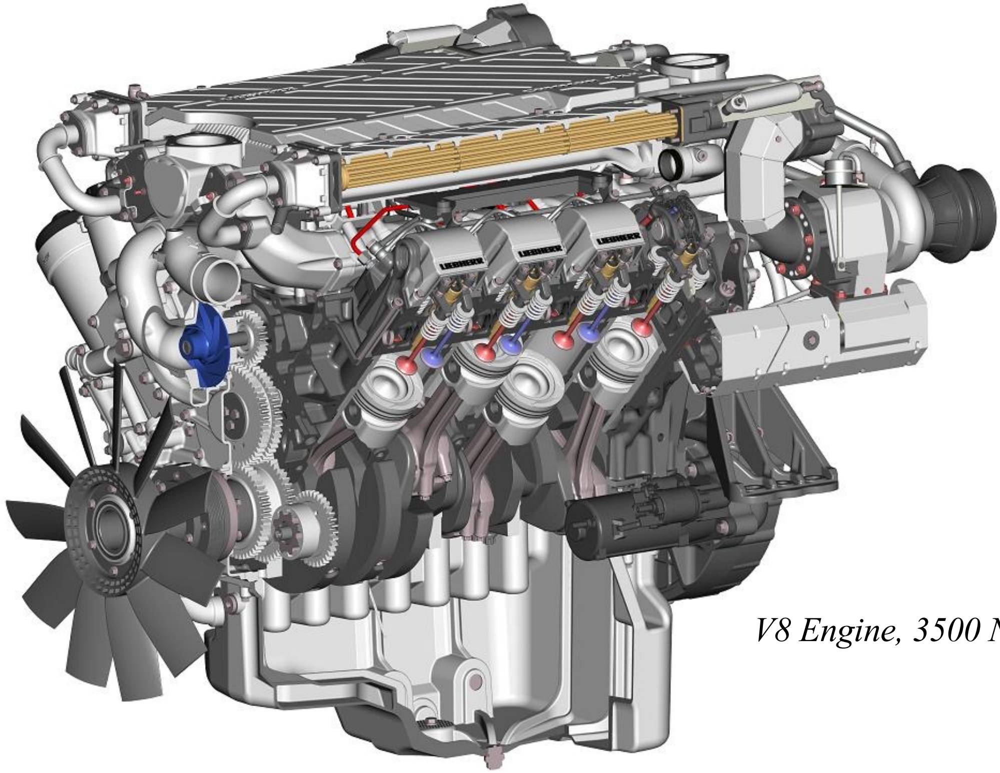
Chapter 4:
Compression Ignition Engines
(Diesel engines)



V8 Engine, 3500 Nm, 1500 rpm, 500 kW

Learning objectives of chapter 4

⇨ know the fuel properties required for a Diesel cycle and recognize the different phases of the combustion process

⇨ know the operating principles of the most common injection systems used in Diesel engines

⇨ understand the **load regulation** strategy of Diesel engines and the influence of key control parameters on performances

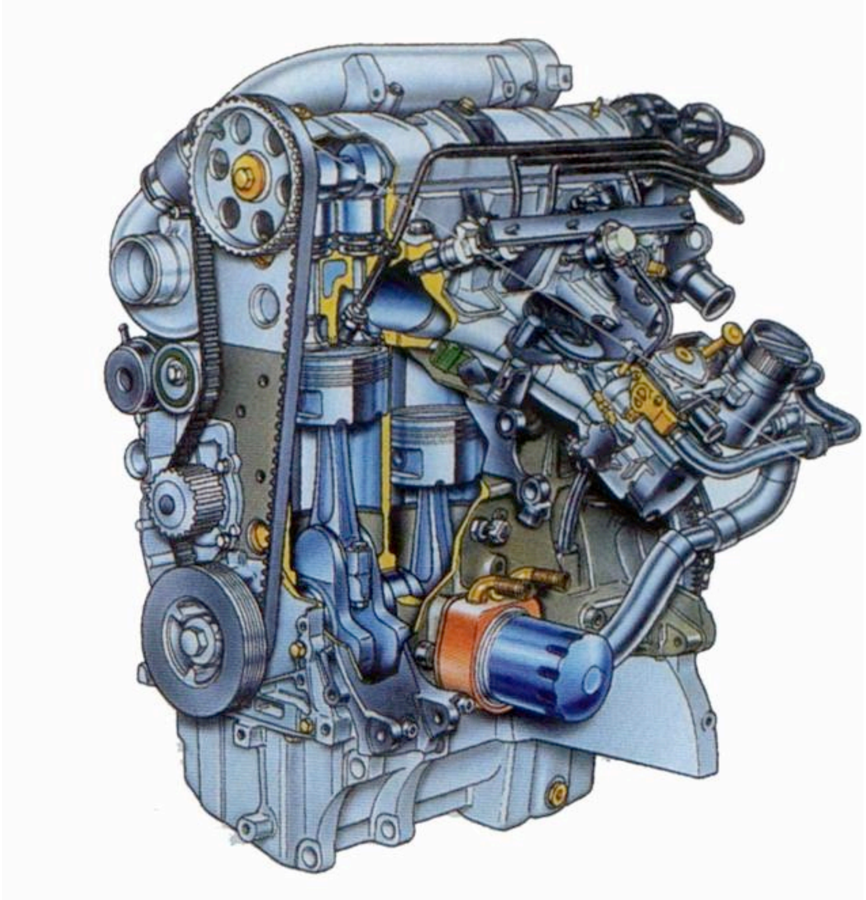
⇨ establish the thermal balance and energy distribution in a Diesel engine for (i) transport or (ii) cogeneration (=> Exercise)

Content of Chapter 4

- Application range
- Operating principle
  - Fuel properties of “Diesel” fuel
  - Injection system
  - Injection process
  - Diesel combustion process
- Load regulation parameters
  - Partial load operation
  - Full load operation
- Energy distribution in Diesel engines
  - Conventional engines
  - Cogeneration engines

Application range

- Common use and applications
  - 4-stroke cycle (< 10 MW)
    - Passenger cars (≈ 50%)
    - Heavy duty vehicles (≈ 100%)
    - Off-road vehicles
    - Agricultural machines (high V_{cyl})
    - Medium-speed marine engines
    - Generators (electricity)
  - 2-stroke cycle (> 10 MW)
    - Low-speed marine engines
    - Generators



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Operating principle

General

- The combustion process in a DIESEL engine starts thanks to the <u>spontaneous</u> ignition of the fuel injected to highly pre-compressed air in the cylinder

  ⇒ Firing by **auto-ignition** of the fuel in (highly) compressed air

  ⇒ Injection of fuel under very high pressure close to TDC

  ⇒ High compression ratio ($\varepsilon > 15$)

  ⇒ to obtain the required conditions for **auto-ignition** (initiation of combustion process)

  ⇒ Combustion in **<u>heterogeneous</u>** fuel-air mixture

- φ_{inj} : injection (crank) angle

  ⇒ defines the «timing» of the combustion in the engine cycle

- Rate of fuel injection : (it takes time!) $\boxed{\dot{M}_F = f(\varphi_{c.a})}$

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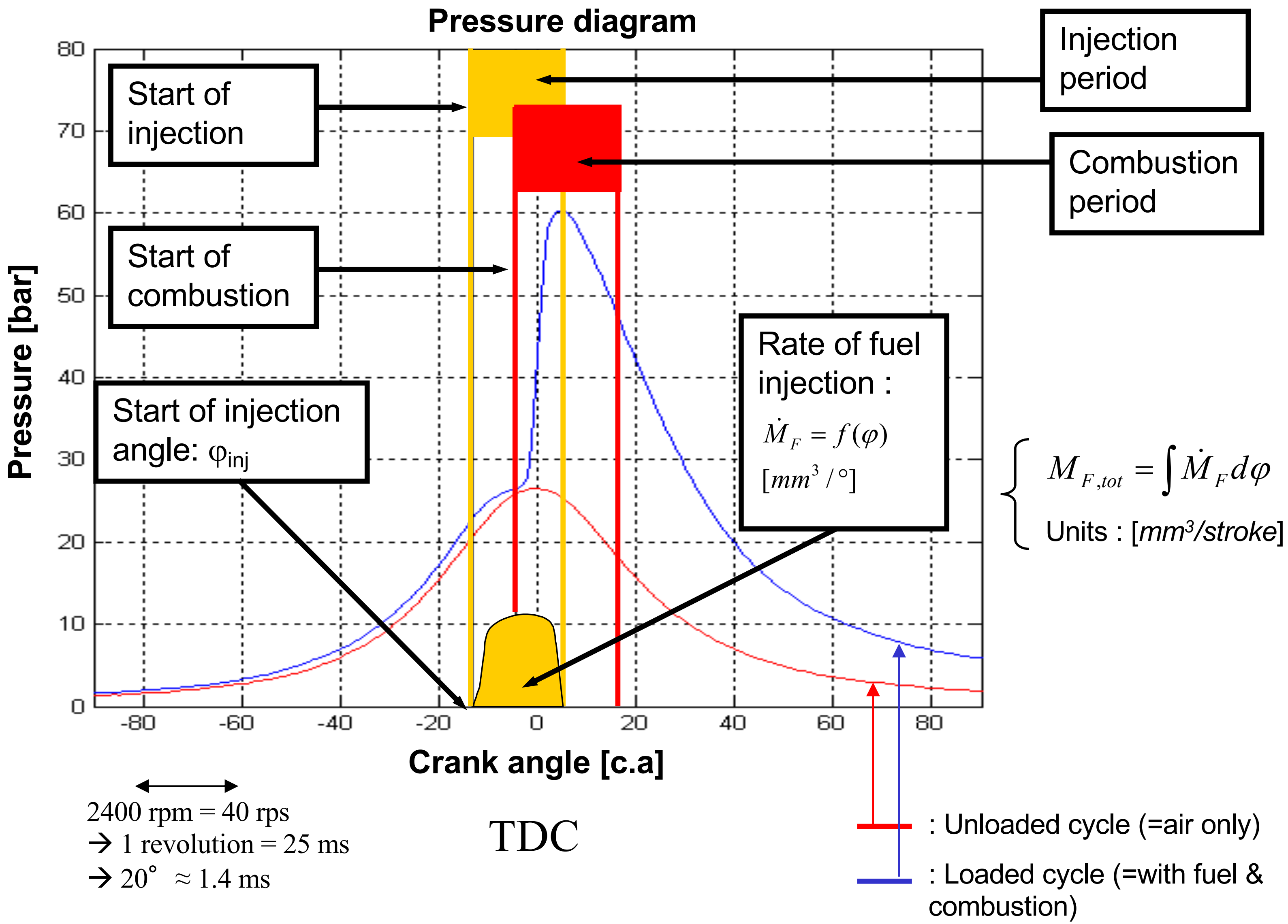
Combustion must be as fast as possible, else the exhaust T will be too high

