-4 : isentropic expansion: $E^+ < 0$

4-1 : isochoric heat-removal: $Q^+ < 0$

2-3: The system receives heat at const. pressure => v, s increase

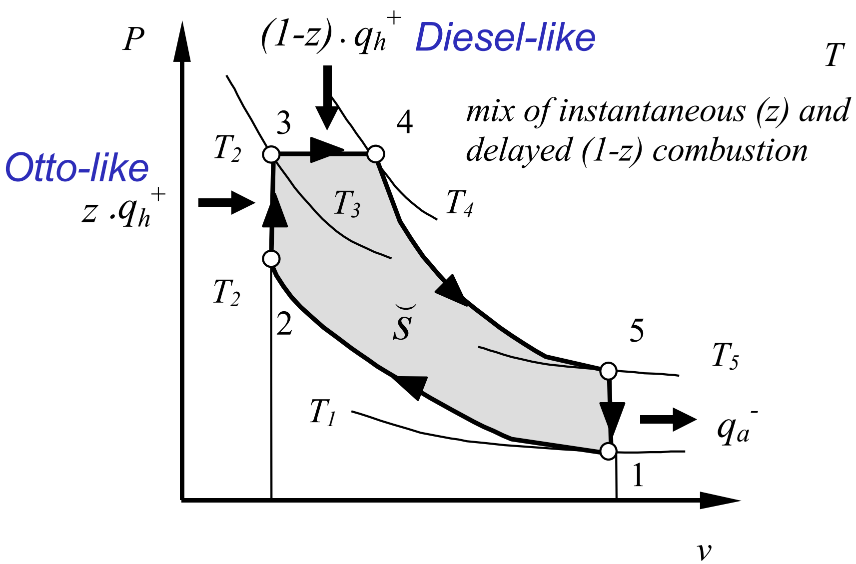
(but the s (and T) increases are less than with heat addition at const. volume)

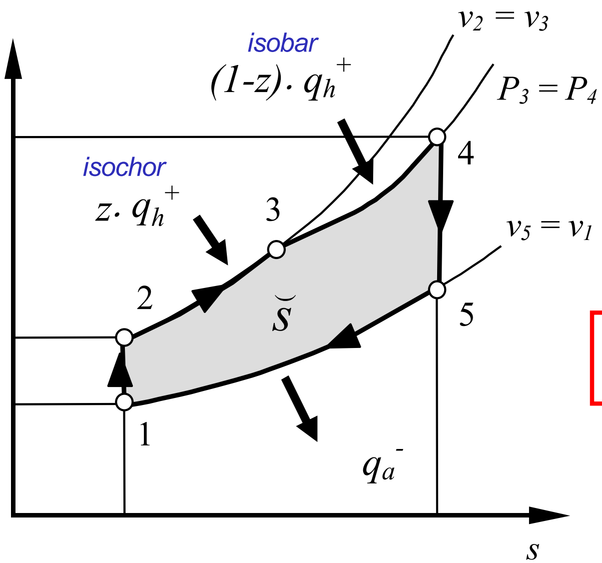
Ideal cycles

5. Sabaté “combined” cycle (closer to a real Diesel cycle)

⇒ bi-thermal reversible power cycle

⇒ **2 isentropic + 2 isochoric** *(Otto-like)* **+ 1 isobaric** *(Diesel-like)* **processes**





Diesel-like

$$\eta_I = z \cdot \eta_{I\,V=cte} + (1-z) \cdot \eta_{I\,P=cte}$$

Otto-like

Description

1-2 : isentropic compression (adiabatic and reversible): $E^+ > 0$

2-3 : isochoric heat supply (isochoric compression) : $Q^+ > 0$

3-4 : isobaric heat supply (isobaric expansion) : $Q^+ > 0$

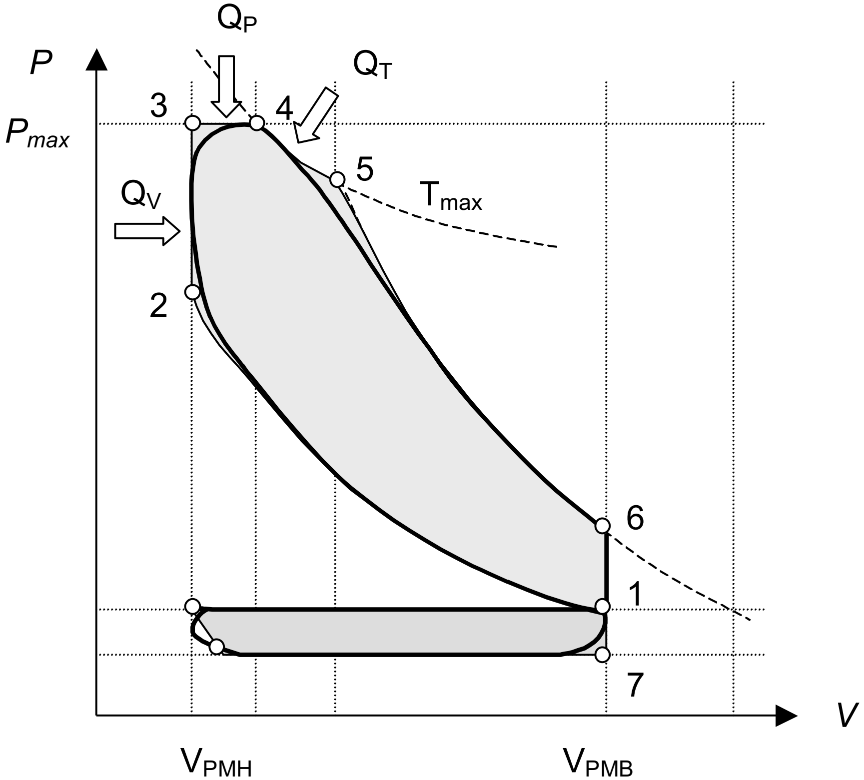
4-5 : isentropic expansion: $E^+ < 0$

5-1 : isochoric heat-removal: $Q^+ < 0$

Ideal cycles

6. “Wrapped” cycle

- Approximation of a real cycle by an ideal « wrapped » cycle



Description :

