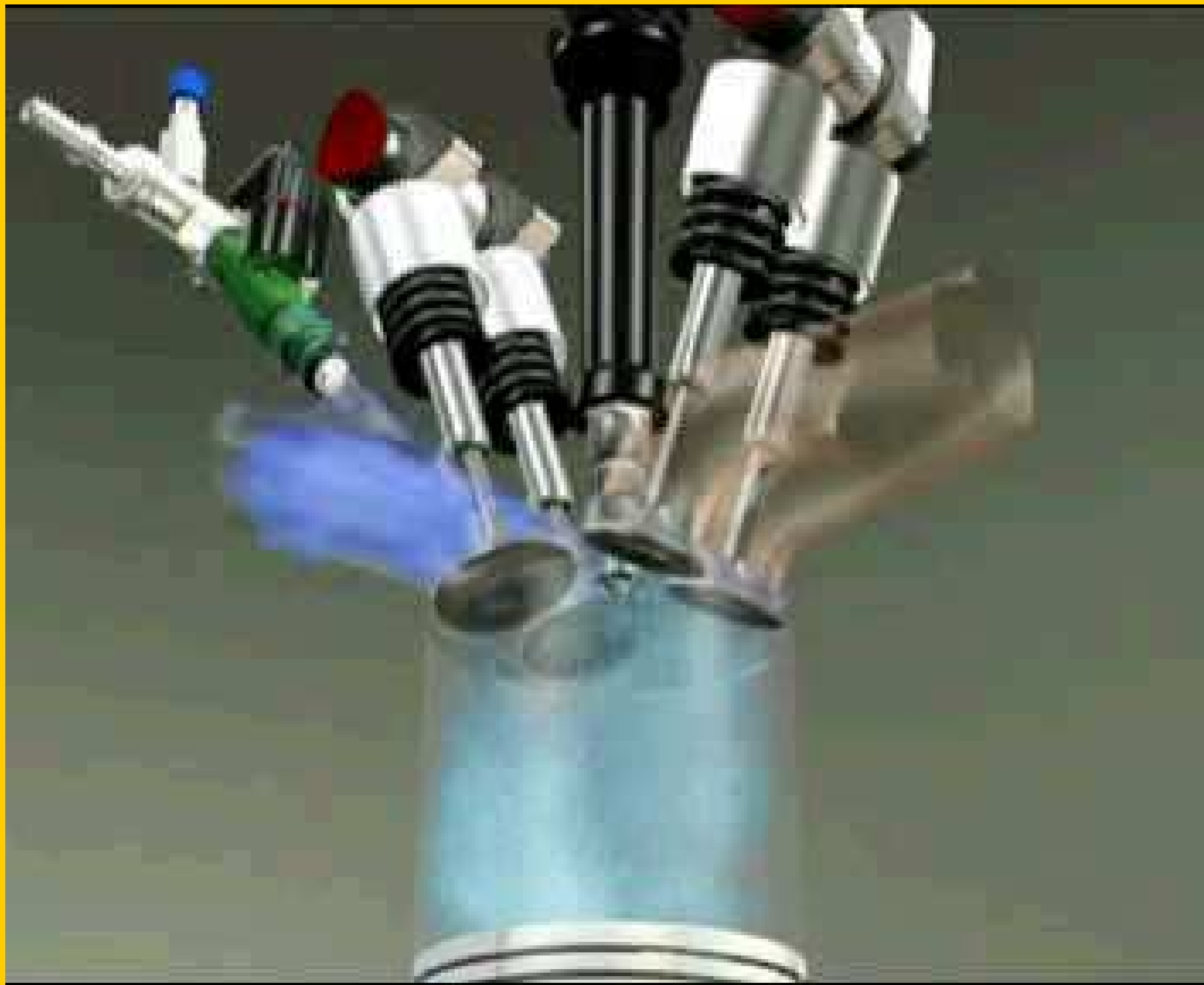
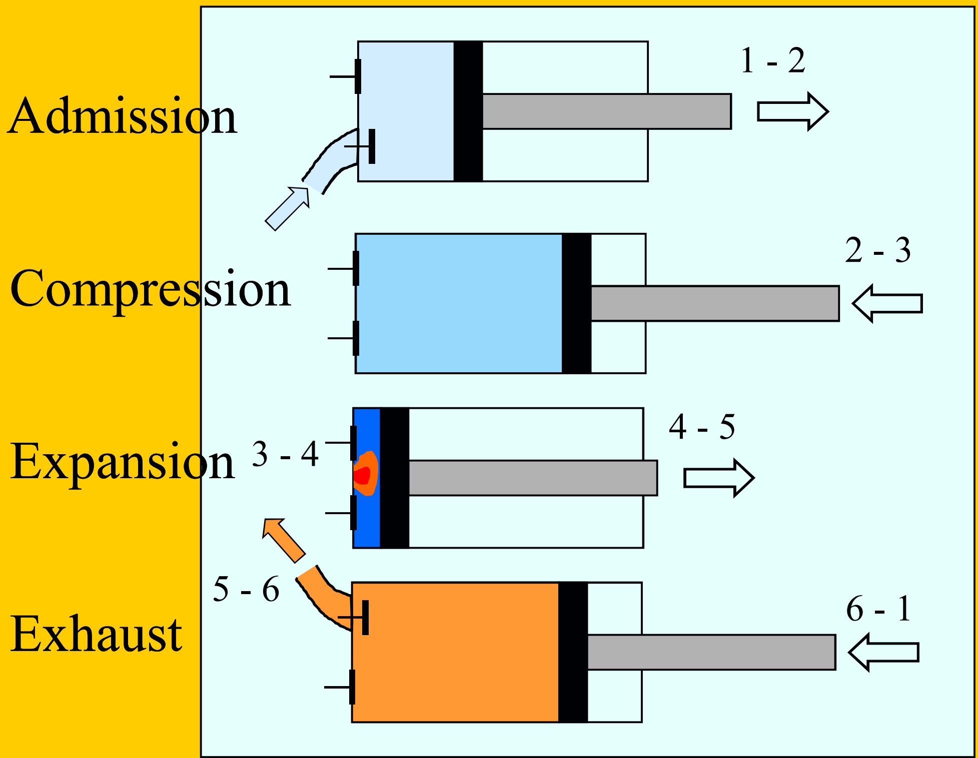
Internal Combustion Engines

By Dr. Akos Bereczky, BME

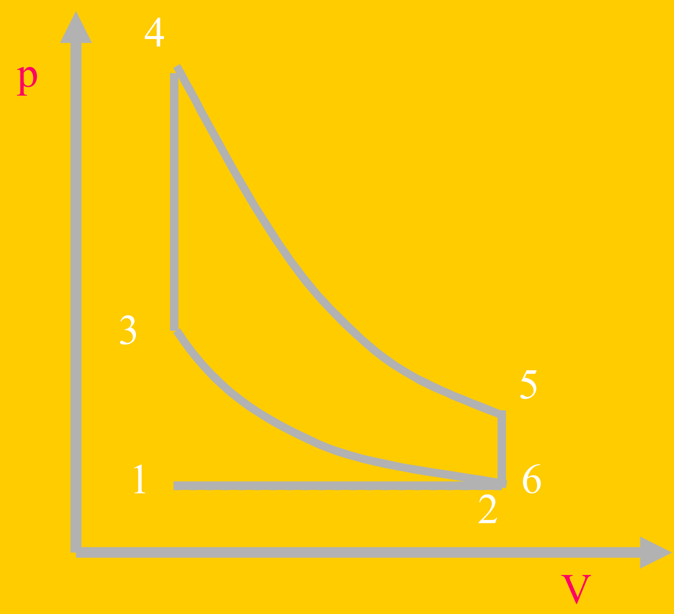
MPI (Ford)



OTTO CYCLE

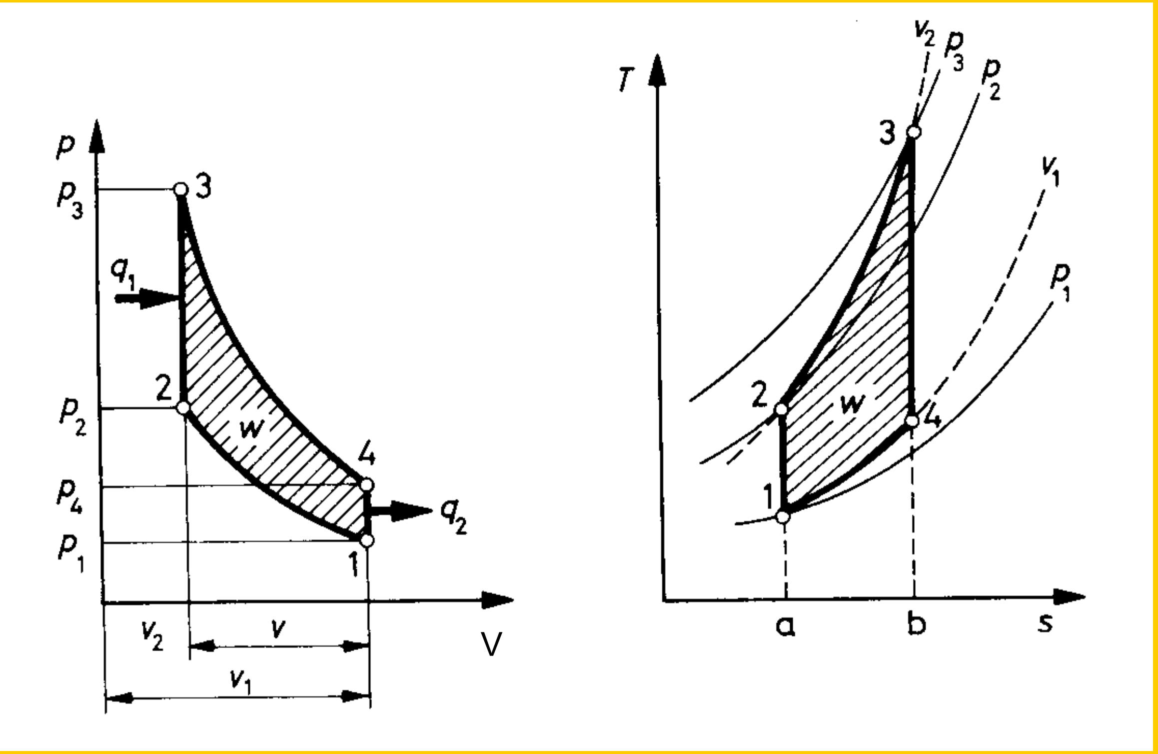


Admission
1 - 2
Compression
2 - 3
Expansion
3 - 4
4 - 5
5 - 6
Exhaust
6 - 1



p
4
3
5
1
6
2
V

The ideal air standard Otto cycle:



$$\varepsilon = \frac{V_1}{V_2}$$

$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_v(T_3 - T_2) - c_v(T_4 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1}{\varepsilon_4^{\kappa - 1}}$$

COMPRESSION RATIO VALUES

$$\left(\frac{V_{cil.}+V_{c.c.}}{V_{c.c.}}\right)$$

1 – **atmospheric engine**

3 – **Otto engine**

4 – **side valves**

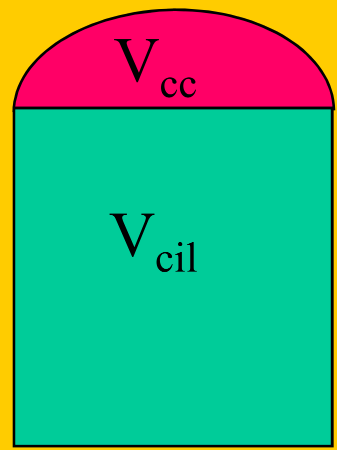
6 – **Ricardo (turbulance) head**

7 – **head valves**

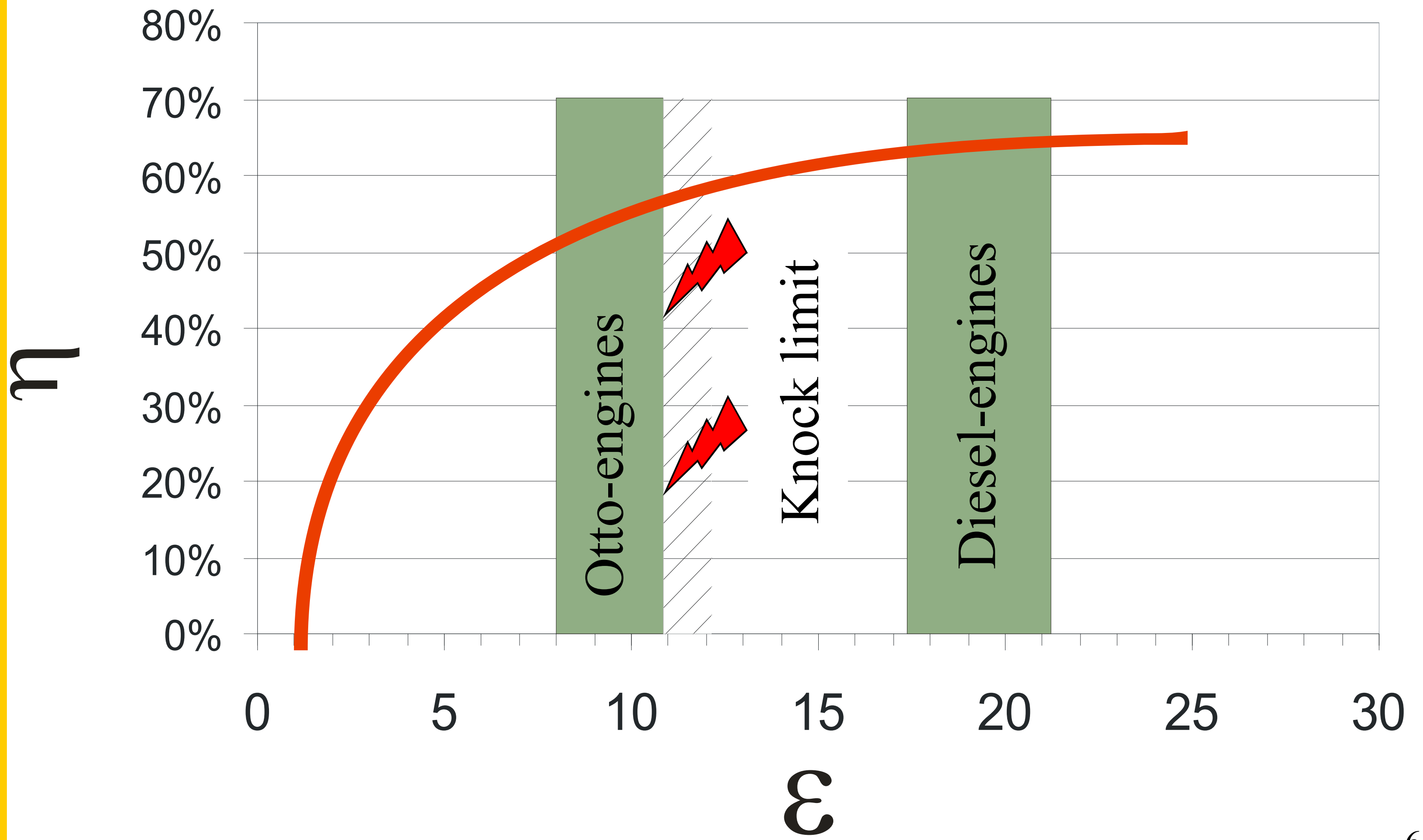
9 – **leaded petrol (5 star)**

10 – **electronic injection MPI**

11 – **detonation control**

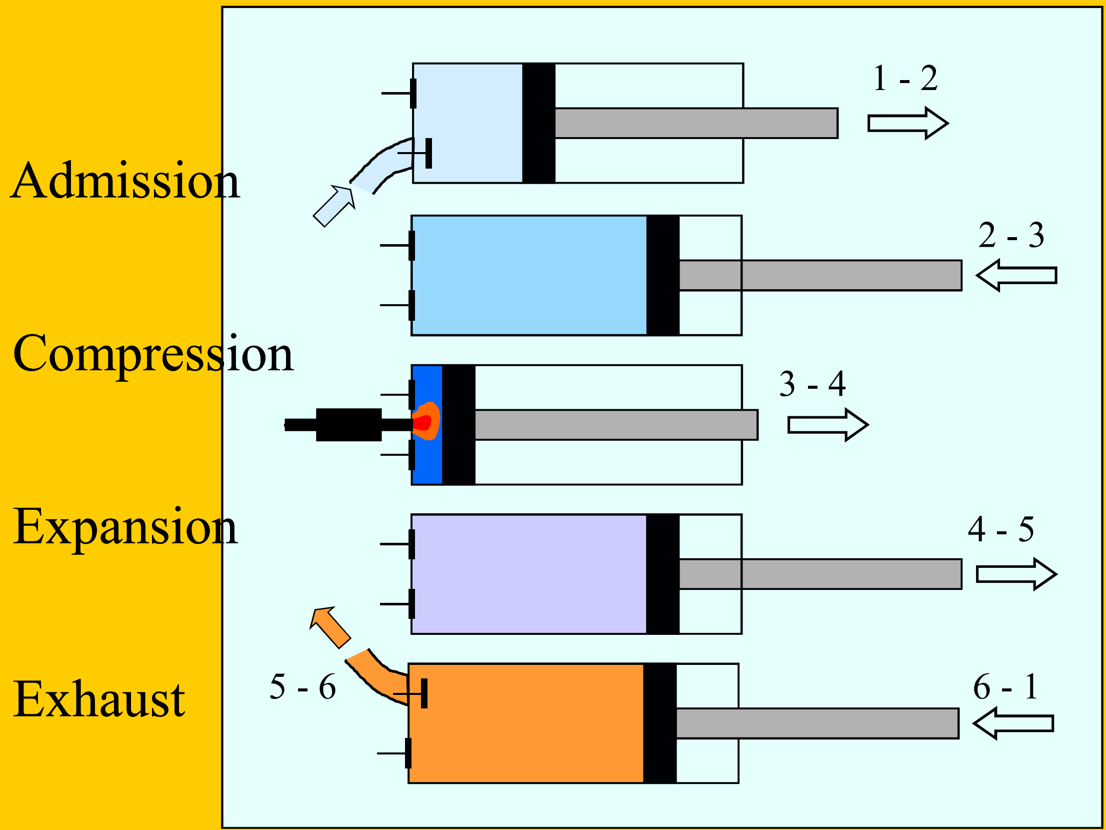


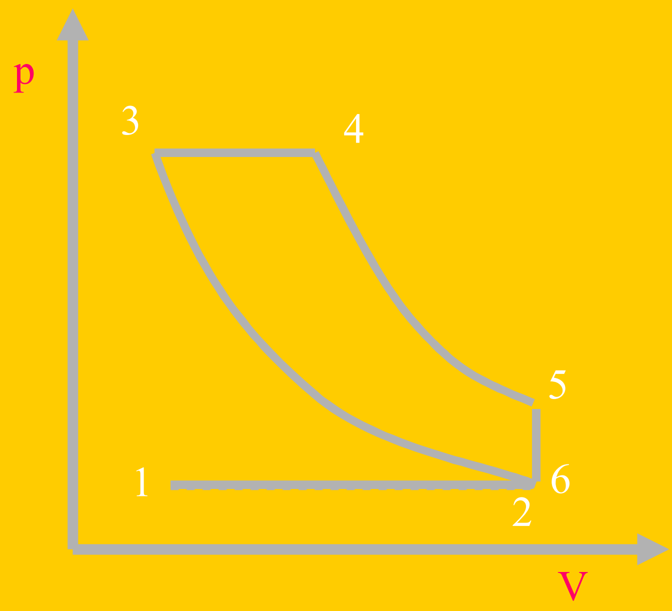
Efficiency in the function of the compression ratio



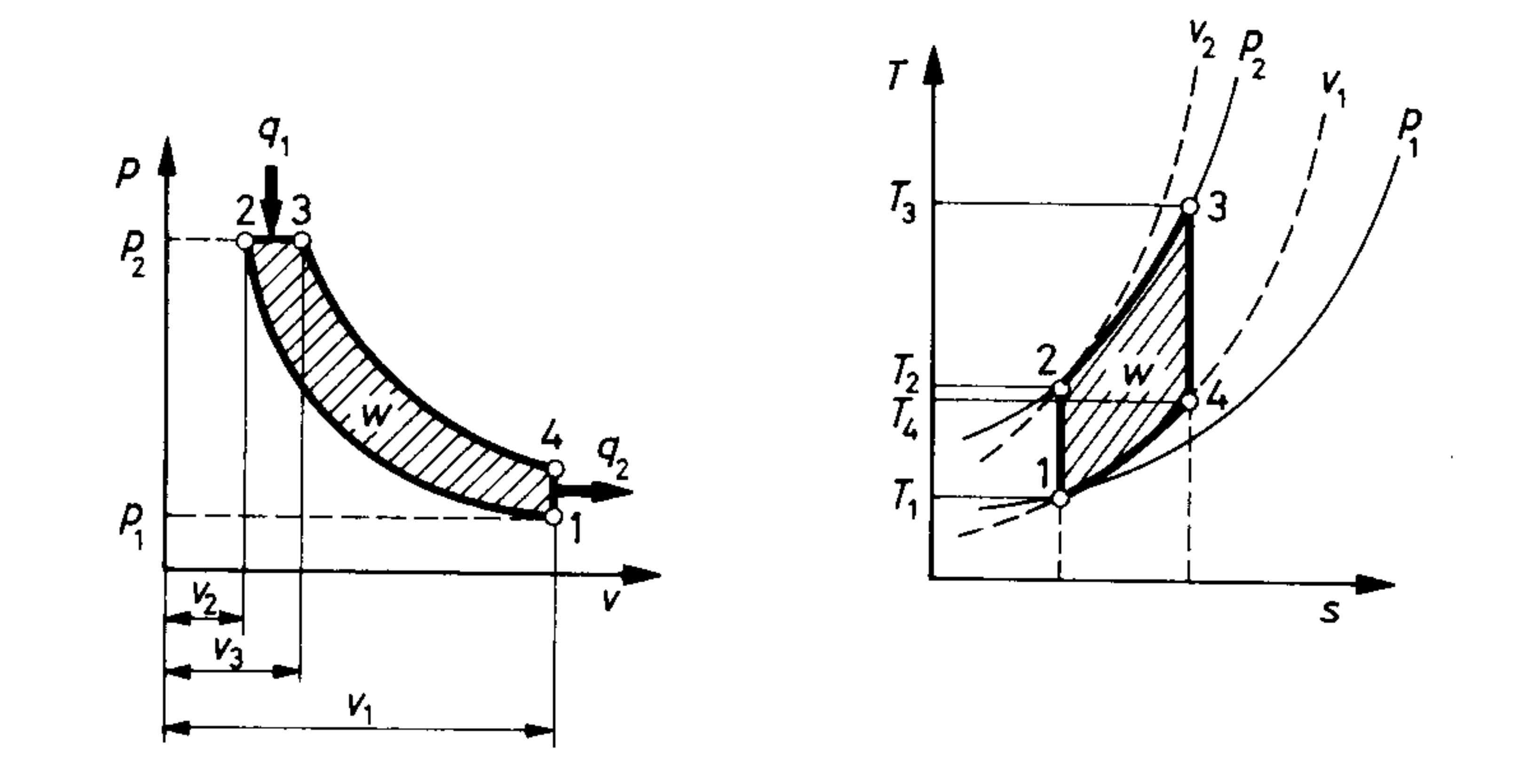
Direct Injection Combustion

DIESEL CYCLE



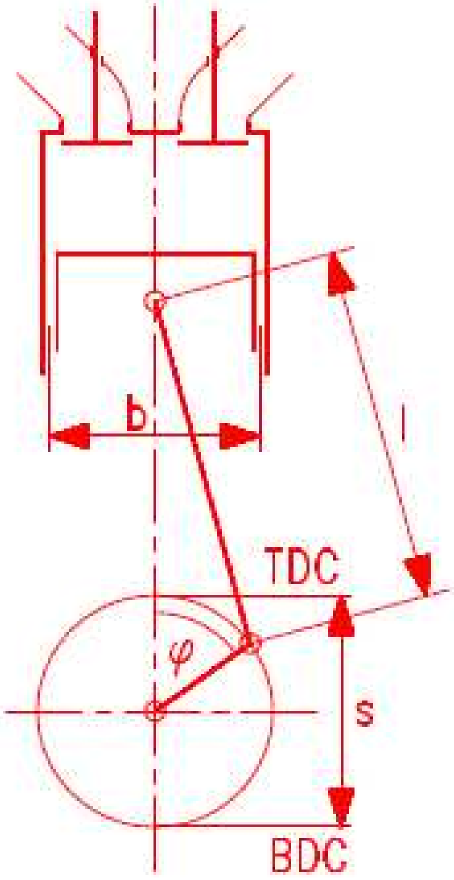


Diesel cycle



$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} - 1}{\kappa \cdot (\rho - 1)} \qquad \rho = \frac{\boldsymbol{v}_3}{\boldsymbol{v}_2}$$

SUMMARY OF DEFINITIONS AND RELATED EQUATIONS



b: bore [mm] (1 in = 25.4 cm)

l: connecting rod length [mm]

s: stroke [mm]

φ: crank angle from the TDC

Displacement volume (one)

$$V_d = \frac{b^2 \cdot \Pi}{4} \cdot s \quad [\mathrm{cm}^3]$$

Clearance volume $\quad V_c \quad [\mathrm{cm}^3]$

Cylinder volume $\quad V = V_d + V_c$

Compression ratio

$$\varepsilon = \frac{\text{max. volume}}{\text{min. volume}} = \frac{V}{V_c} = \frac{V_d + V_c}{V_c} \quad [-]$$

Friction power :

The difference between the indicated and the brake power i.e. the power required to overcome the frictional resistance of the engine parts.

$$P_f = P_i - P_b \qquad [W]$$

Mechanical efficiency :

$$\eta_M = \frac{P_b}{P_i} = \frac{bmep}{imep} \qquad [\text{ - }]$$

Volumetric efficiency :

$$\eta_V = \frac{\dot{V}}{\dot{V}_s} = \frac{\dot{m}_a + \dot{B}}{V_s \cdot \rho_i \cdot n \cdot i} \qquad [\text{-}]$$

ρ_i : fuel-air mixture density in the intake manifold

Indicated efficiency :

$$\eta_i = \frac{P_i}{\dot{B} \cdot H_i} \qquad [\text{ - }]$$

H_i : available energy content of fuel [kJ/kg]

$\dot{B}$: mass flow rate of fuel [kg/s]

Brake thermal efficiency :

$$\eta_{eff} = \frac{P_b}{\dot{B} \cdot H_i} \qquad [\text{ - }]$$

Criteria of performance:

Torque :

The torque measured by dynamometers. Obtained by reading off a net load (F [N]) at known radius (k [m]) from the axis of rotation.

$$M = F \cdot k \qquad \text{[Nm]}$$

Indicated power :

The rate of work done by the gas on the piston evaluated from the indicator diagram obtained from the engine. P_i [W]

Brake power :

The power delivered by the engine. $P_b = 2 \cdot \pi \cdot n \cdot M$ [W]

Indicated mean effective pressure (imep) :

It is defined as

$$p_i = \frac{P_i}{V_s \cdot n \cdot i} \qquad \text{[bar]}$$

n : engine revolution [rev/s]

i : 1 if two stroke engine

2 if four stroke engine

Break mean effective pressure (bmep) :

It is defined as

$$p_e = \frac{P_b}{V_s \cdot n \cdot i} \qquad \text{[bar]}$$

Delivery ratio :

$$\lambda = \frac{m_a}{V_s \cdot \rho_\infty \cdot n \cdot i} = \frac{p_\infty + \Delta p_i}{p_\infty} \cdot \frac{T_\infty}{T_\infty + \Delta T_i} \quad [\,-\,]$$

$p_\infty, \rho_\infty, T_\infty$: ambient density, pressure and temperature

$\Delta p_i, \Delta T_i$: pressure and temperature change through intake

Excess air factor :

$$\lambda_m = \frac{\dot{m}_a}{B \cdot \mu} \quad [\,-\,]$$

$\dot{m}_a$: mass flow rate of air [kg/s]

μ : stoichiometric air-fuel ratio

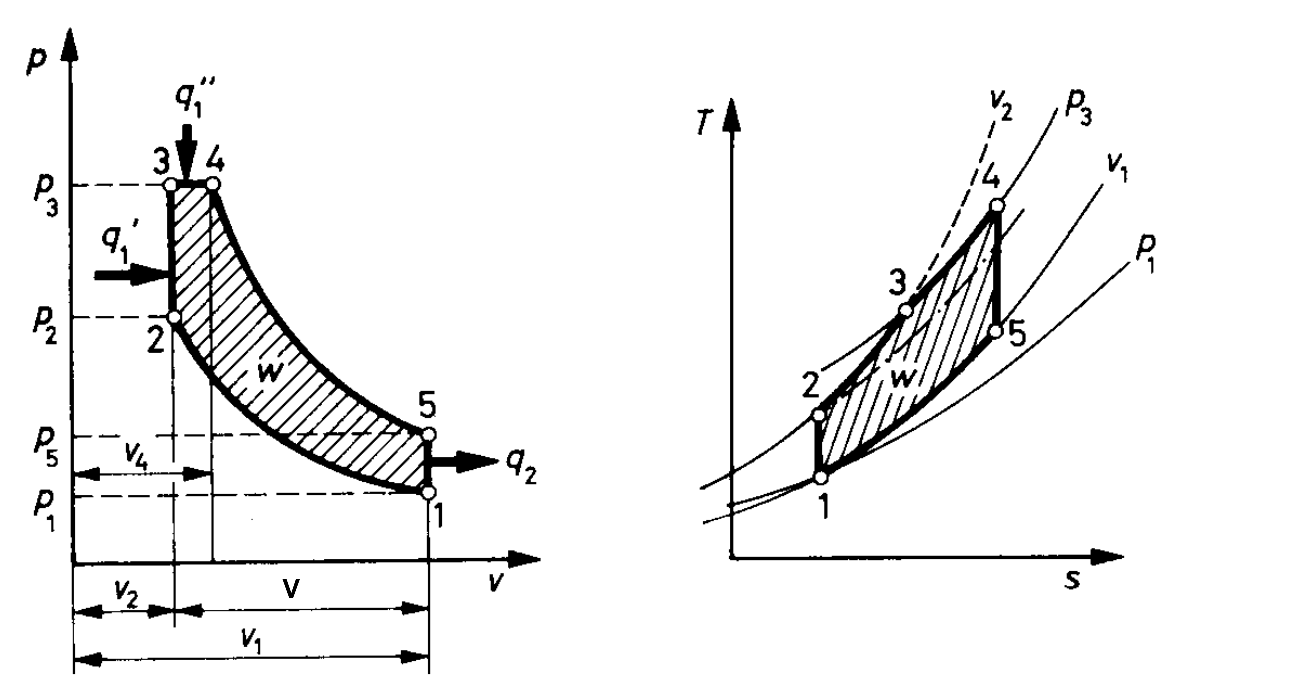
Brake specific fuel consumption (bsfc):

$$b_e = \frac{B}{P_b} = \frac{1}{H_i \cdot \eta_{eff}} \quad [\text{g/kWh}]$$

Mean piston speed :

$$\overline{u}_p = 2 \cdot s \cdot n \quad [\text{m/s}]$$

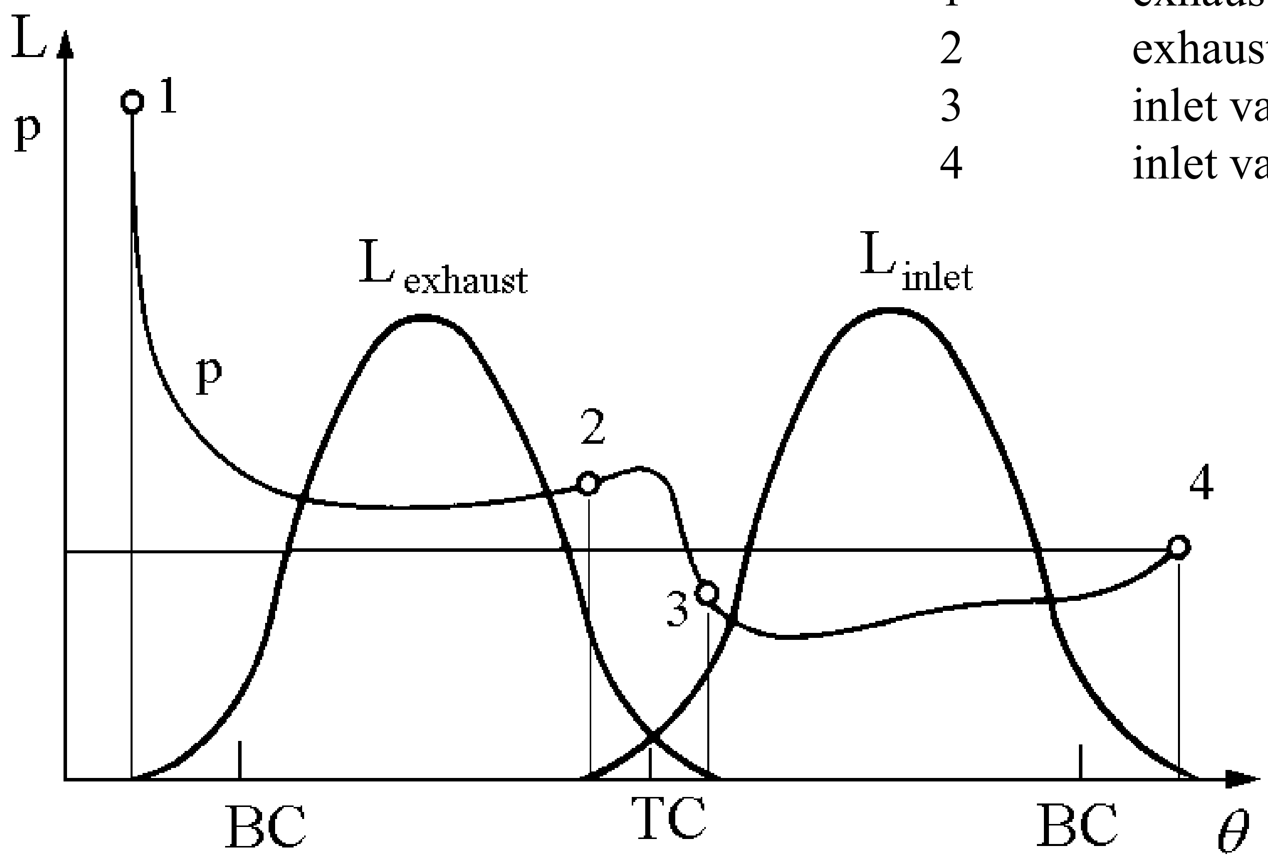
The Dual-combustion cycle



$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) + c_v(T_4 - T_3) - c_v(T_5 - T_1)}{c_p(T_3 - T_2) + c_v(T_4 - T_3)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} \cdot \lambda - 1}{(\lambda - 1) + \kappa \cdot \lambda \cdot (\rho - 1)}$$

$$\rho = \frac{v_4}{v_3} \qquad \lambda = \frac{p_3}{p_2}$$

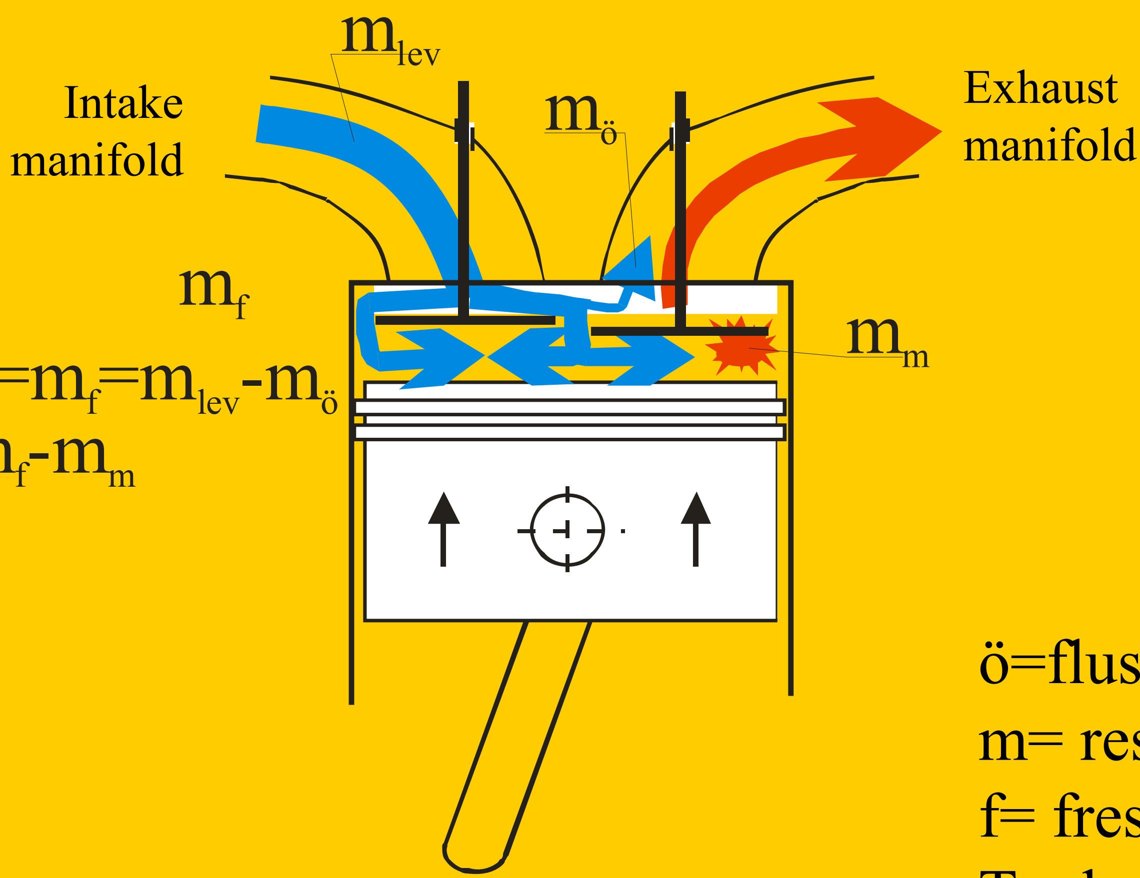
- **LOSSES IN INTERNAL COMBUSTION ENGINES**:
  - **Intake and exhaust losses (Fresh mixture (air) loss, rest gases, valve loss, ...)**
  - Heat transfer (non isentropic) compression
  - Incomplete combustion
  - Limited combustion speed (+ Heat loss of Combustion)
  - Gas losses (Blow-by)
  - Friction is not internal loss



L	valve lift
BC	bottom dead center
TC	top dead center
1	exhaust valve opens
2	exhaust valve closes
3	inlet valve opens
4	inlet valve closes