Operating principle

■ General



Operating principle

- **Fuel properties of "Diesel"**
  - Different types of fuel have "Diesel" designation
    - ⇒ **commercial Diesel**, Biodiesel, domestic fuel, heavy fuel oil (HFO – marine)

Diesel

ρ [kg/L]	Distillation range [° C] initial T°	final T°	Nb of carbon atoms	$R_{A/F}$ [kg_{air}/kg_{fuel}]	LHV by mass [kJ / kg]	LHV by volume [kJ / L]
0.82 – 0.86	200 - 220	300 - 330	8 - 30	≈ 14.5	≈ 42'600 (*Gasoline* ≈ 42'690)	≈ 35'600 (*Gasoline* ≈ 32'000)

⚠ ⇒ The flammability range is not specified because A/F mixture is heterogeneous !

$$\frac{LHV_{vol,DIESEL}}{LHV_{vol,GASOLINE}} \cong 1.11 \, {}^{L_{gasoline}}\!/_{L_{Diesel}}$$

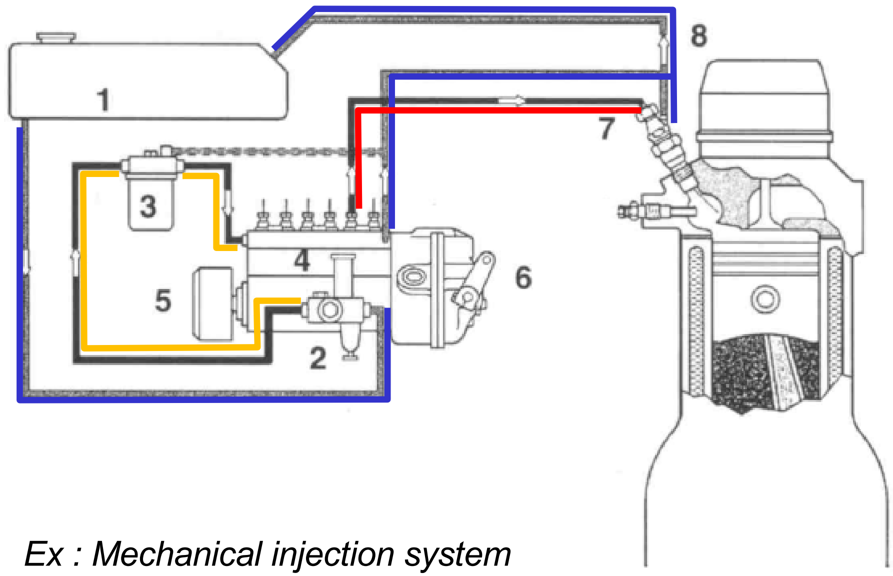
- **Cetane number : CN** ('cetane' = $C_{16}H_{34}$, hexadecane)
  - characterizes the auto-ignition capacity
  - CN xx ⇒ corresponds to a binary mixture of 2 pure HC fuels with the same behavior than the one which is analyzed (measured on a reference engine *CFR**)
    - CN 0 : corresponds to a fuel-mix identical to 100% α-methylnaphtalene (very resistant to auto-ignition)
    - CN 100 : corresponds to a fuel-mix identical to 100% n-cetane (very favorable to auto-ignition => fast combustion)
  - commercial fuels : CN ≈ 40 to 60

**CFR cooperative fuel research (definition of fuel mixtures)*

Operating principle

- Injection system (=key) for DIESEL engines
  - Composed of 2 main circuits
    - low-pressure circuit ⇒ from 3 to 6 bar
    - (very) high-pressure circuit ⇒ from 1000 to 2000 bar (and more) !
  - Injection fuel-delivery control
    - mechanical
    - electronic

1. Tank
2. Supply pump (low p)
3. Fuel filter
4. Injection pump (high p)
5. Timing device
6. Governor (fuel rate)
7. Nozzle and holder assembly
8. Overflow line



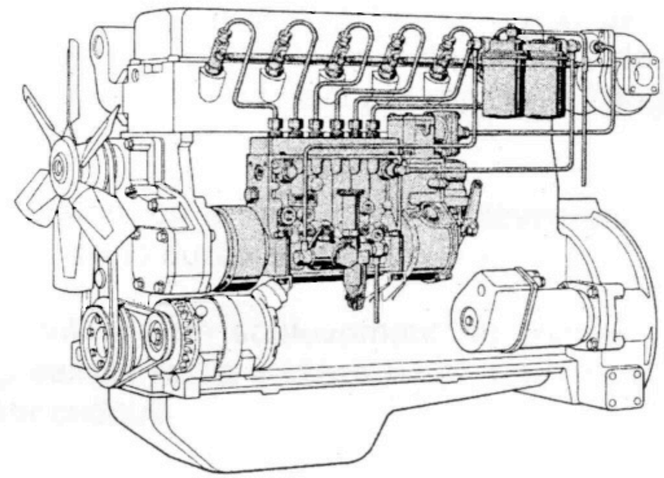
Ex : Mechanical injection system

A failing injector (or a failing turbocharger) may lead to average lambda below 1.2 and hence black smoke.

Operating principle

- ## Injection system for DIESEL engines (oldest to newest)

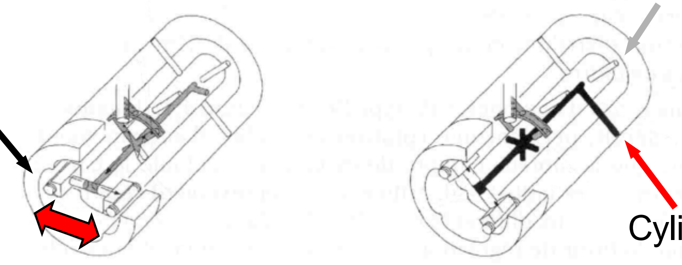
a) in-line pump

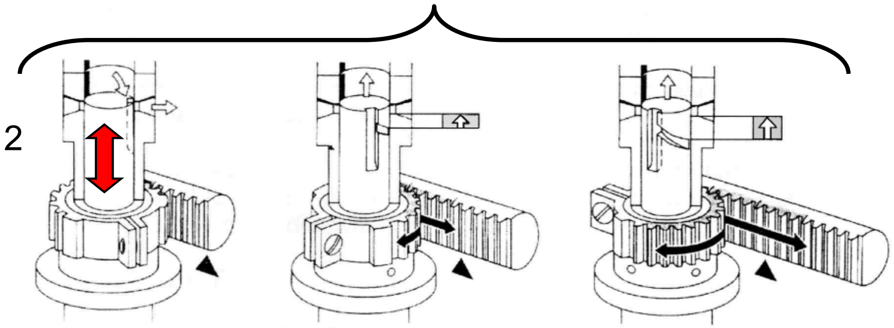


- 1 pump *"plunger"* for each cylinder
- Mechanical regulation:
  - ⇒ pump : camshaft, plunger & barrel, governor
  - ⇒ nozzle : pre-tensioned spring at $P_{\text{injector opening}}$
- Electronic regulation
  - ⇒ electro-actuators

b) distributor-type pump

- 1 pump *"plunger"*
  for all injection nozzles





Fuel quantity regulation on an in-line mechanical pump

- Mechanical *or* electronic regulation

Operating principle

- Injection system for DIESEL engines (oldest to newest)

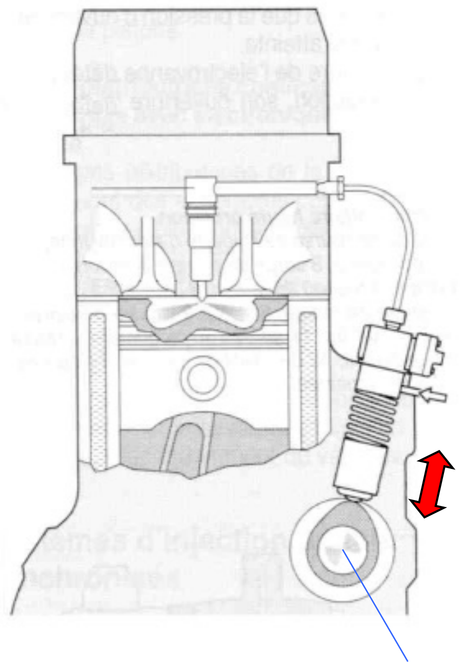
  c) Unitary pump (1980'ies)

  - 1 *"pumping unit"* per cylinder / nozzle
  - No specific camshaft *or* cam-disc is required
  - Driven by the engine camshaft
  - Very high injection pressure (⇒ 1800 bar)
  - Electronic regulation

    ⇒ Electro-actuators are mounted
    on the high-pressure circuit side
  - Injector / nozzle

    ⇒ Mechanical device with spring ($P_{discharge}$)

    activates the injector opening

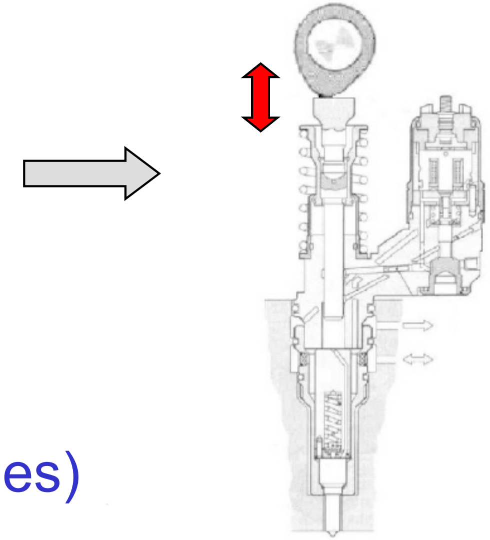


Operating principle

- Injection system for DIESEL engines

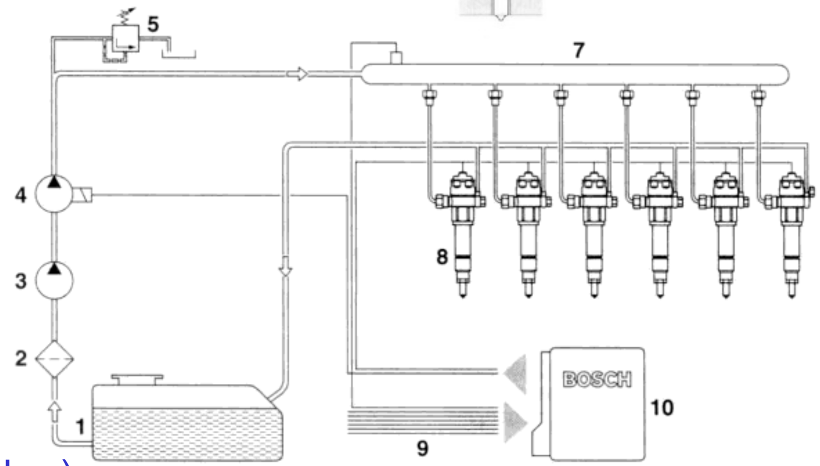
  d) Unit injector (>yrs 2000'ies)