1-2 : isentropic *or* polytropic process

2-3 : isochoric process

3-4 : isobaric process

4-5 : isothermal process

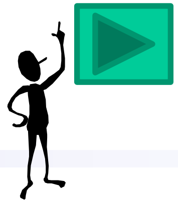
5-6 : isentropic *or* polytropic process

6-1 : isochoric process

…etc

→ *modeling of engine cycles*

Ideal cycles

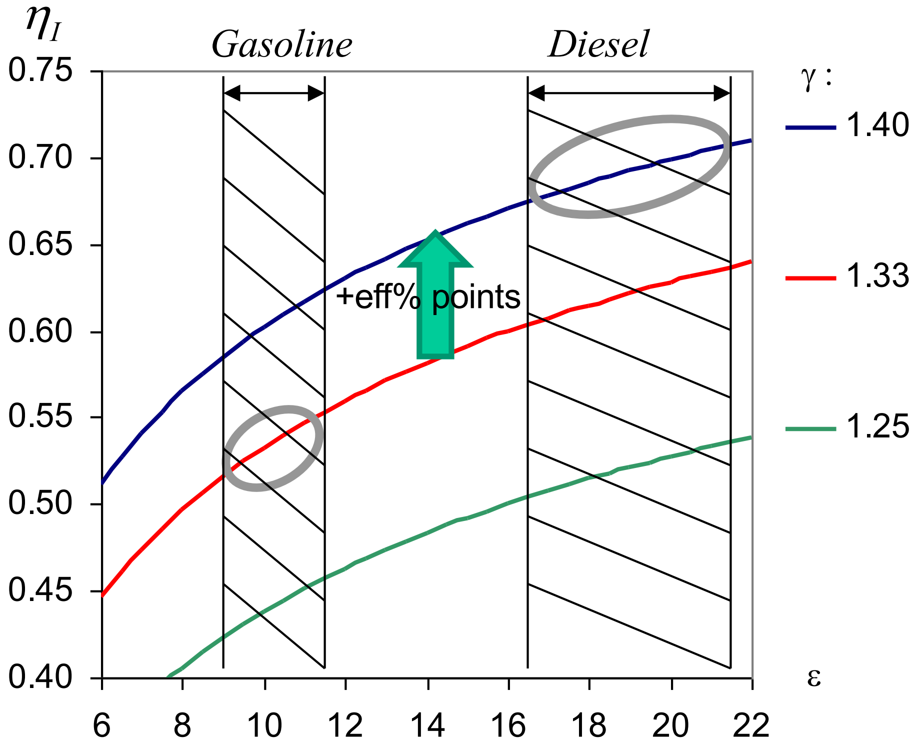


Efficiency of ideal cycles

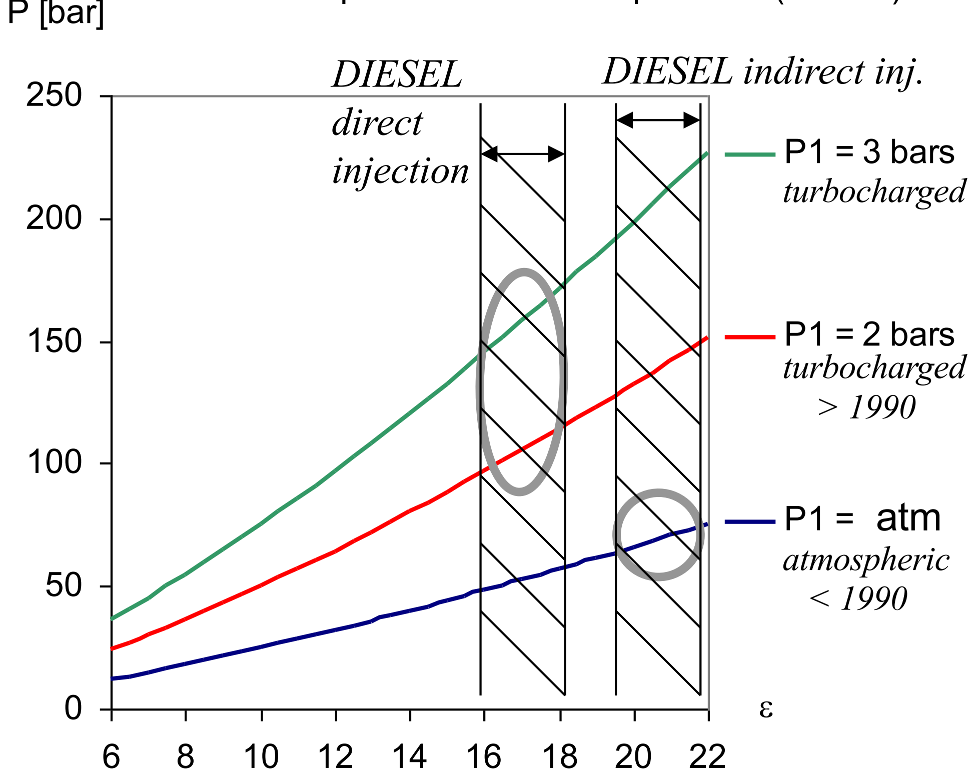
- $\eta_V \Rightarrow$ theoretical thermodynamic efficiency at ***V* = const**. :

$$\eta_I = 1 - \frac{1}{\varepsilon^{\gamma - 1}}$$

Theoretical thermodynamic efficiency at V = constant



End of compression maximum pressure (at TDC)



maximize ε, maximize γ (*curves without combustion process*)