2-3: overlap period

aerodynamic losses during intake



$m_{valós}=m_f=m_{lev}-m_ö$

$m_t=m_f-m_m$

ö=flushing air

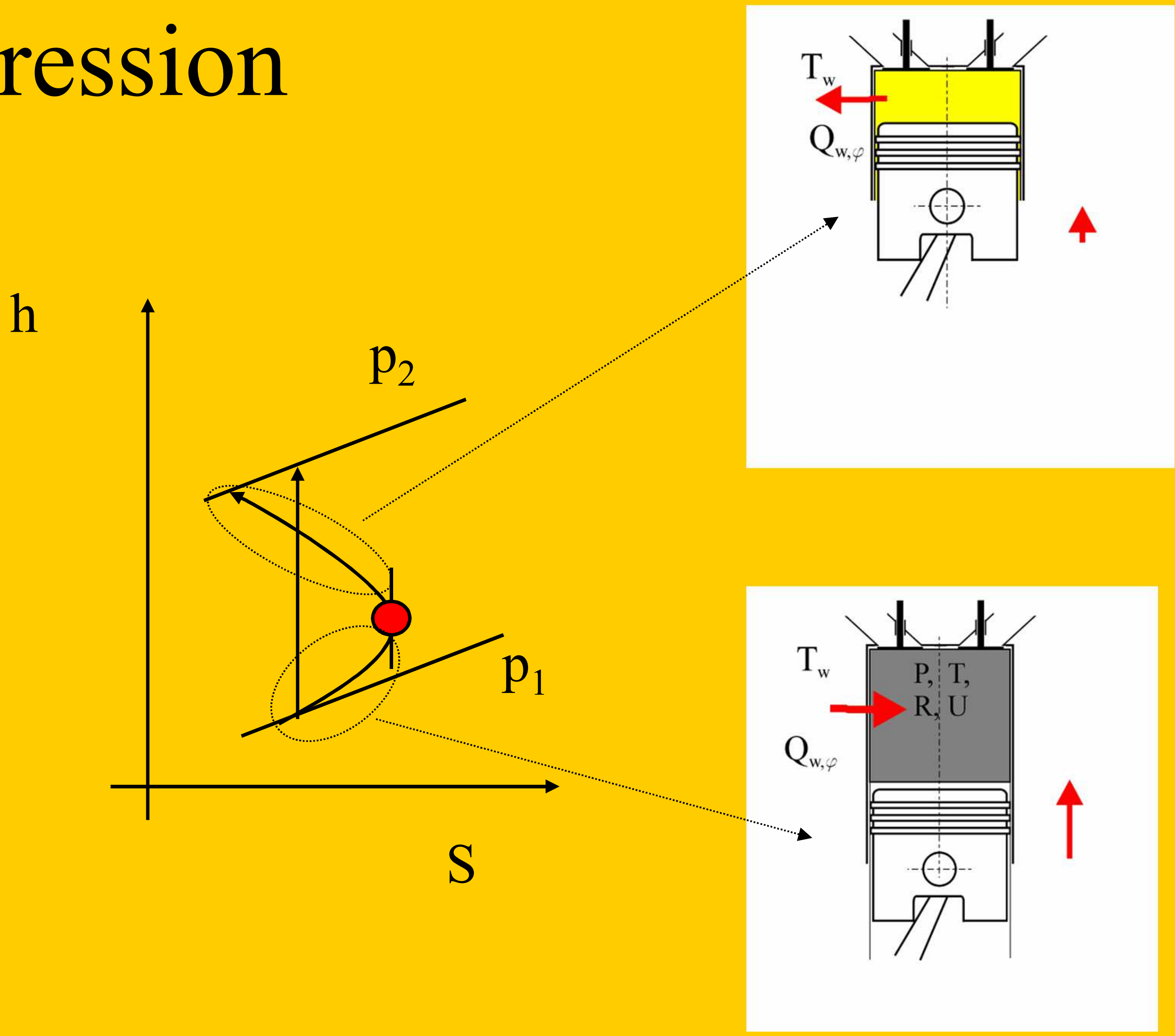
m= rest gases

f= fresh air

T=charge

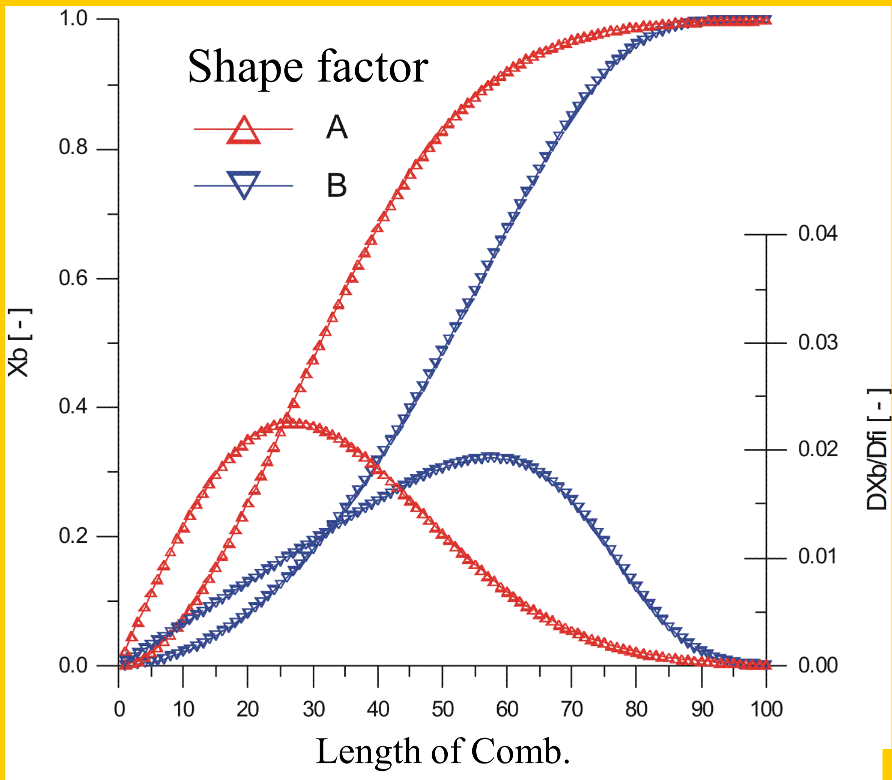
- **LOSSES IN INTERNAL COMBUSTION ENGINES**:
  - Intake and exhaust losses (Fresh mixture (air) loss, Rest gases, valve loss, ...)
  - **Heat transfer (non isentropic) compression**
  - Incomplete combustion
  - Limited combustion speed (+ Heat loss of Combustion)
  - Gas losses (Blow-by)
  - Friction

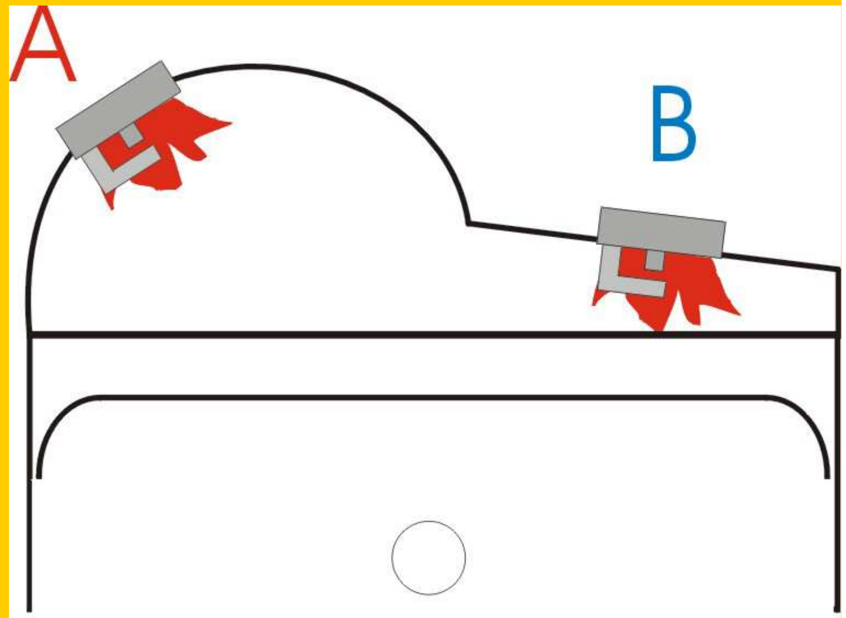
Compression



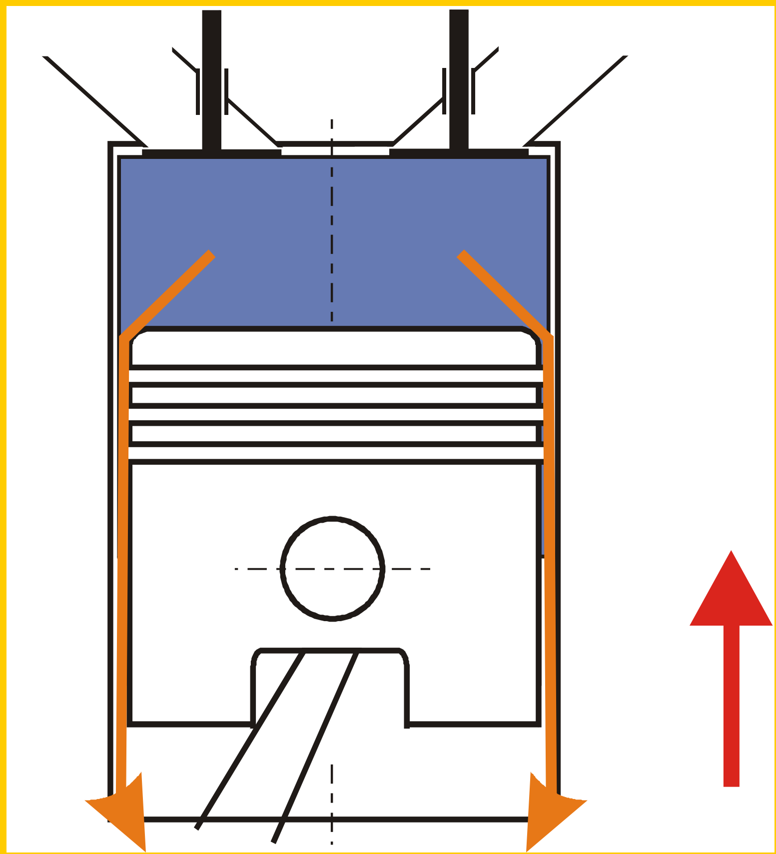
- **LOSSES IN INTERNAL COMBUSTION ENGINES**:
  - Intake and exhaust losses (Fresh mixture (air) loss, Rest gases, valve loss, ...)
  - Heat transfer (non isentropic) compression
  - **Incomplete combustion**
  - **Limited combustion speed (+ Heat loss of Combustion)**
  - Gas losses (Blow-by)
  - Friction

Influence of Geometry on the Combustion process





Gas losses (Blow-by)



- **LOSSES IN INTERNAL COMBUSTION ENGINES**:
  - Intake and exhaust losses (Fresh mixture (air) loss, Rest gases, valve loss, ...)
  - Heat transfer (non isentropic) compression
  - Incomplete combustion
  - Limited combustion speed (+ Heat loss of Combustion)
  - Gas losses (Blow-by)
  - **Friction is not internal loss**

LOSSES IN INTERNAL COMBUSTION ENGINES I

During the operation of the internal combustion engines only a fraction of the chemical energy is converted into mechanical work. The "lost work" can mainly be attributed to the following:

- **Heat transfer**

Heat transfer occurs between the cylinder wall and working fluid. The most significant phenomenon is the heat loss of the hot burned gases, which occurs during combustion and expansion.

- **Mass loss**

A fraction of the high pressure unburned gases flows from the combustion chamber into the crankcase (blowby) thus the cylinder pressure drops and the output work decreases. This mass loss is about one percent of the charge.

- **Incomplete combustion**

The exhaust gases usually contain unburned particles (H_2, CO, CH) carrying a fraction of the fuel's chemical energy (SI engine : 5%, CI engine : 1-2%).

LOSSES IN INTERNAL COMBUSTION ENGINES II

- **Limited combustion speed**

In an ideal SI engine the combustion time is zero i.e.: the combustion speed is infinitive. In a real case the combustion process requires certain time (order of milliseconds in passenger cars) therefore the ignition starts before the TC and complete after the TC. Thus the peak pressure will be less than the one of the perfect cycle and the extracted work will be less, too.

- **Exhaust blow down loss**

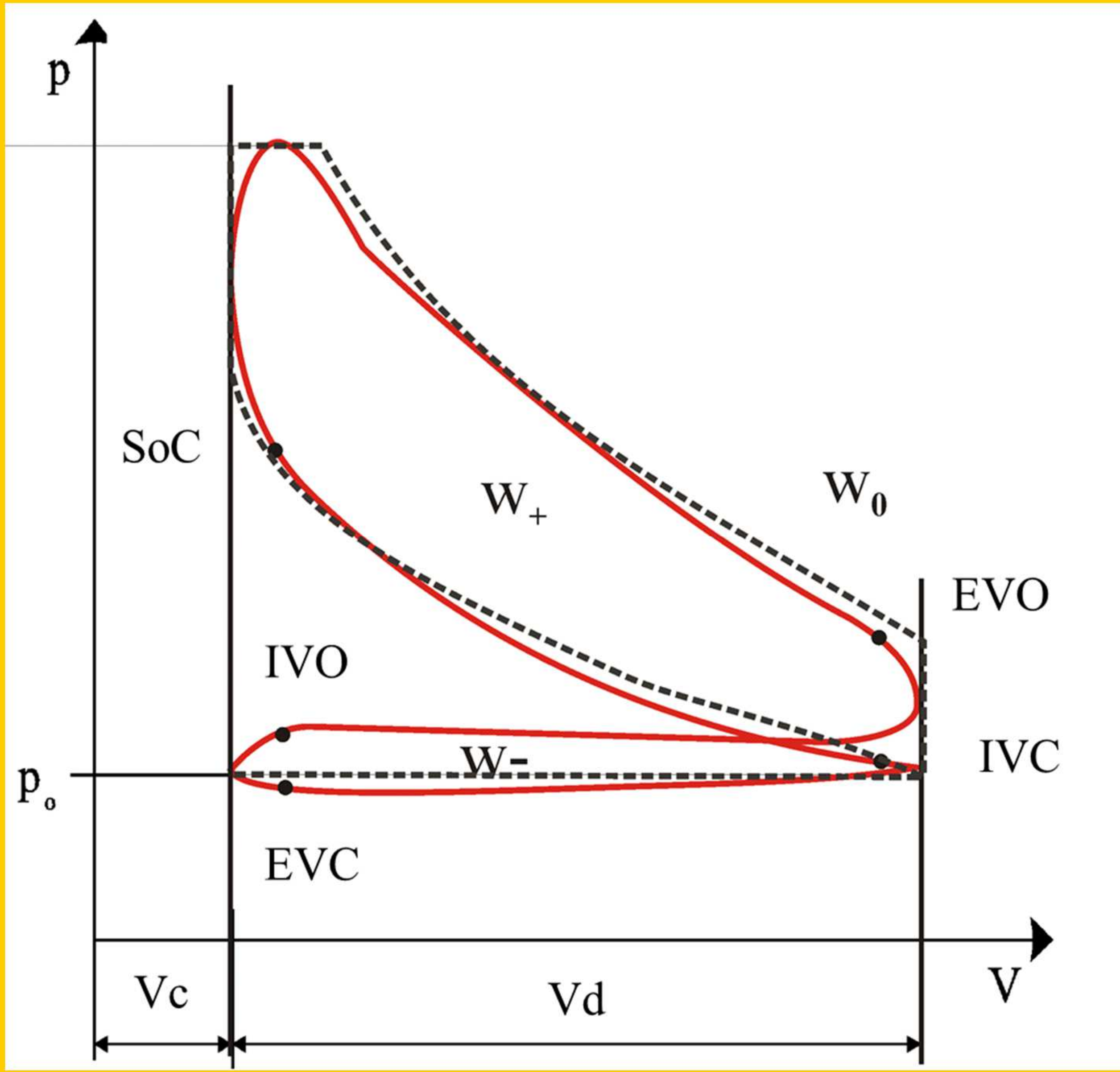
Considering that the blow down process takes time the exhaust valve must be opened before the BC thus the expansion stroke will be uncompleted and work will be lost.

- **Pumping work**

The friction of the streaming gases and the aerodynamic losses during intake cause pressure drop in the cylinder before compression and sequentially lower peak pressure and less output work. The blowdown process of the exhaust gases requires work, too. The pumping loss is most superior in quantity governed (SI) engines at part load.

- - Friction

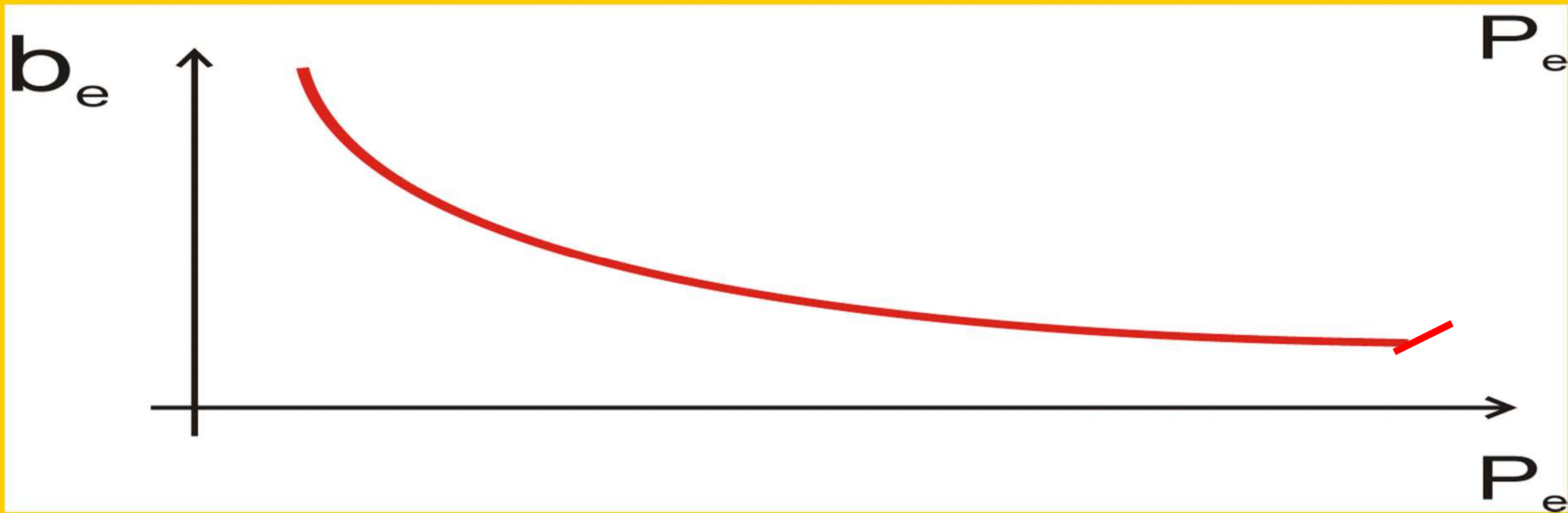
The most significant source of this loss is the friction between the piston skirt, rings and the cylinder (about 60-80% of the total frictional work). Usually it is higher in diesel engines, because of the stronger piston rings. The other sources of frictional losses are the crankshaft, camshaft, valve mechanism, gears, etc..



Theoretical (dot line) and real indicator diagramm (cont. line)

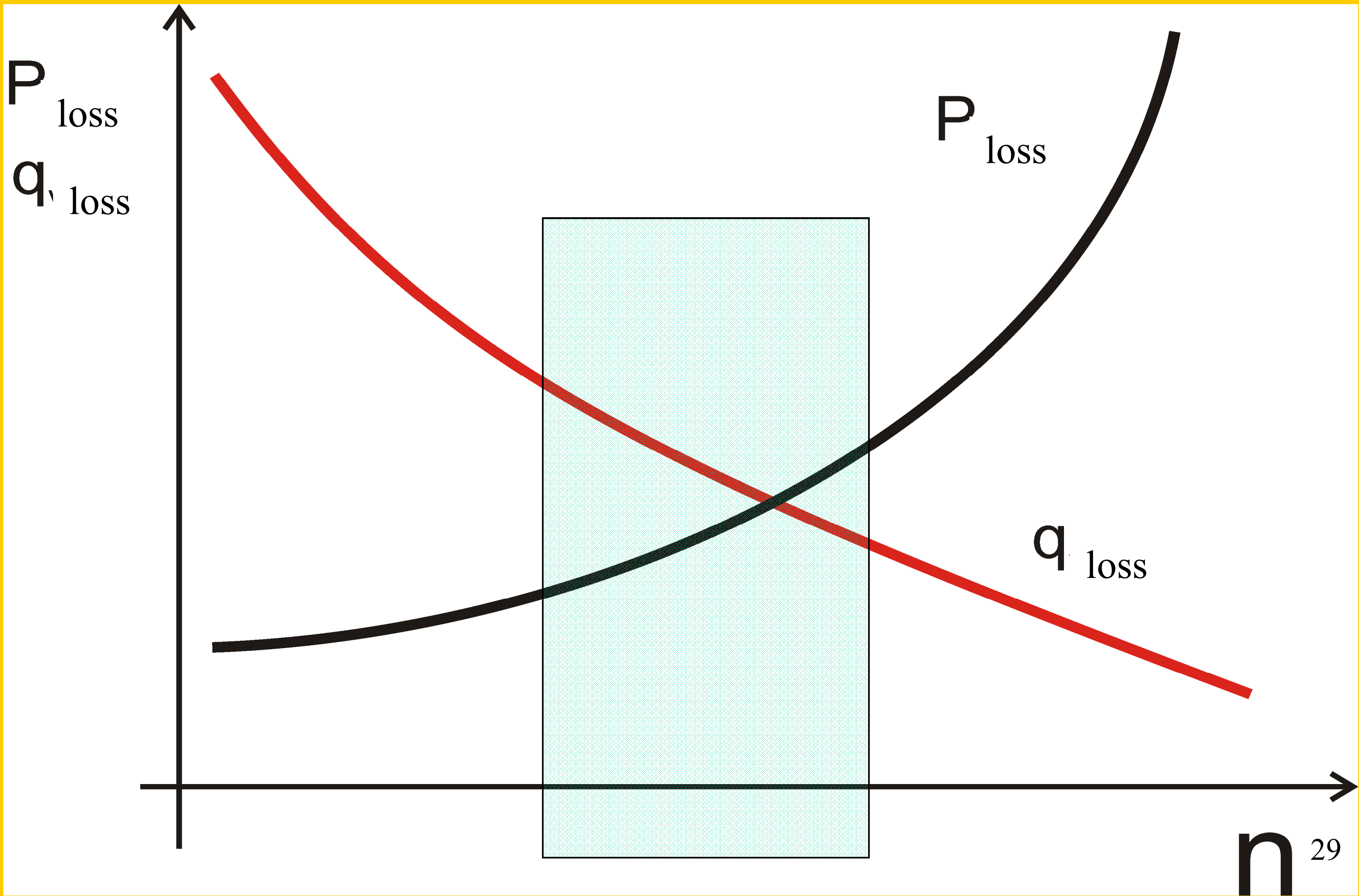
Characteristic of ICE Engines

Brake Specific fuel consumption at different loads, constant speed

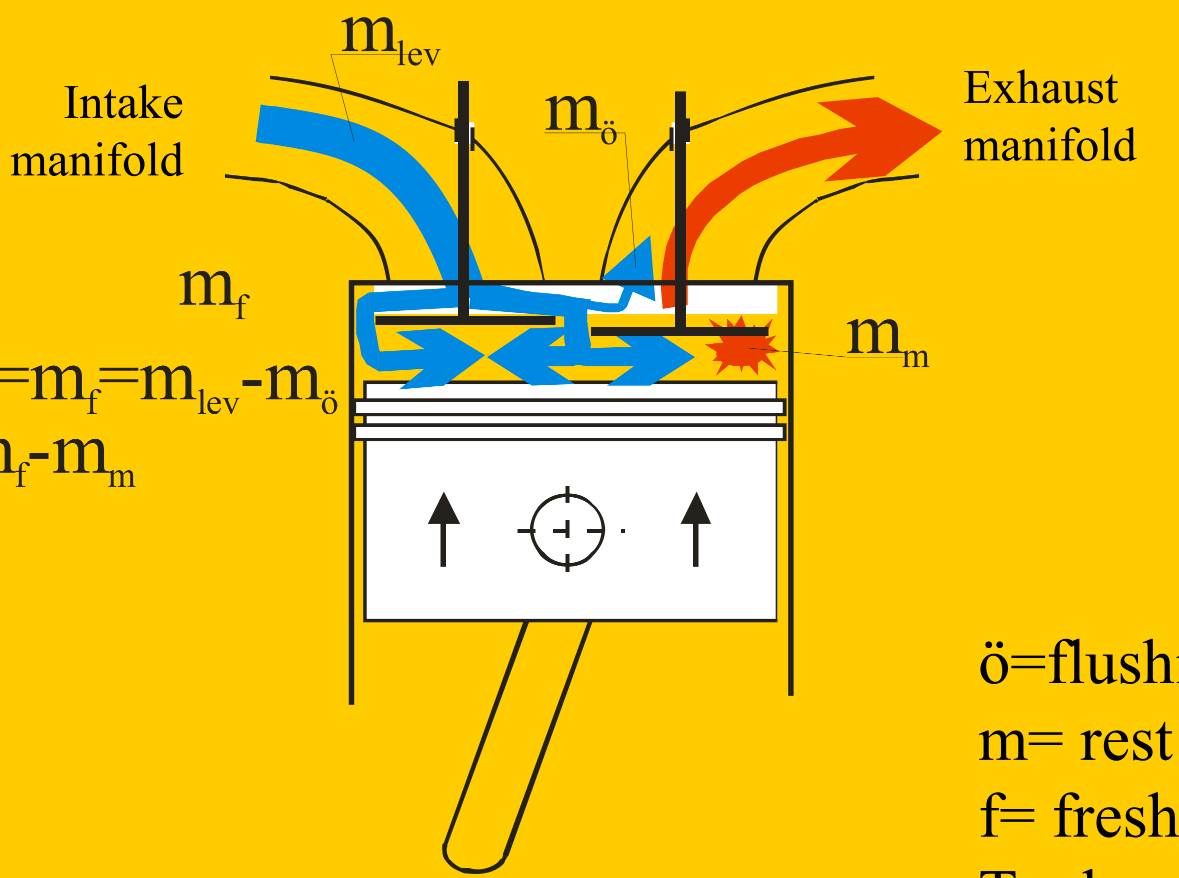


$$b_e = \frac{\dot{B}}{P_b} = \frac{1}{H_i \cdot \eta_{eff}}$$

Losses in the Function of the Speed



aerodynamic losses during intake



$m_{valós}=m_f=m_{lev}-m_ö$

$m_t=m_f-m_m$

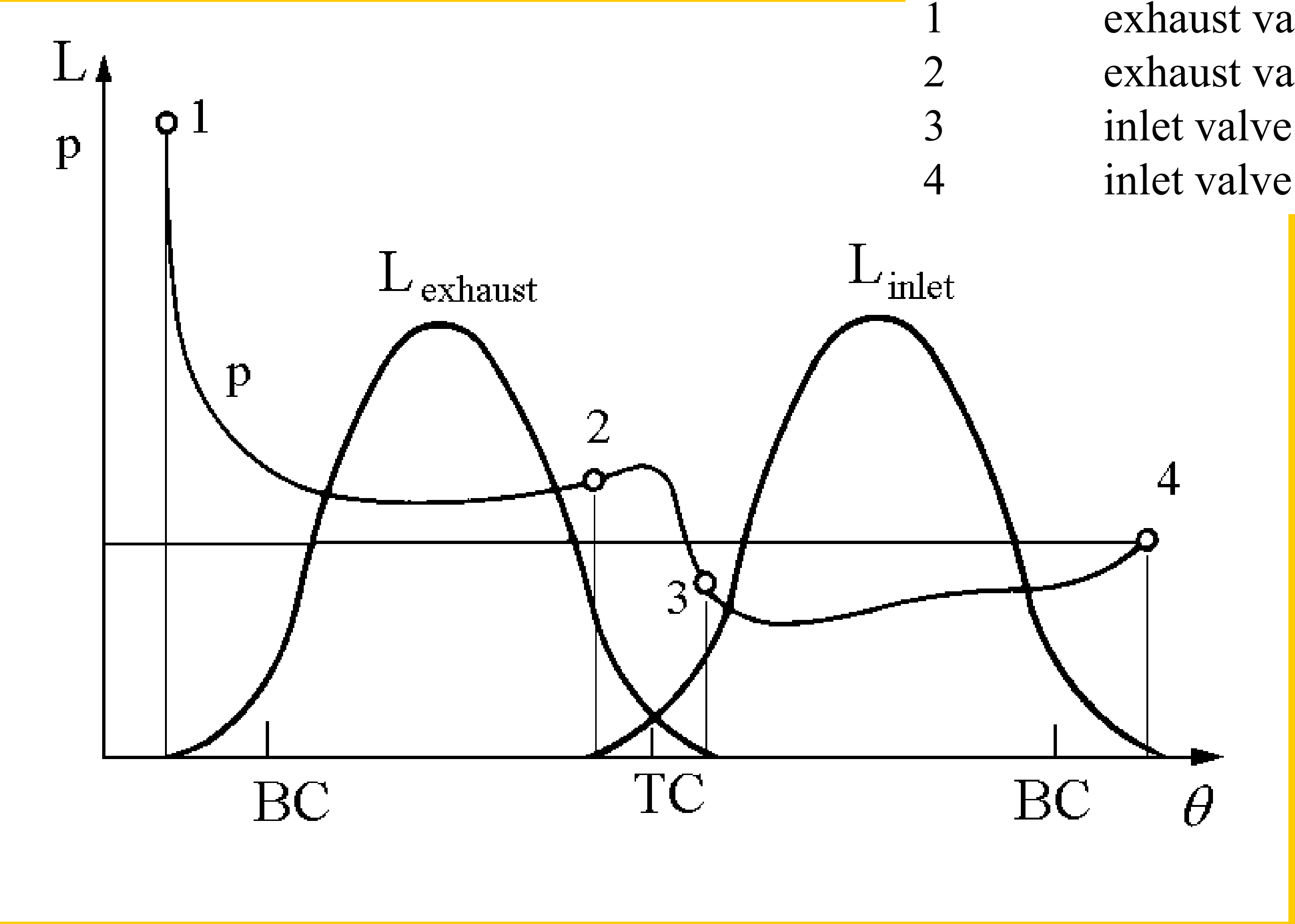
ö=flushing air

m= rest gases

f= fresh air

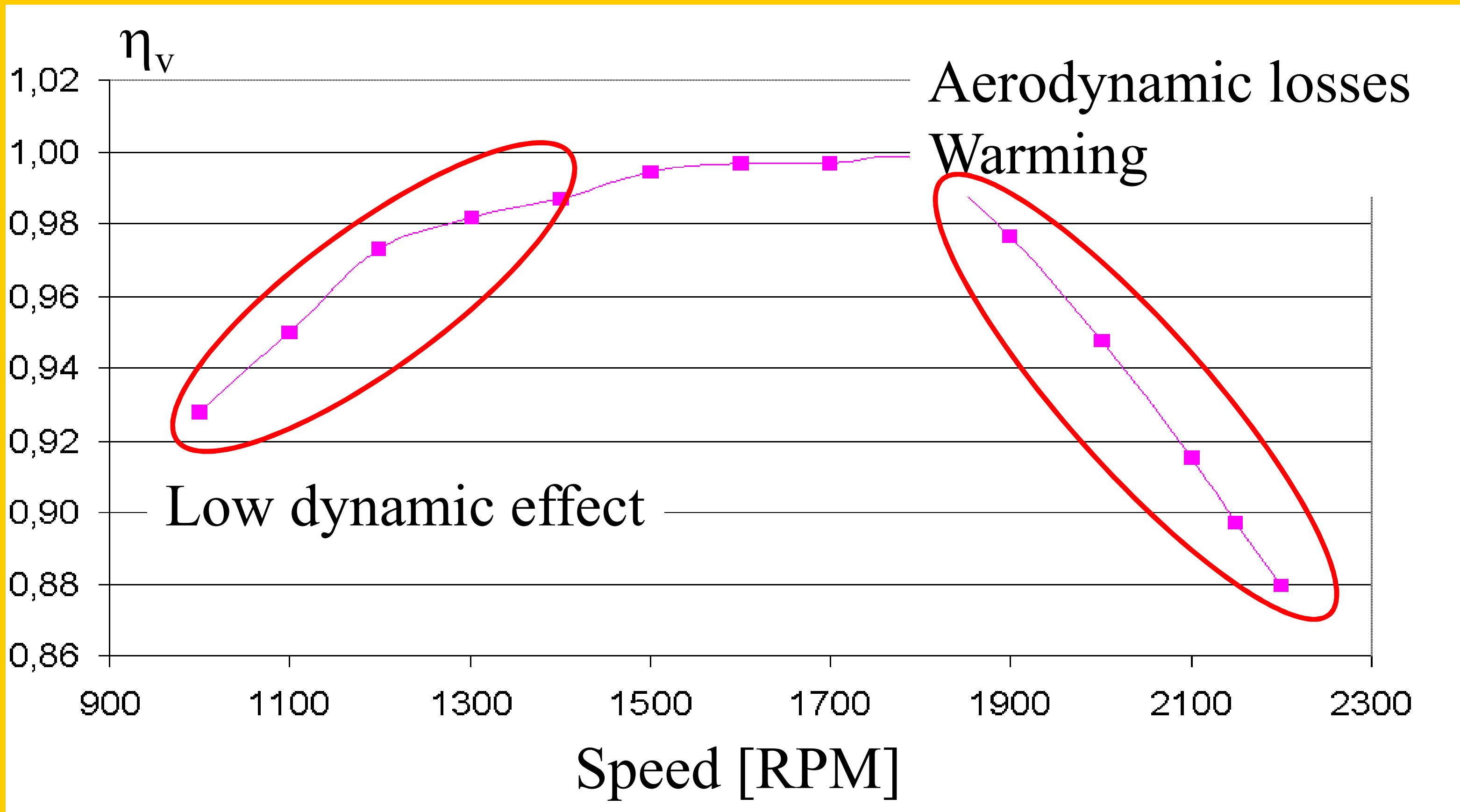
T=charge

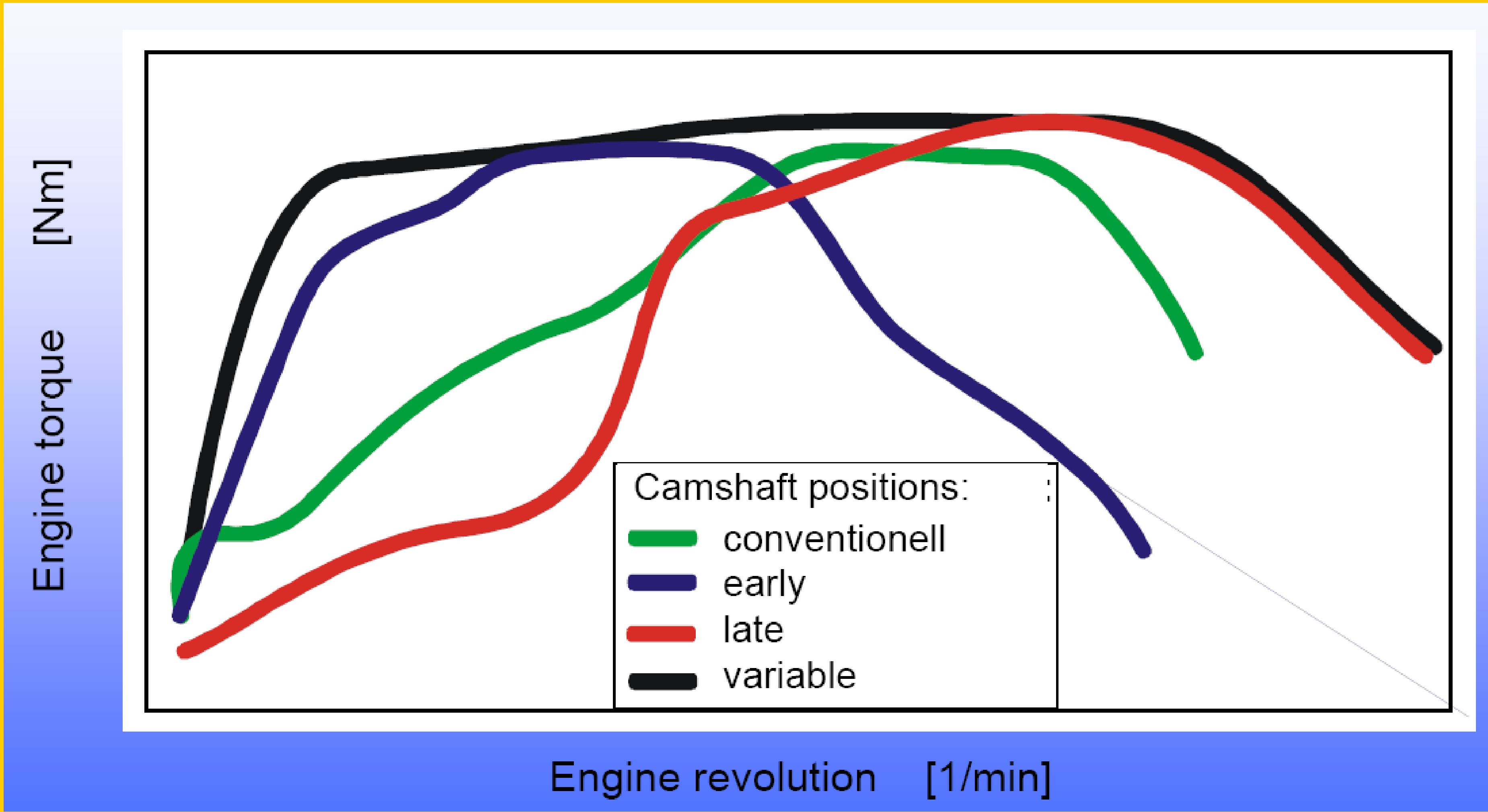
L	valve lift
BC	bottom dead center
TC	top dead center
1	exhaust valve opens
2	exhaust valve closes
3	inlet valve opens
4	inlet valve closes



2-3: overlap period

Volumetric efficiency in the Function of the Speed




Engine torque [Nm]
Camshaft positions:
conventionell
early
late
variable
Engine revolution [1/min]

Charging systems

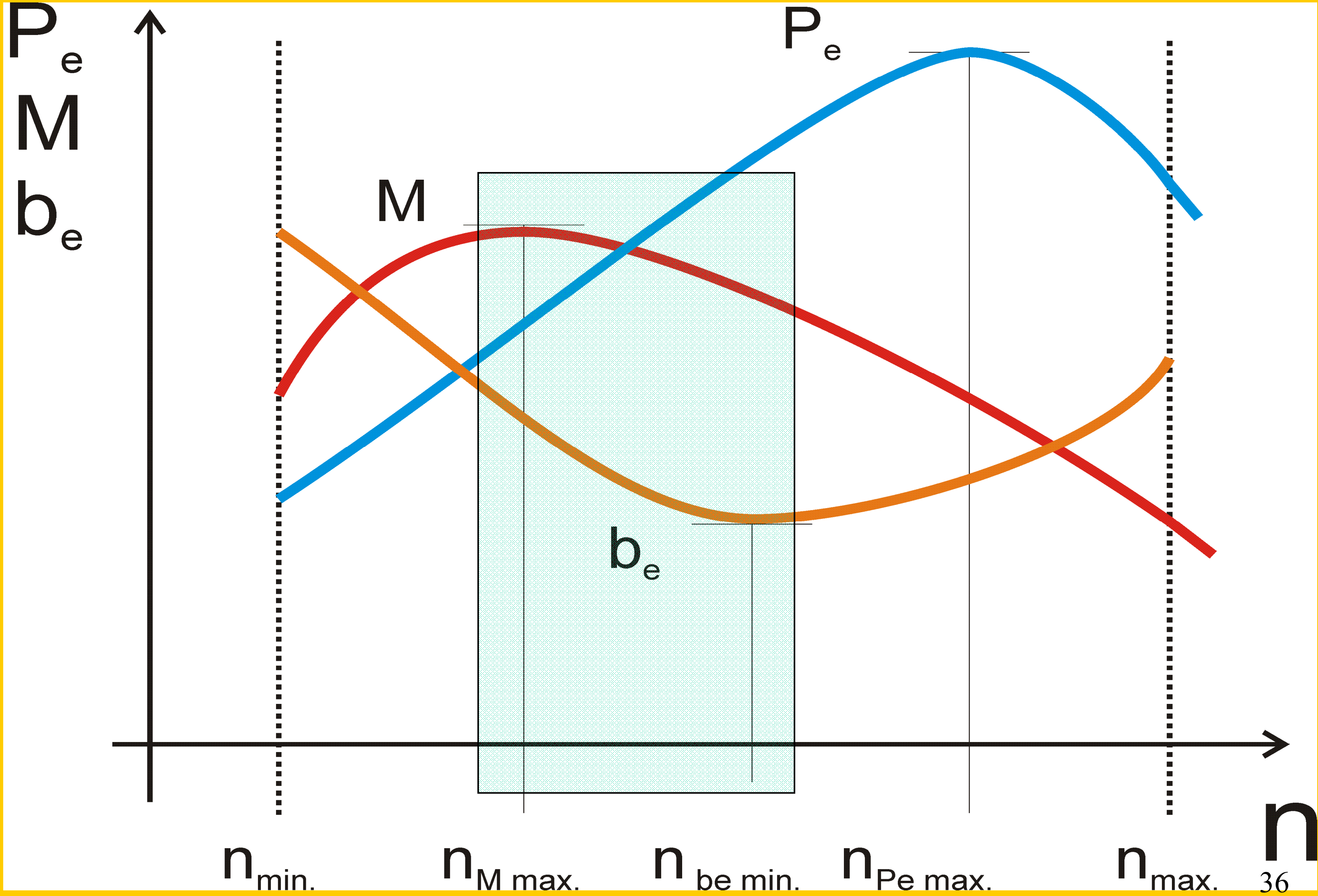
– Naturally aspirated
– Mechanically charged
– Turbo charged
– Acoustical charged