  - Pump and injector are one unique component driven by the engine camshaft
  - «high pressure» connectors are suppressed
  - Very high injection pressure ⇒ > 2000 bar



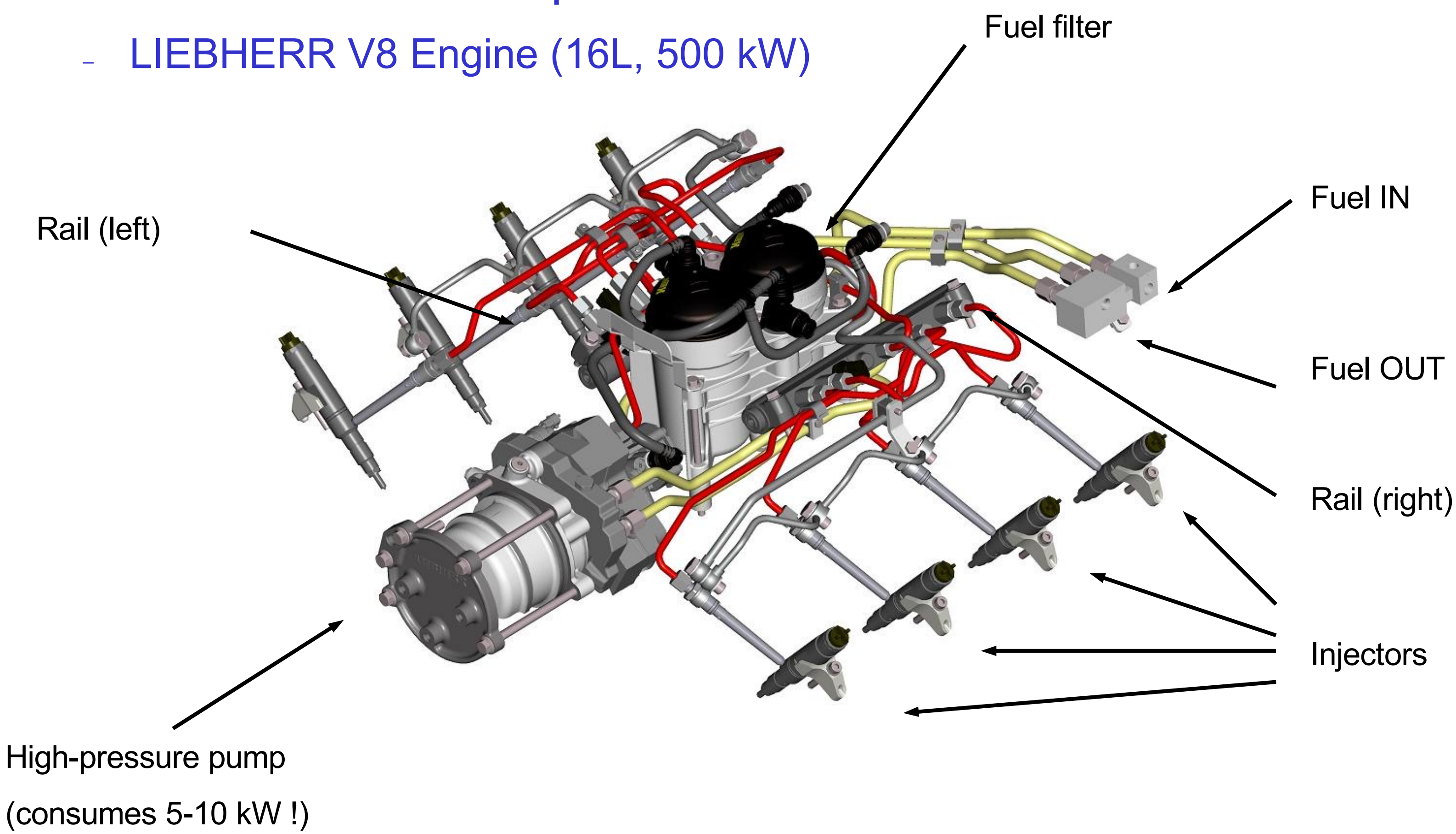
  e) **<u>Common Rail</u>** (now 99% of diesel engines)

  - *Rail* ⇒ pressure accumulator
  - Injectors with electronic actuators
    - **Multiple injections are feasible**
  - Pressure generation is obtained by:
    - In-line pump
    - Distributor-type pump
  - Variable injection pressure of 400 to 1800-2000 bar (since 2010 : > 2200 bar)



Operating principle

- Injection system for DIESEL engines
  - **Common Rail** : example
    - LIEBHERR V8 Engine (16L, 500 kW)



Operating principle

- Types of DIESEL combustion systems
  - INDIRECT-injection IDI : (old technology)
    - Combustion chamber divided into 2 regions

      a) Prechamber (into the cylinder-head)

      b) Combustion (main) chamber
    - a) and b) are connected via a nozzle
    - Creation of high turbulence (v_{flow} ⬀ ⬀)
    - $\varepsilon \approx$ 21:1 to 24:1 (very high T and NOx)

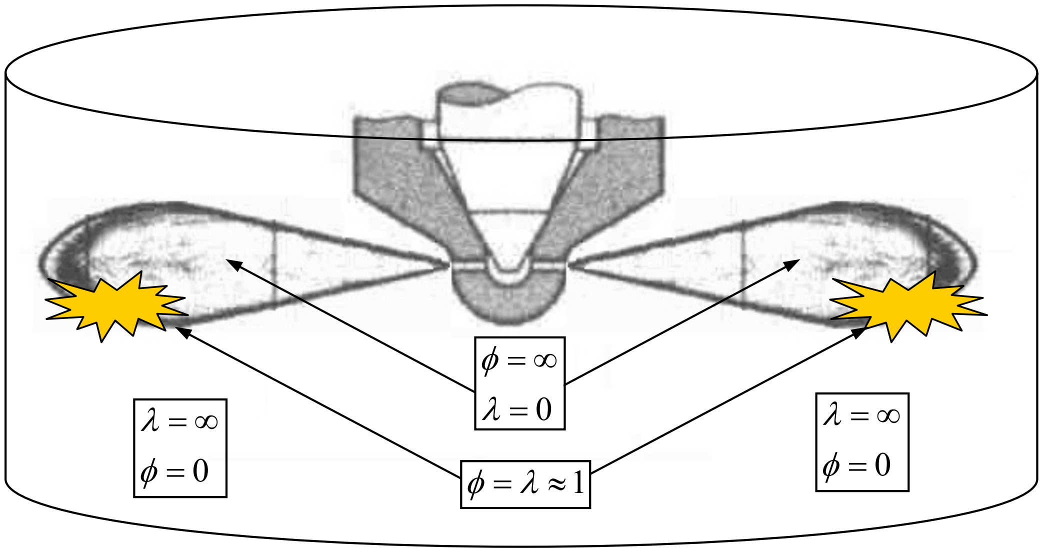
  - DIRECT-injection DI : (all types of Diesel)
    - Combustion chamber = 1 unique volume
      defined by the piston shape (bowl) and cylinder head.
    - The control of internal air flow motion (**swirl**)
      and the **design** of injector (hole **geometry**, number)
      are CRITICAL for good operation
    - $\varepsilon \approx$ 15:1 to 21:1 (lower T and NOx)
    - always with a turbocharger (2-3 bar air pressure)

Operating principle

- DIESEL combustion process
  - The injection system of Diesel engines induces a strong gradient of the air-fuel mixing into the combustion chamber

    $\Rightarrow$ Diesel combustion process occurs in a highly **<u>heterogeneous</u>** environment:

$$\boxed{\begin{aligned}\lambda_{global} &= \overline{\lambda} \\ \phi_{global} &= \overline{\phi}\end{aligned}}$$



! $\Rightarrow$ We use the notion of a <u>global</u> Air/Fuel ratio (or Fuel/air ratio) which corresponds to the mean value of the mixing !

Operating principle

- DIESEL combustion process
  - The combustion sequence takes place in 3 different stages:

I. Ignition delay

segment AB

Determined by the cetane number CN.
The higher CN, the faster the combustion

II. Premixed (or rapid) combustion phase (uncontrolled)

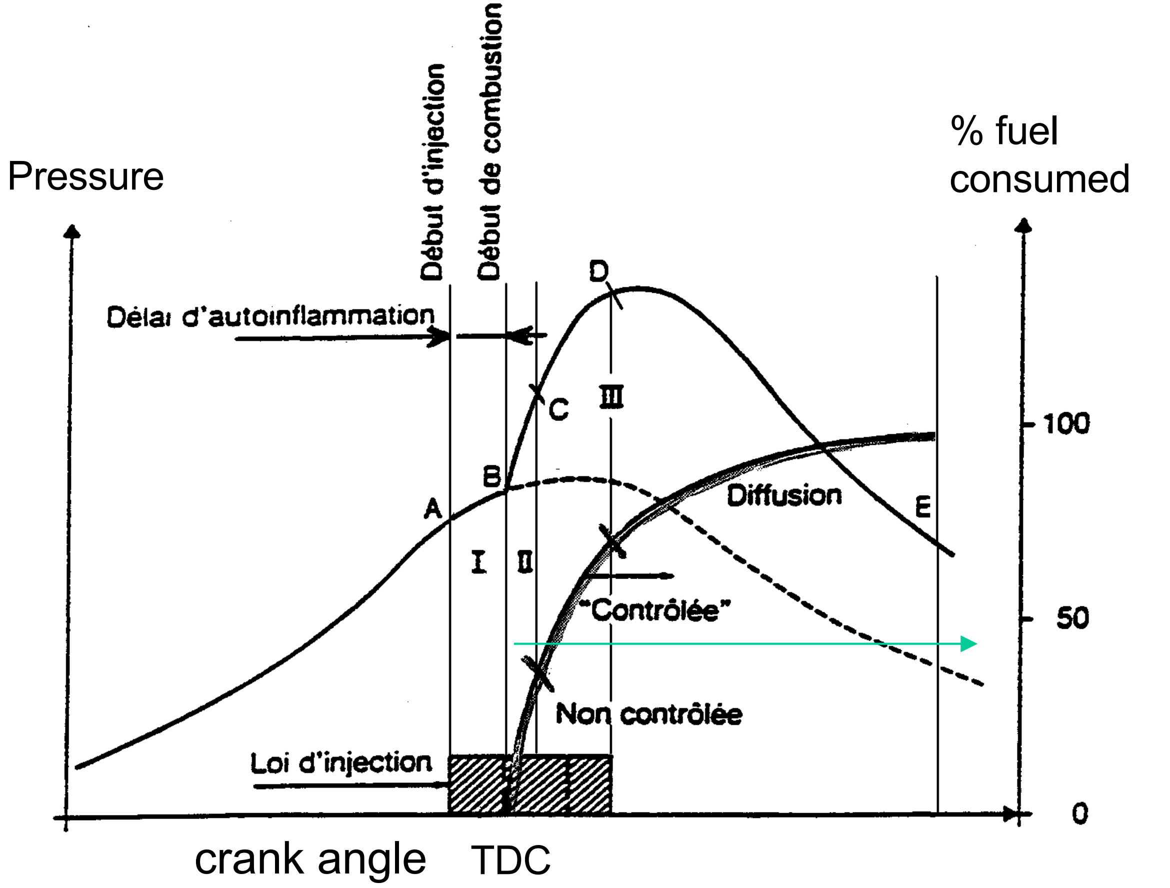
segment BC

steep P-gradient (noise)