Ideal cycles

Efficiency of ideal cycles

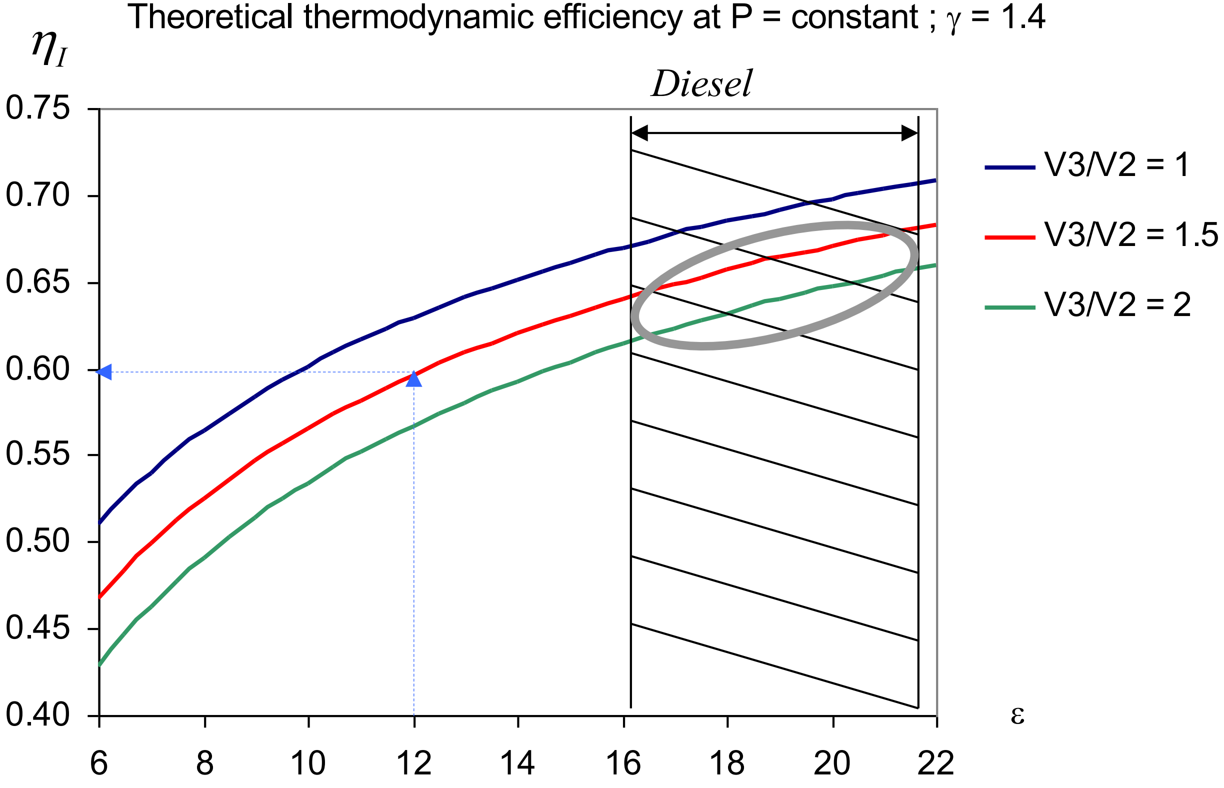
- $\eta_P \Rightarrow$ theoretical thermodynamic efficiency at ***P* = cnst.** :

$$\eta_I = 1 - \frac{1}{\varepsilon^{\gamma-1}}\left[\frac{\delta^{\gamma}-1}{\gamma(\delta-1)}\right]$$

$$with \quad \delta = \frac{V_3}{V_2}$$

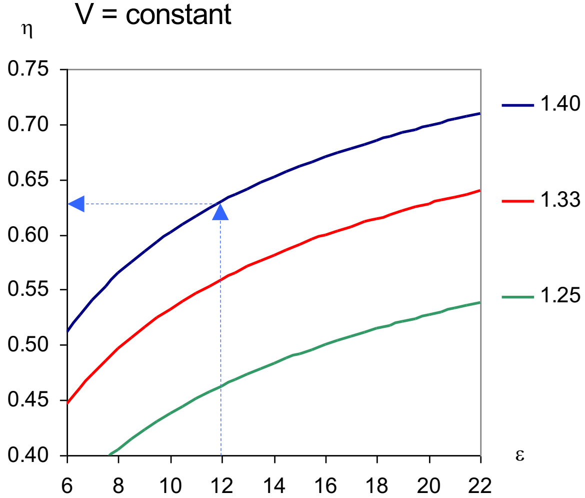
≈ combustion delay

Theoretical thermodynamic efficiency at P = constant ; γ = 1.4



Green curve combustion process is twice as long as blue process, 7% points penalised in efficiency.