Advantages and Disadvantages

- Smaller Engine Dimensions (Down-sizing)
- Higher Power/mass ratio
- Higher efficiency
  - Pe/Pm ration better
  - Positive pumping work (W(-) -> W(+))
- Smaller Cooler

- Thermically and Mechanically Load increases

Characteristic Curves

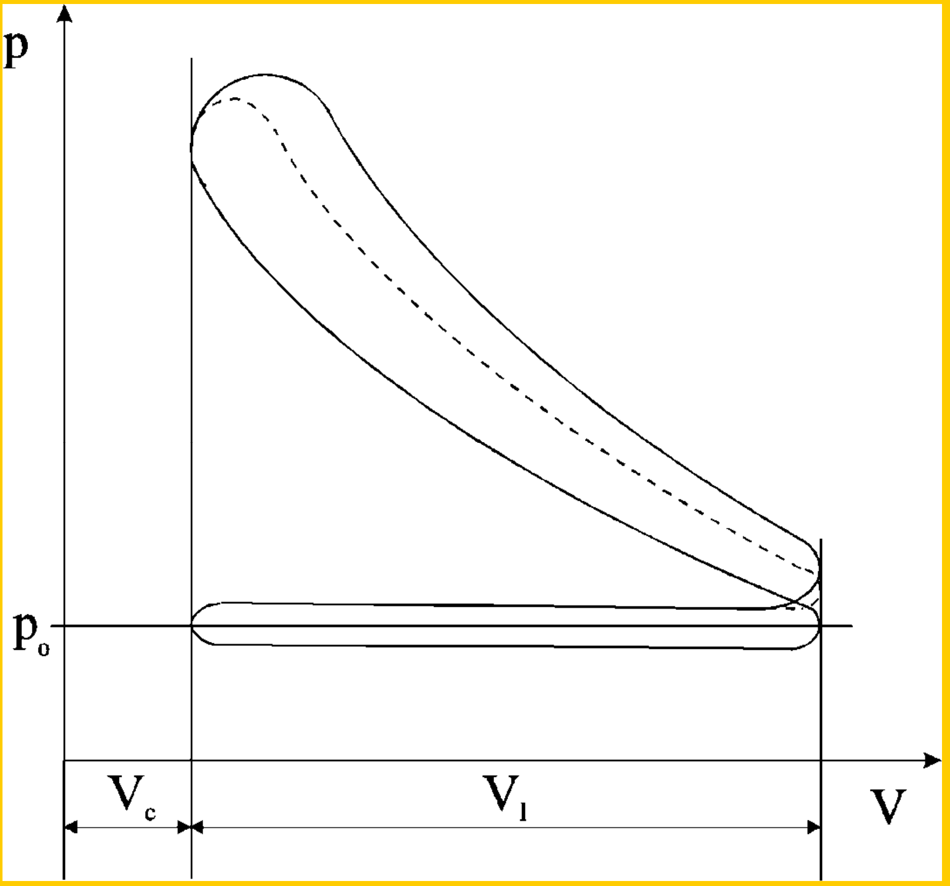


- $0\text{-}n_{min.}$:
  - Flywheel does not store enough energy,
  - Wrong mixing,
  - Big heat losses,
- $n_{min.}\text{-}n_{M\ max.}$:
  - Better mixing,
  - Growing volumetric eff.,
  - Decreasing Heat Losses,
- $n_{M\ max} – n_{be\ min.}$:
  - Decreasing volumetric eff.,
- $n_{be\ min.} – n_{pe\ max.}$:
  - Worse mixing
  - Power losses
- $n_{pe\ max.} – n_{max}$
  - The growth of friction is higher ($f[n^2]$) than the effect of speed growth ($f[n]$)

Control systems of IC Engines

Qualitative Control (CI ICE)

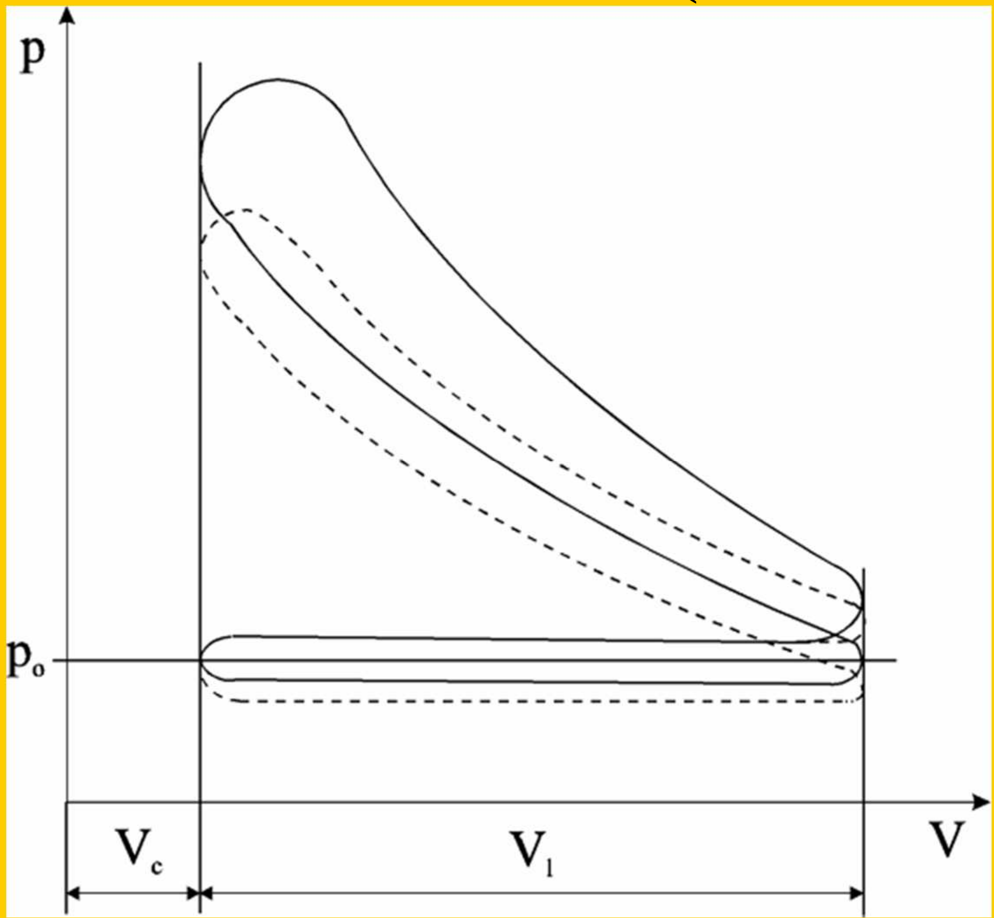
$\lambda \neq$ constant



Control possess of compression ignition engines
(— full load, - - - part load

Quantitative Control (SI ICE)

$\lambda \approx$ constant

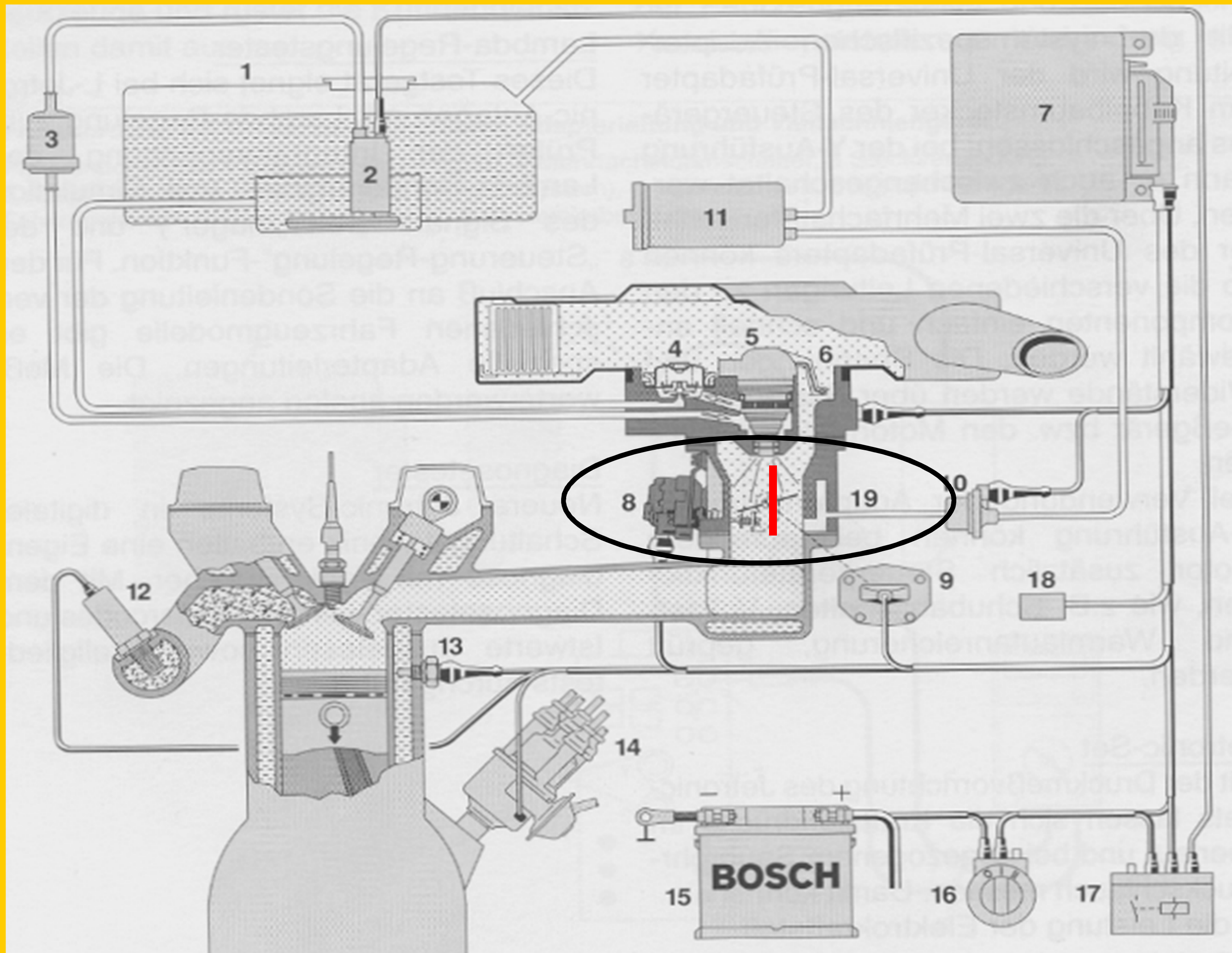


Control possess of spark ignition engines

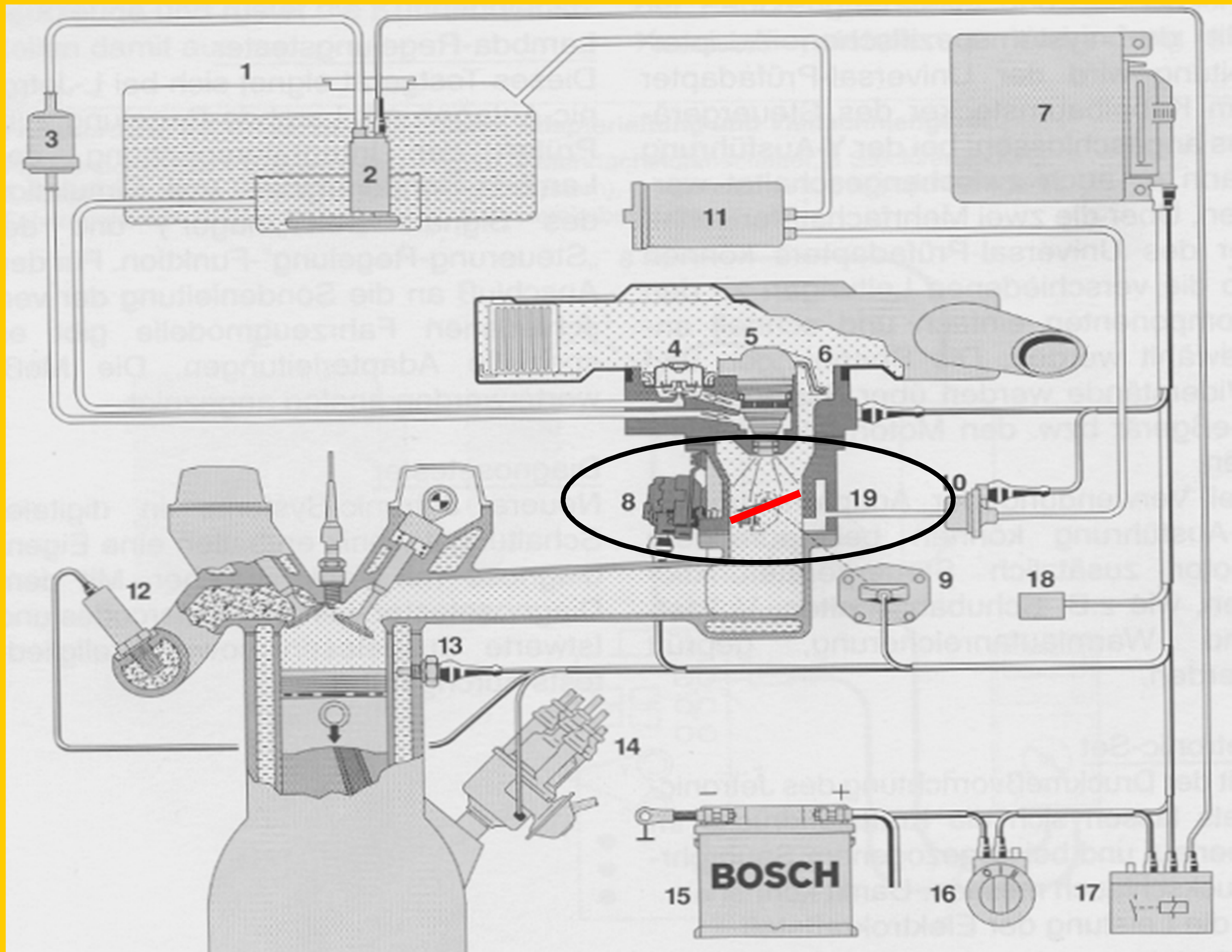
(— full load (throttle is open) , - - - part load (throttle is partially closed)

Engine Maps
(part load characteristic)

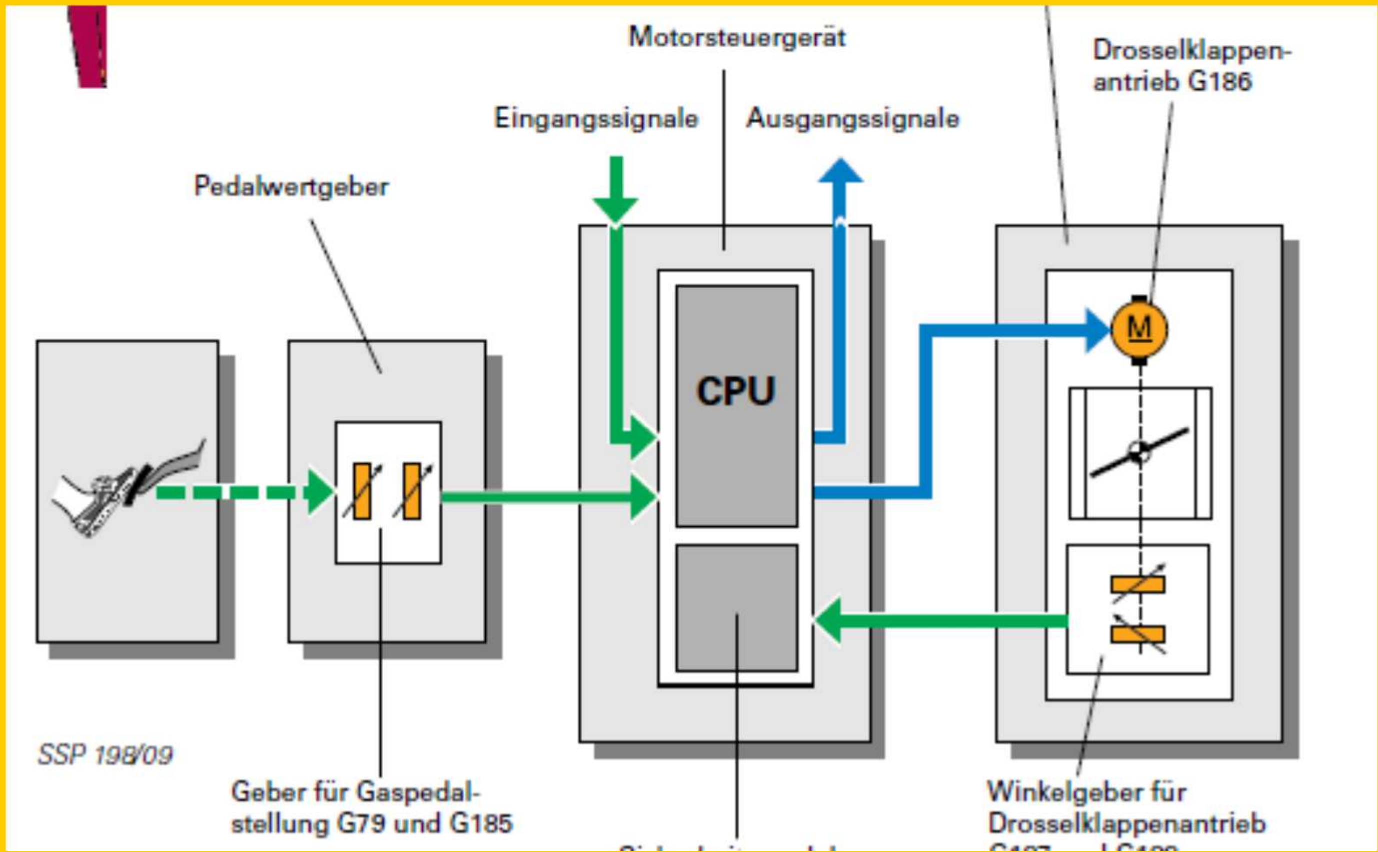
Quantitative control with throttle-valve, full-load



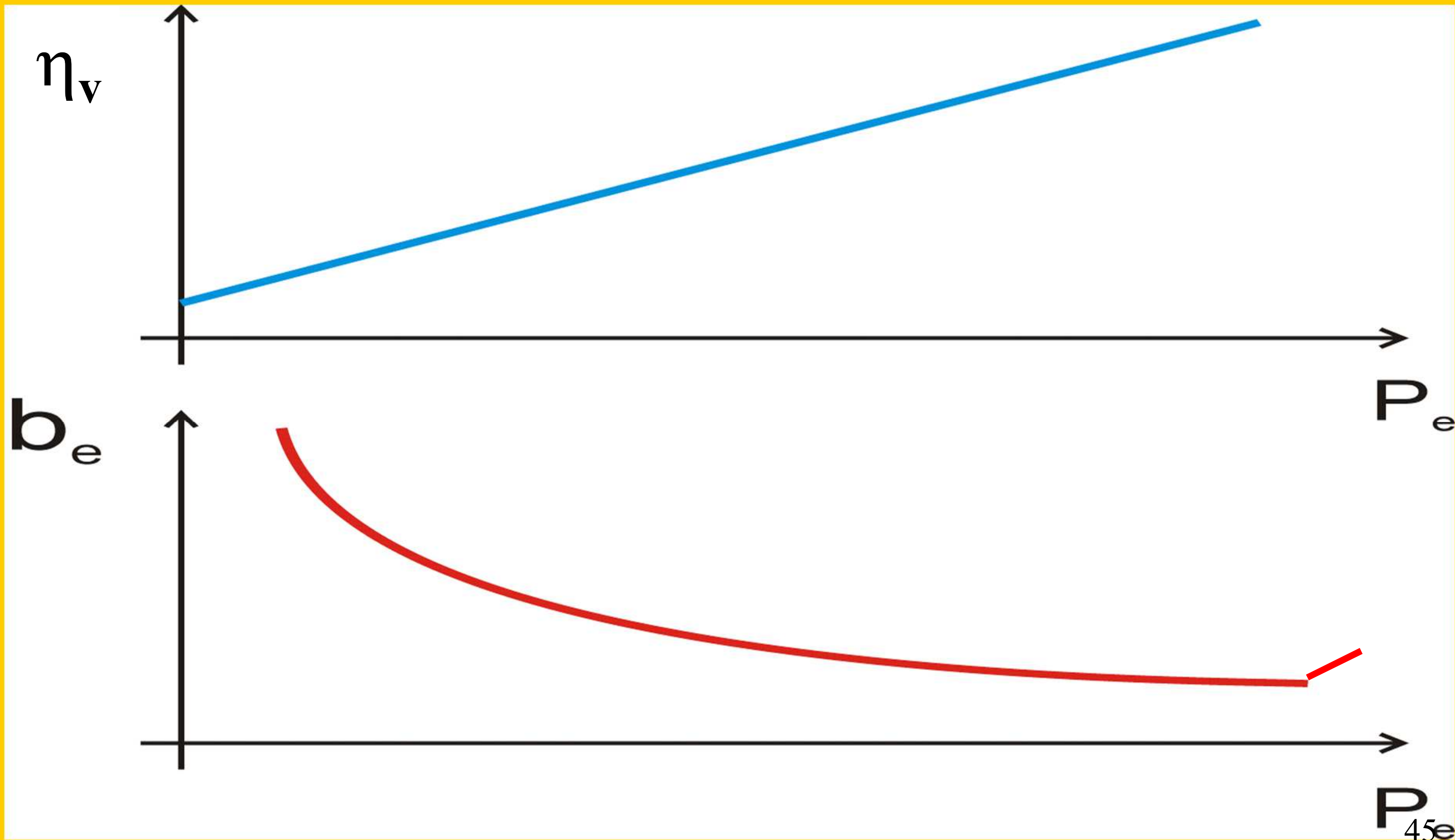
Quantitative control with throttle-valve, partial-load



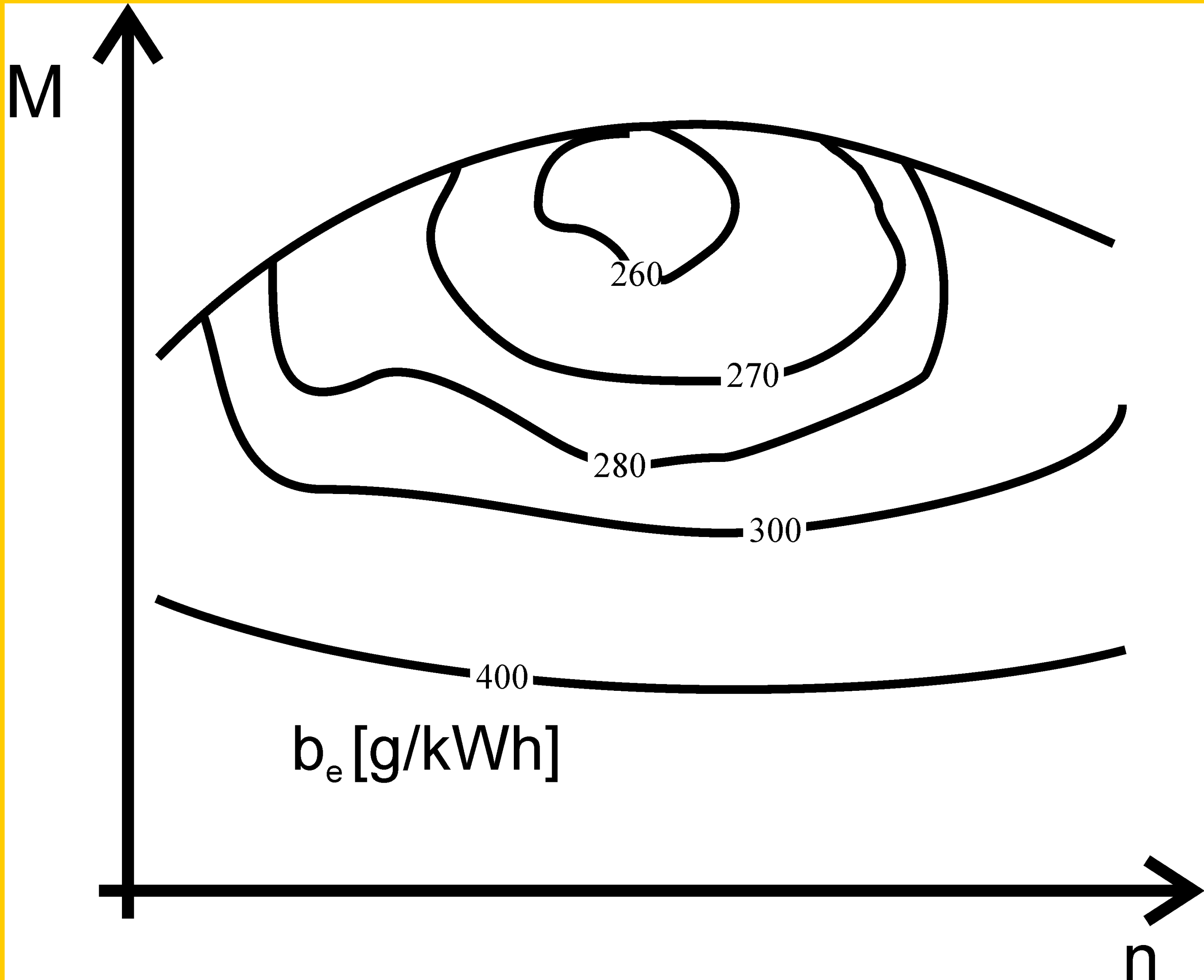
Electronic Throttle Control (ETC)



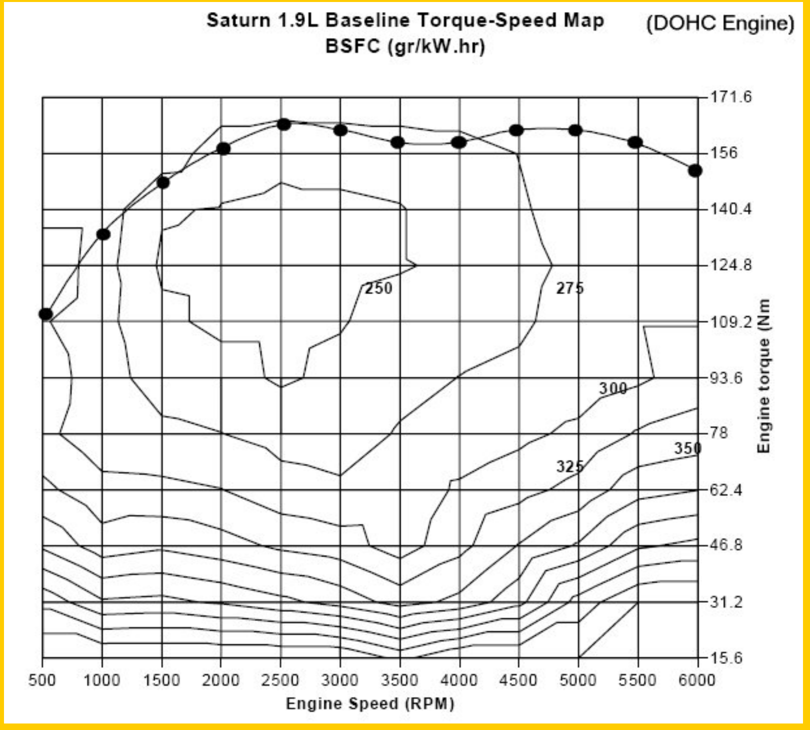
Volumetric efficiency and the Brake Specific Fuel Consumption at different loads, constant speed (S.I. ICE)

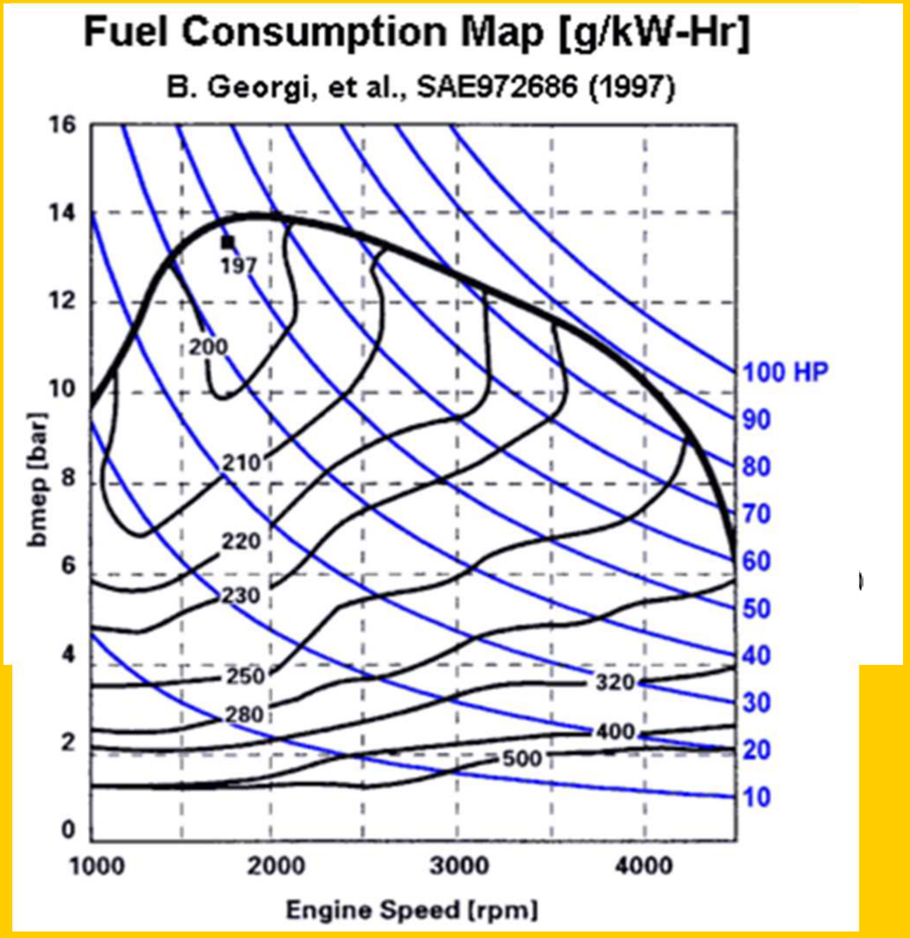


Engine Map (the BSFC in the function of the speed and Torque)-1



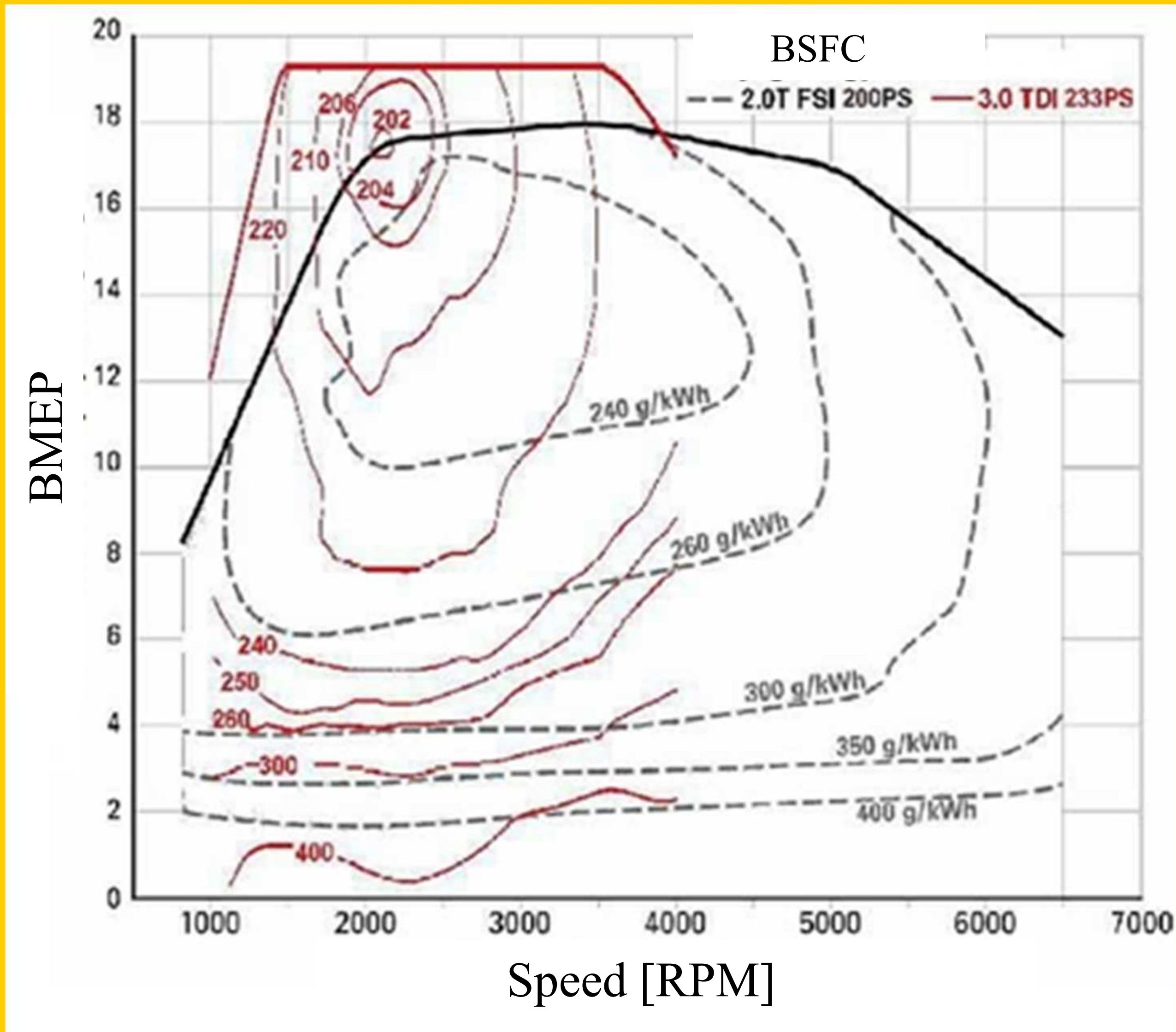
Engine Map (the BSFC in the function of the speed and Torque or BMEP)-2



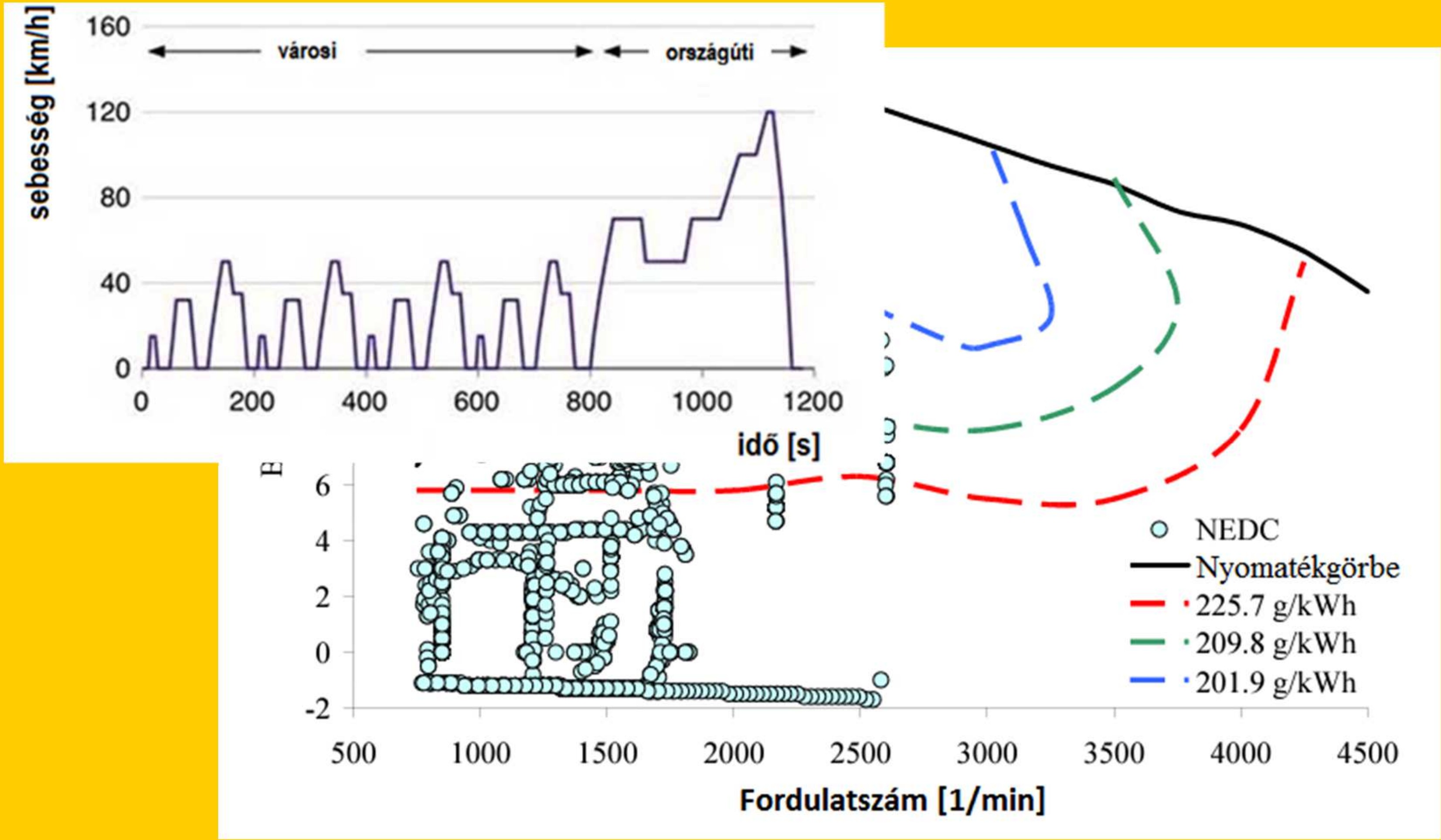


TDI 1.9L ALH 1999.5-2003

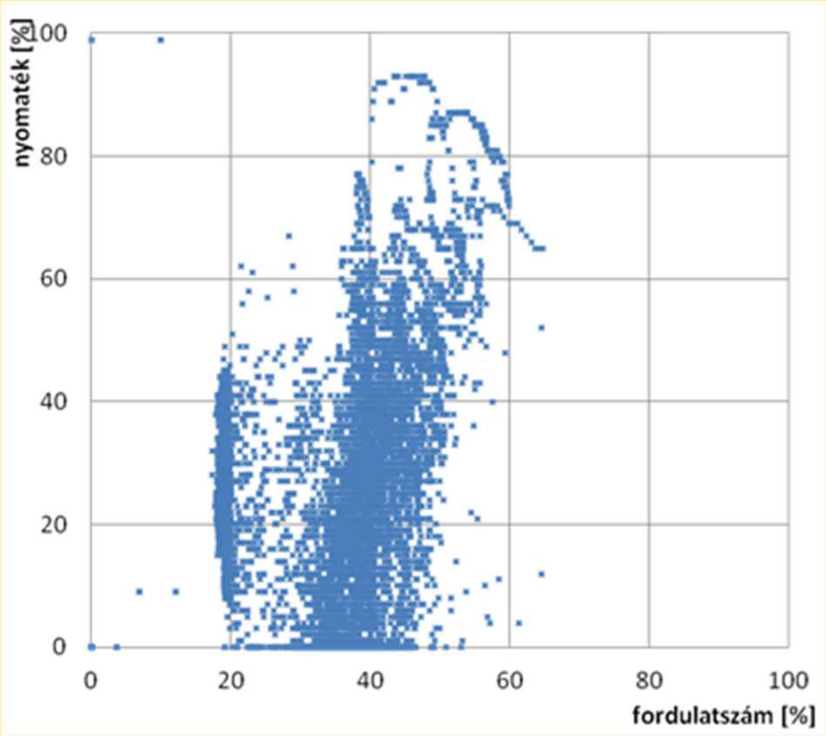
TFSI - TDI

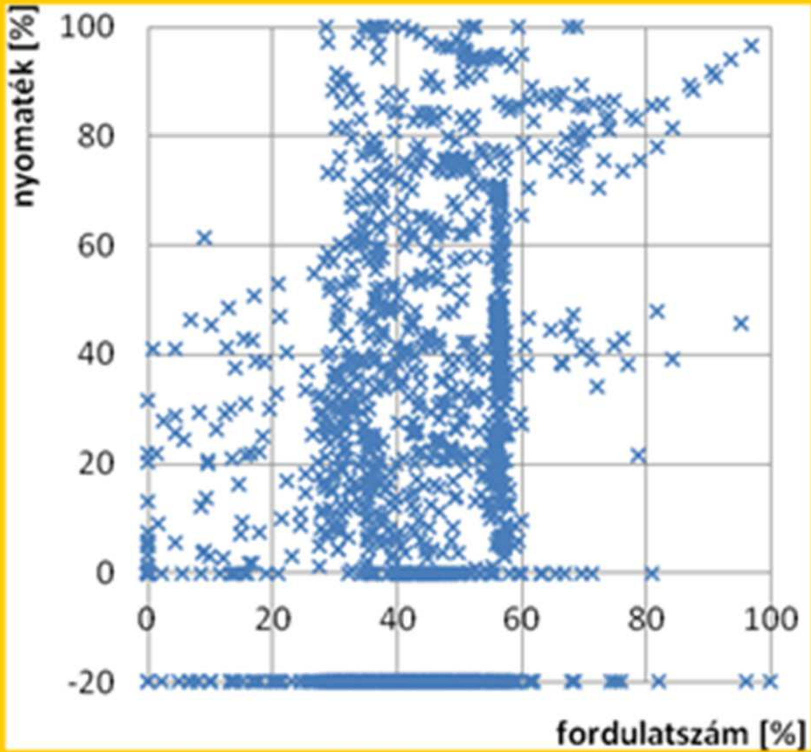

BSFC
2.0T FSI 200PS
3.0 TDI 233PS
BMEP
Speed [RPM]
240 g/kWh
260 g/kWh
300 g/kWh
350 g/kWh
400 g/kWh
202
204
206
210
220
240
250
260
300
400
0
2
4
6
8
10
12
14
16
18
20
1000
2000
3000
4000
5000
6000
7000