III. Diffusion (or mixing-controlled) combustion phase

segment CE

~ 'constant' P-regime, characteristic of diesel cycle (see Chapter 2)



Operating principle

- ## DIESEL combustion process

I) Ignition delay : (segment AB)

The period between the start of fuel injection and the start of the combustion (determined from the change in slope on P-φ). Depends on:

- **Physical delay** (atomization, vaporization and diffusion of fuel before reaching the conditions of auto-ignition)
  - $\Rightarrow$ jet characteristics (hole shape (conical), number of holes, $P_{injection}$)
- **Chemical delay** (chemical process which leads to the auto-ignition)
  - $\Rightarrow$ Fuel properties (type, cetane number CN)
  - $\Rightarrow$ Thermodynamic conditions (pressure and temperature)
  - $\Rightarrow$ <u>local</u> A/F ratio

II) Premixed combustion phase : (segment BC)

After the injection, the vaporized fuel, which has mixed with air, burns rapidly in a few crank angle degrees and <u>without any control</u>.

$\Rightarrow$ phase conditioned by the auto-ignition delay & rate of fuel injection

$\Rightarrow$ strong pressure gradient (rise in bar per crank angle° = source of **noise** for Diesel engines)

Operating principle

DIESEL combustion process

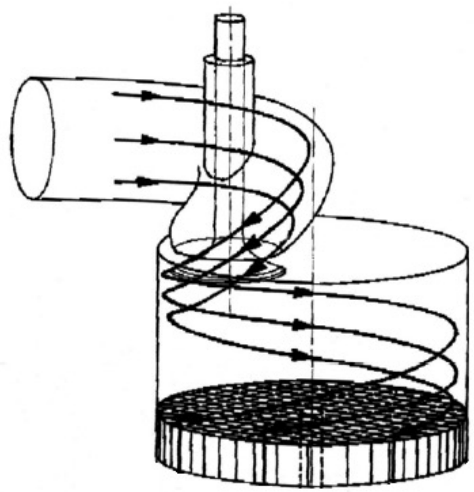
III) Diffusion combustion phase : (segment CDE)

Progressive and <u>controlled</u> fuel combustion by diffusion flame as and when the fuel is injected and meets oxygen necessary for the combustion.

Diffusion phase includes 2 parts :

1. Mixing-controlled phase. Controlled by the processes of <u>vaporization</u> and <u>mixing</u> during injection: *segment* CD
2. Late combustion phase. Small fraction of fuel may not yet have burned and combustion continues after the end of the injection during expansion: *segment* DE

⇒ Internal air flow motion (**turbulence**) generated by ***swirl*** accelerates the combustion process by diffusion (and also to prevent fuel jets from hitting the cylinder walls (sooth formation, cavitation)



Stream lines: *swirl*

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Load regulation parameters

- Relations
  - **Monocylinder :** $\boxed{\dot{E}_i = n_c \cdot E_i}$ and $\eta_i = \frac{\dot{E}_i}{\dot{Y}_{comb}} \approx \frac{\dot{E}_i}{\dot{M}_F \cdot LHV}$

    - $\dot{E}_i$ as a function of $\boxed{M_F}$: $\boxed{\dot{E}_i = n_c \cdot \eta_i \cdot M_F \cdot LHV}$ with $\dot{M}_F = n_c \cdot M_F = n_c \cdot \frac{M_A}{R_{A/F}} \cdot \frac{1}{\lambda}$

      $\dot{M}_F$: Fuel quantity introduced per cycle (and per cylinder)
      n_c : number of engine cycles per second

    - $\dot{E}_i$ as a function of $\boxed{M_A}$: $\boxed{\dot{E}_i = n_c \cdot \eta_i \cdot M_A \cdot \underbrace{\frac{LHV}{R_{A/F} \cdot \lambda}}_{q}}$ with $q = \begin{cases} \frac{LHV}{R_{A/F}} \cdot \frac{1}{\lambda} & if \quad \lambda \geq 1 \\ \frac{LHV}{R_{A/F}} & if \quad \cancel{\lambda \leq 1} \end{cases}$

      (λ<1 never the case with Diesel, always air excess)

      M_A : Air mass introduced per cylinder
      q : Energy content of the mixture: [kJ/**kg**]

    - M_A as a function of P_{coll}, T_{coll} : $\boxed{M_A = \eta_{vol} \cdot \frac{P_{coll}}{r \cdot T_{coll}} \cdot V_u}$ ideal gas law or $M_A = \eta_{vol} \cdot \rho_{coll} \cdot V_u$

      η_{vol} : Volumetric efficiency
      V_u : Cylinder displacement

      $\boxed{\eta_{vol} = \frac{M_{A,real}}{M_{A,ideal}}}$ with $M_{A,ideal} = \rho_{coll} \cdot V_u$

‘collector’ = ‘intake’, ‘adm’,…

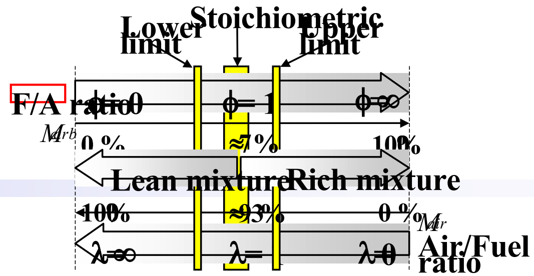
Load regulation parameters

- Relations
  - **Monocylinder** :
    - $\dot{E}_i$ as a function of P_{coll}, T_{coll} : $$\boxed{\dot{E}_i = n_c \cdot \eta_i \cdot \eta_{vol} \cdot \frac{P_{coll}}{r \cdot T_{coll}} \cdot V_u \cdot q}$$
    - $\dot{E}_i$ as a function of M_A : $\boxed{\dot{E}_i = \eta_i \cdot \dot{M}_A \cdot q}$ with $$\boxed{\dot{M}_A = n_c \cdot \underbrace{\eta_{vol} \cdot \frac{P_{coll}}{r \cdot T_{coll}} \cdot V_u}_{M_A}}$$

    Note: for naturally aspirated Diesel engine ⇒ $P_{coll} \approx P_{atm} \rightarrow \boxed{\dot{M}_A = f(N)}$
  - **Multicylinder** : $$\boxed{\dot{E}_i = \eta_i \cdot \underbrace{\frac{N}{60 \cdot n_{TM}}}_{n_c} \cdot \underbrace{\eta_{vol} \cdot \frac{P_{coll}}{r \cdot T_{coll}} \cdot V_{cyl}}_{M_A} \cdot q}$$ with $V_{cyl} = n \cdot V_u$ ($n_{TM} = 2$ for 4-stroke)
    - Effective / Break power : $\dot{E}_e = \eta_{org} \cdot \dot{E}_i$ ⇒ $$\boxed{\dot{E}_e = \underbrace{\eta_{org} \cdot \eta_i}_{\eta_e} \cdot \underbrace{n_c}_{N/120} \cdot M_A \cdot q}$$

Massflow in diesel engine is a fct of the engine speed N (for atm. intake; else, in addition, also on the turbocharge compression)

Load regulation parameters



- **Summary**
  - Effective power depends on ⇒ $\dot{E}_e = f(\cancel{\eta_{org}}, \cancel{\eta_i}, \cancel{N}, M_{air}, q)$

Means of action on effective power $\dot{E}_e$:

Mass of air : M_A

$$M_{air} = f(\eta_{vol}, T_{adm}, P_{adm})$$

Energy content of the mixture : q

$$q = f(\cancel{LHV}, \cancel{R_{A/F}}, \lambda)$$

Pressure variation in the manifold

Nat. aspirated engines: $P_{adm} \approx P_0$

Turbocharged engines: $P_{turbo\ min} < P_{adm} < P_{turbo\ max}$

Fuel/Air ratio variation

$\lambda_{min} < \lambda < \lambda_{max}$

$\lambda_{smoke\ limit} < \lambda < \infty$

P_{adm} ⇒ **Potential to increase the maximal power of the engine (FULL LOAD)**

= act on the air turbocharger (P)

λ ⇒ **Means of action to change the engine's load (PARTIAL LOAD)**

= act on the fuel injector (P)

Load regulation parameters

Partial load operation

- Action on M_F :

$$\dot{E}_e = \eta_{org} \cdot \eta_i \cdot \underbrace{\dot{M}_A \cdot q}$$

with $q = \frac{LHV}{R_{A/F}} \cdot \frac{1}{\lambda}$

and $\dot{M}_A = \dot{M}_F \cdot R_{A/F} \cdot \lambda$

$$\dot{E}_e = \eta_e \cdot \underbrace{\frac{N}{120} \cdot M_F}_{\dot{M}_F} \cdot LHV$$

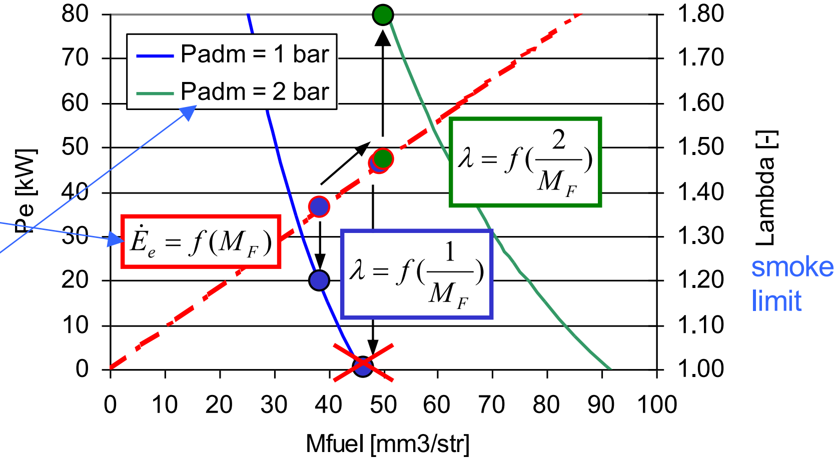
if N = const.

$$\boxed{\dot{E}_e = f(M_F)}$$

$\Rightarrow$ The power regulation in **partial load** is done by acting on the fuel quantity injected into the cylinder M_F

$$\left.\begin{aligned} V_{cyl} &= 2\ L \\ \eta_e &= 40\ \% \\ N &= 2000\ rpm \end{aligned}\right\} \longrightarrow \dot{E}_e \approx cte \cdot M_F$$

$$\left.\begin{aligned} T_{adm} &= 25° \\ \eta_{vol} &= 100\ \% \end{aligned}\right\} \longrightarrow \lambda = cte \cdot \frac{M_A(f(P_{adm}))}{M_F}$$



(100 mm³/stroke = 0.1 ml)

Verification:
2000 rpm = 33 rps = 16.5 stroke/s
50 mm3/stroke = 0.05 ml => 16.5 stroke/s = 0.82 ml/s = 0.65 g/s = 28 kJ/s
=> 40% efficiency = 11 kW => 4 cylinders = 44 kW.