Theoretical thermodynamic efficiency at V = constant



*Note that for **same** ε, Otto efficiency is **higher** than Diesel efficiency with combustion delay **(δ > 1)***

Comparison SI / CI engines

	Spark Ignition (Otto)	Compression Ignition (Diesel)
Cycle	Constant volume heat addition	Constant pressure heat addition
Fuel	Gasoline, very volatile. (''C8'') High self-ignition T (resistant)	Diesel, low volatility. (''C16'') Lower self-ignition T (auto-ignition)
Fuel intake	Fuel-air mixture	Fuel added to highly compressed air by high pressure pump & injector
Load control	Throttle for fuel-air mixture quantity	Fuel quantity only, no air quantity control
Ignition	Spark (battery or magneto)	Self-ignition (due to high P and T)
Compression (CR)	8-11. Limited by fuel antiknock quality	16-20. Limited by engine weight (mechanical limit)
(Engine) Speed	High (lighter weight; **homogeneous** combustion). E.g. car 3000 rpm	Low (heavier weight; **heterogeneous** combustion). E.g. car 2000 rpm
Thermal efficiency	Lower (due to low CR)	Higher (due to high CR)
Weight	Lighter (lower peak pressure)	Heavier (higher peak pressure)

Why thermal efficiency C.I. > S.I.

Despite higher P and T, the cooling upon expansion is more important for a C.I. exhaust than for a S.I. exhaust, because of the higher compression ratio :

		Gasoline	Diesel				
γ	gamma	1,35	1,4				
Γ	GAMMA	0,259	0,286				
ε	epsilon (C.R.)	10	16				
	v1/v2	10	16				
	P1 - in	1	1	bar			
	P2	22,4	48,5	isentropic relation Pv^gamma = constant			
	T_adiab	2000	2200				
	T_exhaust K	1101	996	isentropic relation P^GAMMA/T_s = constant			
	T_exhaust °C	828	723				

Content Chapter 2

- Thermodynamic basics
  - ***P-v*** and *T-s* diagrams
  - Thermodynamic cycles
- **Ideal** cycles
  1. Carnot cycle
  2. Stirling cycle
  3. Otto cycle
  4. Diesel cycle
  5. Combined cycle
  6. Wrapped cycle
  7. Efficiency of ideal cycles
- **Real** cycles
  - Measurement method of a thermodynamic cycle on an engine
  - Difference between **real** cycle ⇔ **ideal** cycle

Real cycles

- Measurement method of the thermodynamic cycle on an engine

  P-v cycle acquisition requires the measure of $P_{cylinder}$ and $\varphi_{crankshaft}$

  - Instrumentation (1): ***pressure***

    $P_{cylinder} \Rightarrow$ cooled pressure sensor located <u>inside</u> the combustion chamber

Signal outlet: P = f(t)

Cooling circuit

Pressure sensor is fitted inside the combustion chamber

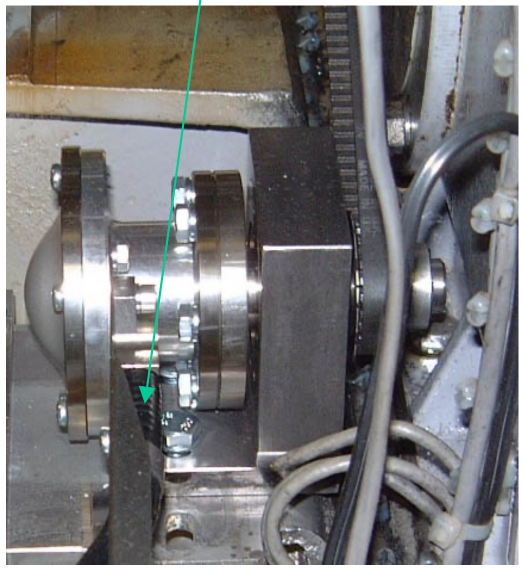
Once P, v determined (2 state functions) => everything fixed

Real cycles

- Measurement method of the thermodynamic cycle on an engine
  - Instrumentation (2): ***volume*** (piston displacement)

$\varphi_{crankshaft}$ ⇒ angular encoder fixed on the crankshaft (3600 teeth/rev.= 0.1°)

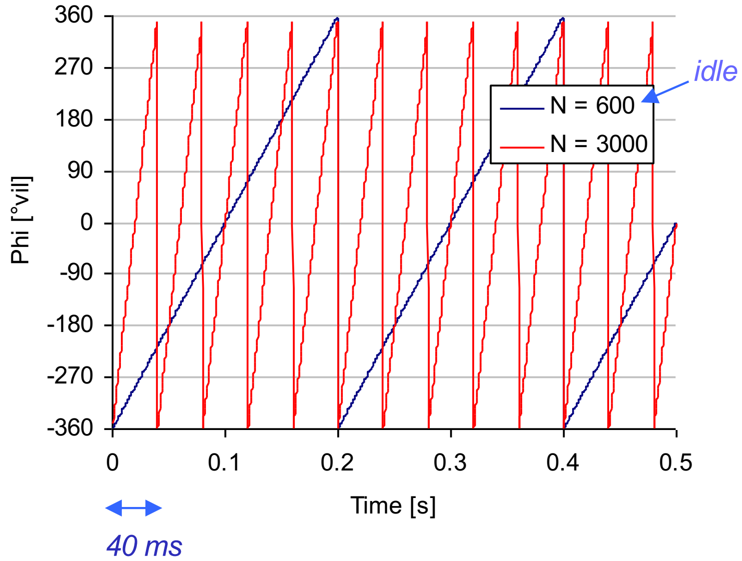
Angular encoder signal: $\varphi = f(t)$
(captured by optical sensor)



Example : $\varphi = f(t)$

$$\omega(rad/s) = \frac{2 \cdot \pi}{60} \cdot N(1/\min)$$

$$\varphi = \omega \cdot t$$



<u>for N = 3000 rpm = 50 rps</u>

1 rev ⇒ 20 ms (1 ↓↑stroke)