NEDC



VAGarena (2011): Ominaiskulutus, hyötysuhde ja polttoaineenkulutus. VAGarena.fi - Das. Forum, Finnország





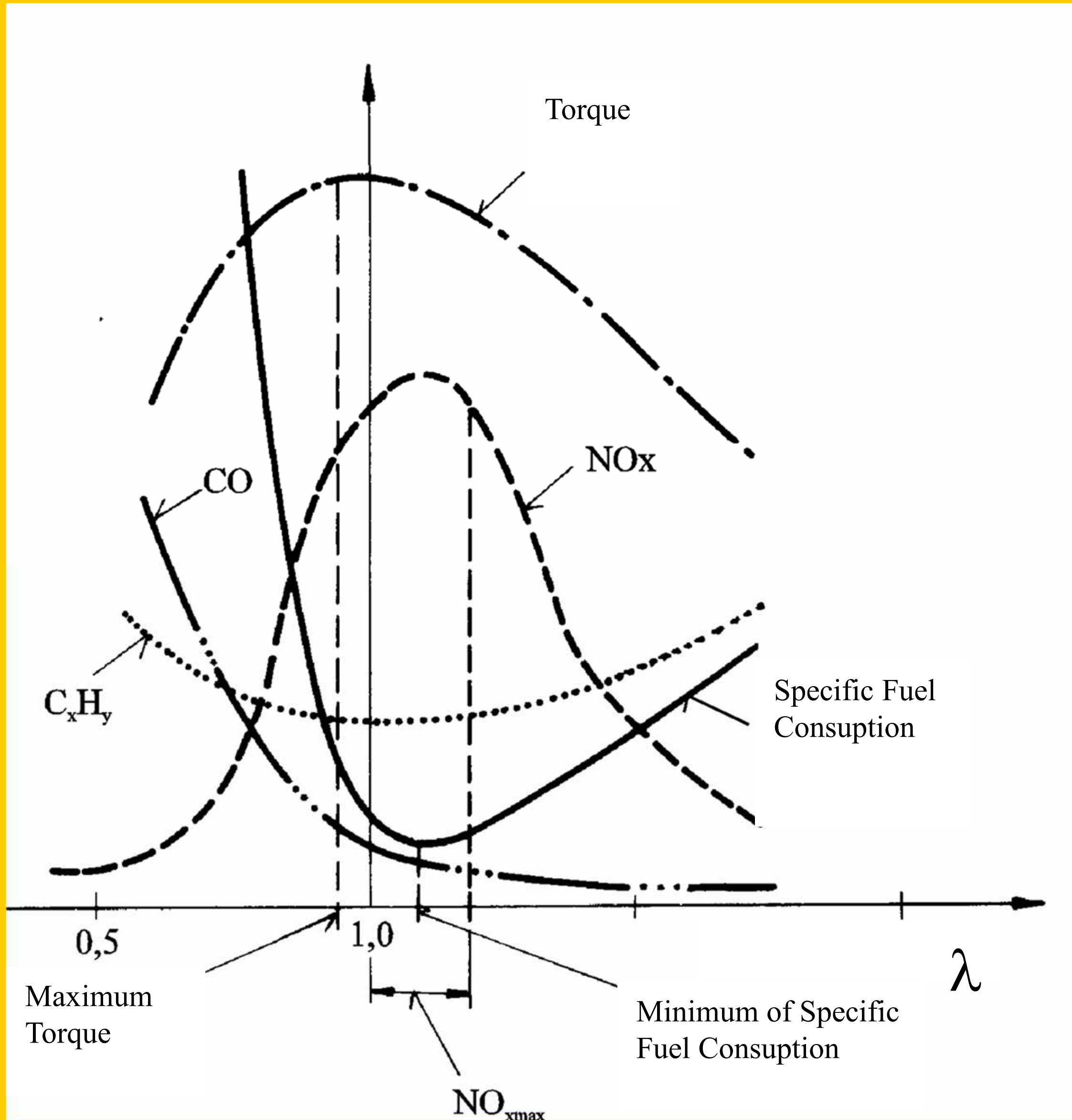
Real Drive

WHTC

Torbágyi Tas: Range extended hibrid jármű szélsőséges esetének vizsgálata, Budapest, 2014

Mixing systems of SI ICE

requirements (type) of mixture used, where is the optimum?



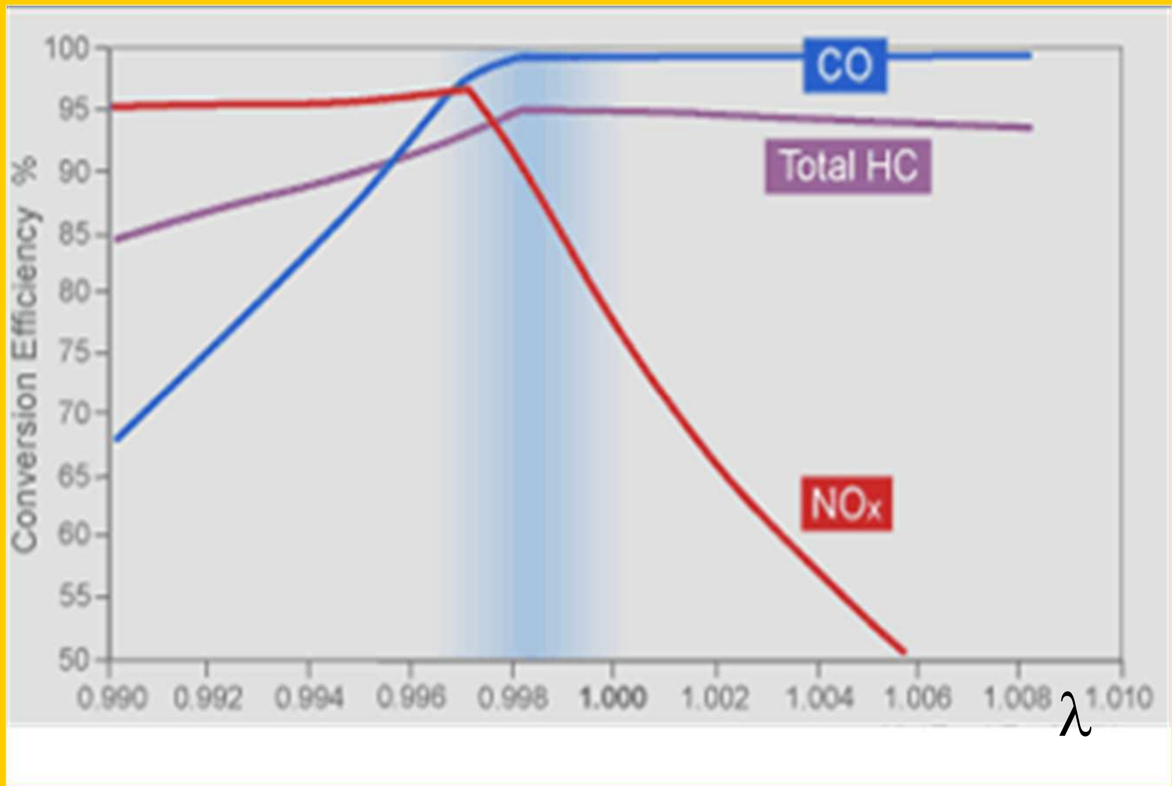
Catalytic Converters

3-way (NSCR) Catalysts (λ=1)

NO_x → N_2+O_2

CO → CO_2

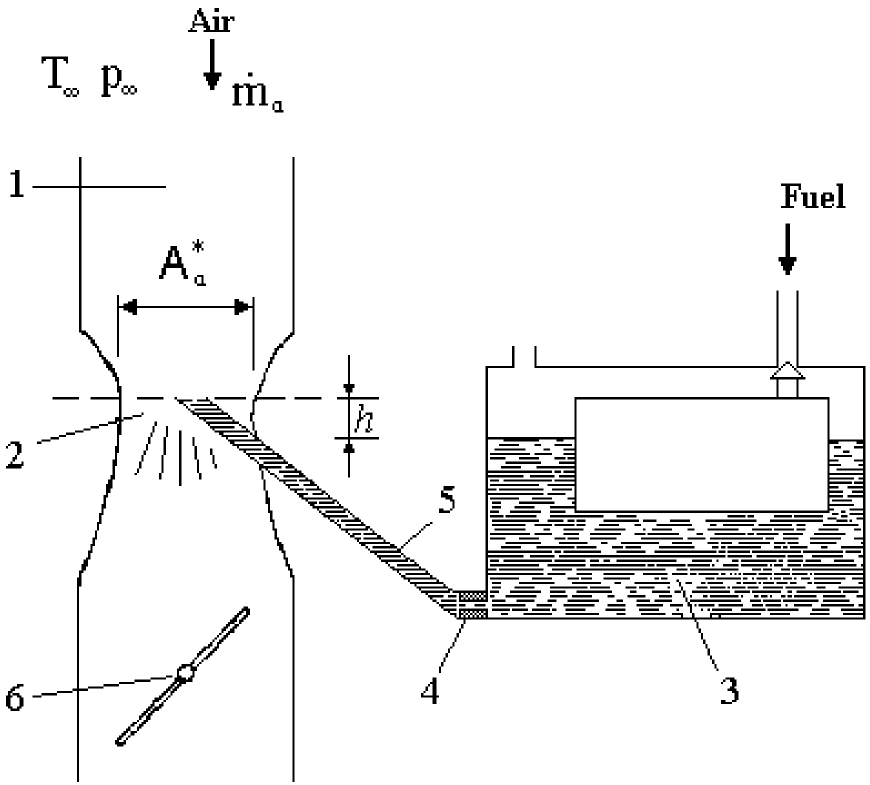
C_xH_y → H_2O+CO_2



Optimum for the NSCR

Additional requirements:

- Cold engine: fuel rich mix. (condensation)
- Idle run: fuel rich mix. (bad mixing)
- Full Load: fuel rich mix. (higher power)
- Acceleration: fuel rich mix. (higher power)



The elementary carburetor
1. Inlet section
2. Venturi nozzle
3. Float chamber
4. Metering orifice
5. Fuel discharge tube
6. Throttle plate

THE CARBURETOR

Massflow of Air across Venturi

$$m_a = \frac{C_{DT} A_T p_o}{\sqrt{R T_o}} \left(\frac{p_T}{p_o}\right)^{\frac{1}{\kappa}} \sqrt{\frac{2\kappa}{\kappa - 1}\left(1 - \frac{p_T}{p_o}\right)^{\frac{\kappa-1}{\kappa}}}$$

[2. 1]

After simplifications:

$$m_a = C_{DT} A_T \sqrt{2 \rho_a \Delta p_a}\,\Phi$$

[2. 2]

Where:

$$\Delta p_a = p_0 - p_T$$

[2. 3]

$$\Phi = \sqrt{\frac{\kappa - 1}{\kappa}\; \frac{\left(p_T / p_0\right)^{2/\kappa} - \left(p_T / p_0\right)^{\kappa+1/\kappa}}{1 - p_T / p_0}}$$

[2. 4]

Massflow of Fuel

$$m_f = C_{DO} A_o \sqrt{2 \rho_f \Delta p_f}$$

[2. 5]

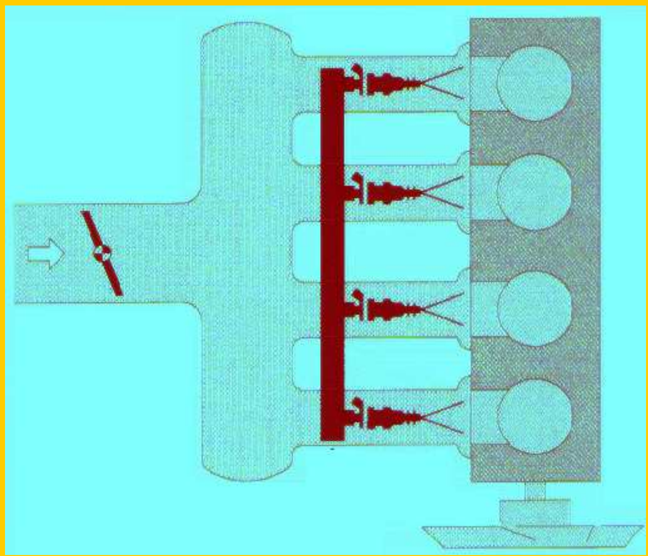
Access air factor:

$$\lambda = \frac{m_a}{m_f L_0} = \frac{\Phi}{L_0} \frac{C_{DT}}{C_{DO}} \frac{A_T}{A_O} \sqrt{\frac{\rho_a}{\rho_f} \frac{\Delta p_a}{\Delta p_a - \rho_f g h}} \approx C \frac{1}{\sqrt{\Delta p_a}}$$

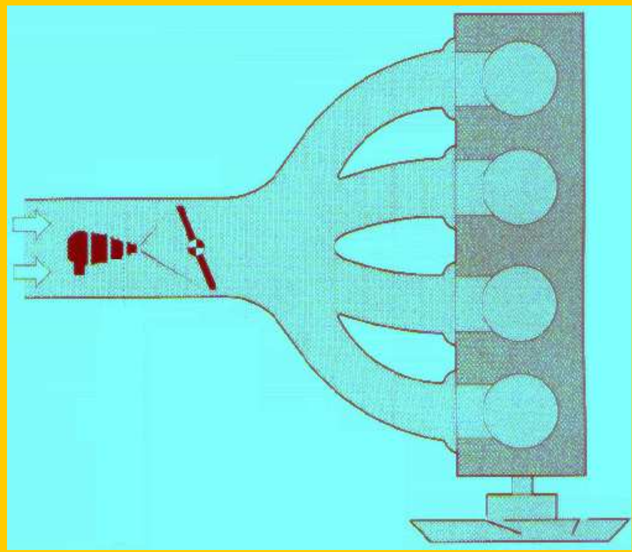
[2. 6]

ahol $L_{0benzin}$=14,7

Injector Types



MPI



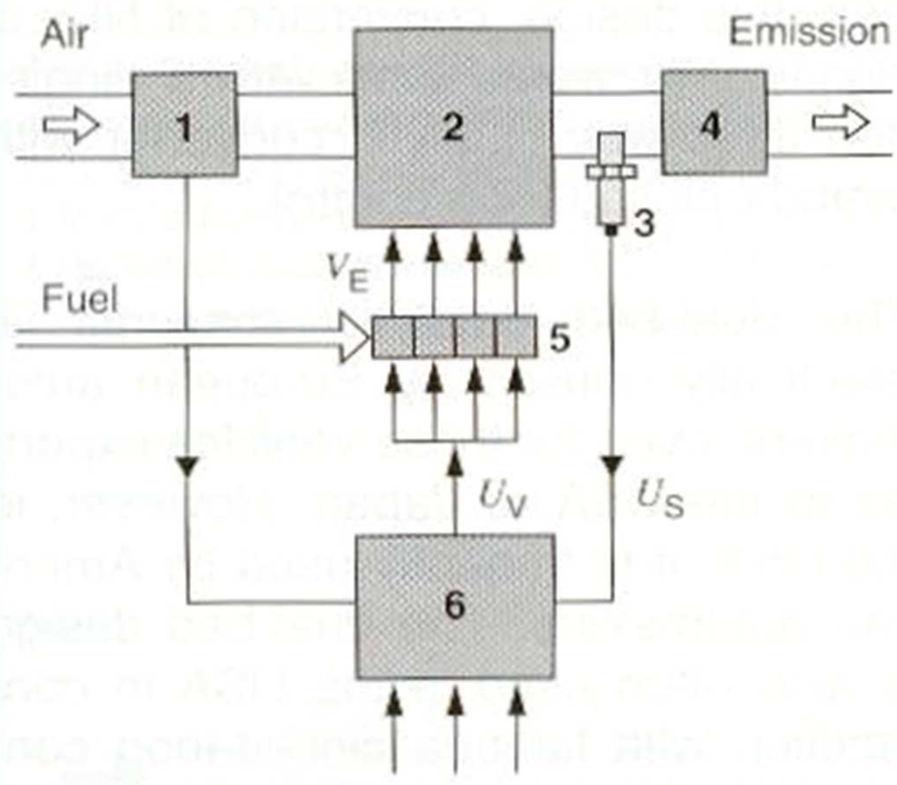
SPI

The advantages of the fuel injection over the carburation

- Homogenous fuel-air ration in all cylinders (MPI)
- Control system for the fuel metering (Excess air factor)
- Increased volumetric efficiency
  - There is no choking caused by the Ventury nozzle
- Higher Compression ratio
  - Knock limit !
- Evaporation Cools the IM
- Higher thermal efficiency

Lambda (Excess air factor) Control system

1 Air-flow sensor, 2 Engine, 3 Lambda sensor, 4 Catalytic converter, 5 Fuel-injection valves (injectors), 6 Lambda closed-loop control, U_S Sensor voltage, U_V Valve-actuation voltage, V_E Injected fuel quantity.



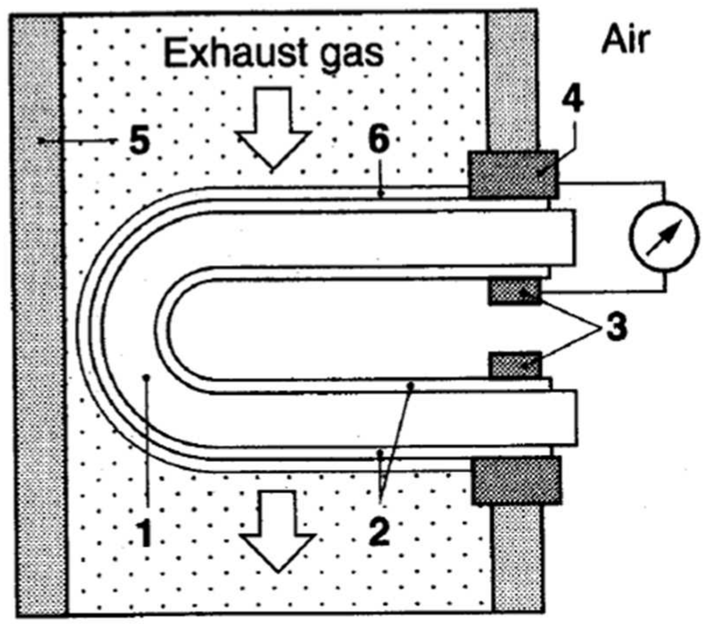
Source: BOSCH

Lambda Oxygen Sensor

- Solid-state electrolyte made of ZrO ceramic material.

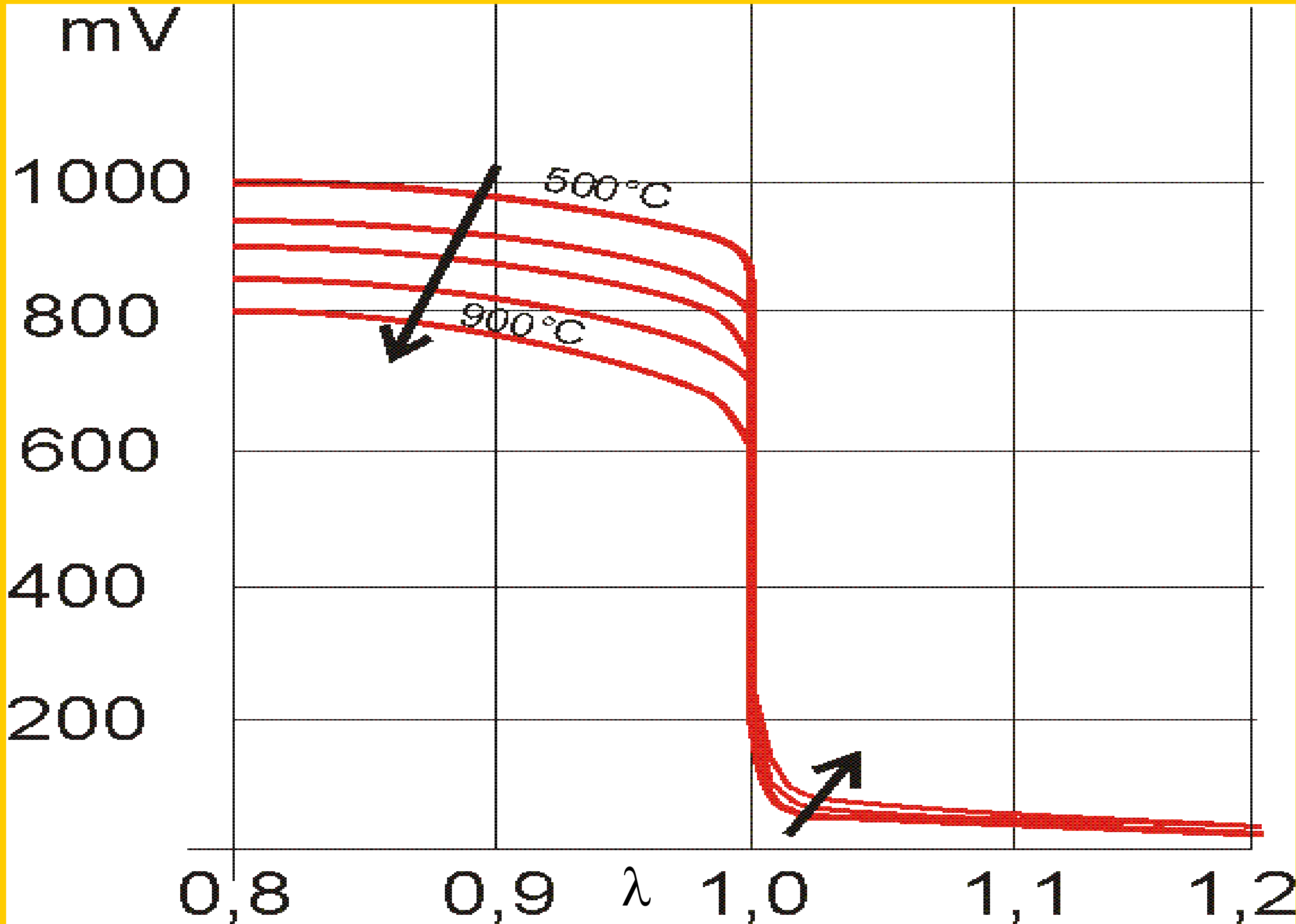
  At high temperatures, the electrolyte becomes conductive and generates a characteristic galvanic charge at the sensor connections this voltage is an index of exhaust gas oxygen content

Lambda oxygen sensor in exhaust pipe
1 Ceramic sensor, 2 Electrodes, 3 Contact, 4 Housing contacts, 5 Exhaust pipe, 6 Protective ceramic coating (porous).

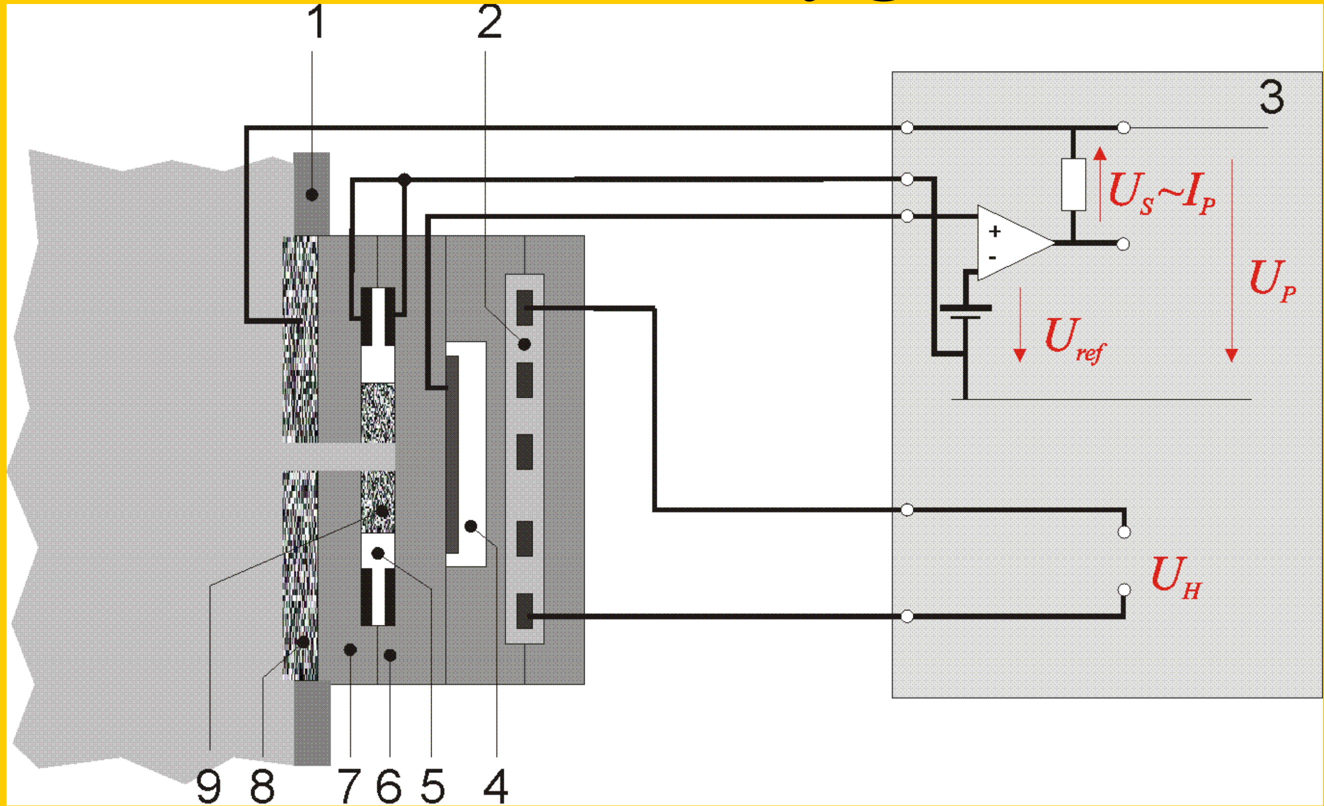


Source: BOSCH

Lambda Oxygen Sensor

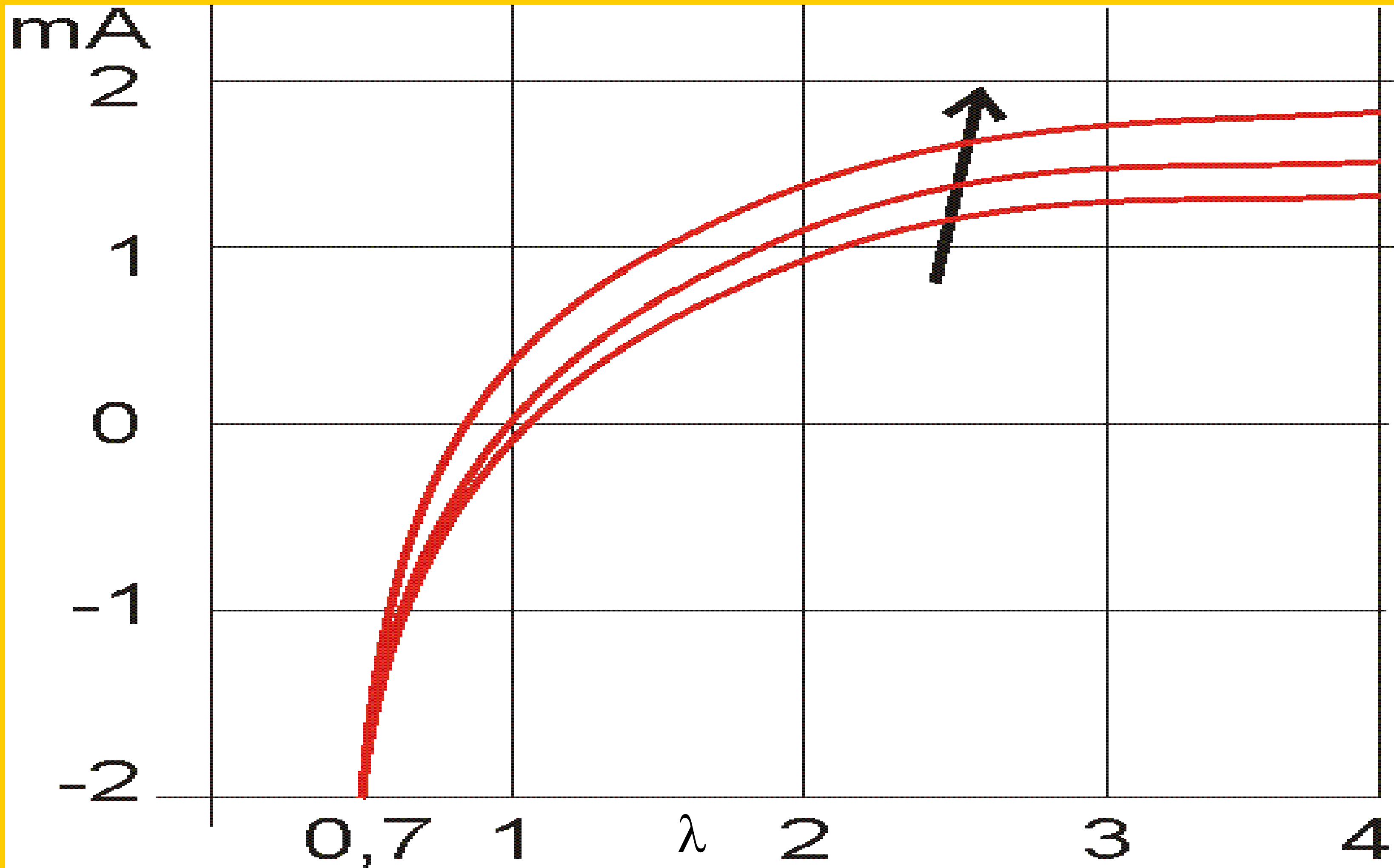


Broadband lambda oxygen sensor

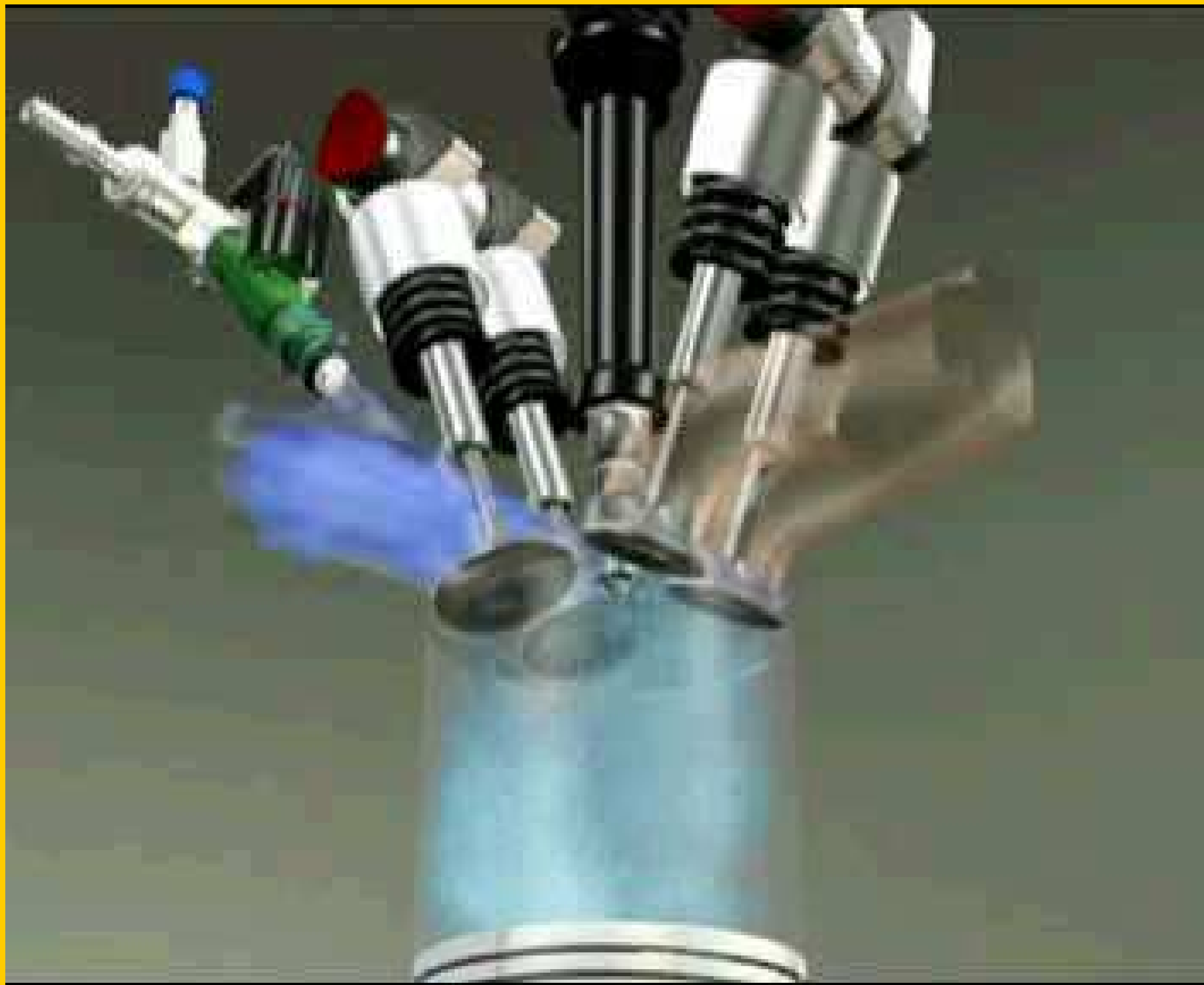


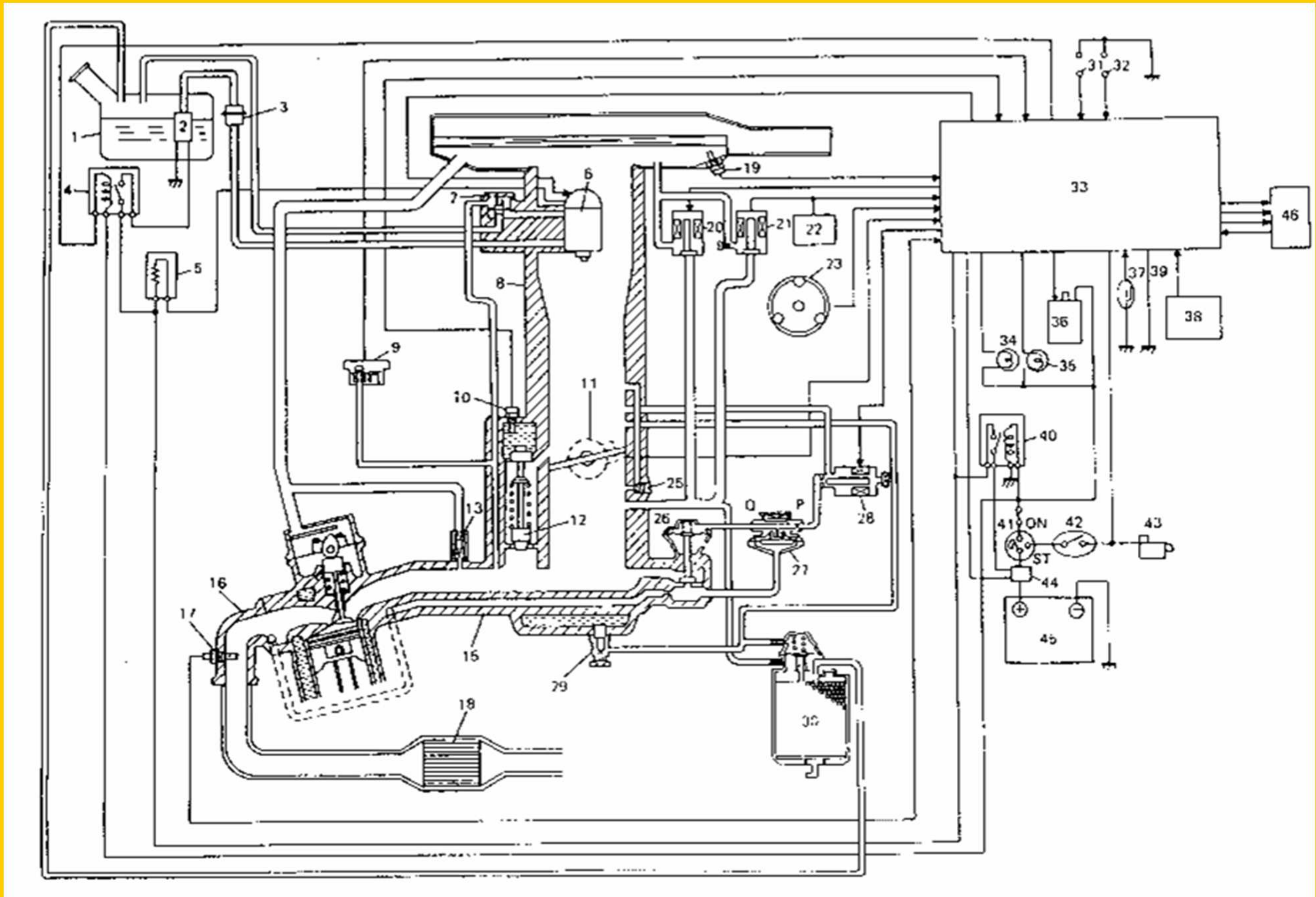
1: Exhaust gases, 2: Heater, 3: Control loop (electronics), 4: Reference air channel, 5: Diffusion gap, 6: Nernst cell, 7:Oxygen pump, 8: Protector, 9: Diffusion gap