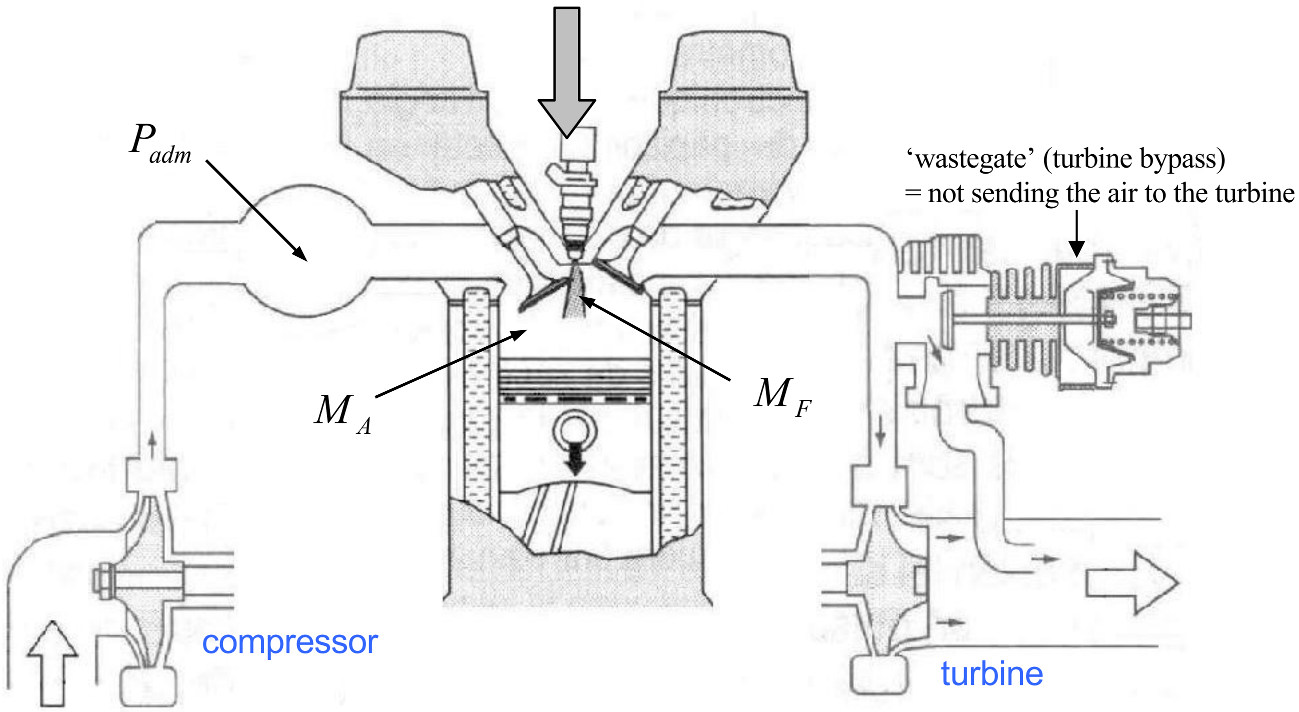
Load regulation parameters

- **Partial** load operation
  - Regulation of E_e by acting on the fuel quantity injected : [mm^3/stroke] (with constant air flow rate!)

$$\boxed{M_F \propto \dot{E}_e}$$

$$\nearrow M_F \rightarrow \searrow \lambda \ (\nearrow \phi) \rightarrow \dot{E}_e$$

M_A	$\rightarrow$	cte
P_{adm}	$\rightarrow$	cte
M_F	$\rightarrow$	var
λ	$\rightarrow$	var



Load regulation parameters

- **Full** load operation
  - Reminder : Effective power depends on $\Rightarrow$ $\dot{E}_e = f(\eta_{org}, \eta_i, N, q, M_A)$

– η_{org} : Increase of friction losses with increase of engine speed

$\eta_{org} = f(N) \Rightarrow$ no major influence

– η_i : The "timing" of the combustion in the engine cycle influences the *SFC*

$\varphi_{inj} = f(SFC) \Rightarrow$ low influence

– *N (rpm)*: $\sigma_{mechanical}$ (stress) and the combustion by diffusion limit the maximal engine speed

Example in automotive applications: $N_{max} < 5000$ rpm $\Rightarrow$ limited influence

– *q* : Limit on λ to avoid the appearance of black smoke ($\lambda_{mean} \approx 1.25$)

$$q_{\max} = \frac{LHV}{R_{A/F}} \cdot \frac{1}{\lambda_{\max}} \rightarrow \approx 2350\,[kJ/kg_{air}]$$ $\Rightarrow$ limited influence

– M_A : Practically, there are no limits to an increase of the air flow

Synthesis $\Rightarrow$ $\boxed{\dot{E}_{e,\max} = f(M_A)}$

Load regulation parameters

- **Full** load operation
  - Action on M_A :

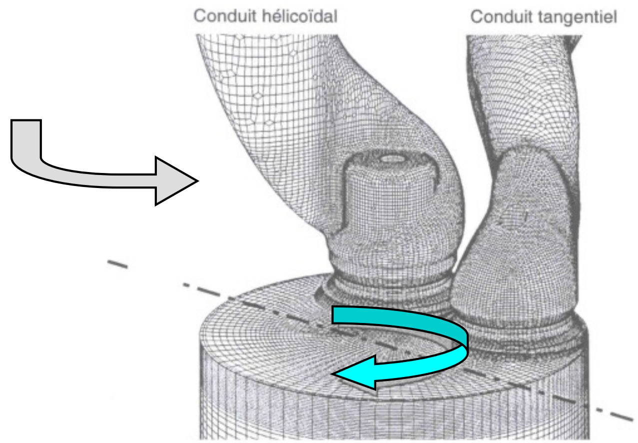
$$\dot{E}_{e,\max} = f(M_A)$$

$$M_A = \eta_{vol} \cdot \frac{P_{coll}}{r \cdot T_{coll}} \cdot V_u$$

$$\boxed{\dot{E}_e = f(\eta_{vol}, P_{coll}, T_{coll})}$$

$\Rightarrow$ Maximal power is obtained by <u>increasing the mass air flow</u> of the engine

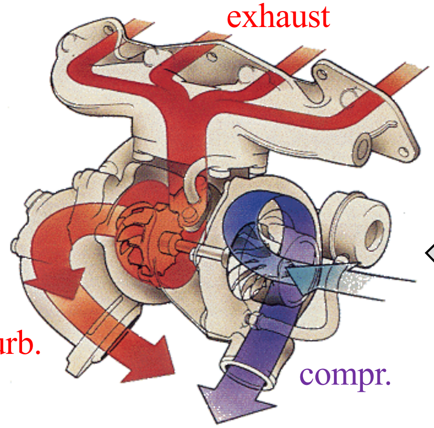
- $\eta_{vol} \Rightarrow$ constraints due to the internal air motion
- $P_{coll} \Rightarrow$ Superchargers and Turbochargers
- $T_{coll} \Rightarrow$ Turbocharged Heat exchangers (intercooled compression)

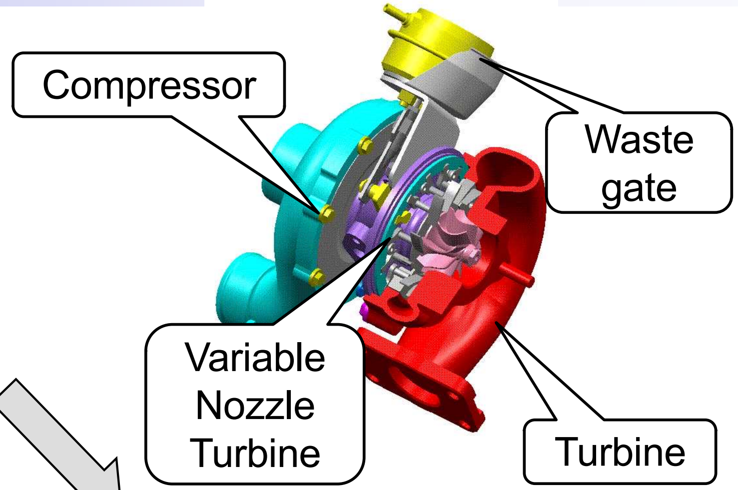


Load regulation parameters

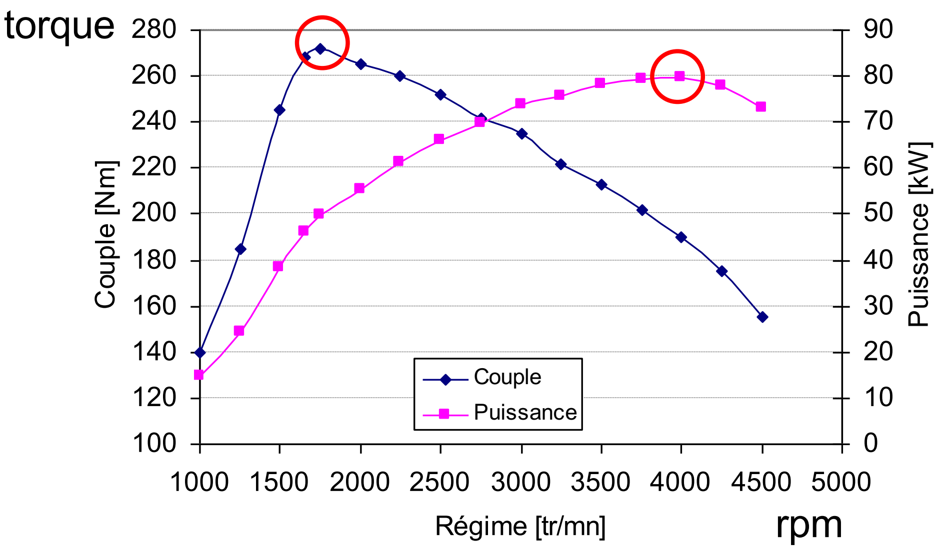
- **Full** load operation
  - Turbochargers

A challenge is to find a well matching compressor map that meets all situations of the engine map.