18° ⇒ *1 ms*

1° ⇒ *0.056 ms*

Real cycles

- Measurement method of the thermodynamic cycle on an engine
  - Acquisition:

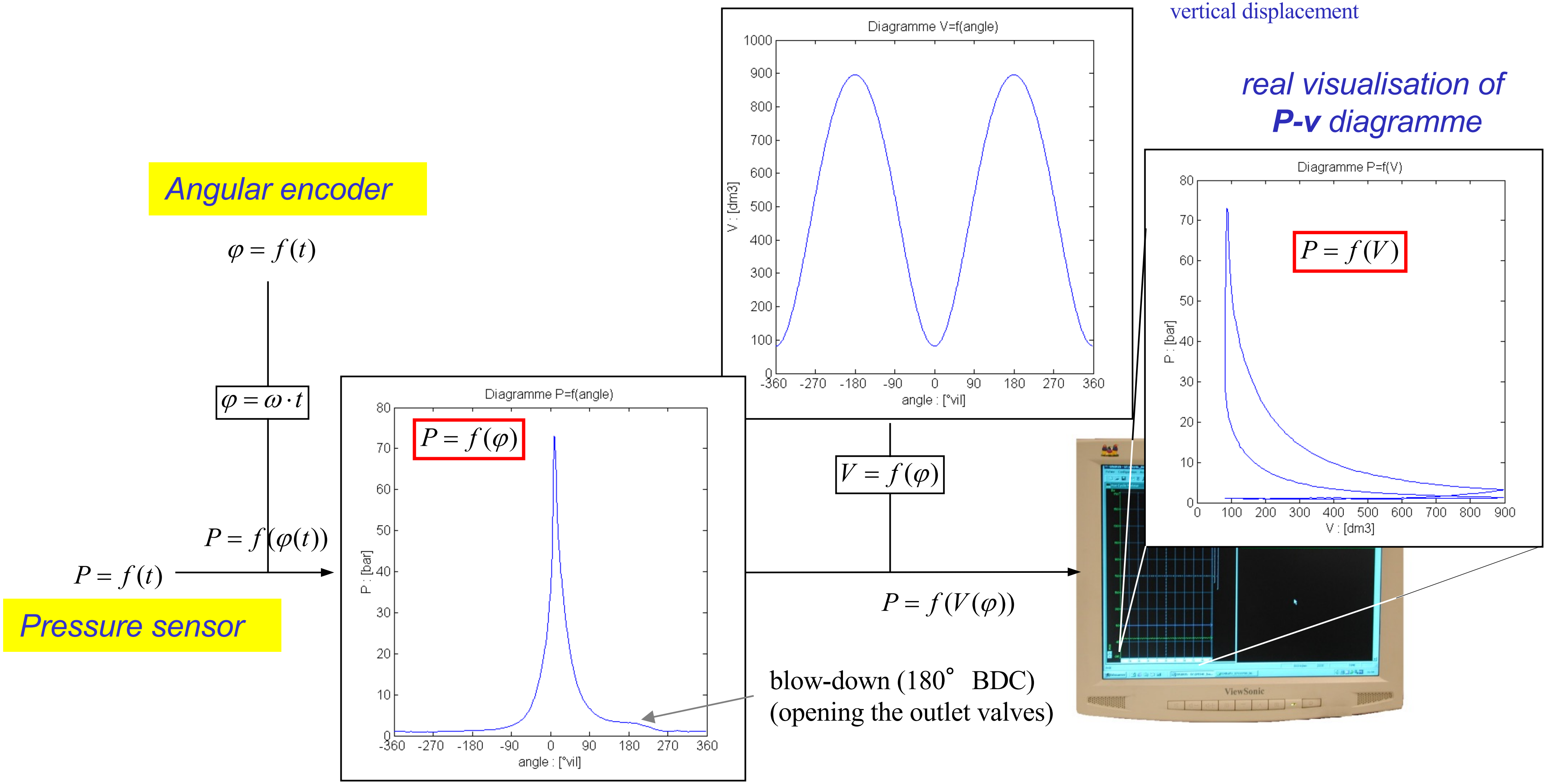
Piston motion equation:

(Chapter 1, p. 24-25)

$$V = V_o + \frac{\pi \cdot D^2}{4} x(\varphi) = \frac{\pi \cdot D^2}{4}\left(\frac{L}{\chi - 1} + x(\varphi)\right)$$

vertical displacement

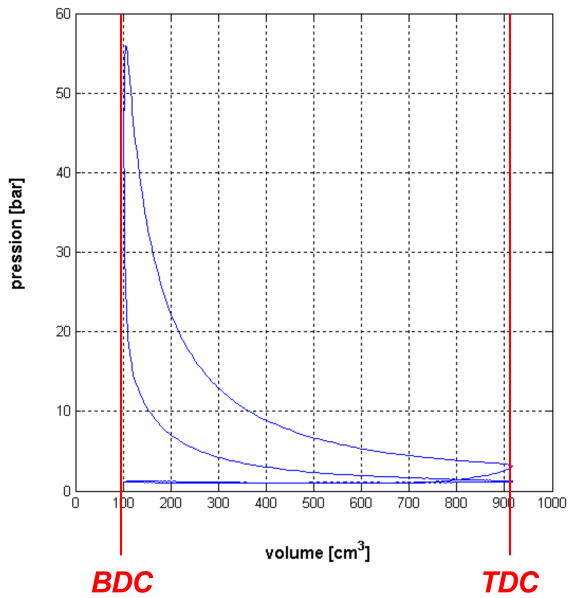
real visualisation of ***P-v** diagramme*