Broadband lambda oxygen sensor



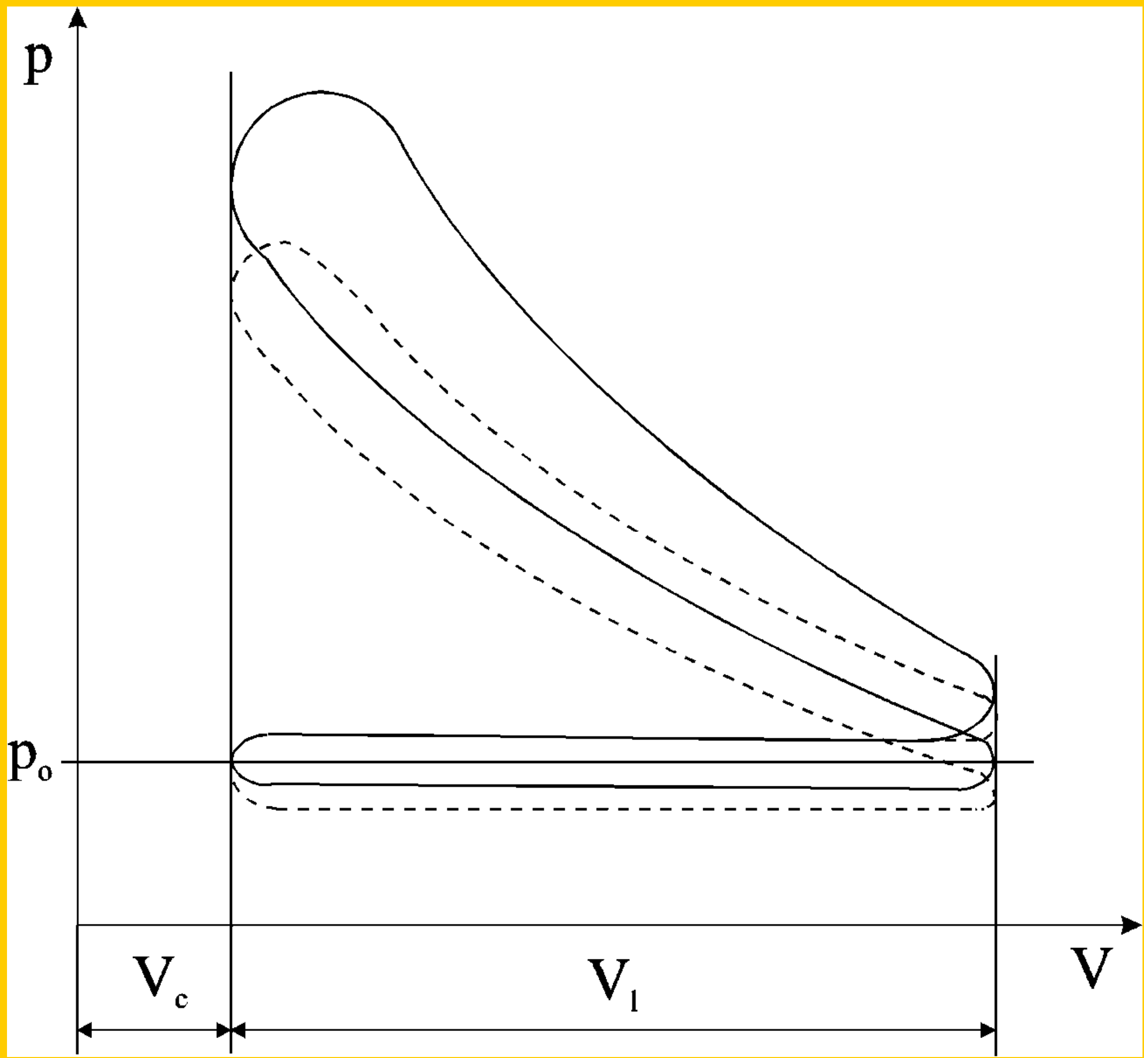
MPI (Ford)



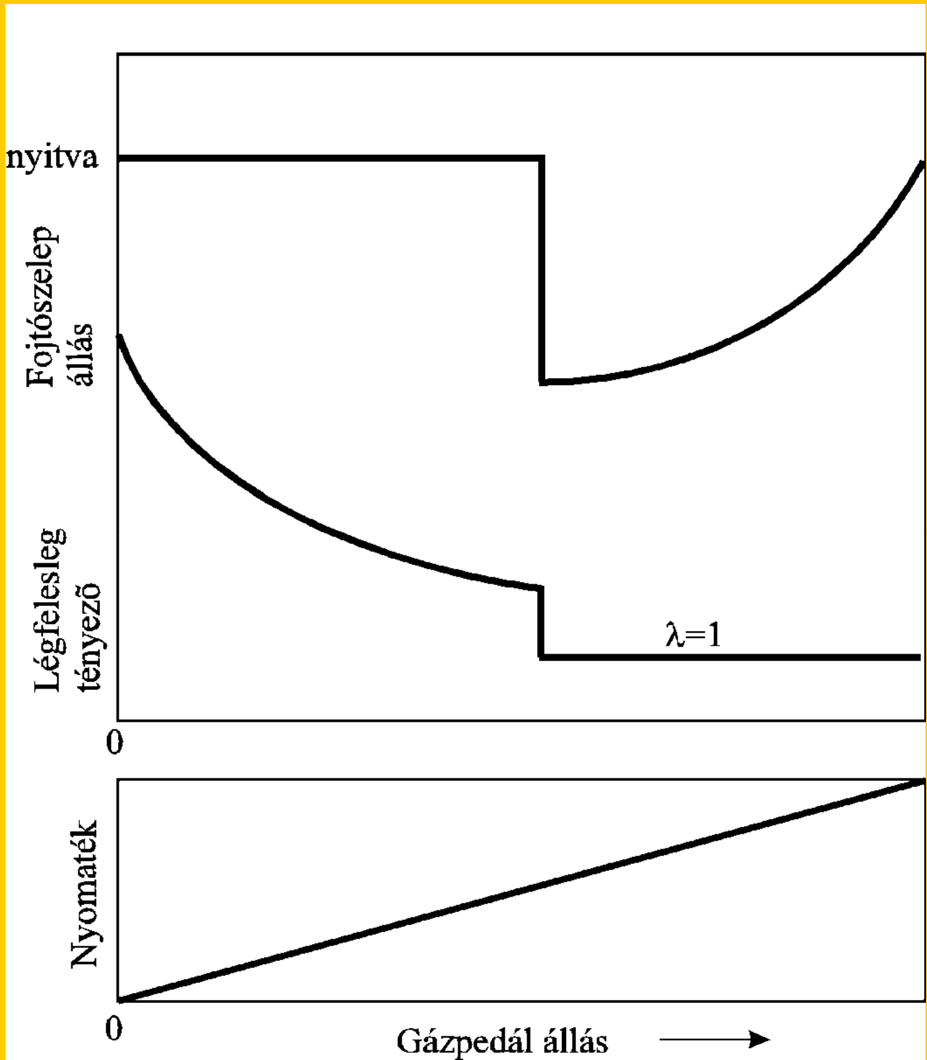


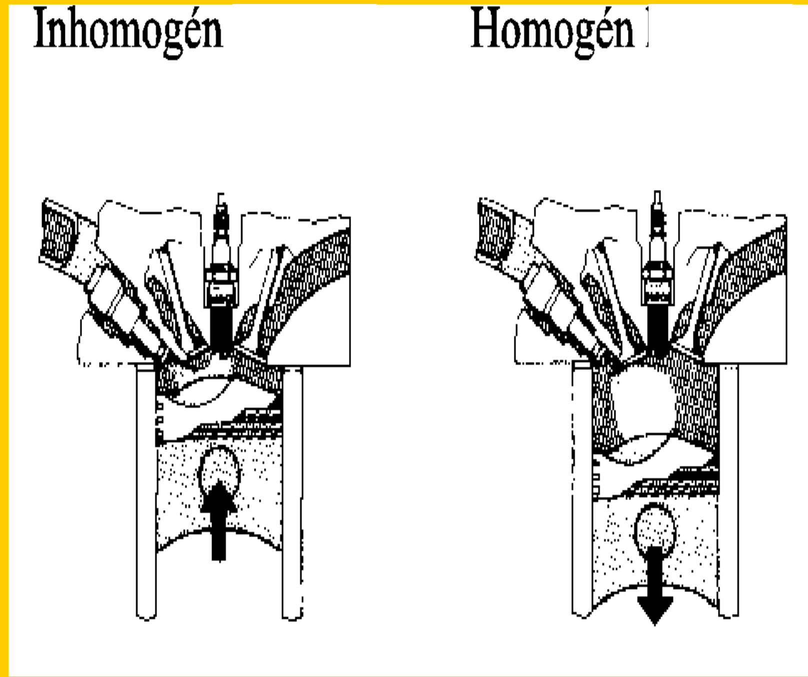
Monotronic system

Direct injection



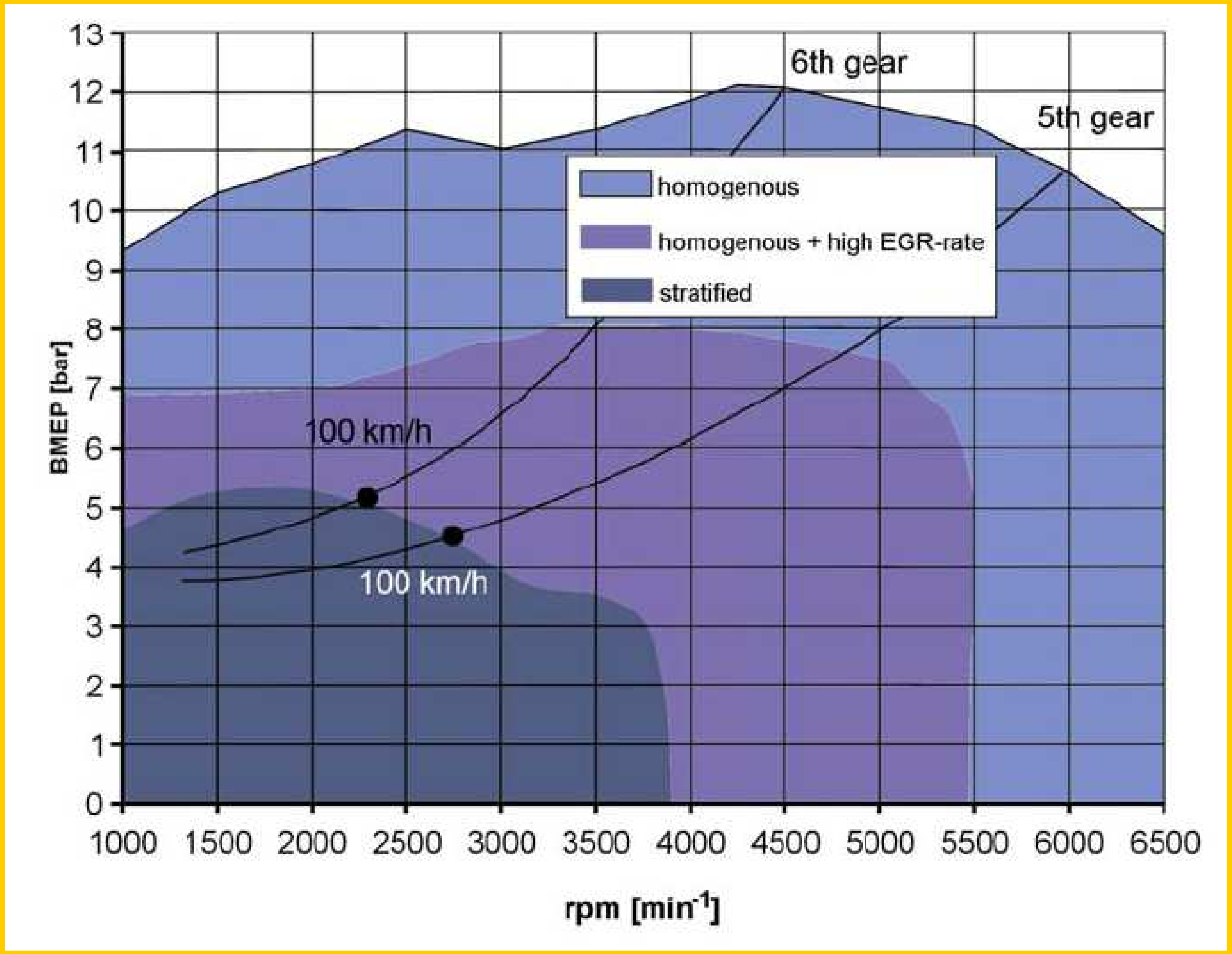
Control of the Otto cycle (- full load, --- partial load)





Bosch direct injection system

FSI (GDI) Engines mixtures



FSI (GDI) Engines Piston

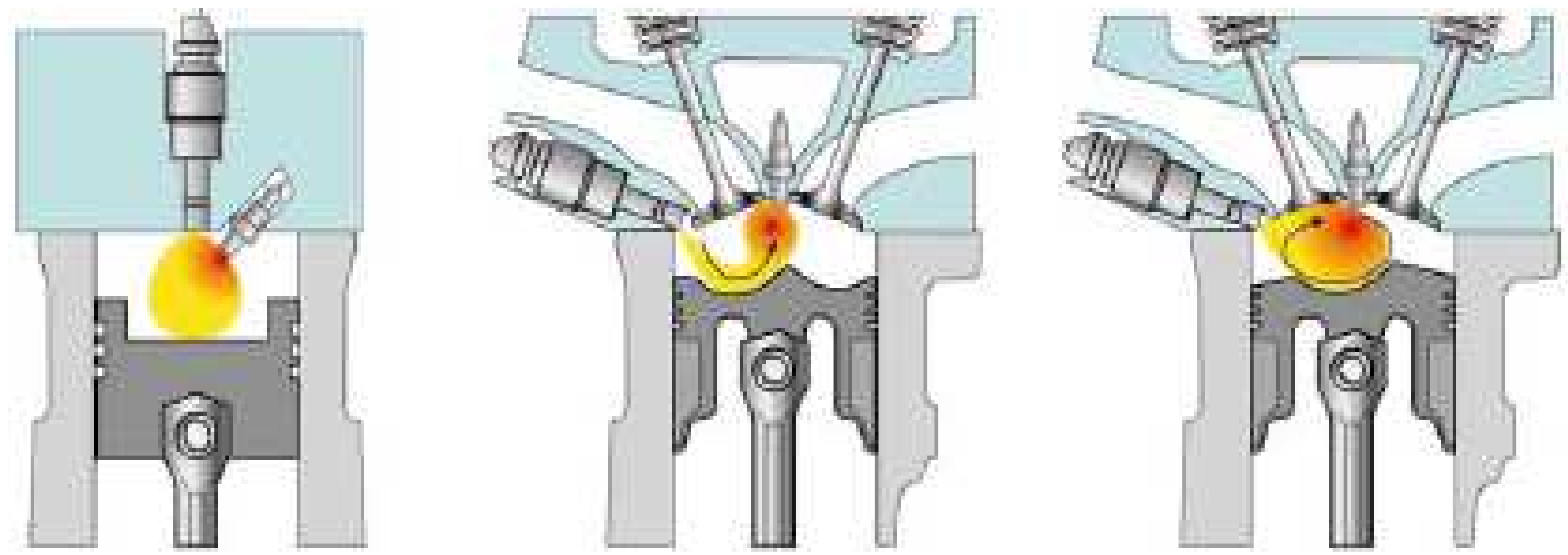
- Wall control type

FSI (GDI) Engines Piston

- Wall control type

FORD

Control of Direct Injection Systems



Spark Cont. Wall Cont. Swirl Cont.

2016

Air-fuel mixing methods

- o Internal (CIE, GDI (SIE))
- o External(SIE)

Combustion chamber design

- single open combustion chamber
- divided combustion chamber
  - o swirl chamber systems
  - o prechamber systems

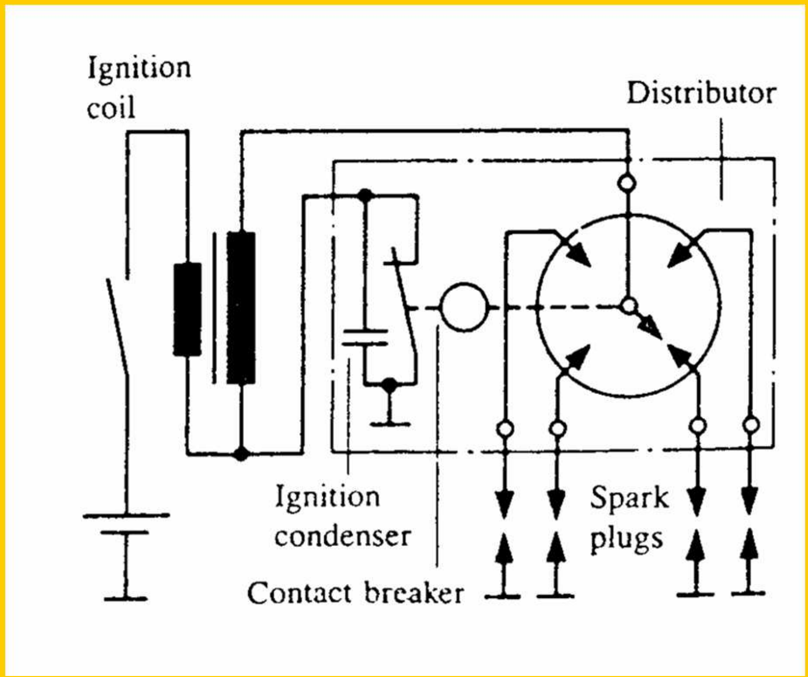
Start of Combustion

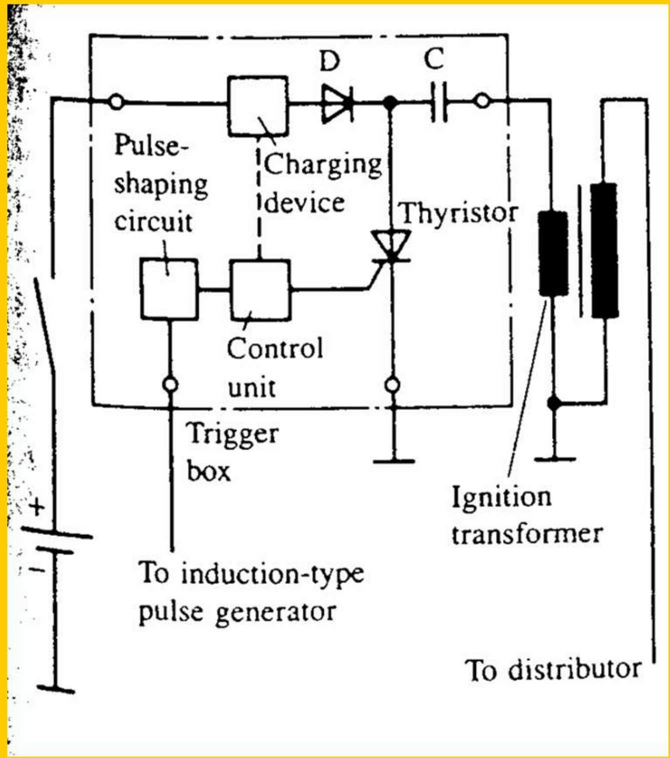
- External energy (Spark)
- Compression (Modell engines)
- Hot Spot

SPARK IGNITION

- In spark-ignition engines, the electrical discharge produced between the spark plug electrodes by the ignition system starts the combustion process close to the end of the compression stroke. The high-temperature plasma kernel created by the spark develops into a self-sustaining and propagating flame front - a thin reaction sheet where the exothermic combustion chemical reactions occur.
- The function of the ignition system is to initiate this flame propagation process, in a repeatable manner cycle-by-cycle, over the full load and speed range of the engine at the appropriate point in the engine cycle.
- A spark can arc from one electrode to another when a sufficiently high voltage is applied. Ignition systems commonly used to provide this spark are: battery ignition systems where the high voltage is obtained with an ignition coil (coil ignition systems); battery systems where the spark energy is stored in a capacitor and transferred as a high-voltage pulse to the spark plug by means of a special transformer (capacitive-discharge ignition systems); and magneto ignition systems where the magneto - a rotating magnet or armature - generates the current used to produce a high-voltage pulse.

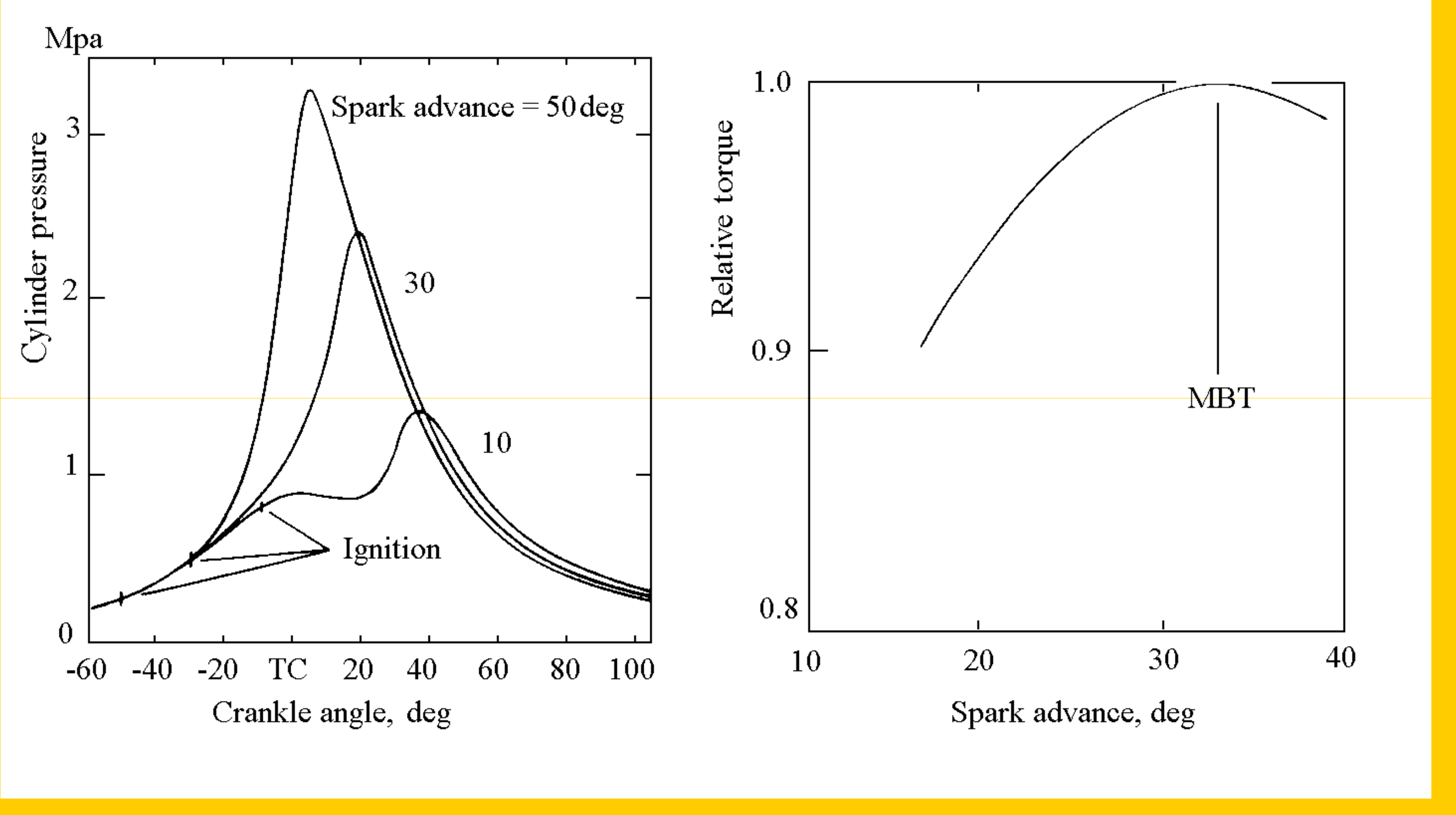
Coil Ignition Systems

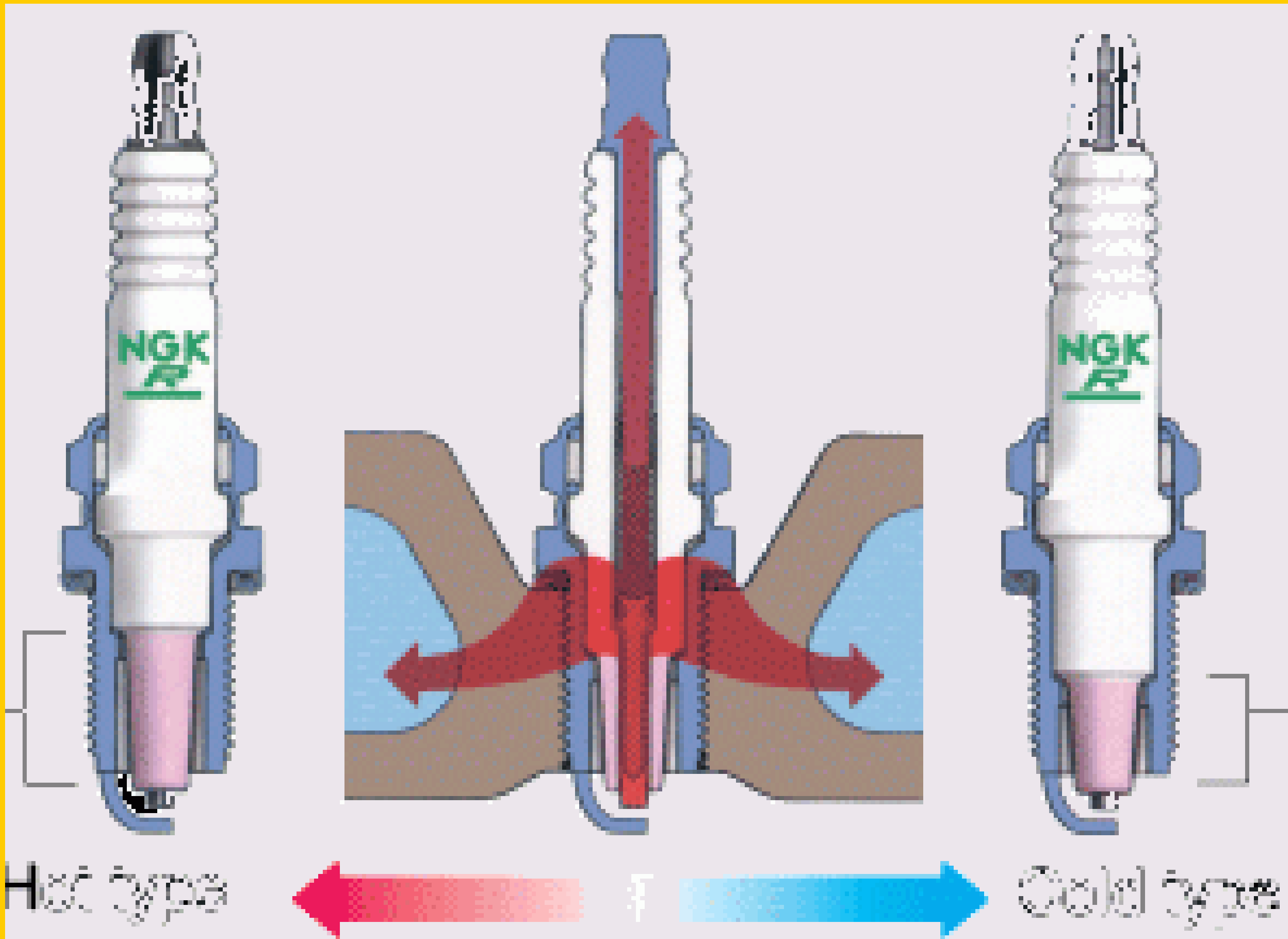




Capacitive-Discharge Ignition (CDI) Systems

Ignition advance

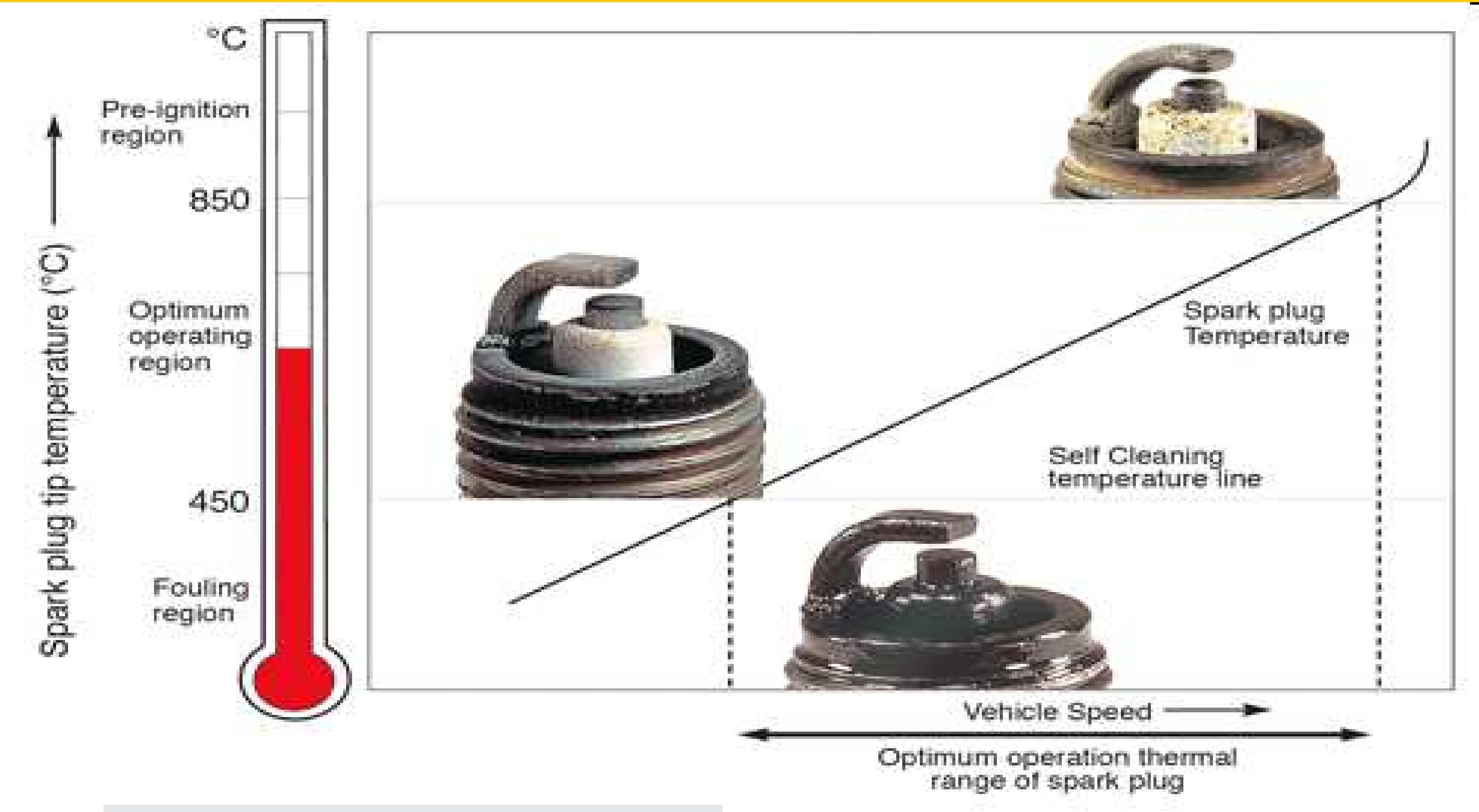




Hot Spark

Cold Spark

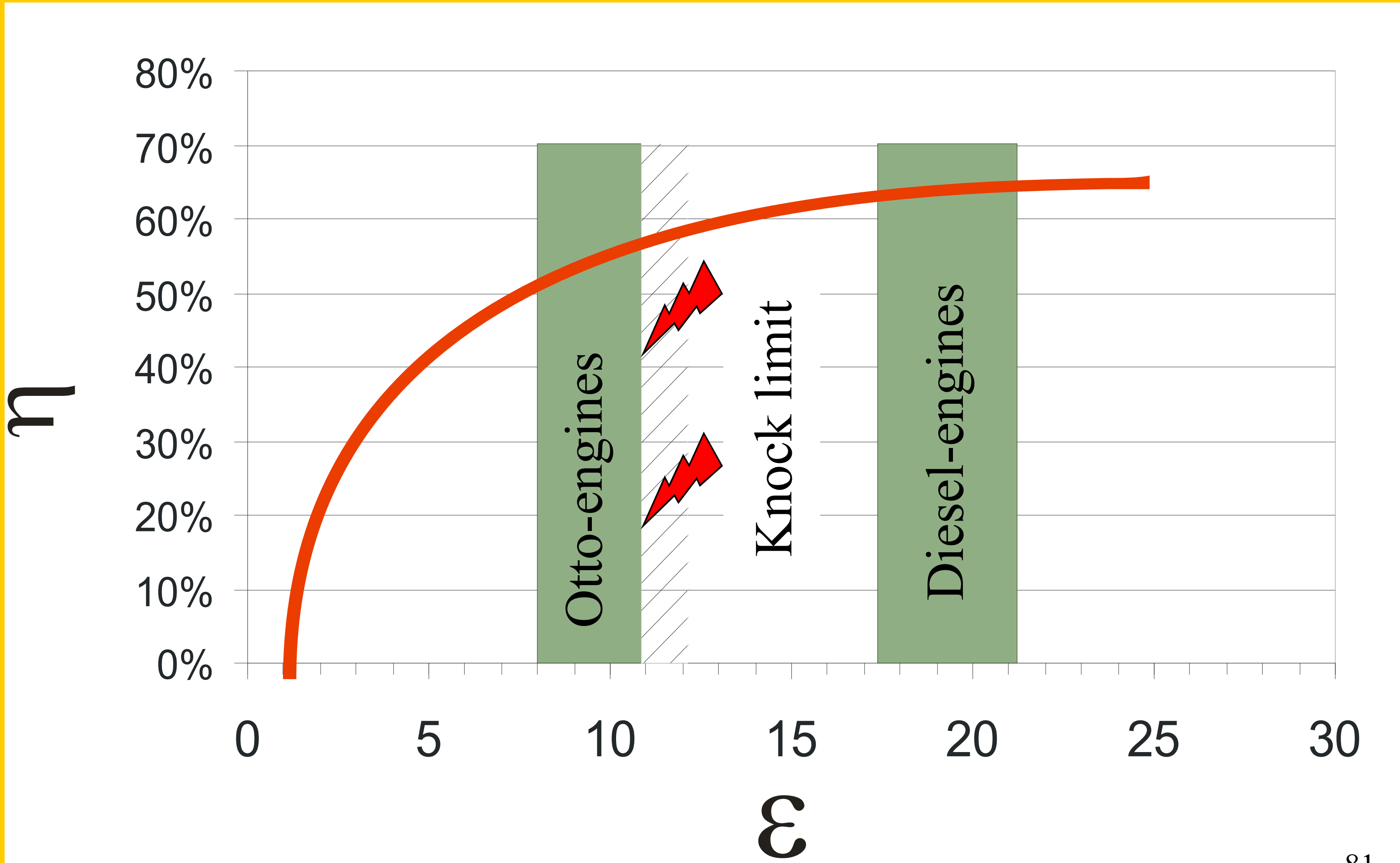
Heat range

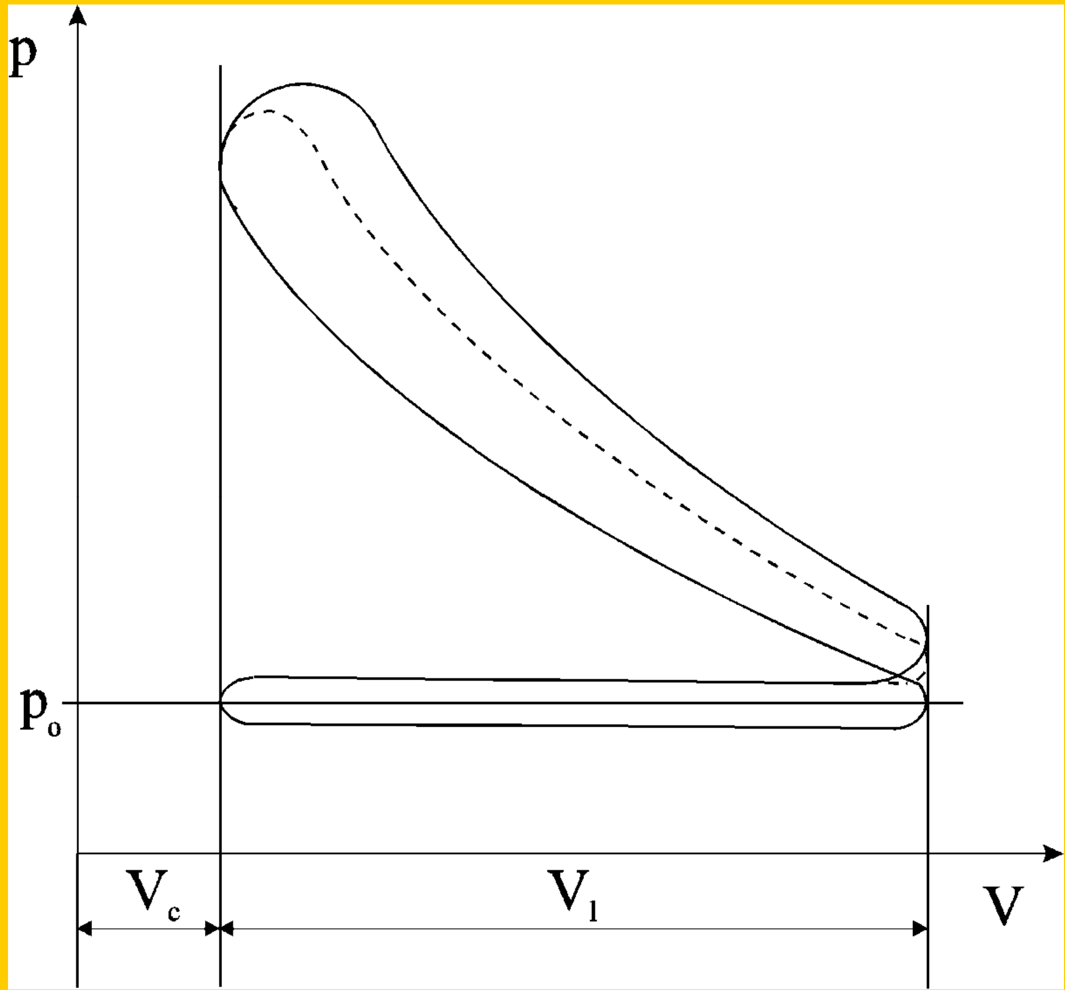


source: NGK

Diesel or CI Engines

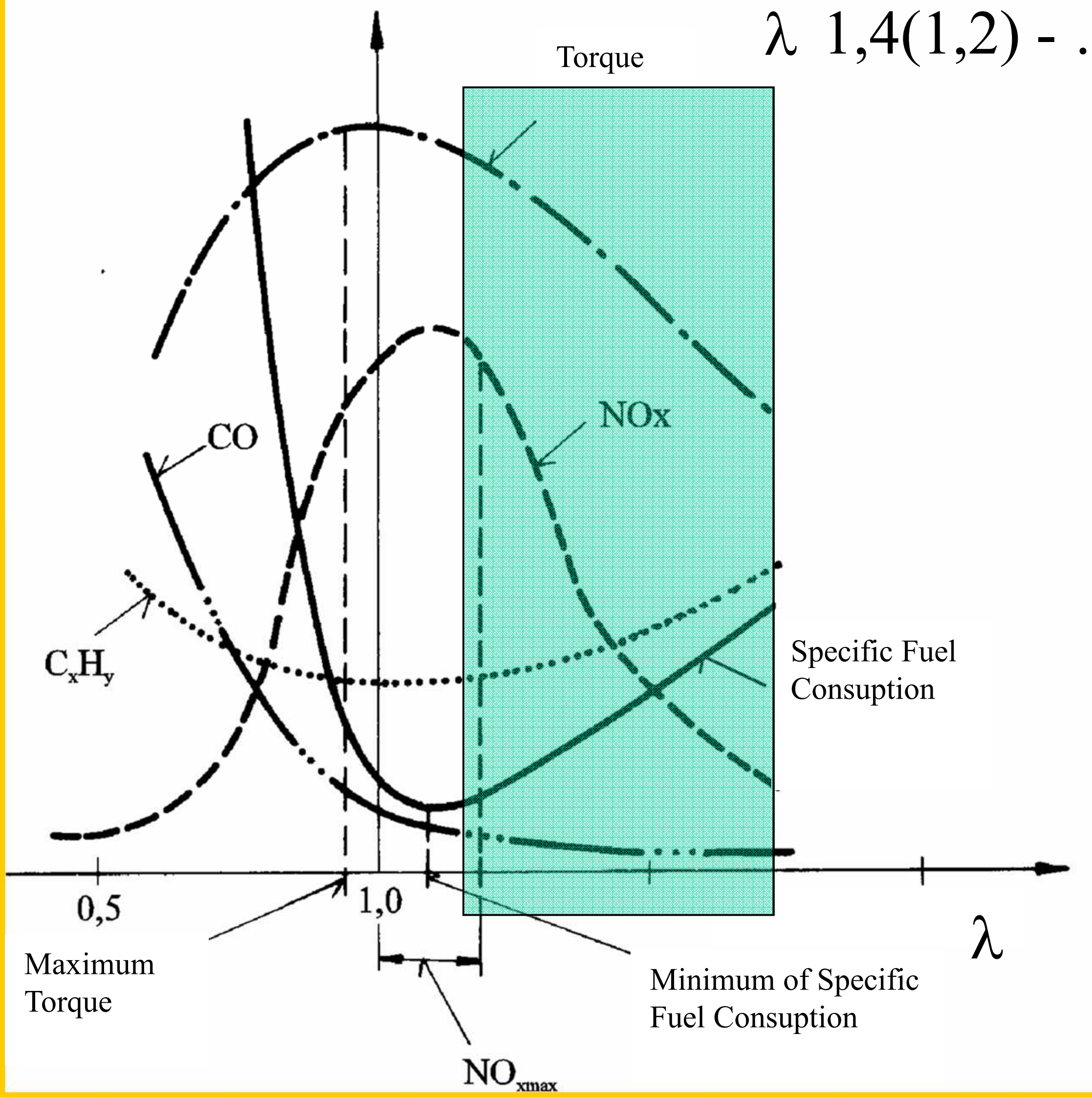
Efficiency in the function of the compression ratio





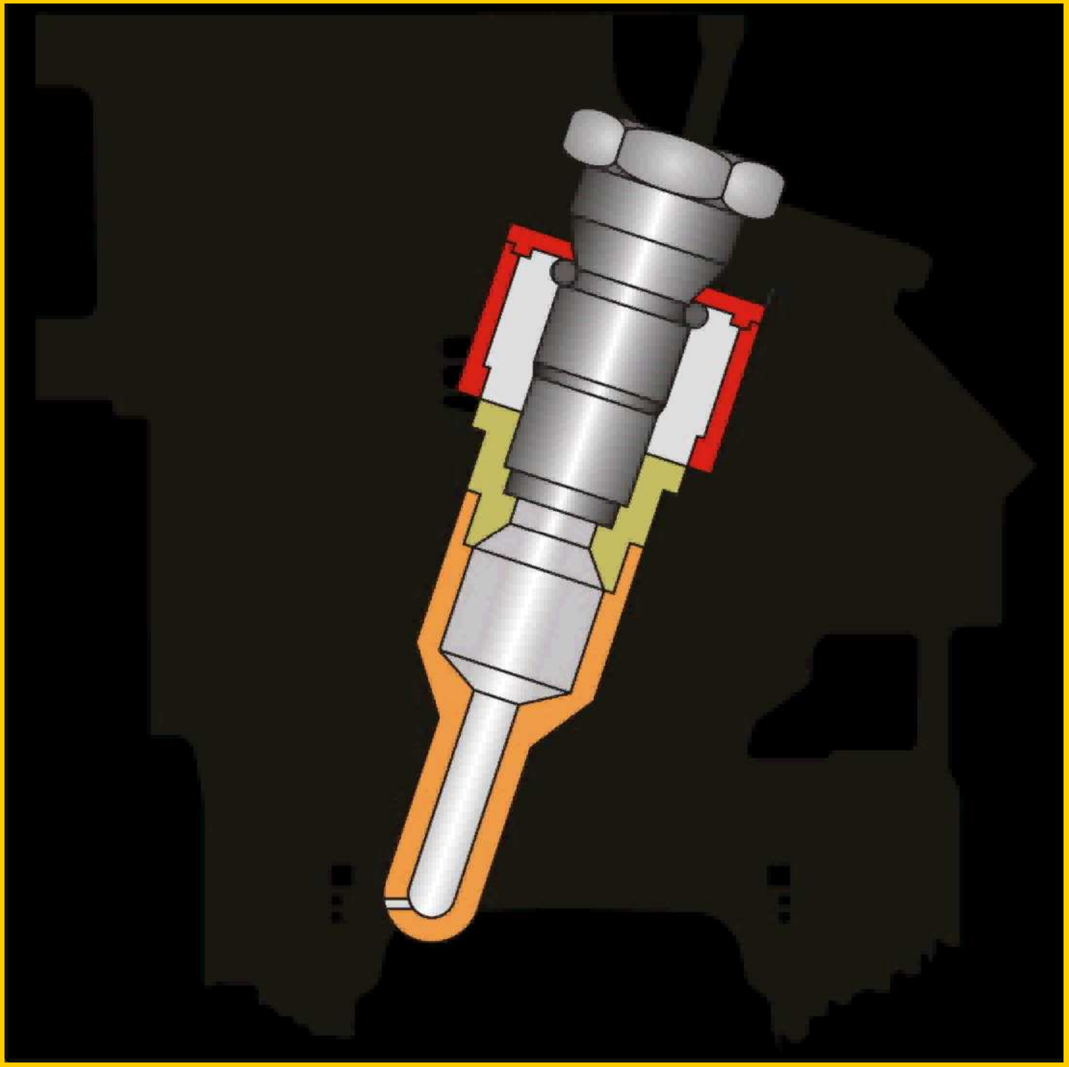
Diesel Control process (- full, --- partial load)

λ 1,4(1,2) - …..