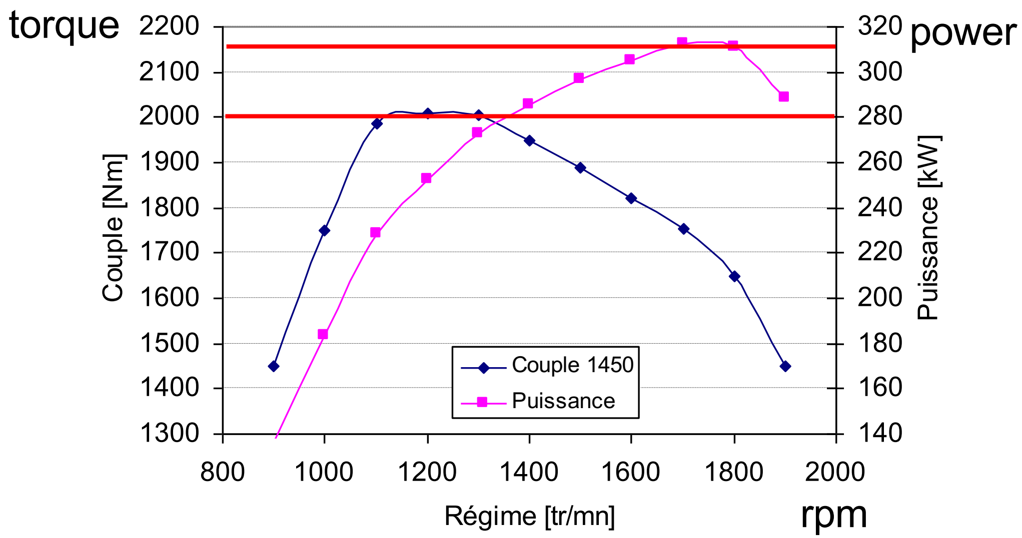


PSA 2.0HDI



car

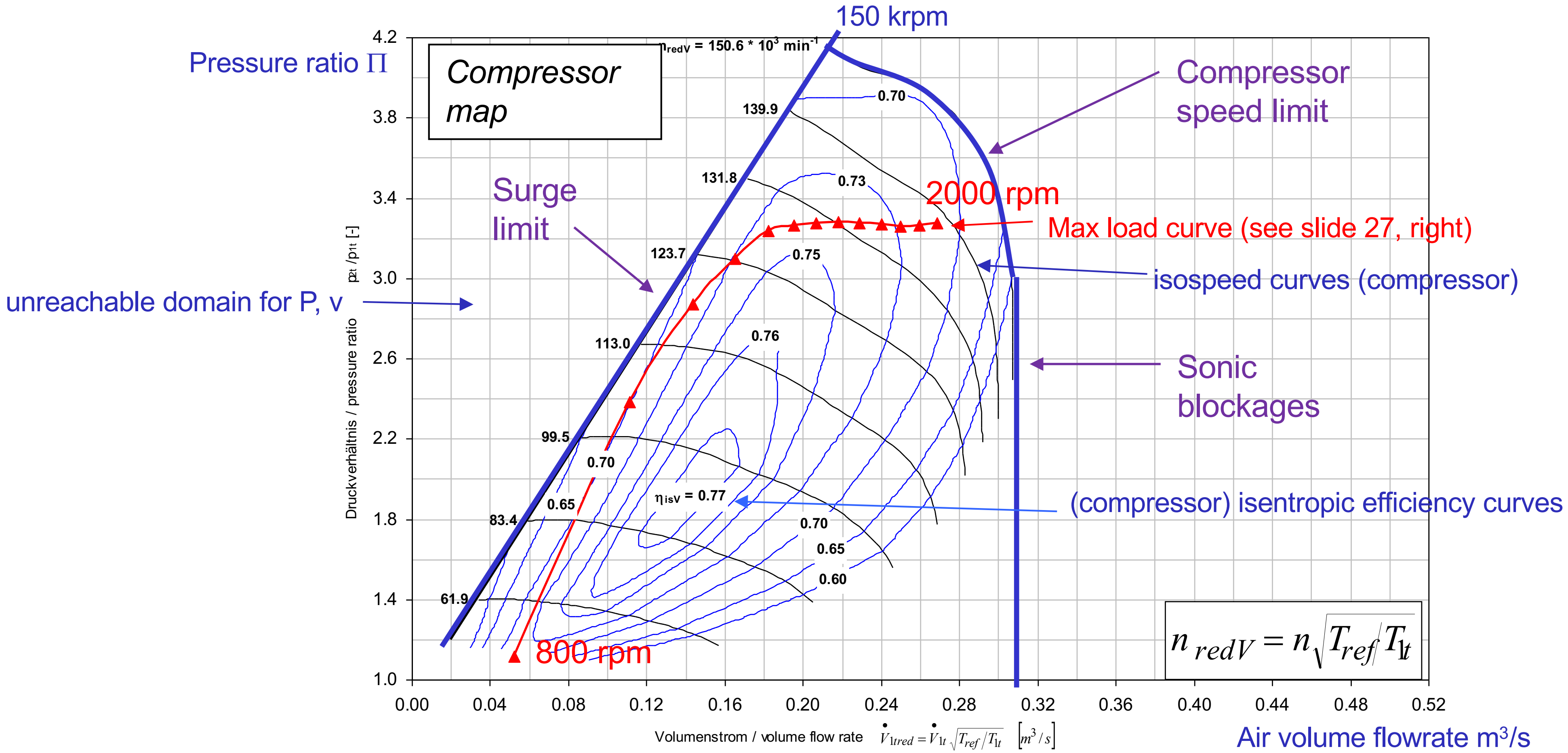
VOLVO D12



truck

Load regulation parameters

- **Full** load operation
  - Turbocharging ⇒ interaction between engine & turbomachinery



Sonic limit: pumping 300 L/s air through a 10 cm² section corresponds to a speed of 300 m/s = 1000 km/h.

Load regulation parameters

- **Full** load operation
  - Increase of $\dot{E}_e$ by acting on the air mass introduced into the cylinder

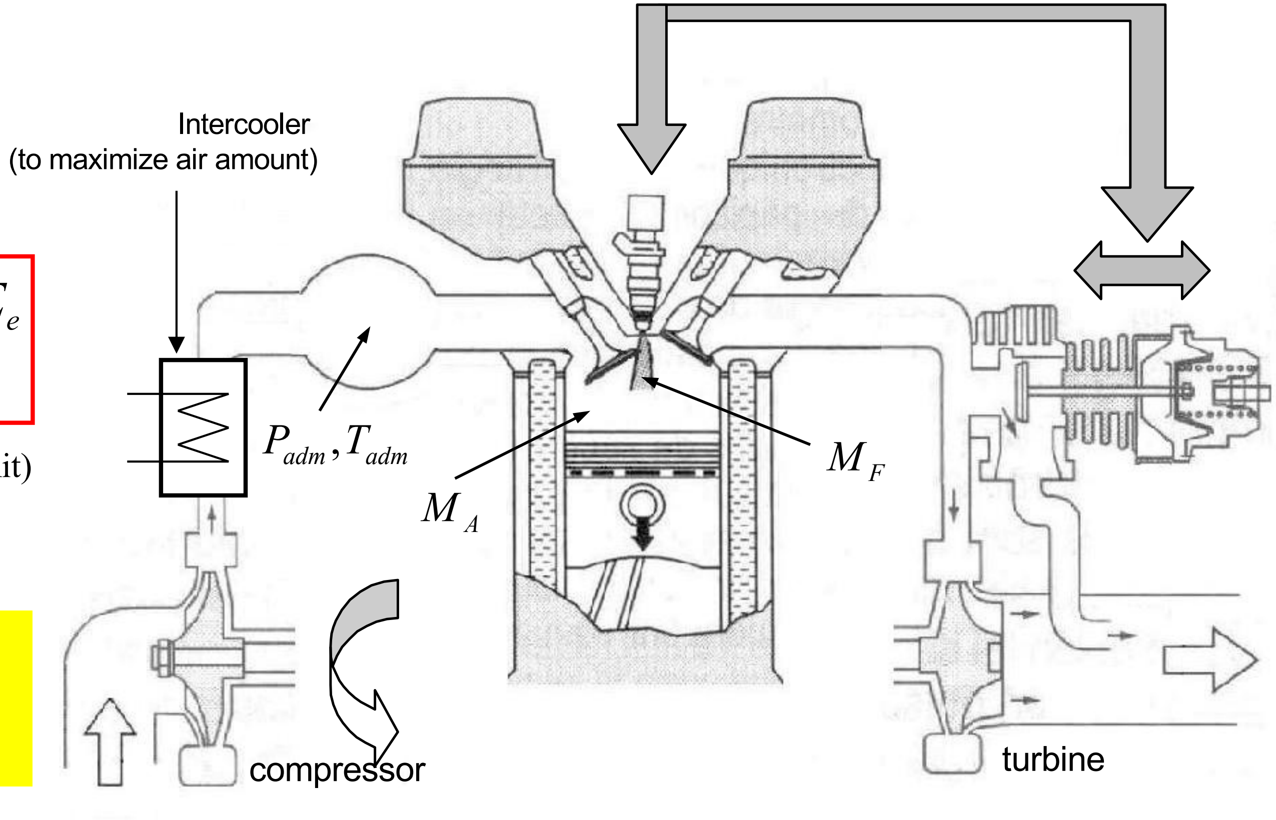
$$\searrow S_{WG} \rightarrow \nearrow E_{comp} \rightarrow \nearrow P_{adm}\ (\searrow T_{adm}) \rightarrow \nearrow M_A\ (\propto \nearrow M_F) \rightarrow \dot{E}_e$$

waste gate section

$$\begin{array}{lcl} \lambda & \rightarrow & cte \\ M_A & \rightarrow & \text{var} \\ M_F & \rightarrow & \text{var} \end{array}$$

$$\boxed{\begin{array}{l} M_A(\propto M_F) \propto \dot{E}_e \\ \lambda = \lambda_{max} > 1.20 \end{array}}$$