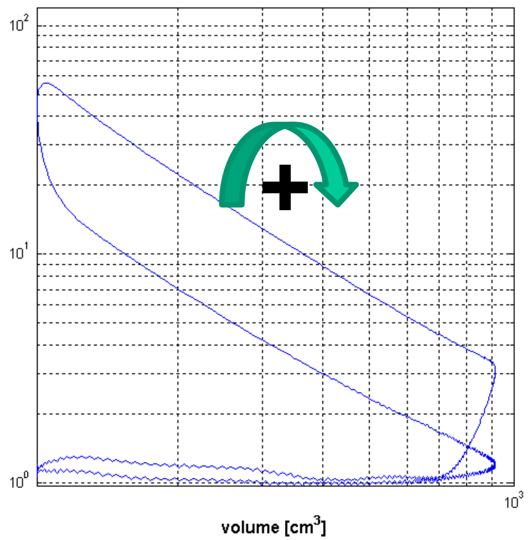


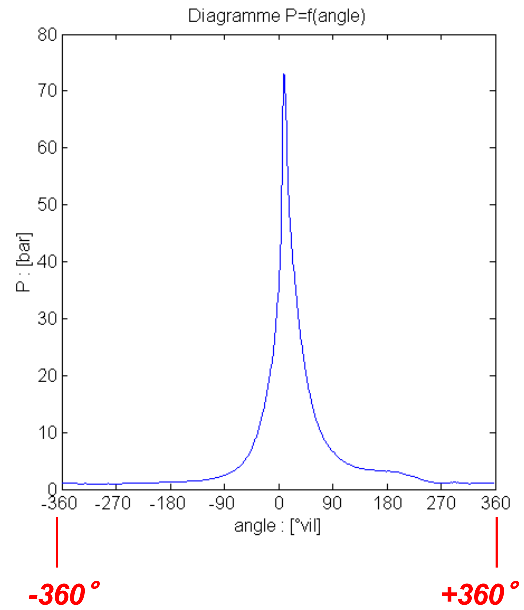
Real cycles

- Measurement method of the thermodynamic cycle on an engine
  - Type of representation of the thermodynamic cycle:

***P* = f (*V*)** **or** **log(*P*) = f (log(*V*))** ***P* = f (φ)**







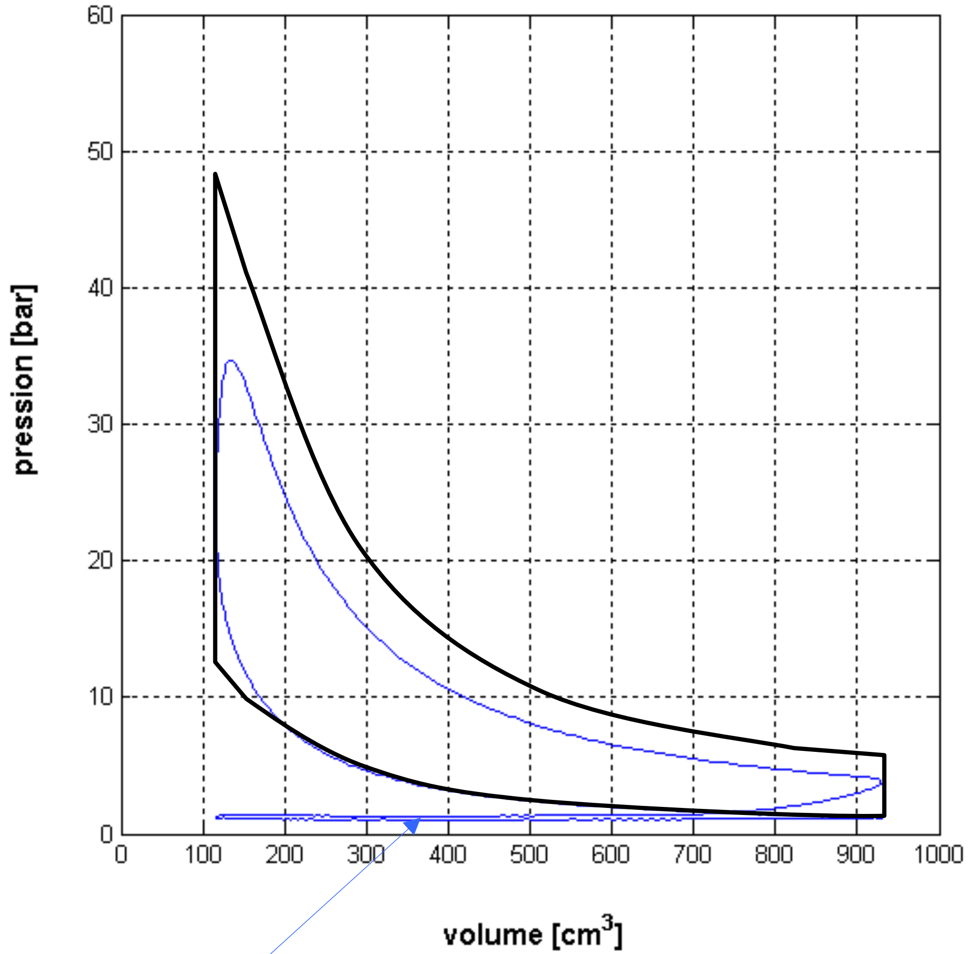
  - results/parameters resulting from the thermodynamic cycle analysis:

$$P_{\max} \;;\; \varphi(P_{\max}) \;;\; \varphi_{spark} \;;\; \varphi_{inj} \;;\; \frac{dP}{d\varphi} \;;\; IMEP \;;\; Q_{5,10,50,90} \;;\; \frac{dQ}{d\varphi}$$