Torque
NOx
CO
CxHy
Specific Fuel
Consuption
0,5
1,0
λ
Maximum
Torque
Minimum of Specific
Fuel Consuption
NOxmax

Combustion in a Pre-chamber

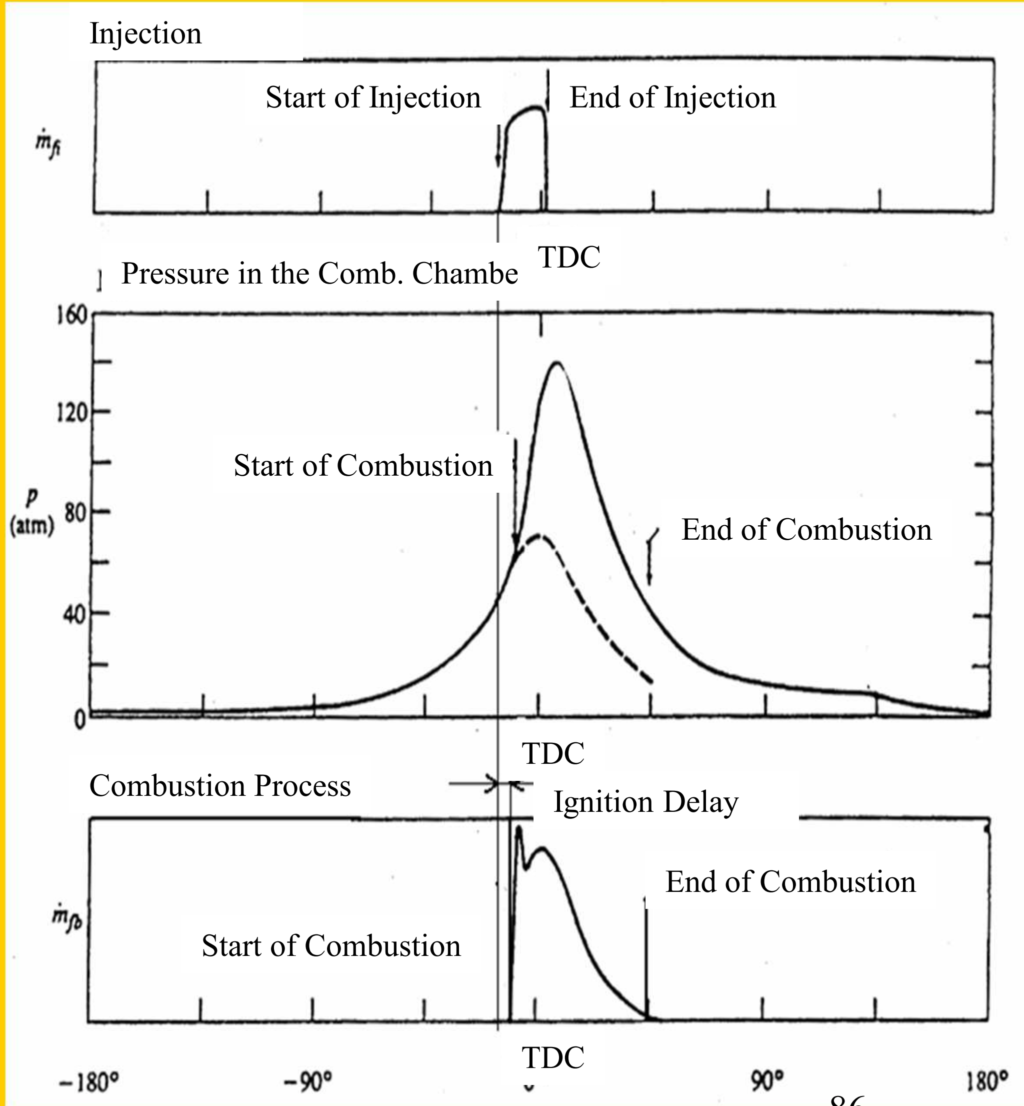


Direct Injection Combustion

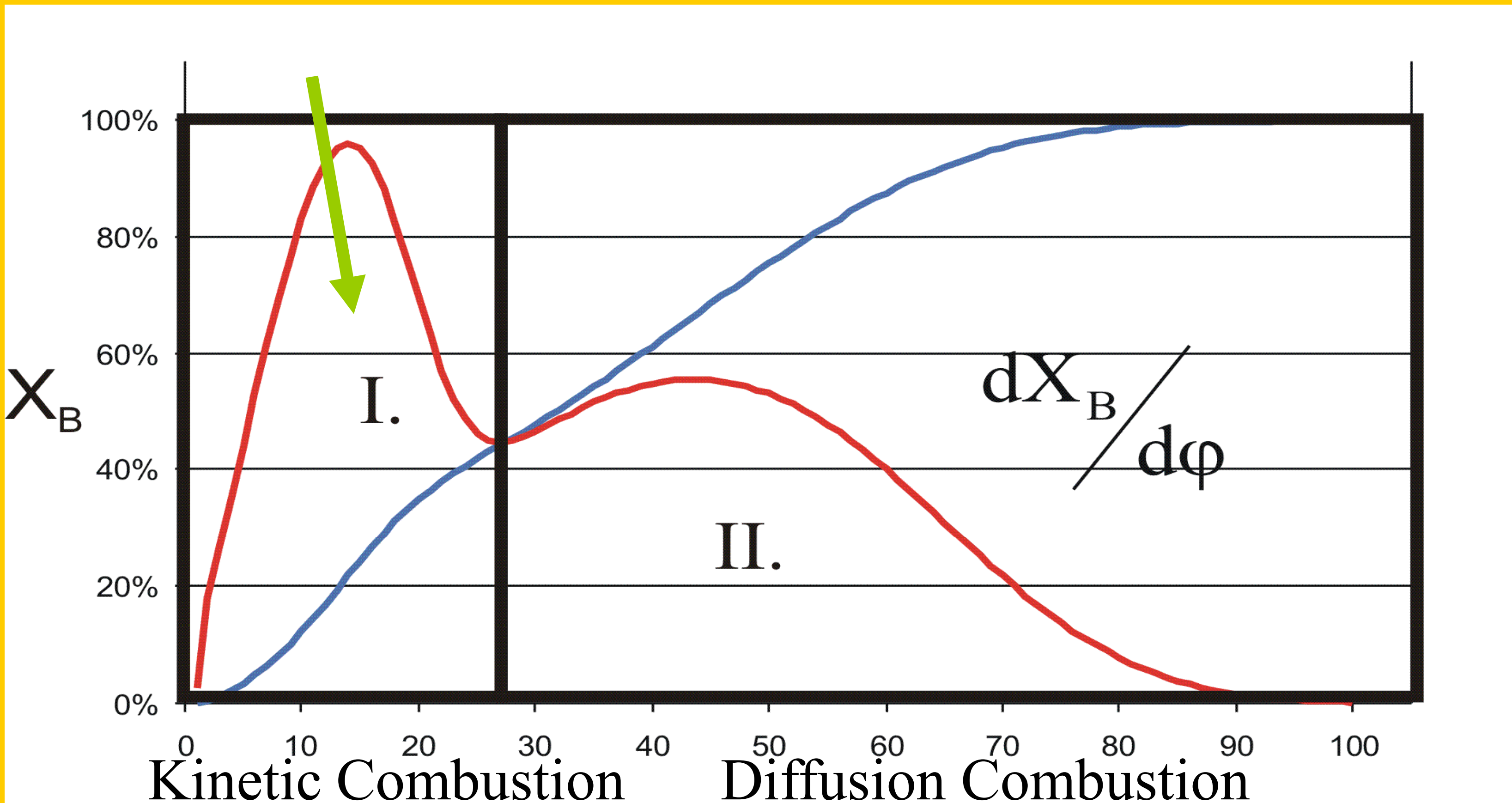
Combustion Process I

- Ignition Delay:

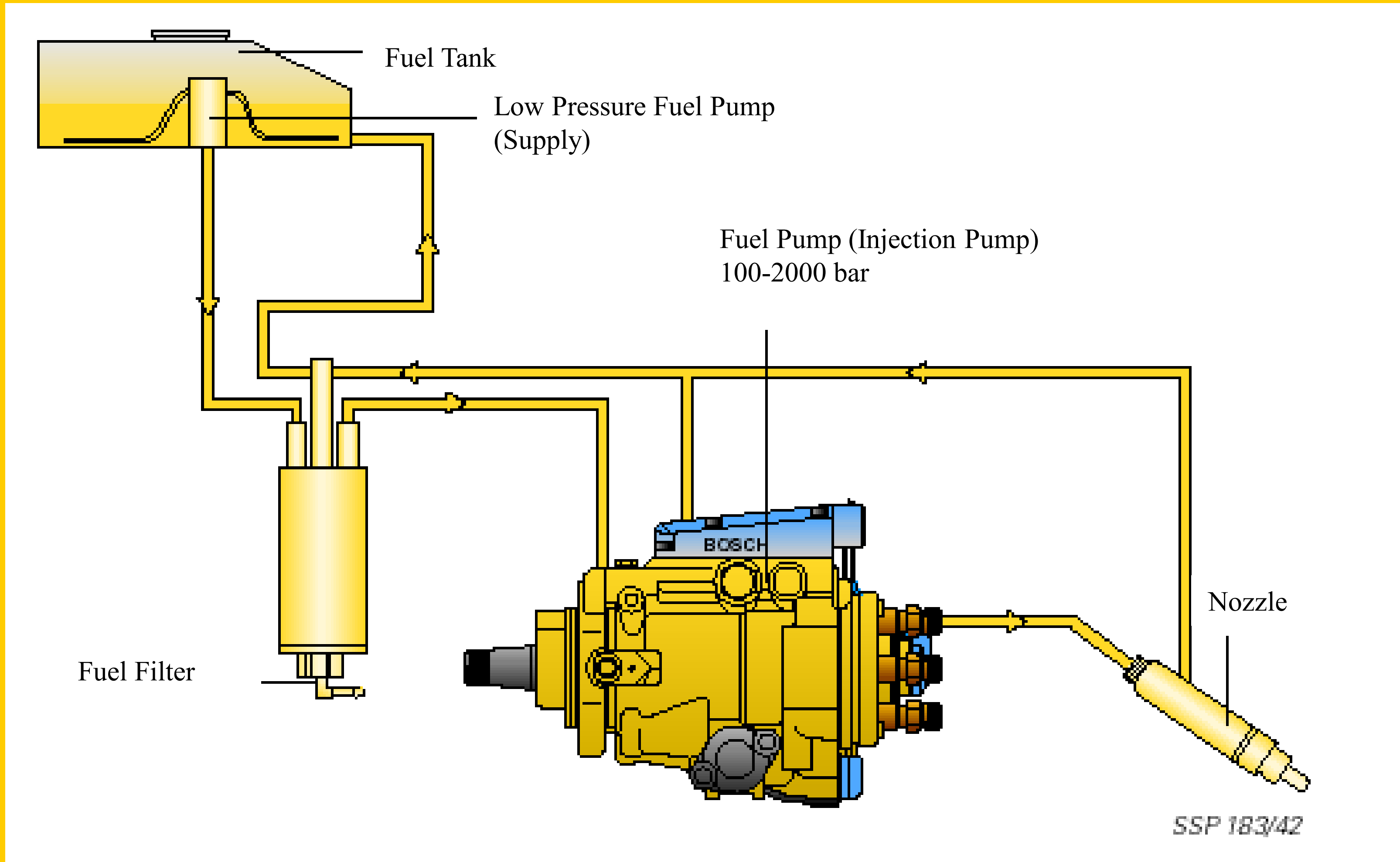
Delay between the Start of Injection and the Start of Combustion (Cetan Number)



Combustion Process II



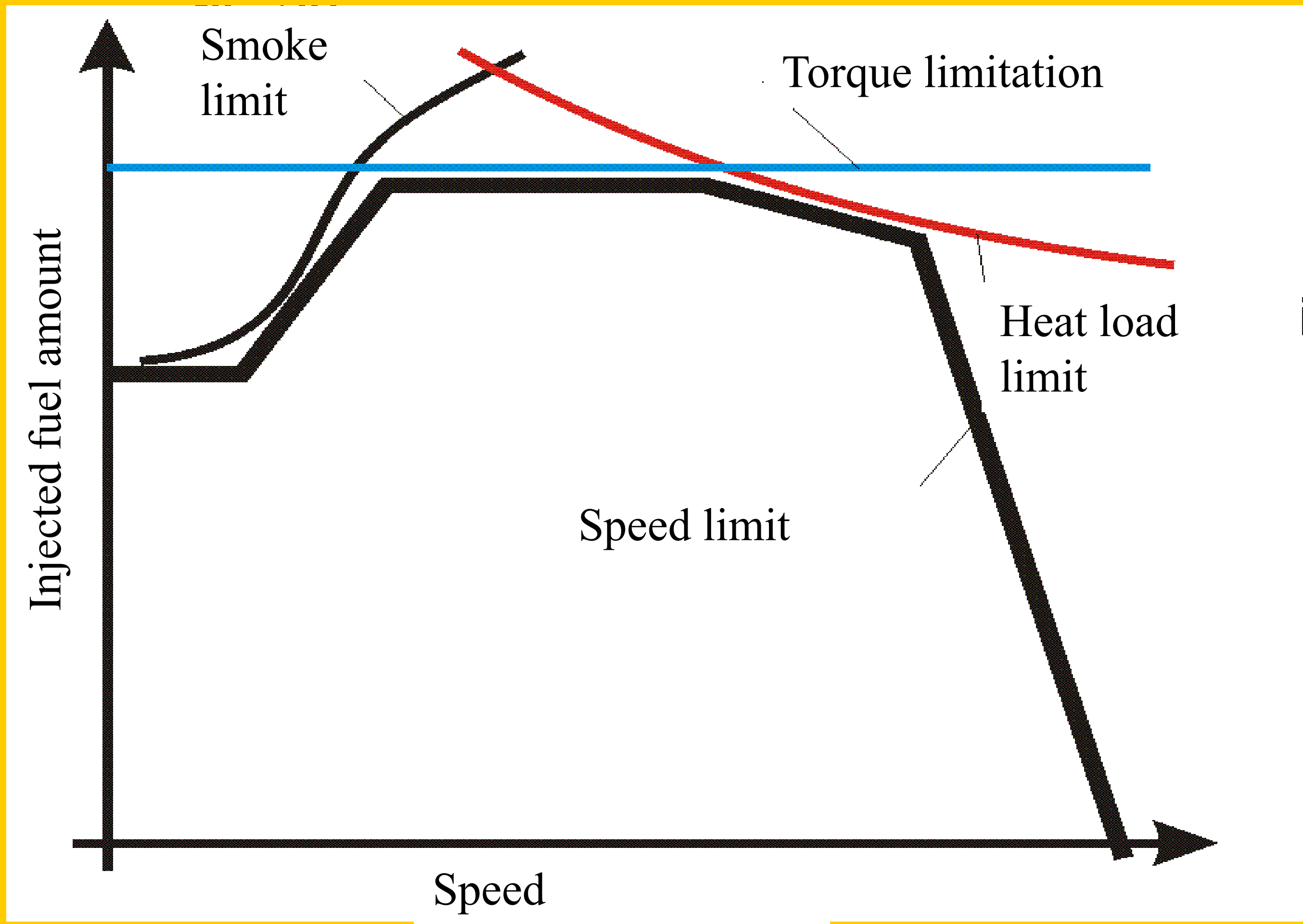
Diesel Fuel System

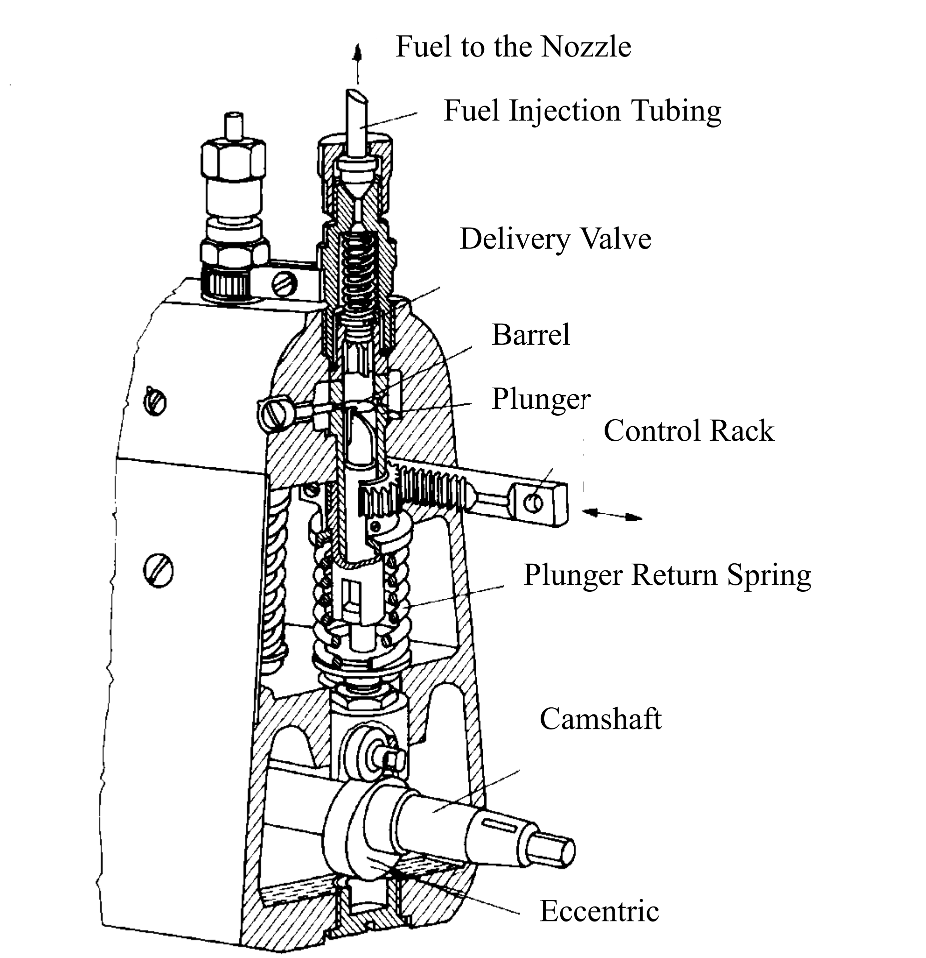


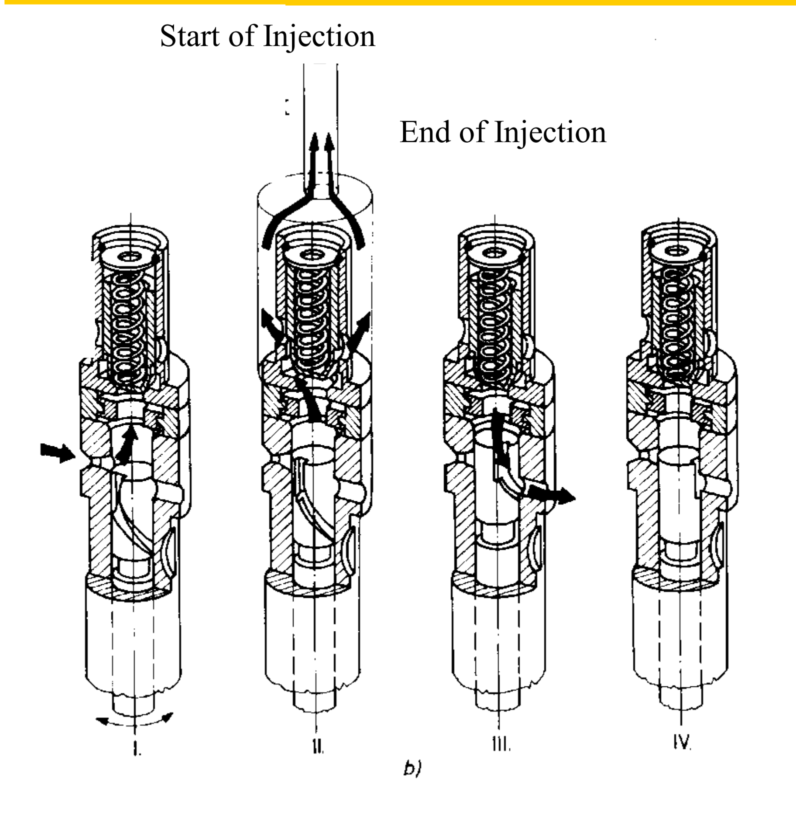
The Main Tasks of the Injector pumps:

- Supply the fuel:
  - At the right time
  - Proper pressure
  - Right propotion

Fuel Limitation



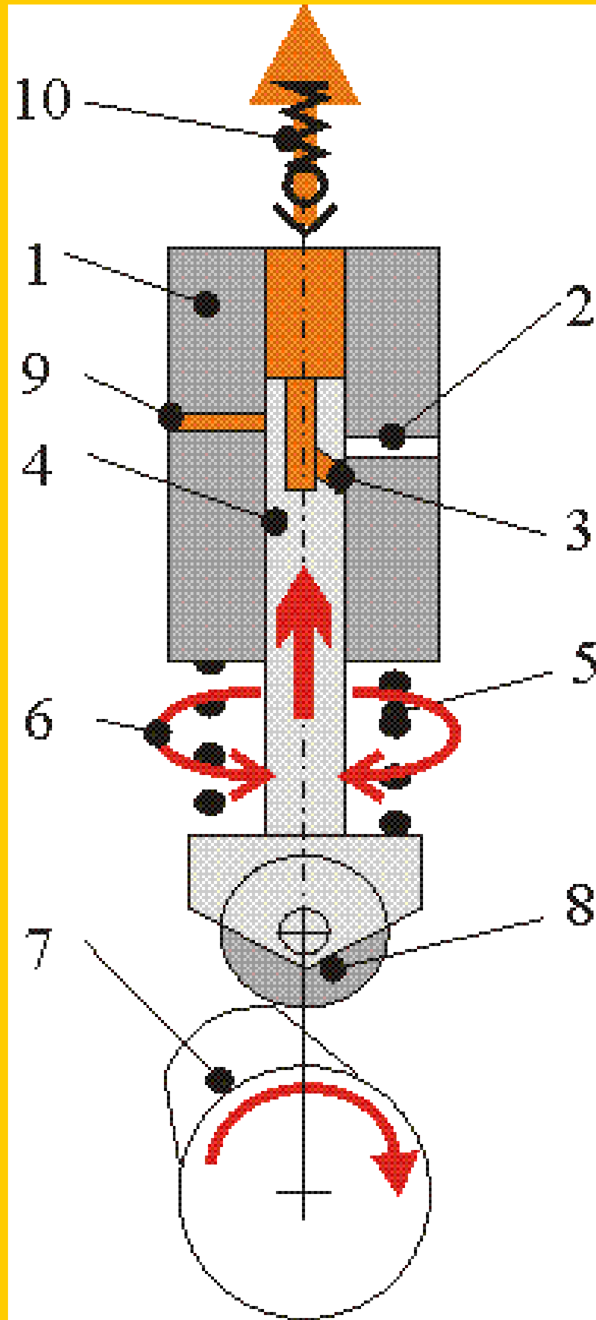




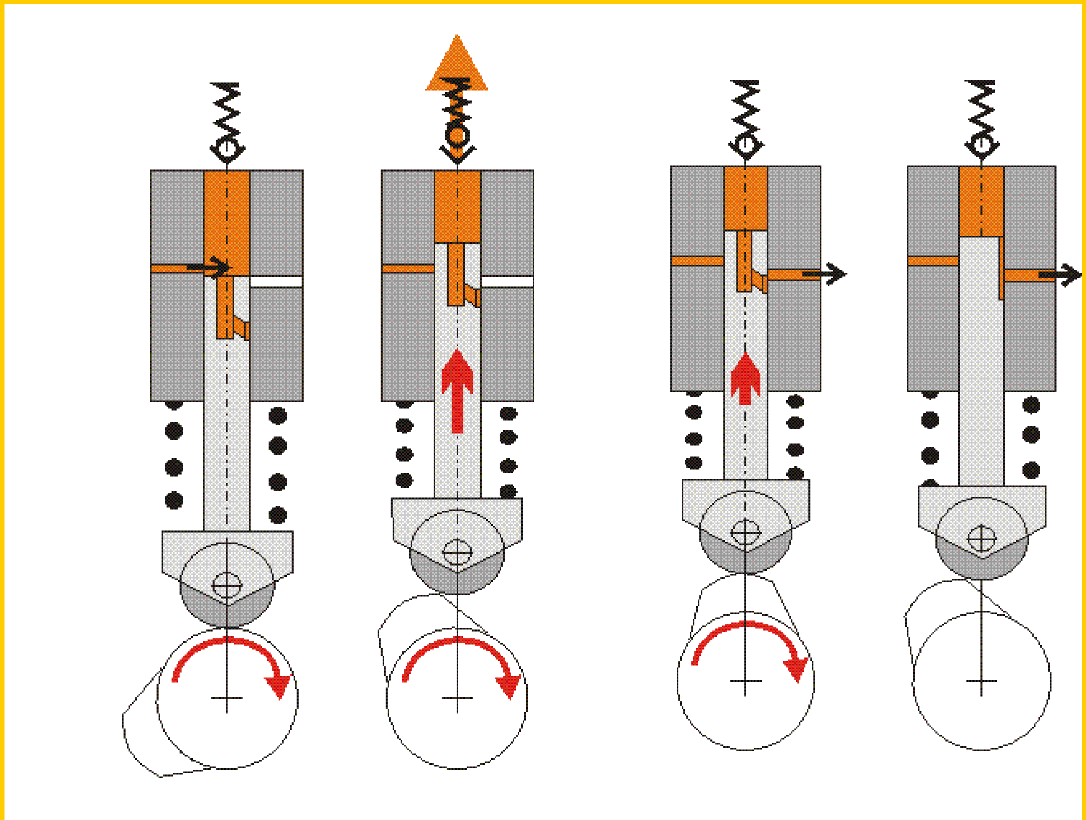
In-Line Fuel Injection Pump (Bosch)

In-Line Fuel Injection Pump

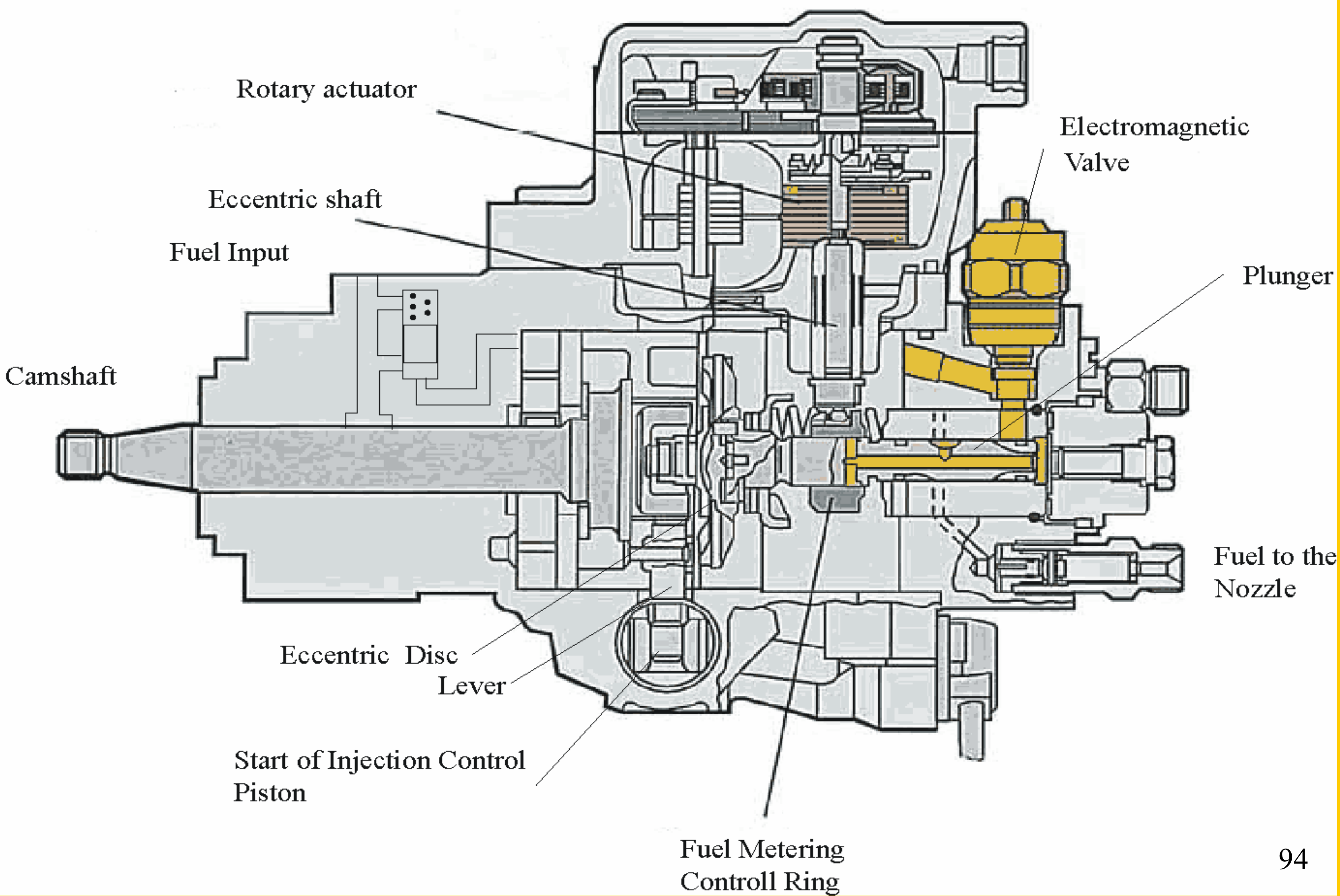
1. Injector house
2. By-pass (overflow port) channel
3. Helical groove
4. Barrel
5. Plunger Return Spring
6. fuel amount controller
7. Eccentric
8. Roller
9. Filling channel
10. Head Valve



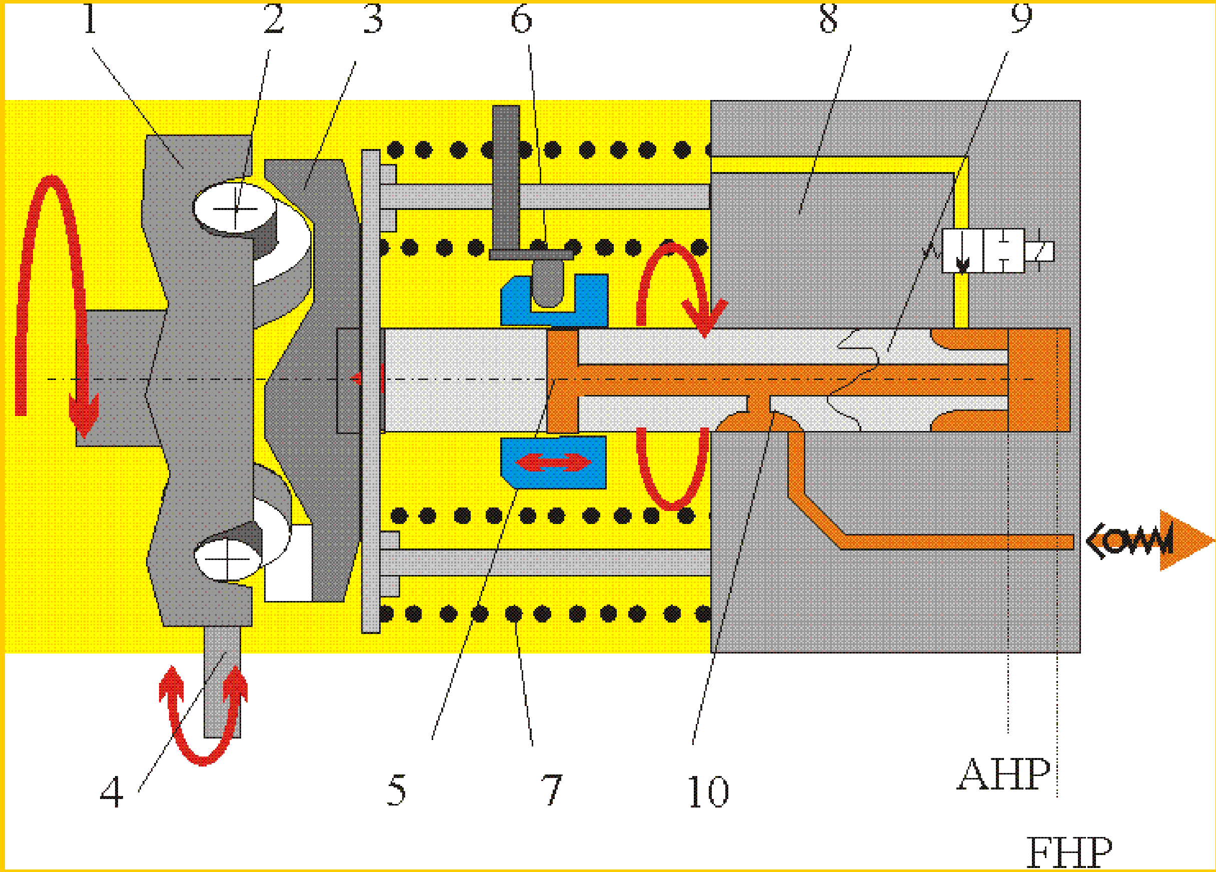
Filling Delivery End of Delivery Idle



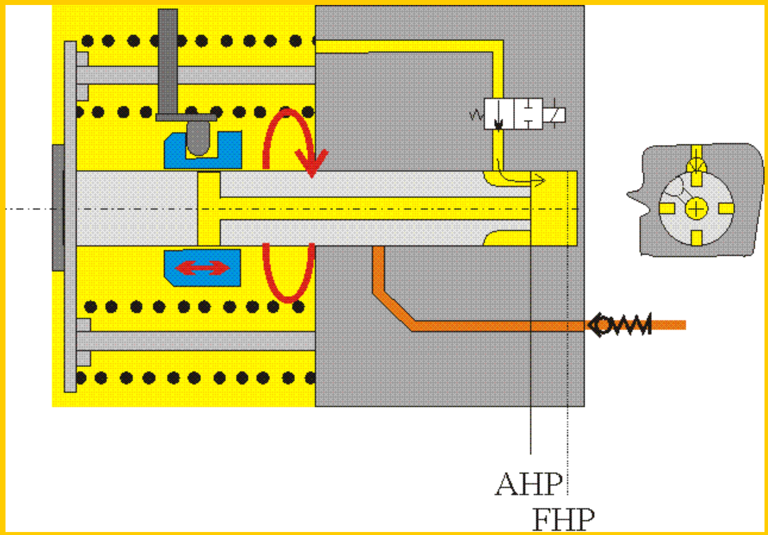
BOSCH VE type Distributor type Injection Pump (radial-piston)