(rem: λ 1.2 = black smoke limit)



Diesel Engine =
operates permanently in air excess (lean mixture)

Content of Chapter 4

- Application range
- Operating principle
  - Fuel properties of a “Diesel”
  - Injection system
  - Injection process
  - Diesel combustion process
- Load regulation parameters
  - Partial load operation
  - Full load operation
- Energy distribution in Diesel engines
  - Conventional engines
  - Cogeneration engines

Energy distribution in Diesel engines

- ## Conventional engines (boundary F)

$\dot{Y}^{+}_{comb}$ = *fuel transformation power*

$\dot{E}^{-}_{e}$ = *work rate (effective power)*

$\dot{Q}^{-}_{G}$ = *heat rate losses of the exhaust gas*

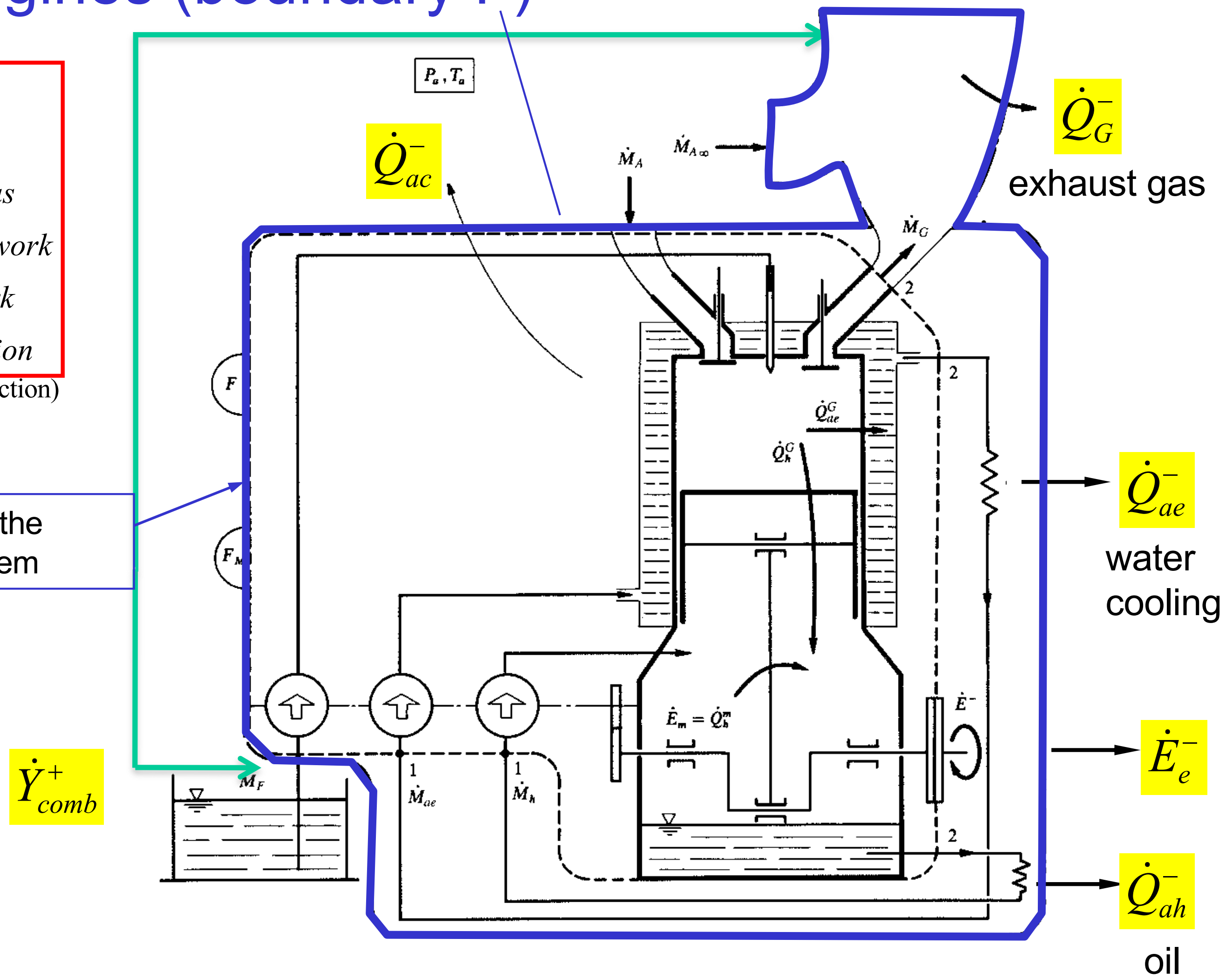
$\dot{Q}^{-}_{ae}$ = *heat rate losses of the water network*

$\dot{Q}^{-}_{ah}$ = *heat rate losses of the oil network*

$\dot{Q}^{-}_{ac}$ = *heat rate deperdition by conduction*

(and radiation, convection)

F : Boundary of the « engine » system



[Scheme of a Diesel engine]()

Energy distribution in Diesel engines

- Conventional engines (boundary F)

classical DIESEL engine

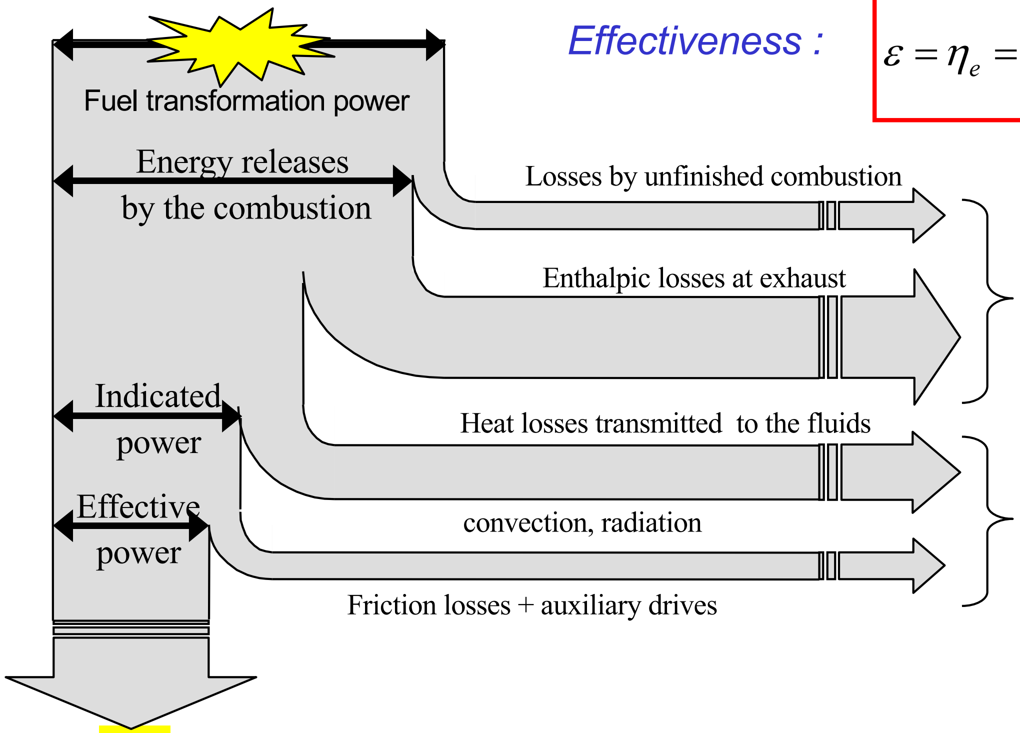
$$\dot{Y}^{+}_{comb} = \dot{M}_F \cdot \Delta h_i^0 + \dot{M}_{cond} \cdot q^0_{vap} + \left(\dot{M}_F \cdot \Delta\hat{h}_F + \dot{M}_A \cdot \Delta\hat{h}_A - \dot{M}_G \cdot \Delta\hat{h}_G\right)$$

$$\boxed{\dot{Y}^{+}_{comb} = \dot{M}_F \cdot \Delta h_i^0 + \dot{M}_{cond} \cdot q^0_{vap}}$$

Total heat energy introduced

Energy balance : $\boxed{\dot{E}^-_e = \dot{Y}^+_{comb} - \dot{Q}^-_G - \dot{Q}^-_{ae} - \dot{Q}^-_{ah} - \dot{Q}^-_{ac}}$

Effectiveness : $\boxed{\varepsilon = \eta_e = \dfrac{\dot{E}^-_e}{\dot{Y}^+_{comb}} = 1 - \dfrac{\dot{Q}^-_G + \dot{Q}^-_{ae} + \dot{Q}^-_{ah} + \dot{Q}^-_{ac}}{\dot{Y}^+_{comb}}}$



$\dot{E}^-_e$

$$\dot{Q}^-_G = \sum\left[\dot{M}_I \Delta h^0_{iI}\right] + \dot{M}_G \hat{h}^0_G$$

$\dot{Q}^-_{ae}$ $\dot{Q}^-_{ac}$ Water, convection

$\dot{Q}^-_{ah}$ $\dot{E}^-_{aux}$ oil, friction, auxiliairies

Energy distribution in Diesel engines

Conventional engines (boundary F)

Energy balance : $$\boxed{\dot{E}_e^- = \dot{Y}_{comb}^+ - \dot{Q}_G^- - \dot{Q}_{ae}^- - \dot{Q}_{ah}^- - \dot{Q}_{ac}^-}$$

Fuel transformation power : $$\dot{Y}_{comb}^+ = \dot{M}_F \Delta h_i^0 + \dot{M}_{cond} \cdot q_{vap}^0$$ HHV

Heat rate losses of the exhaust gases : $$\dot{Q}_G^- = \dot{M}_G \cdot \hat{h}_G + \sum \dot{M}_I \cdot \Delta h_{iI}^0$$

Heat rate losses of cooling water : $$\dot{Q}_{ae}^- = \dot{M}_e \cdot (h_{e2} - h_{e1})$$

Heat rate losses of oil : $$\dot{Q}_{ah}^- = \dot{M}_h \cdot (h_{h2} - h_{h1})$$

Heat rate loss of conduction etc.: $$\dot{Q}_{ac}^-$$

Energy loss of the system

Exergy balance : $$\boxed{\dot{E}_e^- = \underbrace{\dot{E}_{y,comb}^+}_{\dot{M}_F \cdot \Delta k^0} - \dot{L}}$$