SI CI noise Chapter 3

Real cycles

- Difference between **real** and **ideal** cycle



Ideal cycle (Otto)

Real cycle

$$\breve{s}_{REAL} \neq \breve{s}_{IDEAL}$$

$$(\breve{s} = surface)$$

$$\breve{s} = \oint Tds = \oint Pdv$$

negative work
(pumping loss)

Real cycles

Differences between **real** and **ideal** cycle: *reasons*

- gases transfer, pressure drop, pumping work ($\Delta P_{\text{Intake-Exhaust}}$)
  $\Rightarrow$ existence of a low-pressure loop
- valve train system, valve timing
  $\Rightarrow$ delayed compression (does not start exactly at BDC)
  $\Rightarrow$ blowdown expansion* (does not finish exactly at BDC)
- non-instantaneous combustion process (as supposed for Otto cycle)
  $\Rightarrow$ heat-release rate: $Q_F = f\,(\varphi_{\text{crank angle}})$
- wall heat transfer losses
  $\Rightarrow$ compression and expansion are NOT perfectly adiabatic !
- increase of specific heat coefficient, molecular dissociation at high T
  $\Rightarrow$ $\gamma_\sigma \neq$ constant ($\gamma_\sigma \searrow$)
  $\Rightarrow$ endothermal reactions (CO_2, H_2O $\Rightarrow$ CO, H_2 et O_2)
  (they absorb heat, hence prevent the mixture to reach higher T)

$\Rightarrow$ $\eta_{\textbf{real}\ \text{cycle}} \searrow \eta_{\textbf{ideal}\ \text{cycle}}$

*blowdown: expansion of the exhaust from the cylinder into the atmosphere at $P > P_{atm}$ = lost work

Ideal (closed) air cycle: assumptions

- Ideal gas $pV = mRT$, $p = \rho RT$
- No mass change of the working fluid (air)
- Reversible processes
- Heat supply from constant high T hot source (not from chemical reactions)
- Heat rejected to constant low T sink
- No heat loss to surroundings (adiabatic)
- Working fluid has constant Cp, Cv

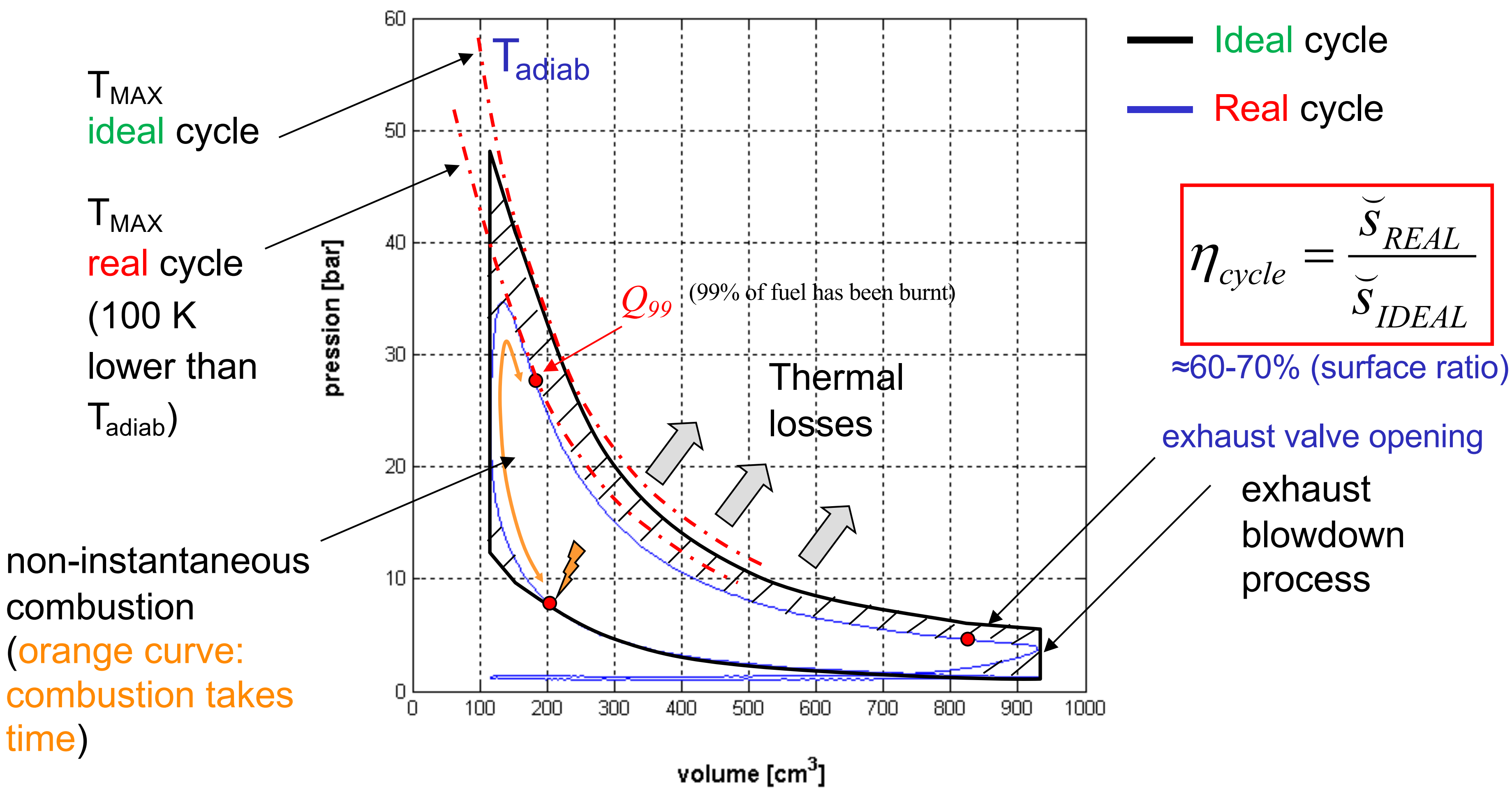
C_p = 1.005 kJ/kg
C_v = 0.717 kJ/kg
M_{air} = 28.84 g/mol

$\gamma = 1.4$

These assumptions will strongly overestimate the cycle performance.