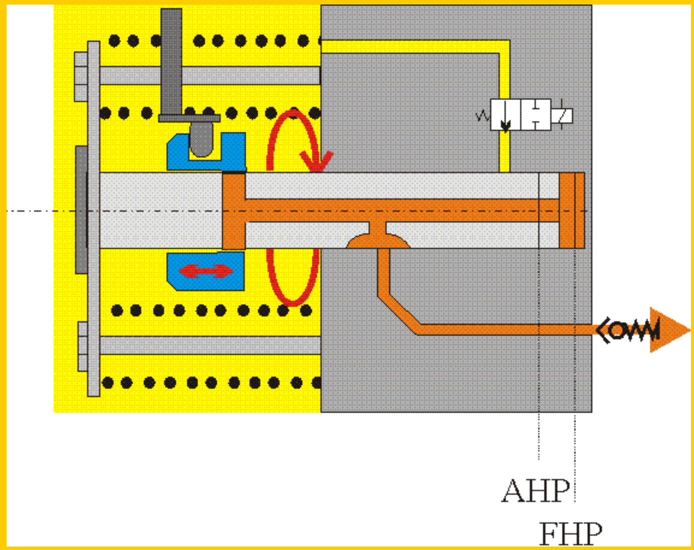


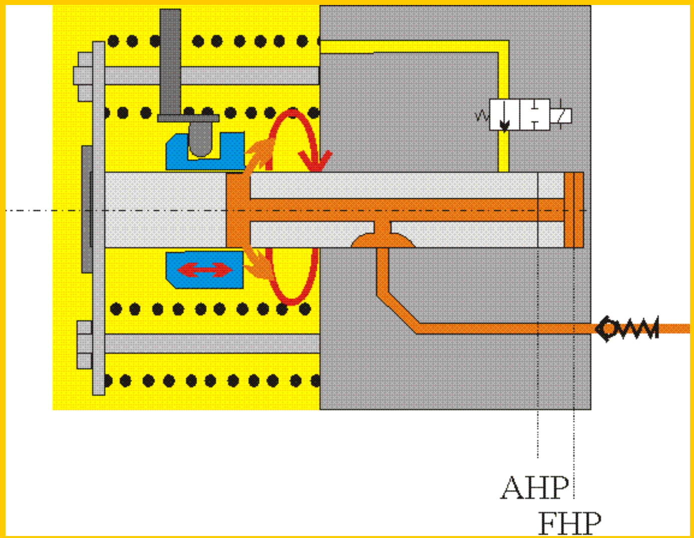
Distributor-type Injector Pump



1. Castor house
2. Castor
3. Eccentric Disc
4. Timing Device
5. By-pass Chanel
6. Fuel Collar (Metering Ring)
7. Spring
8. Shutoff Valve
9. Distributor Plunger
10. Selector Chanel



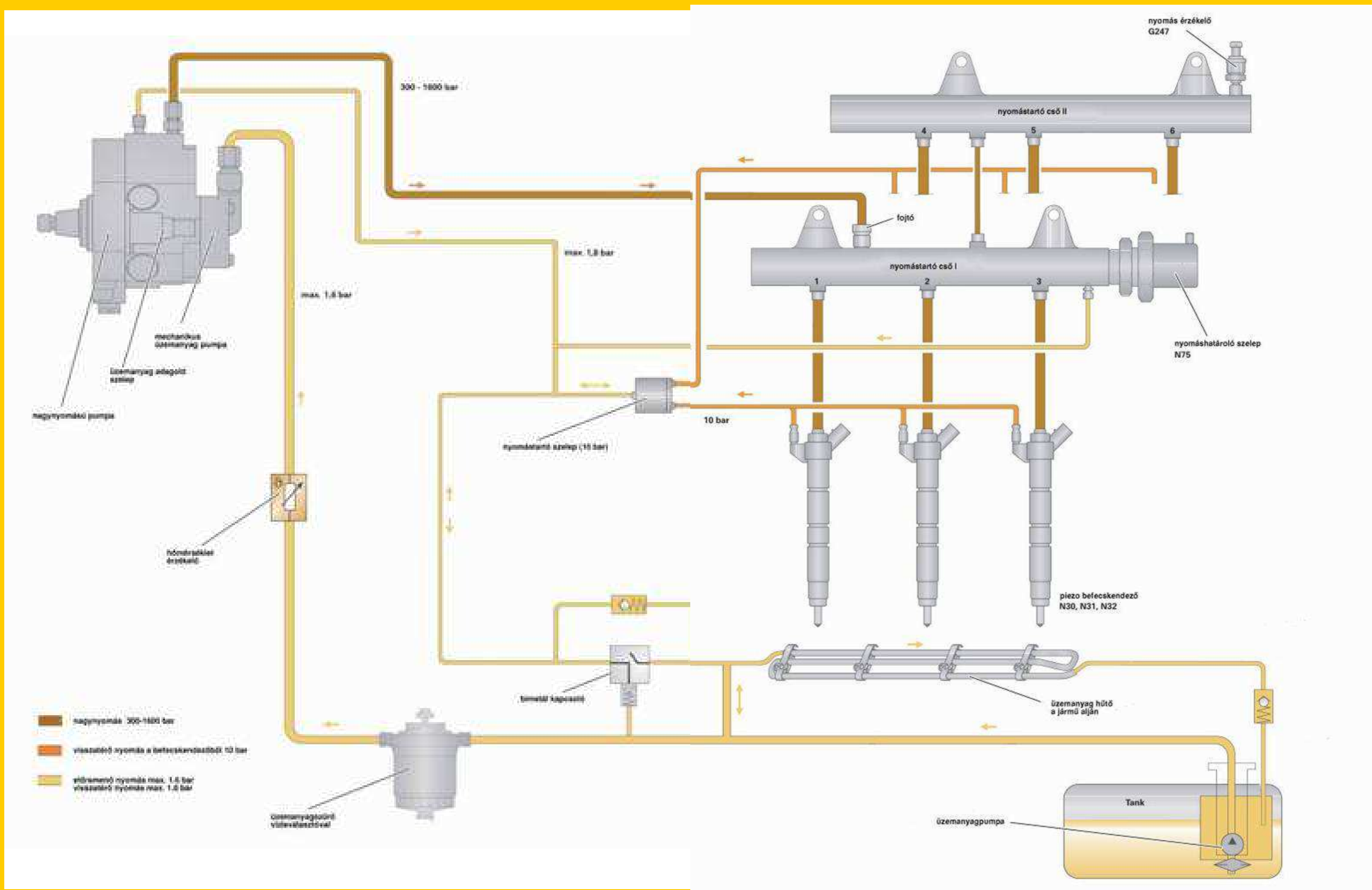




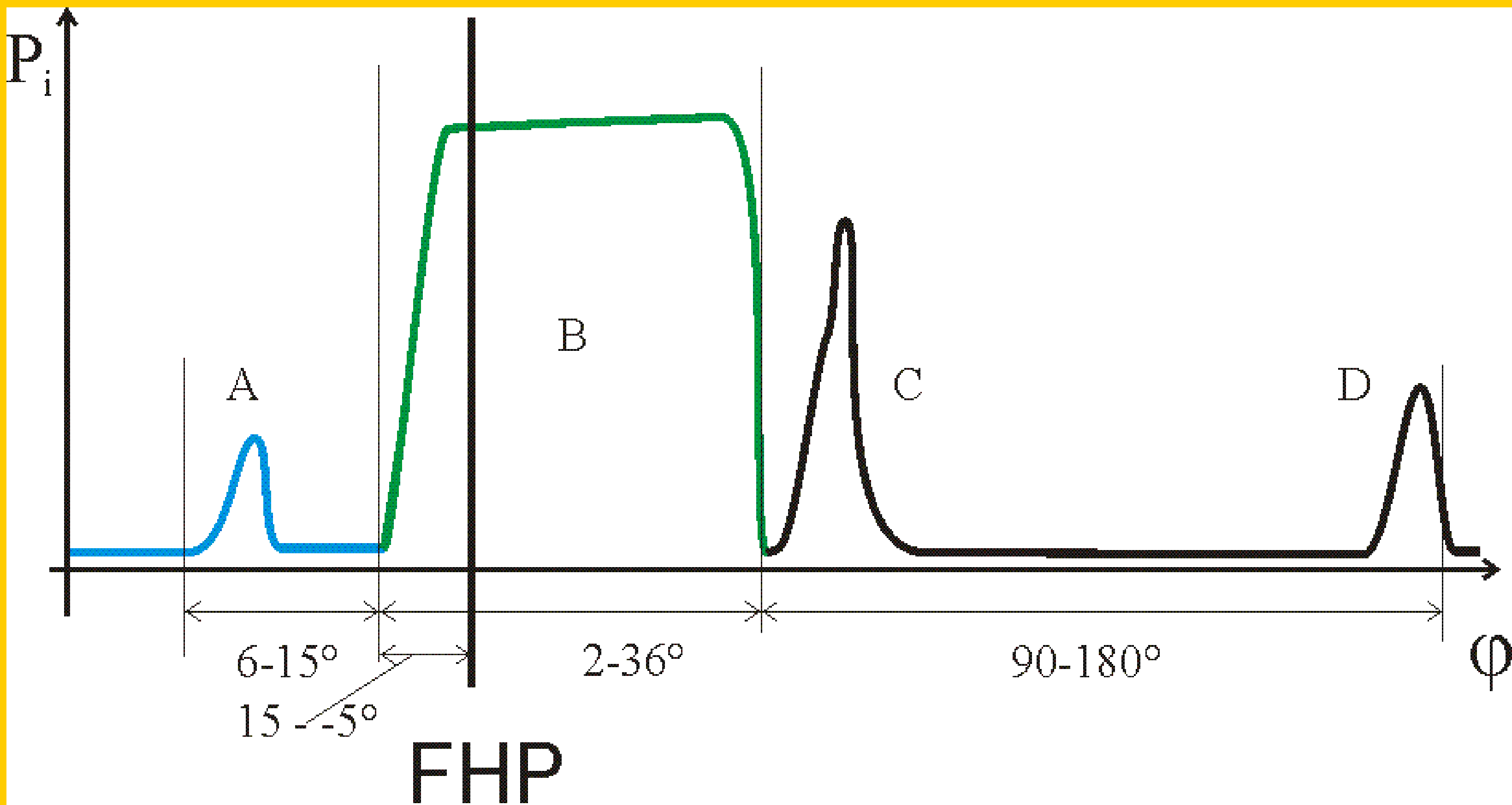
AHP- BDC
FHP- TDC

Common Rail System

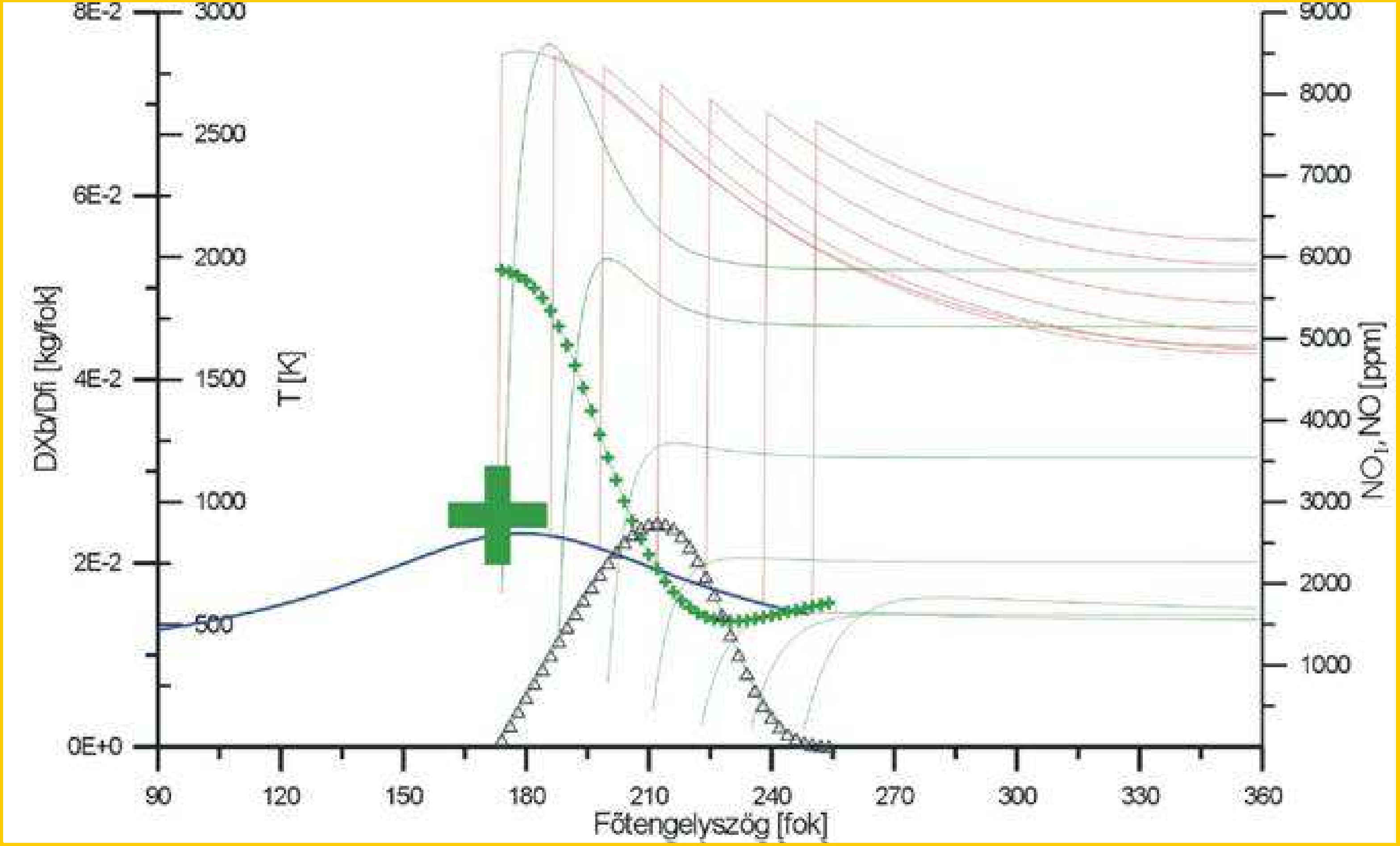
P= 300 –1800 bar



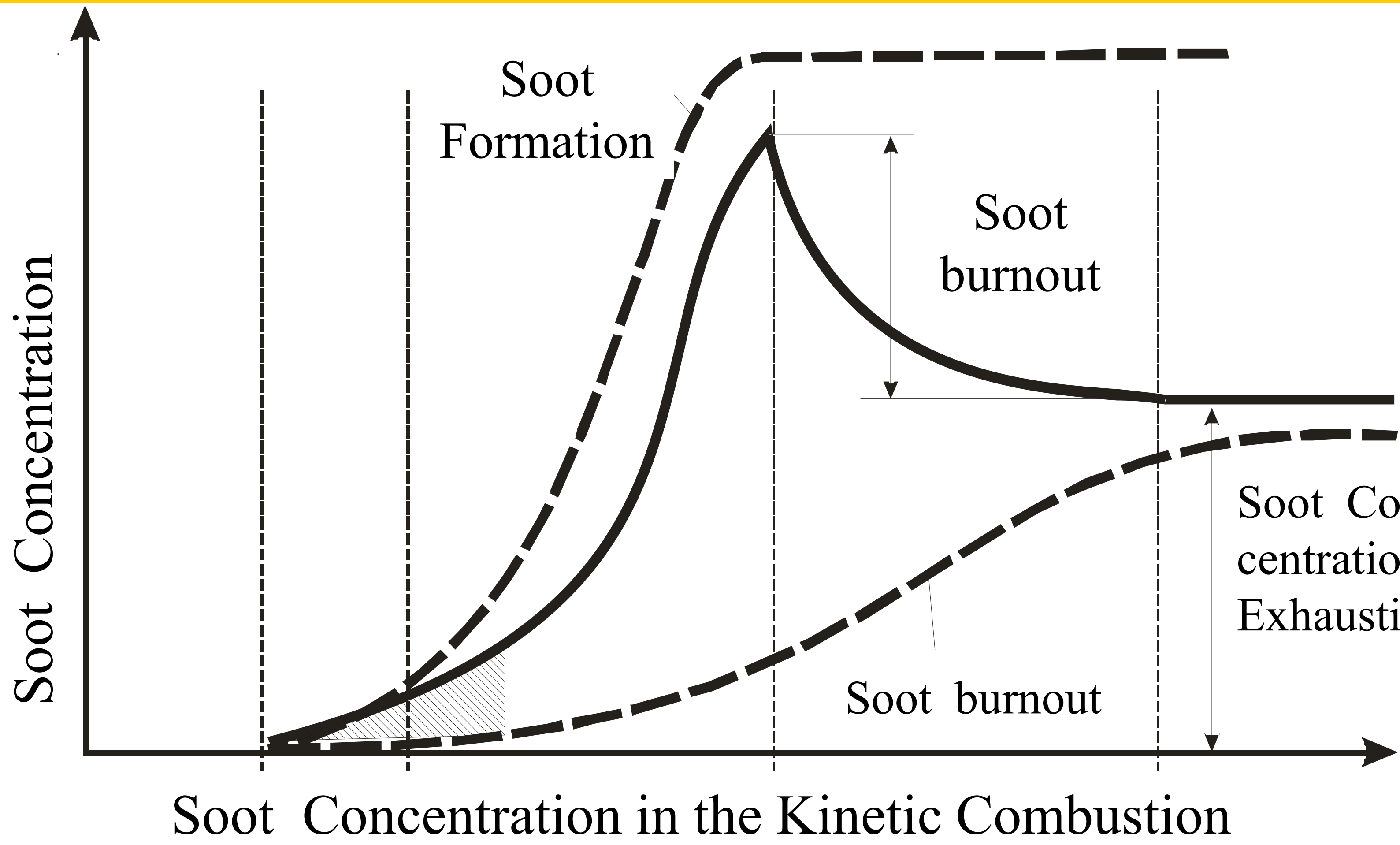
Injection Profiles (CR)



NO formation



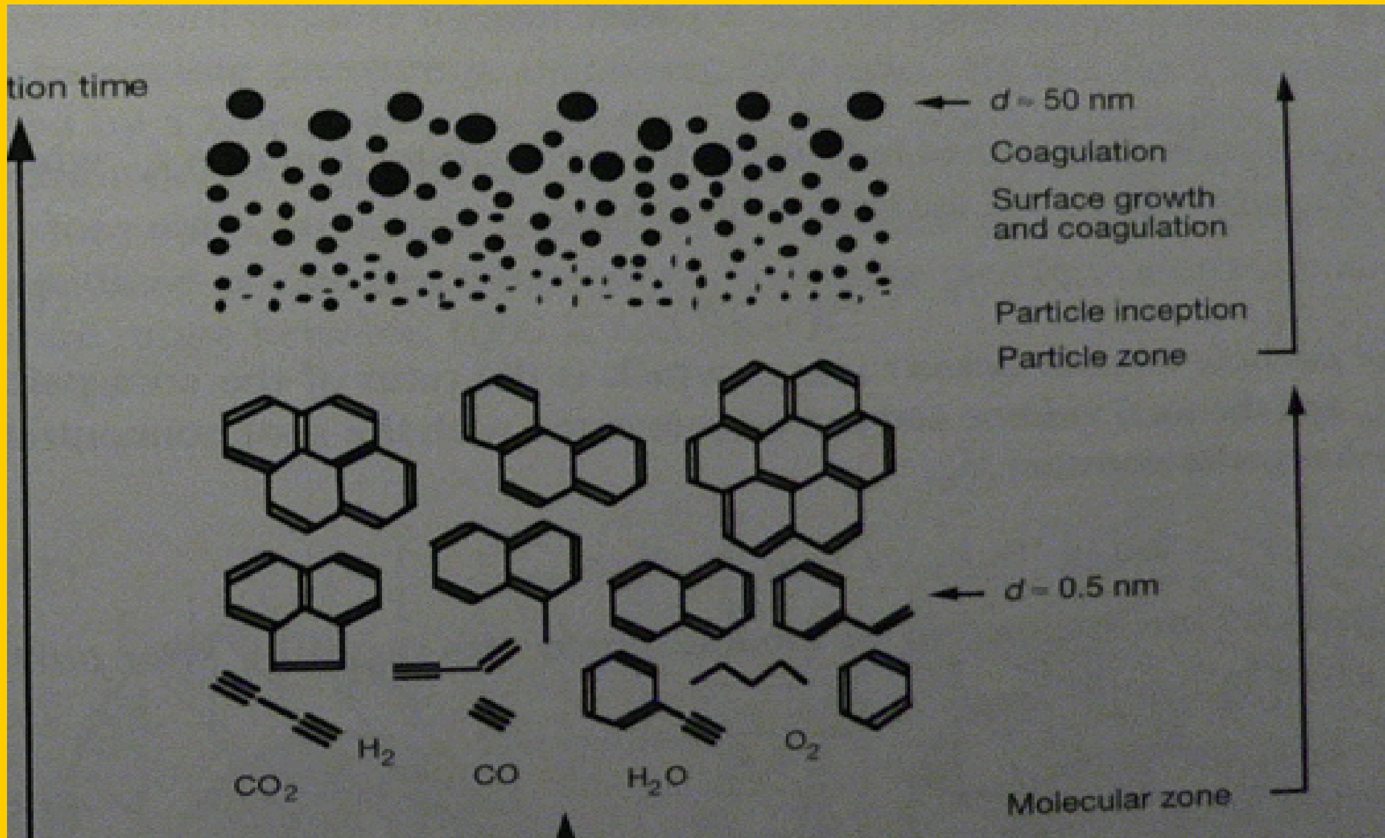
Soot Formation and Burnout



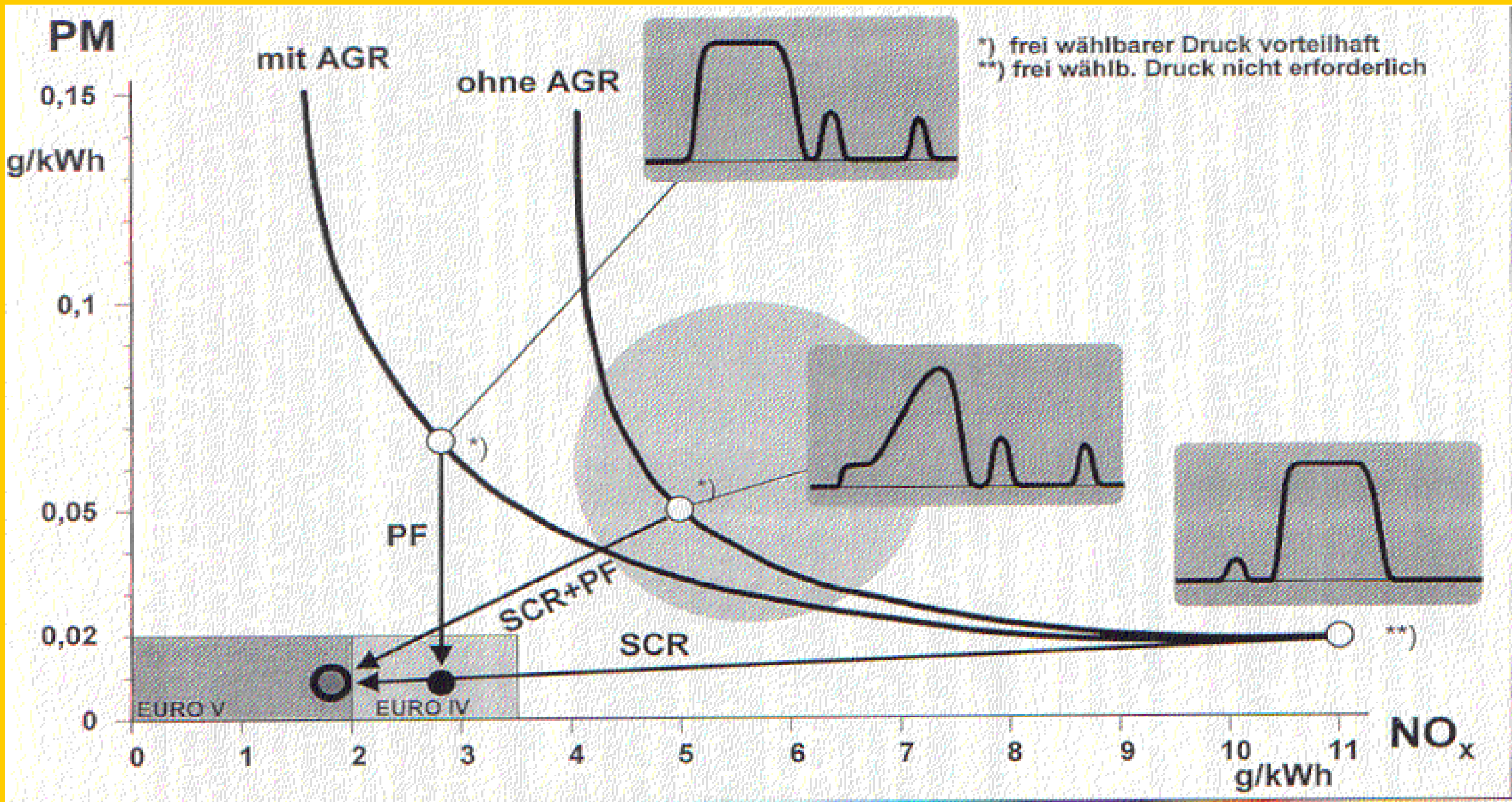
PM formation

Polymerization

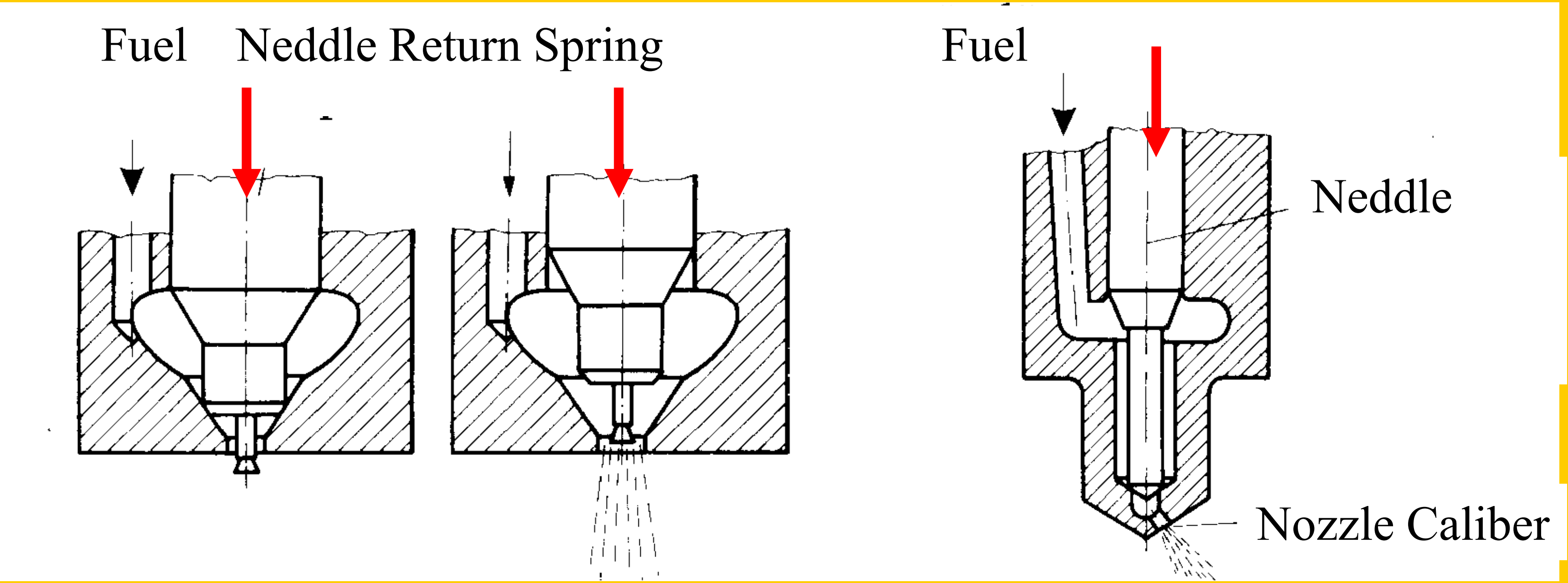
Dehydrogenization
(pirolyzis)



Bockhorn, H. E.: 1994, Soot Formation in Combustion, Springer-Verlag.


PM
0,15
g/kWh
0,1
0,05
0,02
0
mit AGR
ohne AGR
*) frei wählbarer Druck vorteilhaft
**) frei wählb. Druck nicht erforderlich
*)
*)
**)
PF
SCR+PF
SCR
EURO V
EURO IV
0
1
2
3
4
5
6
7
8
9
10
11
g/kWh
NOx

Open Type Injector　　　　Closed Type Injector



The Main Task of the Nozzels:

Provide homogeneous Fuel Jet
Guaranty Satisfactory Droplet Size
Promote Fuel Jet Development
Avoid of Black Flow
………

Air-fuel mixing methods

- Internal (CIE, GDI (SIE))
- External(SIE)

Combustion chamber design

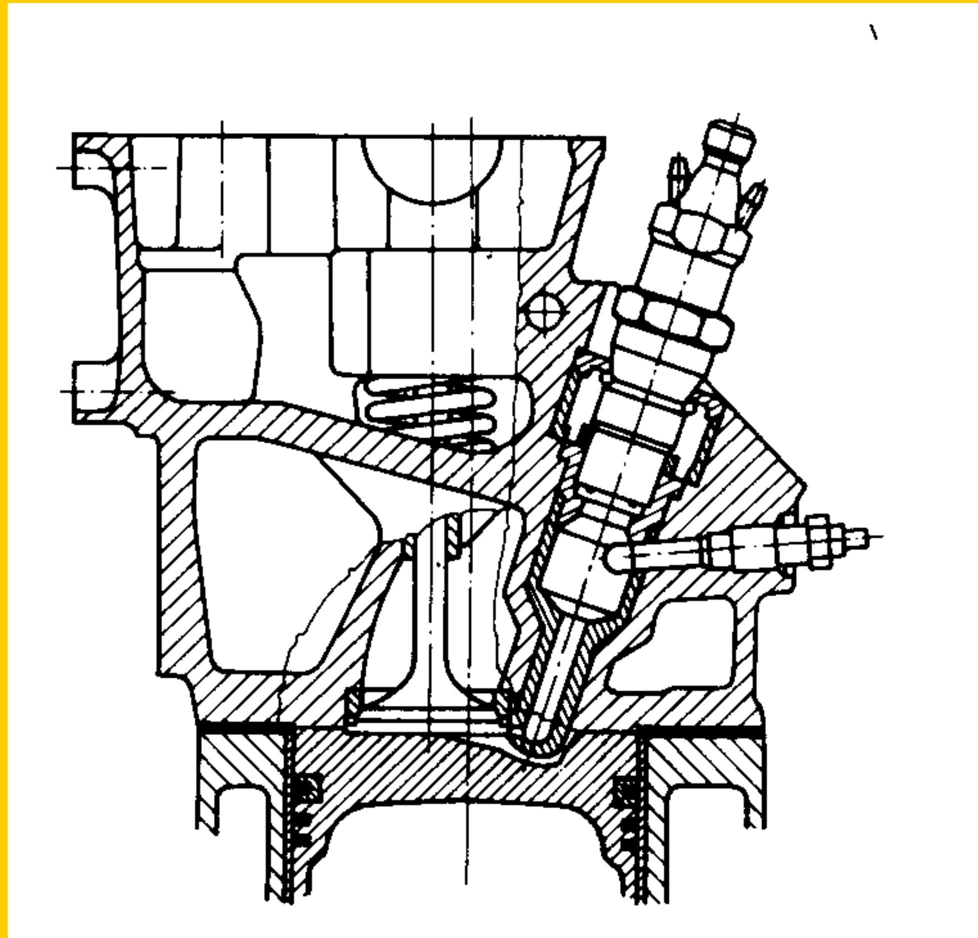
- single open combustion chamber
- divided combustion chamber
-
  - swirl chamber systems
  - prechamber systems

Start of Combustion

- External energy (Spark)
- Compression
- Hot Spot

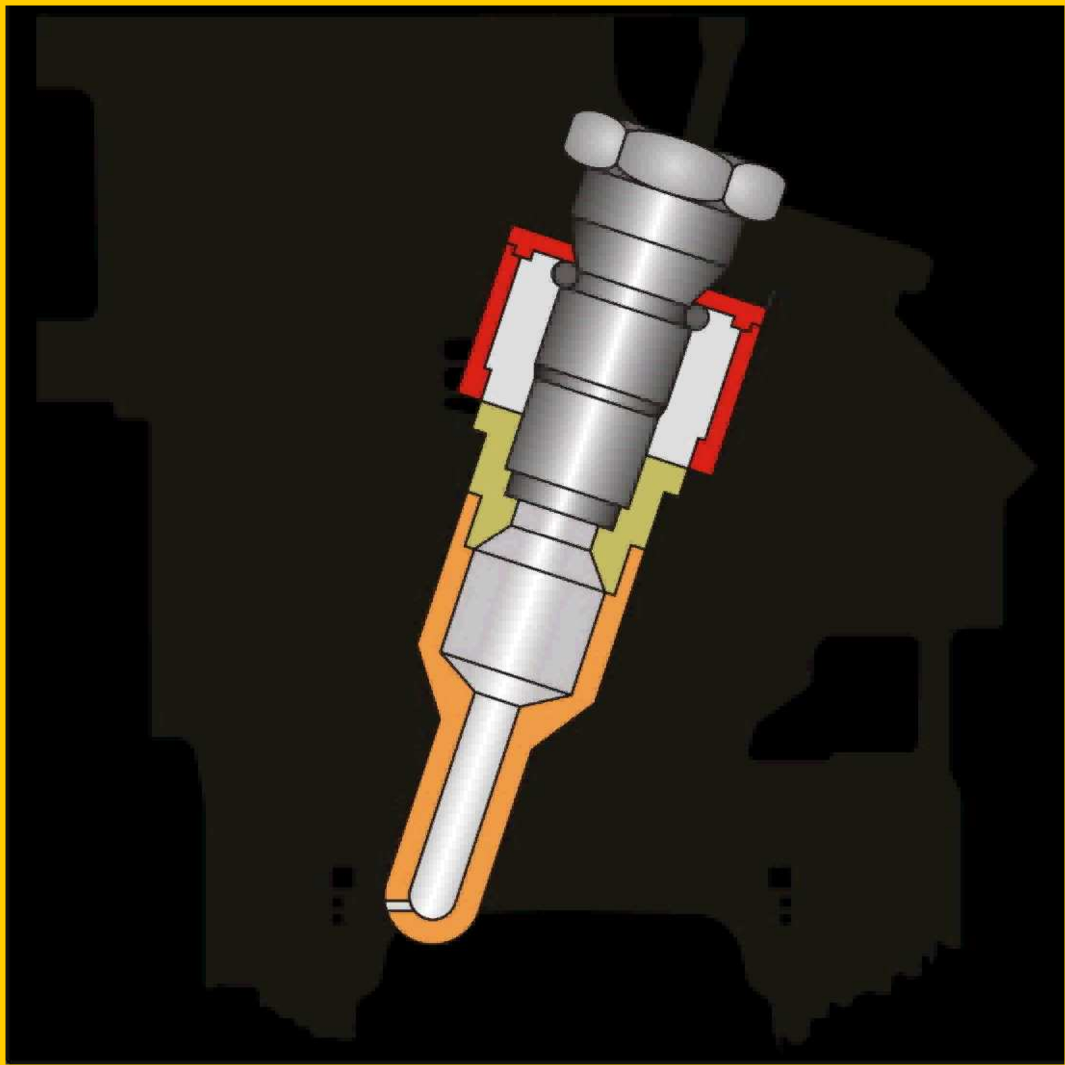
Control methods

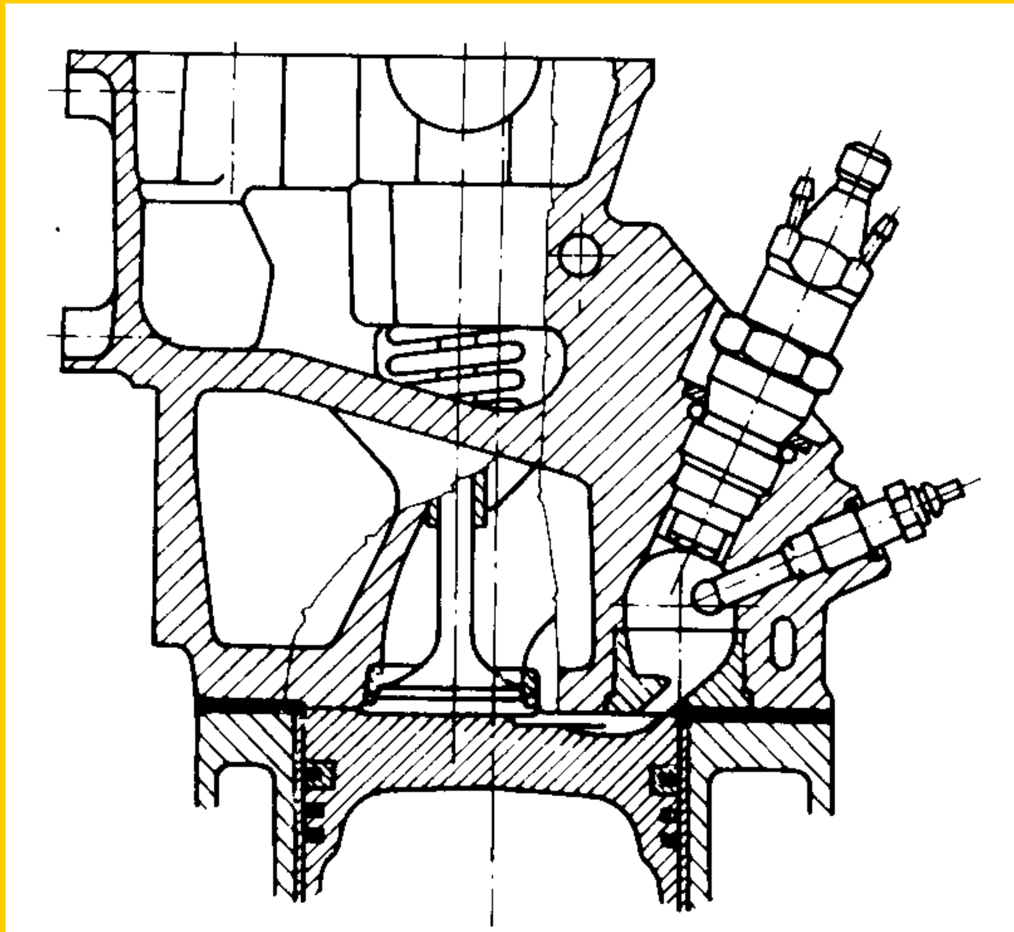
- qualitative (SIE)
- quantitative (CIE, GDI (SIE))



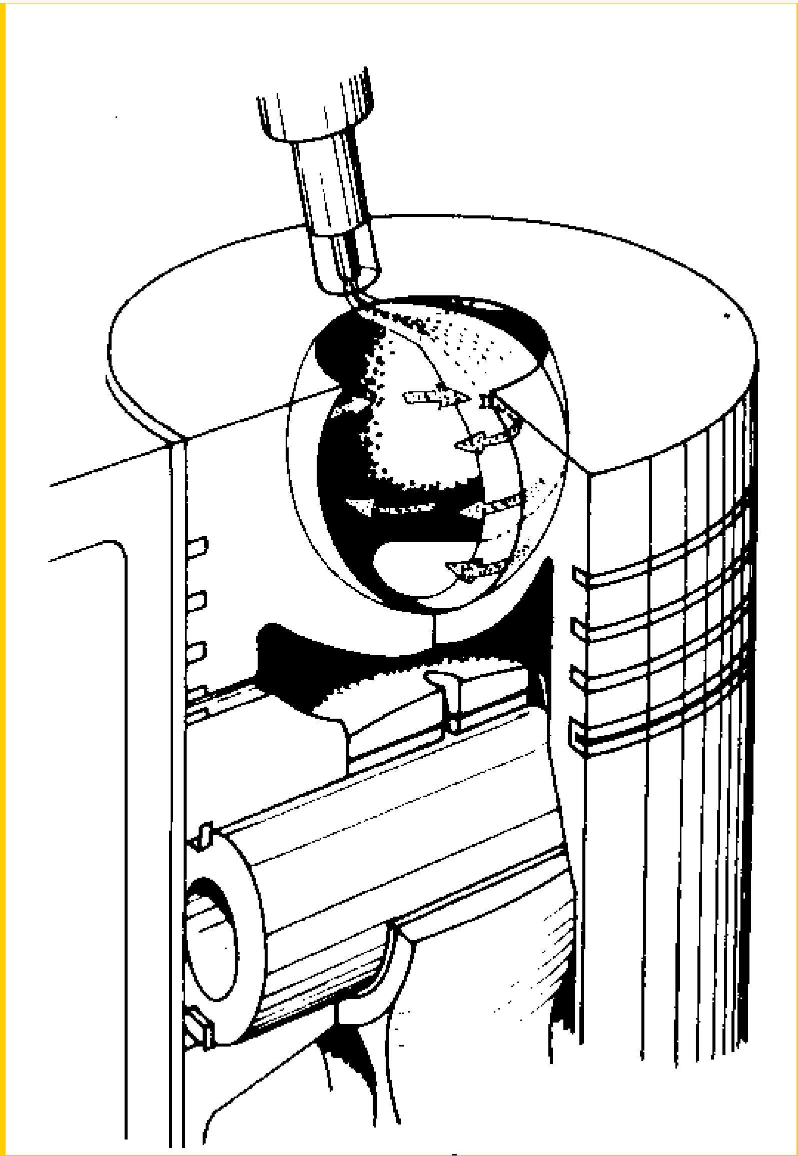
Pre-chamber type C.C.