$\dot{E}_{y,comb}^+ =$ Fuel exergy transformation

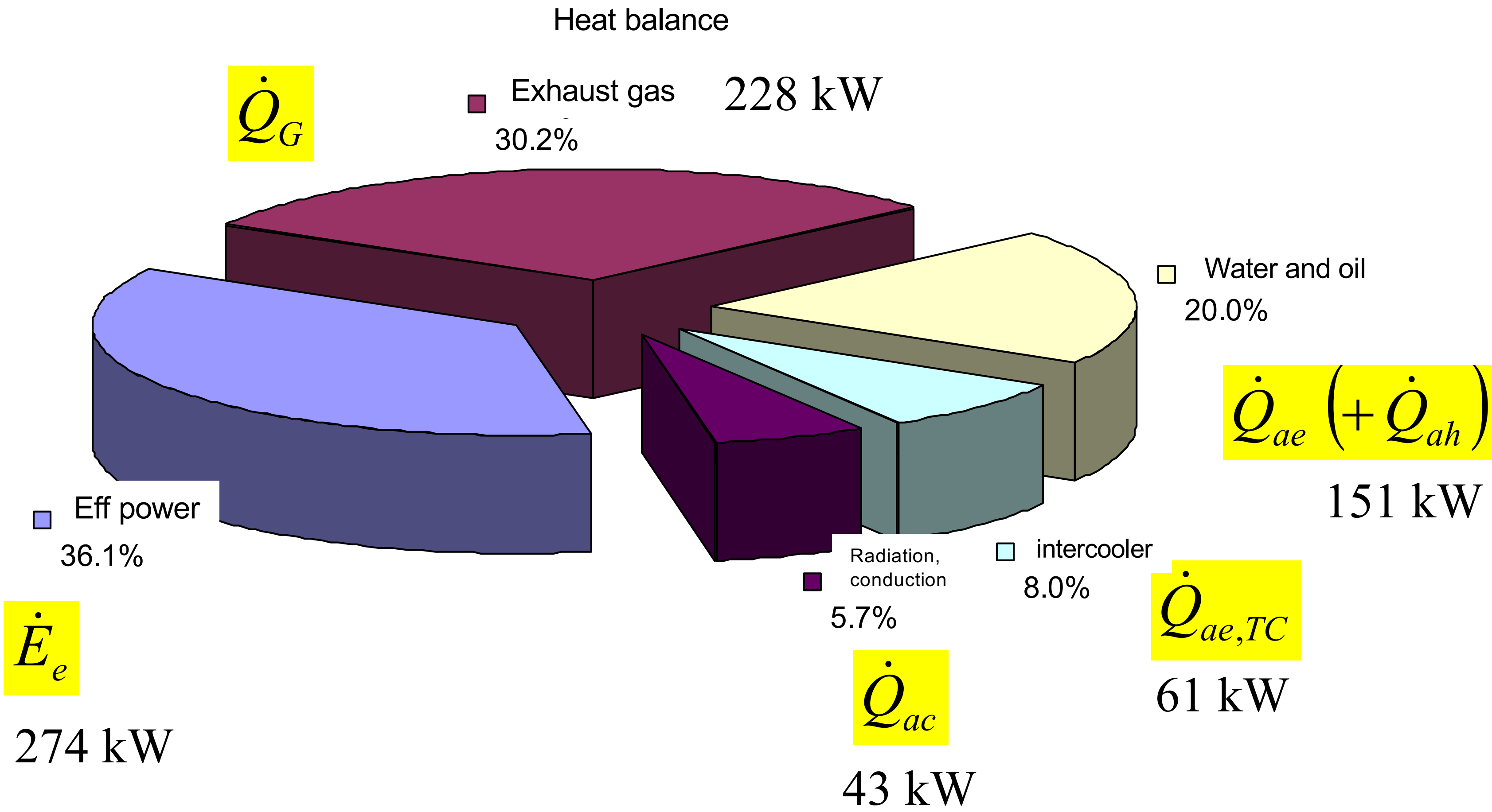
Exergy efficiency : $$\boxed{\eta = \frac{\dot{E}_e^-}{\dot{M}_F \Delta k^0} = 1 - \frac{\dot{L}}{\dot{M}_F \Delta k^0}}$$

$\dot{L} =$ Exergy loss

Energy distribution in Diesel engines

- Conventional engines (boundary F)
  - Energy balance: example

    6 in-line cylinder engine 12 L, 274 *kW* @ 1800 *rpm*



Energy distribution in Diesel engines

■ Cogeneration engines (boundary F_M)

$\dot{Y}^{+}_{comb} = \textit{fuel transformation power}$

$\dot{E}^{-}_{e} = \text{work rate (effective power)}$

$\dot{M}^{-}_{G} \cdot \hat{h}_{G} = \textit{transformation power of the exhaust gas}$

$\dot{Y}^{-}_{ae} = \textit{transformation power of the water network}$

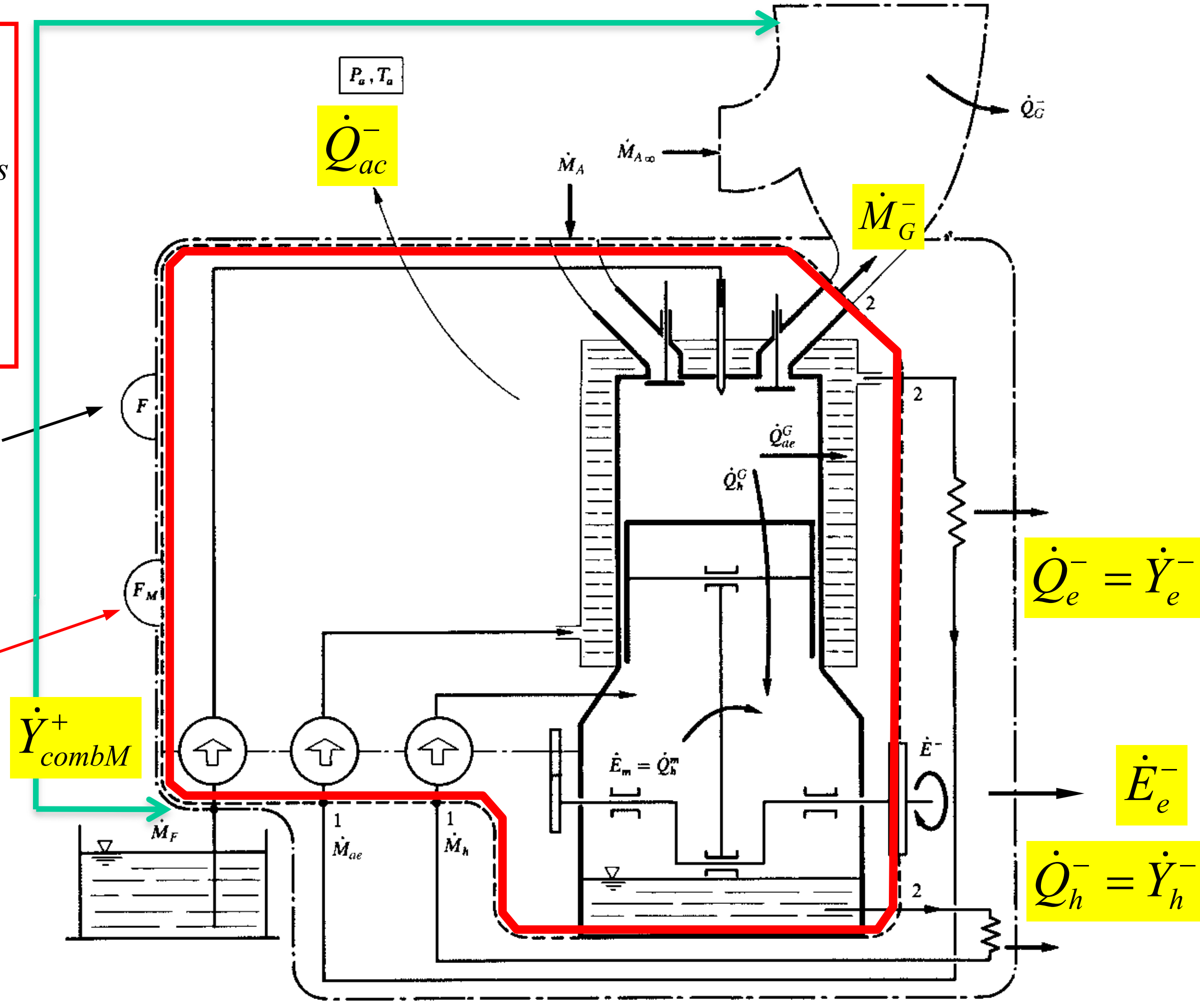
$\dot{Y}^{-}_{ah} = \textit{transformation power of the oil network}$

$\dot{Q}^{-}_{ac} = \textit{heat rate deperdition by conduction}$

F : Boundary of the « engine » system

Scheme of a cogeneration engine

F_M : Boundary of the « cogeneration engine » system



Energy distribution in Diesel engines

- <u>Cogeneration</u> engines (Boundary F_M)

Energy balance :

$$\boxed{\dot{E}_e^- + \dot{Y}_{ae}^- + \dot{Y}_{ah}^- = \dot{Y}_{combM}^+ - \dot{Q}_{ac}^-}$$

Fuel transformation power :

$$\dot{Y}_{combM}^+ = \dot{M}_F \Delta h_i^0 - \dot{M}_I \Delta h_{iI}^0 + \dot{M}_{cond} q_{vap}^0 - \dot{M}_G \hat{h}_G$$

Transformation power of water :

$$\dot{Y}_{ae}^- = \dot{Q}_{ae}^- = \dot{M}_e (h_{e2} - h_{e1})$$

Transformation power of oil :

$$\dot{Y}_{ah}^- = \dot{Q}_{ah}^- = \dot{M}_h (h_{h2} - h_{h1})$$

Potential of energy services provided by the system

Engine effectiveness :

$$\boxed{\varepsilon_M = \frac{\dot{E}_e^- + \dot{M}_G \hat{h}_G + \dot{Y}_{ae}^- + \dot{Y}_{ah}^-}{\dot{M}_F \Delta h_i^0 - \left(\dot{M}_I \Delta h_{iI}^0\right) + \dot{M}_{cond} q_{vap}^0}}$$

Exergy balance :

$$\boxed{\dot{E}_e^- + \dot{E}_{yae}^- + \dot{E}_{yah}^- = \dot{E}_{y,combM}^+ - \dot{L}_M}$$

Exergy efficiency :

$$\boxed{\eta_M = \frac{\dot{E}_e^- + \dot{M}_G \cdot \hat{k}_G + E_{yae}^- + E_{yah}^-}{\dot{M}_F \Delta k^0 - \dot{M}_I \Delta k_I^0} = 1 - \frac{\dot{L}}{E_{y,combM}^+}}$$