Real cycles

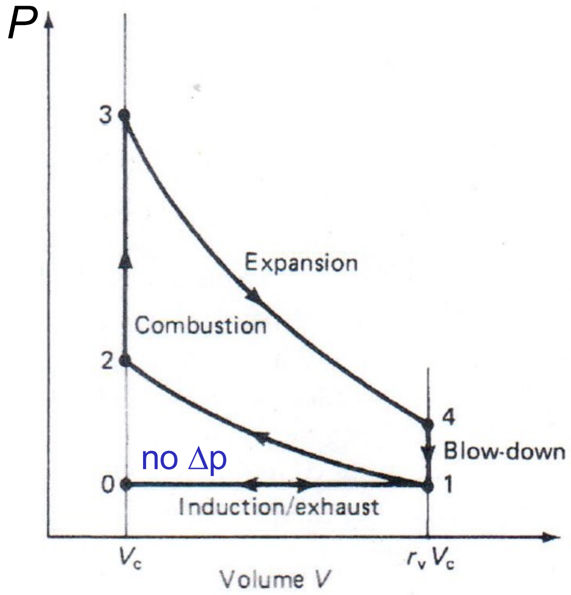
- Difference between **real** and **ideal** cycle: *illustration*



$$\eta_{cycle} = \frac{\breve{S}_{REAL}}{\breve{S}_{IDEAL}}$$

Real cycles

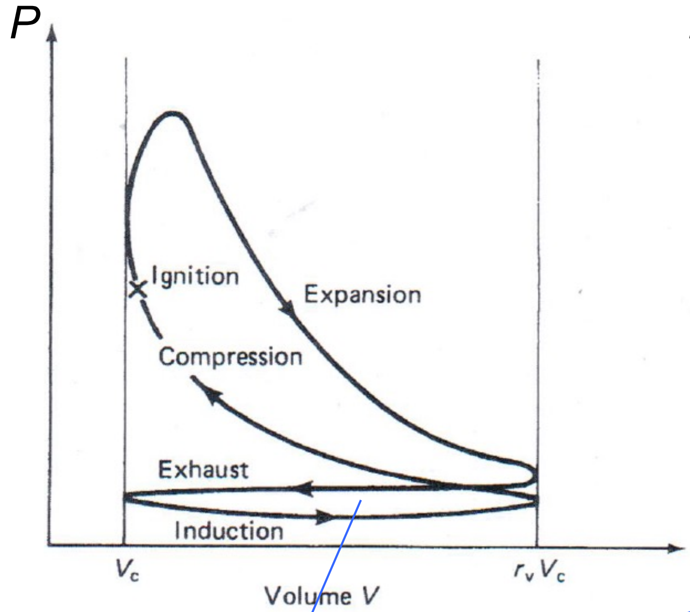
Otto-cycle:
idealised representation



R.Stone Fig. 2.7

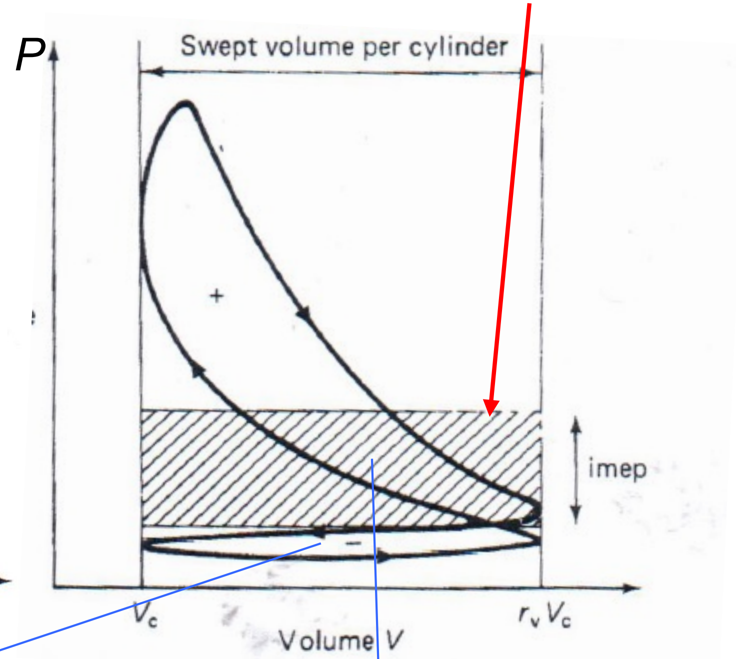
$r_v = \varepsilon$

Otto-cycle: more realistic
approx. representation



R.Stone Fig. 2.6

Otto-cycle: representation
with indicated mean pressure (IMEP)



R.Stone Fig. 2.8

negative low pressure loop

area equivalent
to net cycle surface

Real cycles

Real efficiencies of air-gasoline engine (Otto, S.I.) as f(mixture)