Combustion in a Pre-chamber

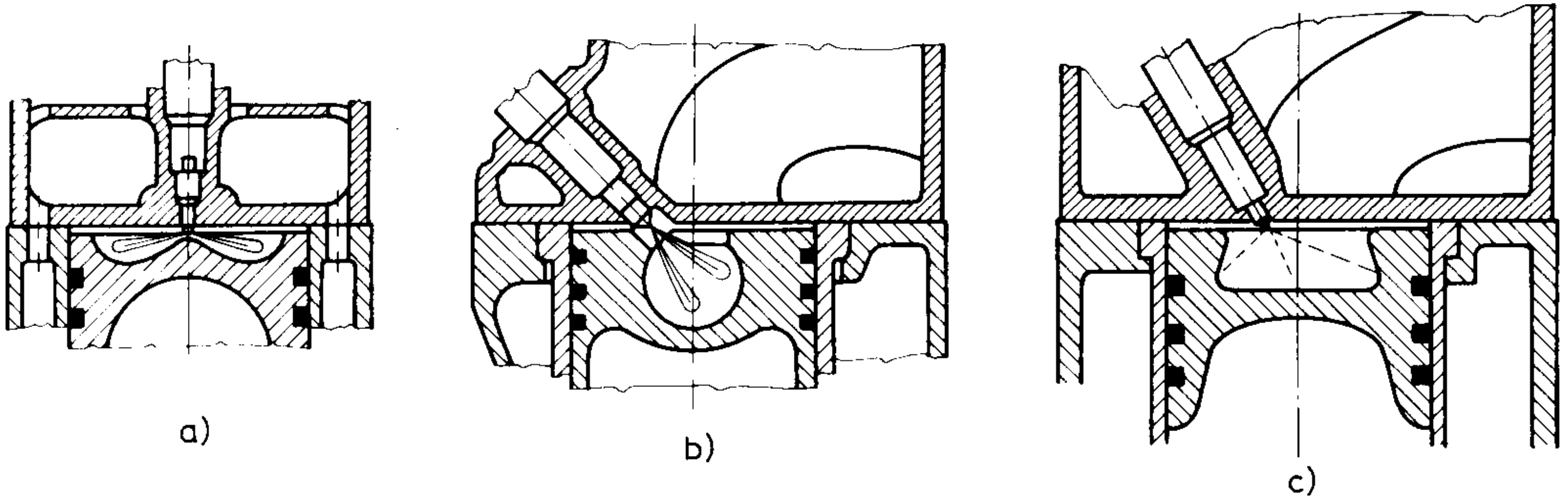




Swirl Chamber type C.C.



Piston Pre-chamber type C.C. (M)



Different Open C.C. Designs (DI)

Air-fuel mixing methods

- o Internal (CIE, GDI (SIE))
- o External(SIE)

Combustion chamber design

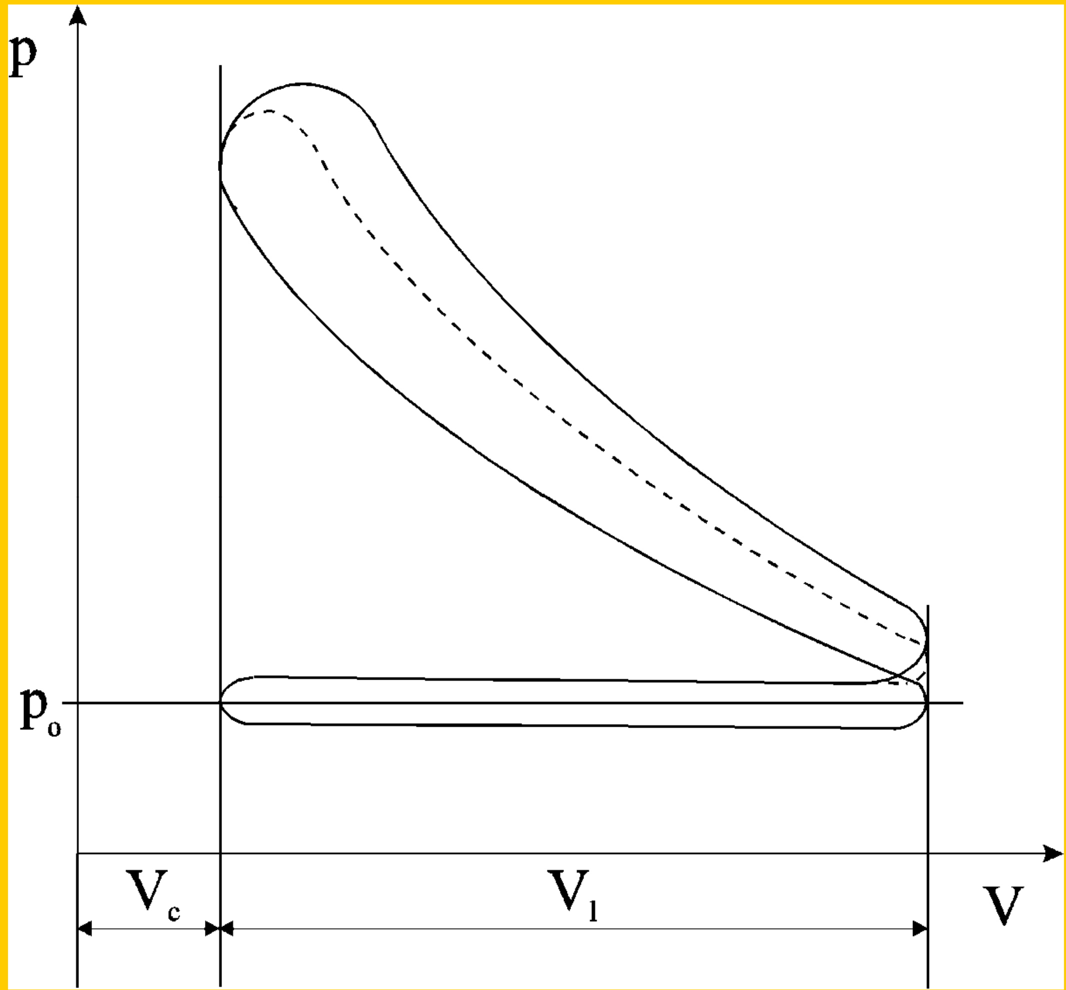
- single open combustion chamber
- divided combustion chamber
  - o swirl chamber systems
  - o prechamber systems

Start of Combustion

- External energy (Spark)
- Compression
- Hot Spot

Control methods

- qualitative (SIE)
- quantitative (CIE, GDI (SIE))



Control of the Diesel cycle (- full load, --- partial load)

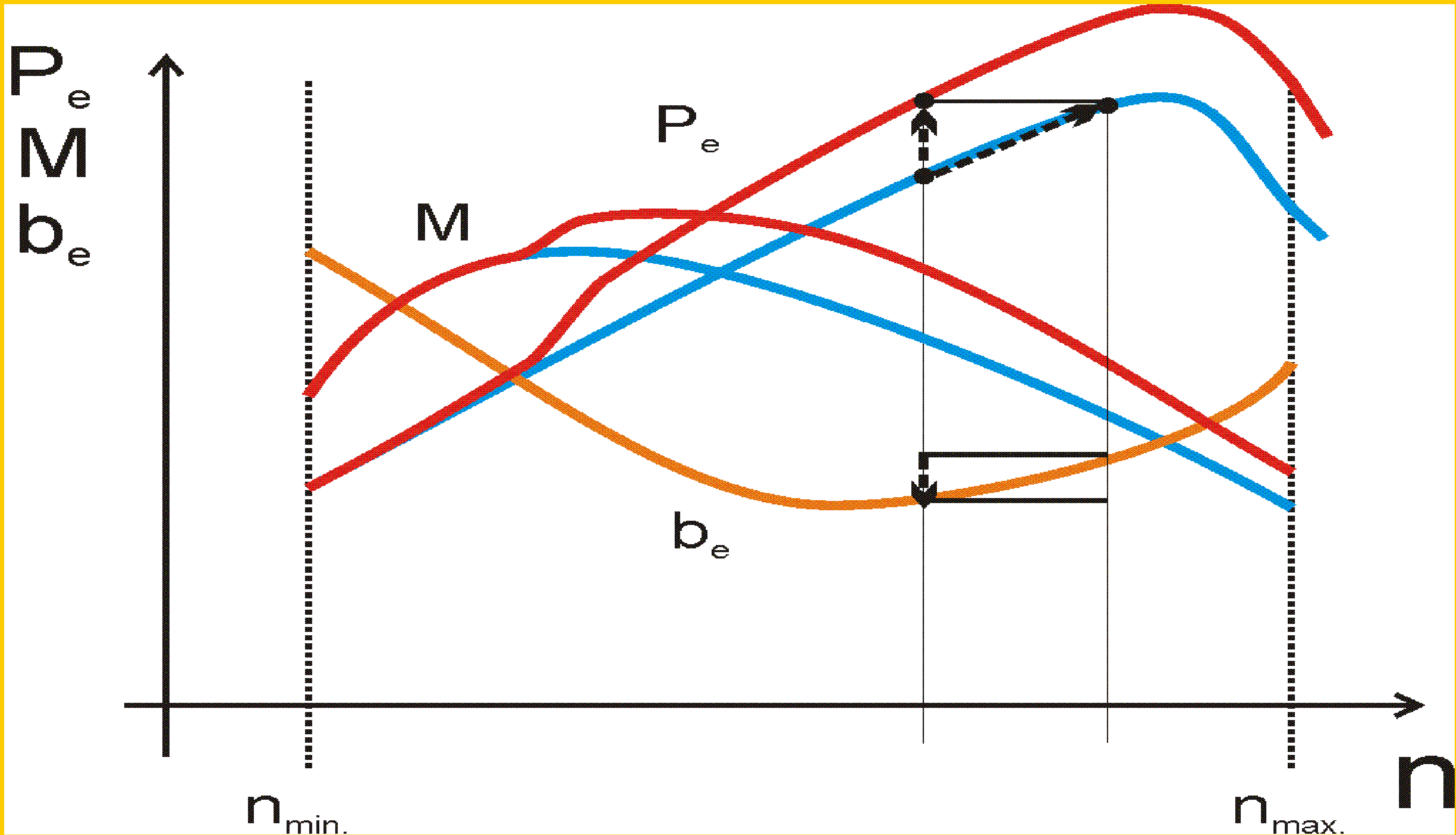
Charging systems

– Naturally aspirated
– Mechanically charged
– Turbo charged
– Acoustical charged

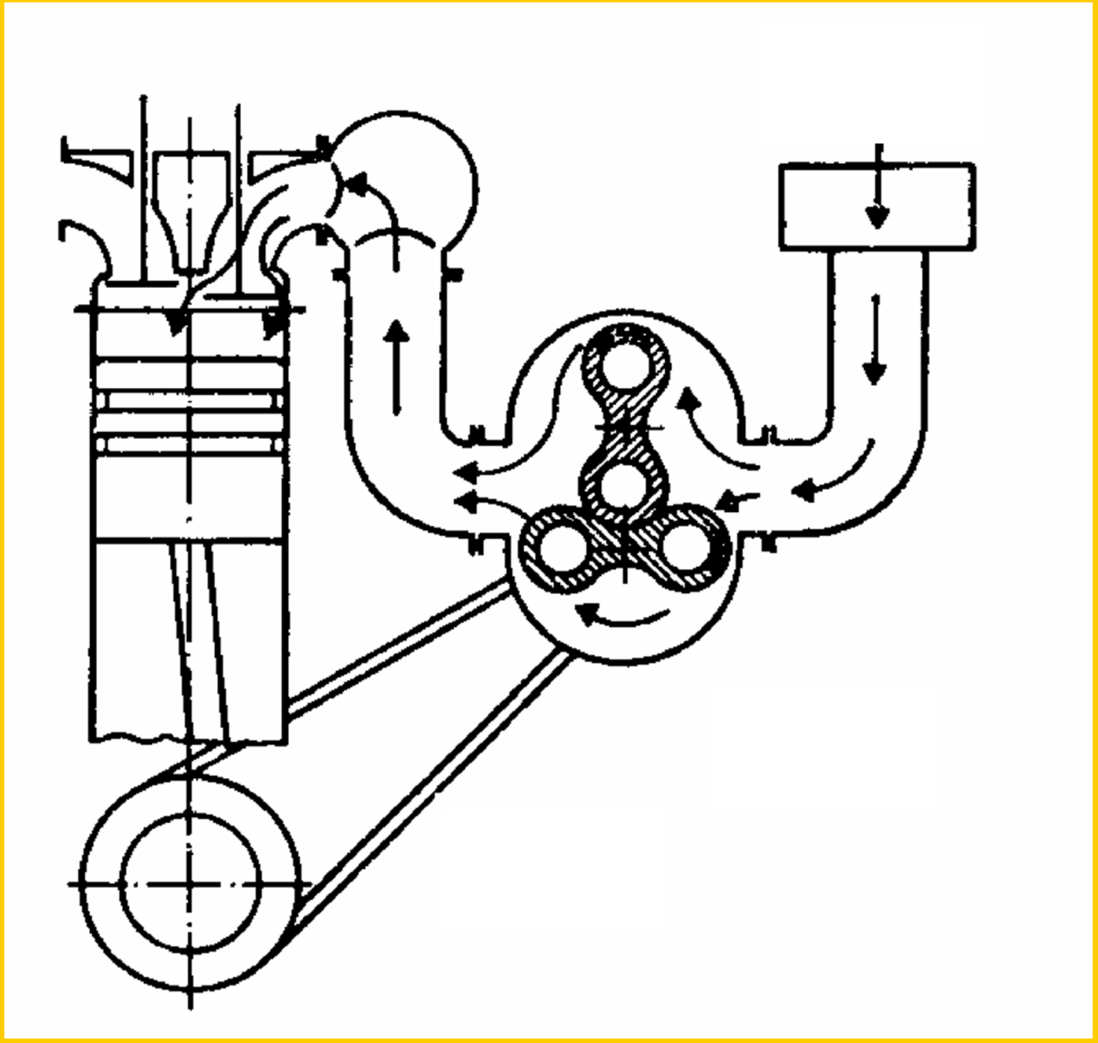
Advantages and Disadvantages

- Smaller Engine Dimensions (Down-sizing)
- Higher Power/mass ratio
- Higher efficiency
  - Pe/Pm ration better
  - Positive pumping work (W(-) -> W(+))
- Smaller Cooler

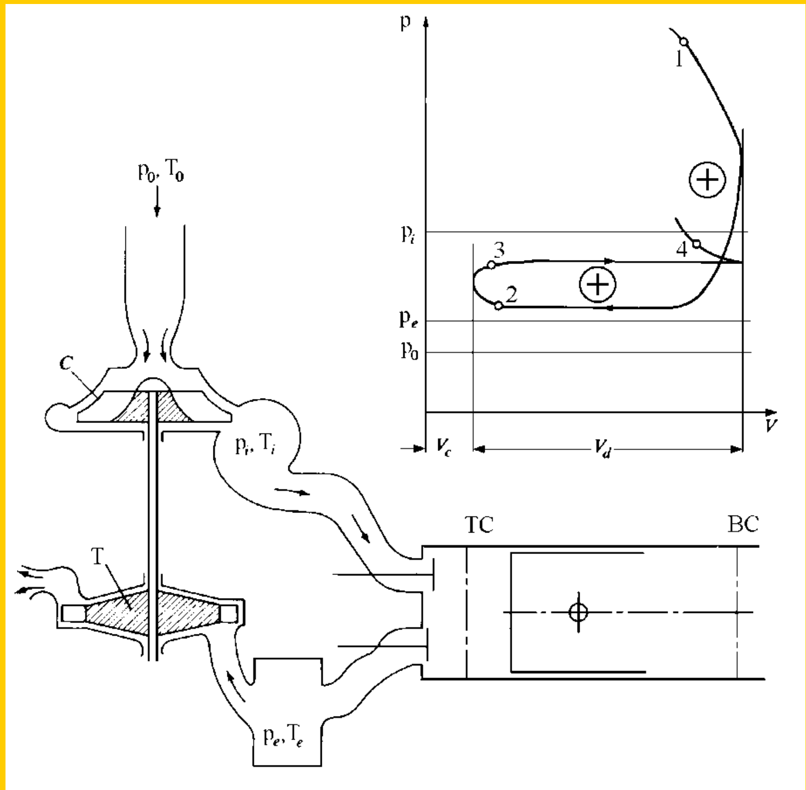
- Thermically and Mechanically Load increases

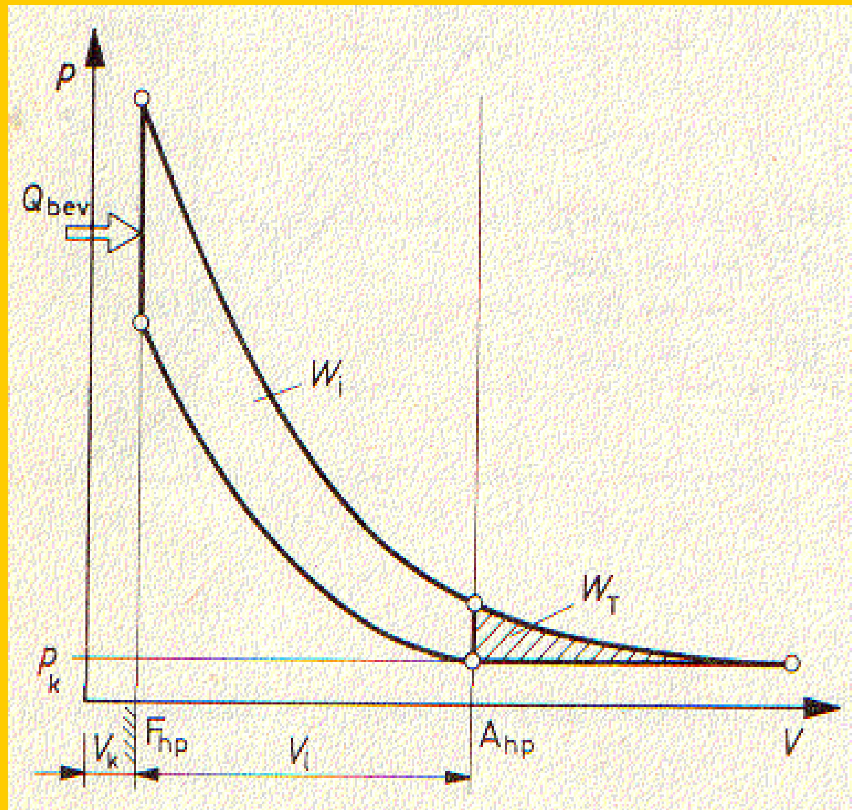


Pe
M
be
Pe
M
be
nmin.
nmax.
n



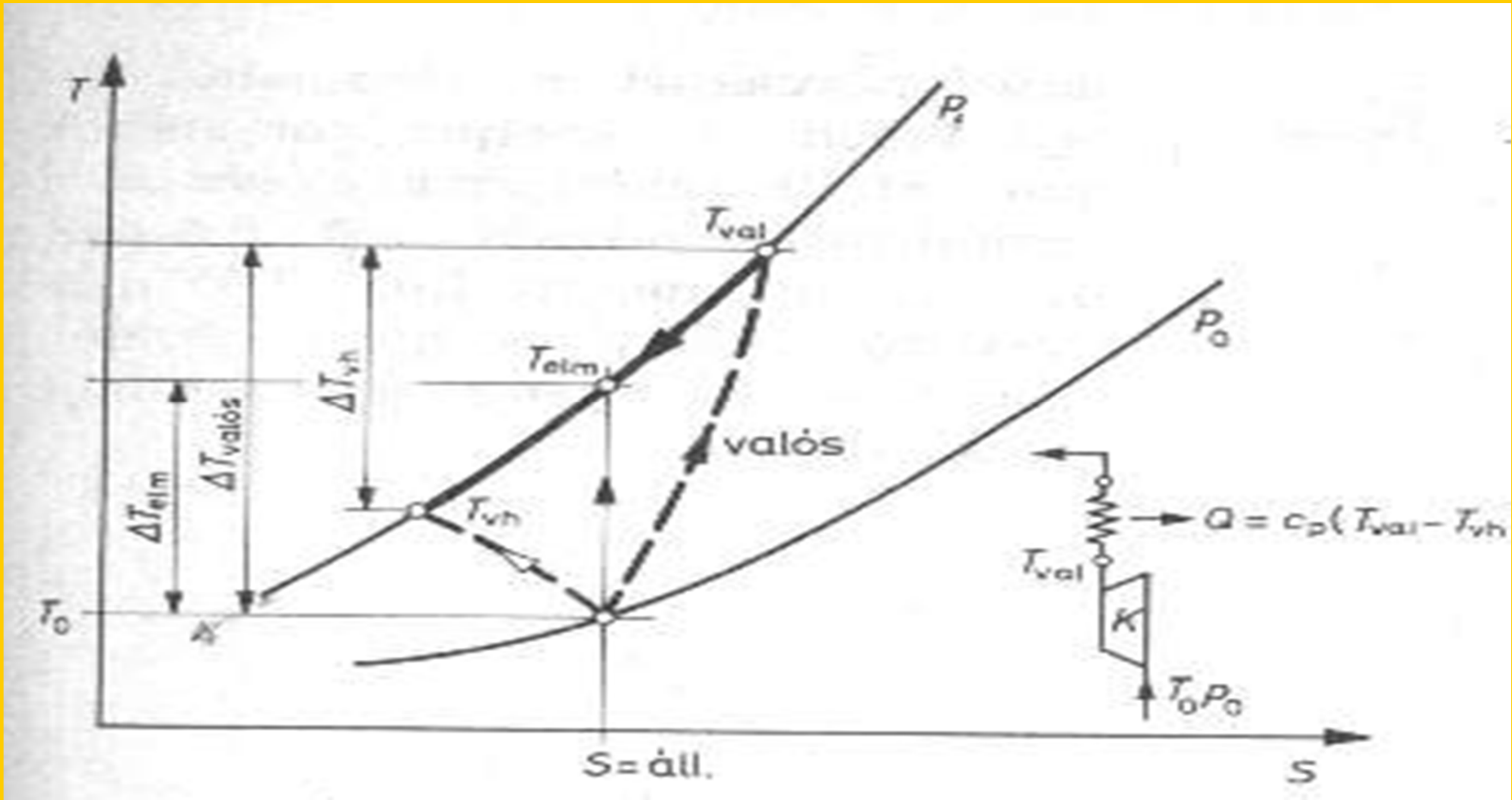
Mechanical Supercharging(Roots)

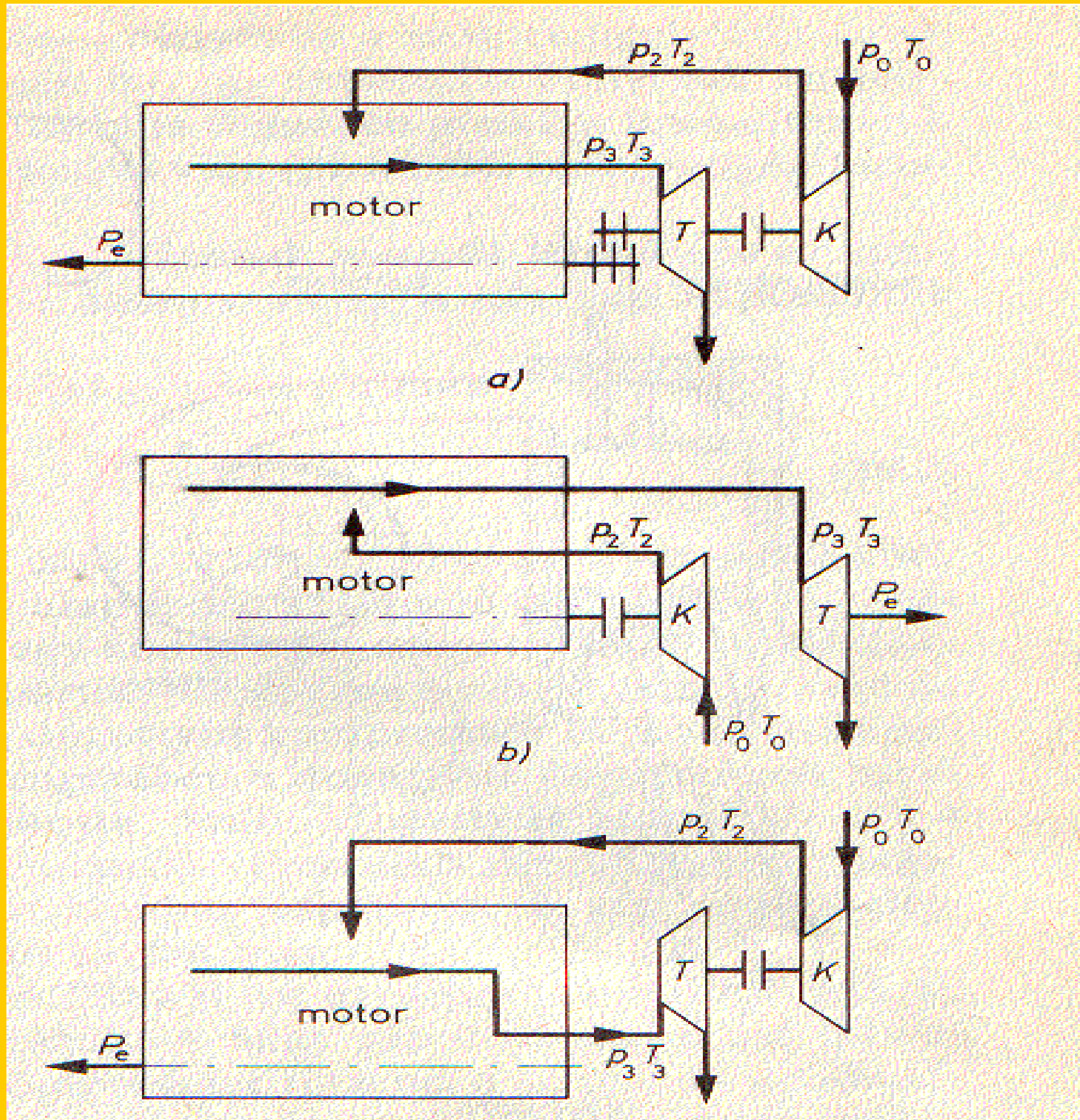




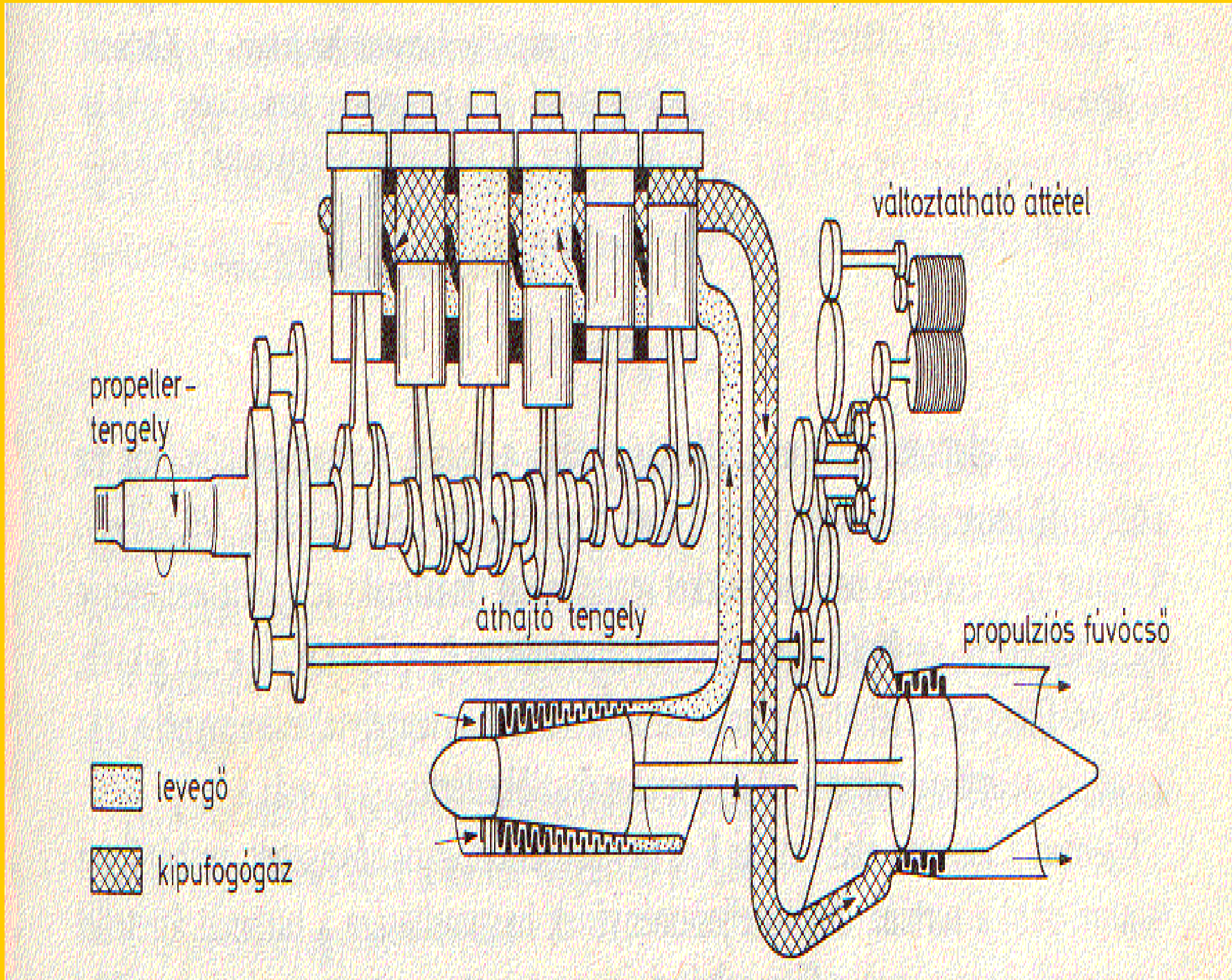
Turbocharging

Sűrítés és visszahűtés



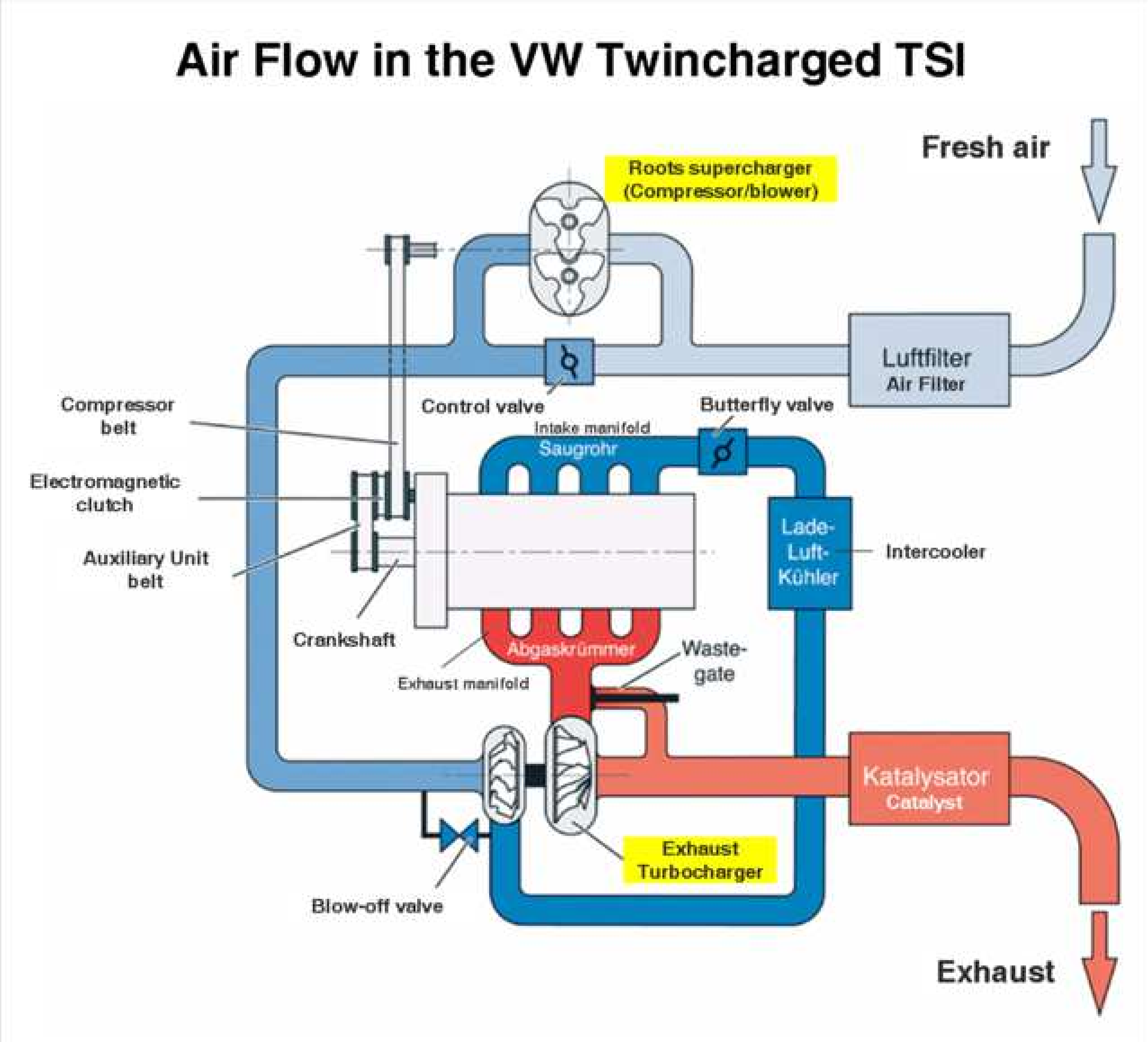


Different Solutions of Turbocharging

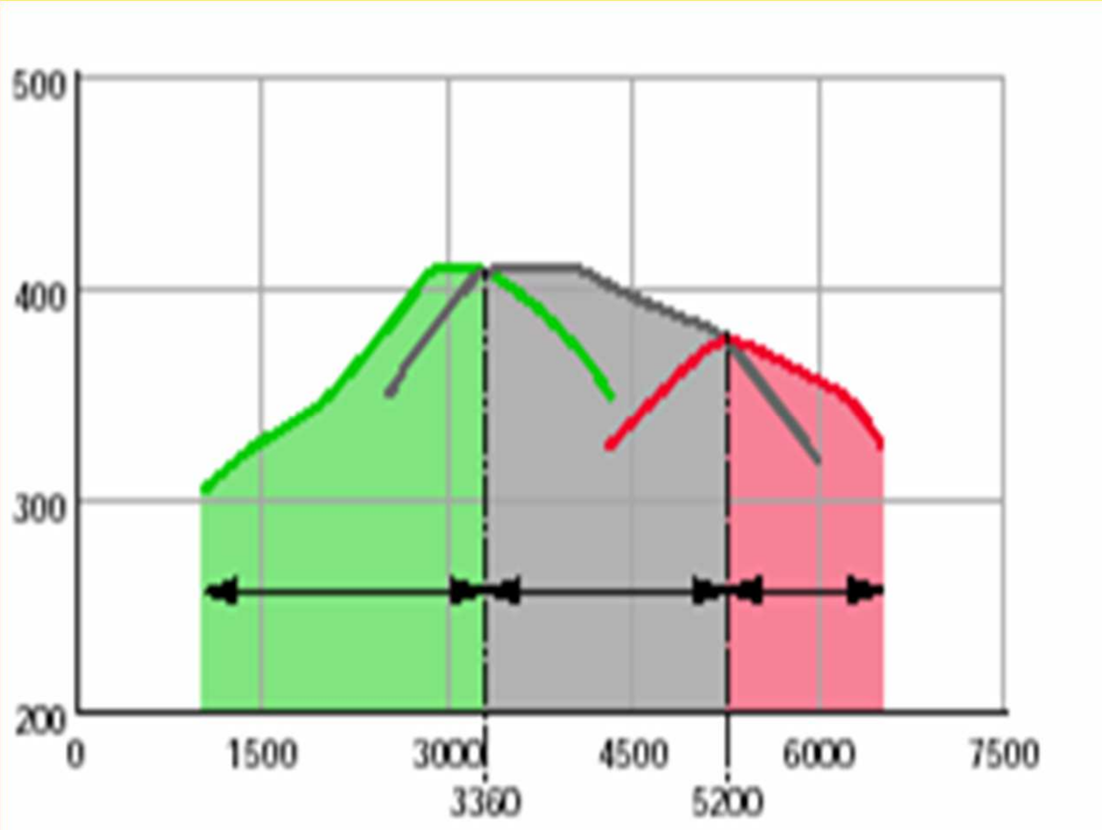


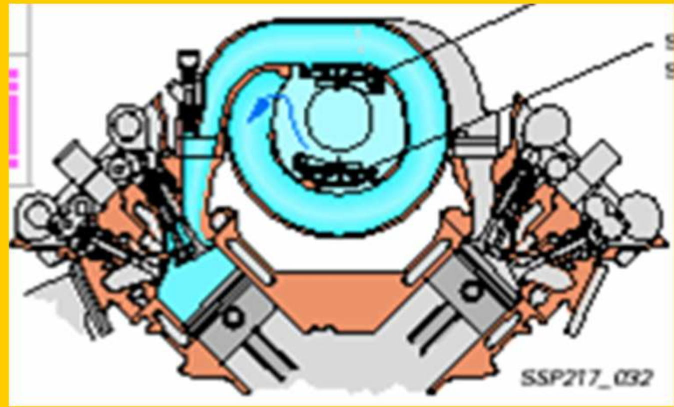
Napier-Nomad Diesel-compaund

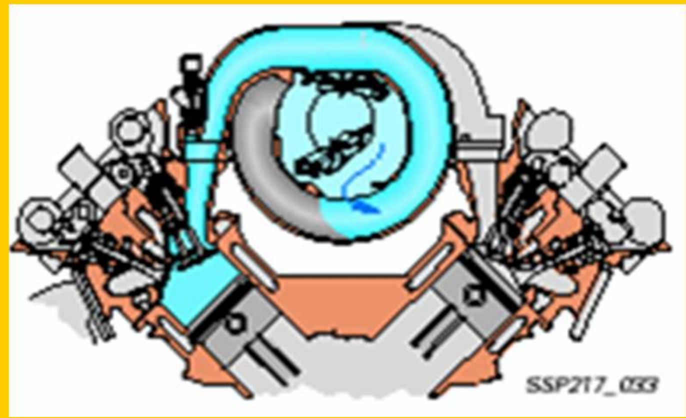
VW FSI

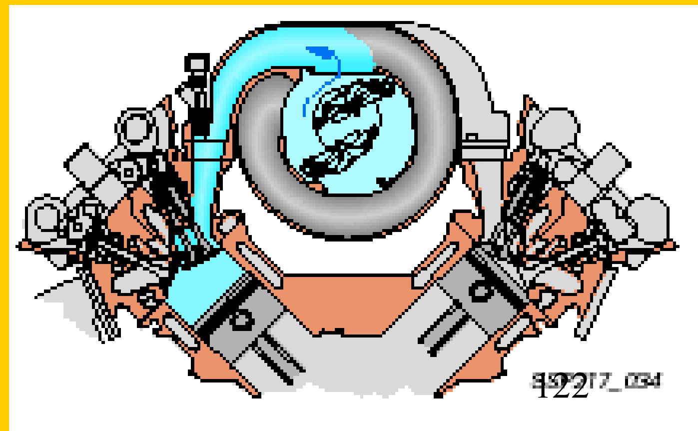

Air Flow in the VW Twincharged TSI
Fresh air
Roots supercharger
(Compressor/blower)
Luftfilter
Air Filter
Compressor
belt
Control valve
Butterfly valve
Intake manifold
Saugrohr
Electromagnetic
clutch
Auxiliary Unit
belt
Lade-
Luft-
Kühler
Intercooler
Crankshaft
Abgaskrümmer
Exhaust manifold
Waste-
gate
Katalysator
Catalyst
Exhaust
Turbocharger
Blow-off valve
Exhaust

Acoustical Charger



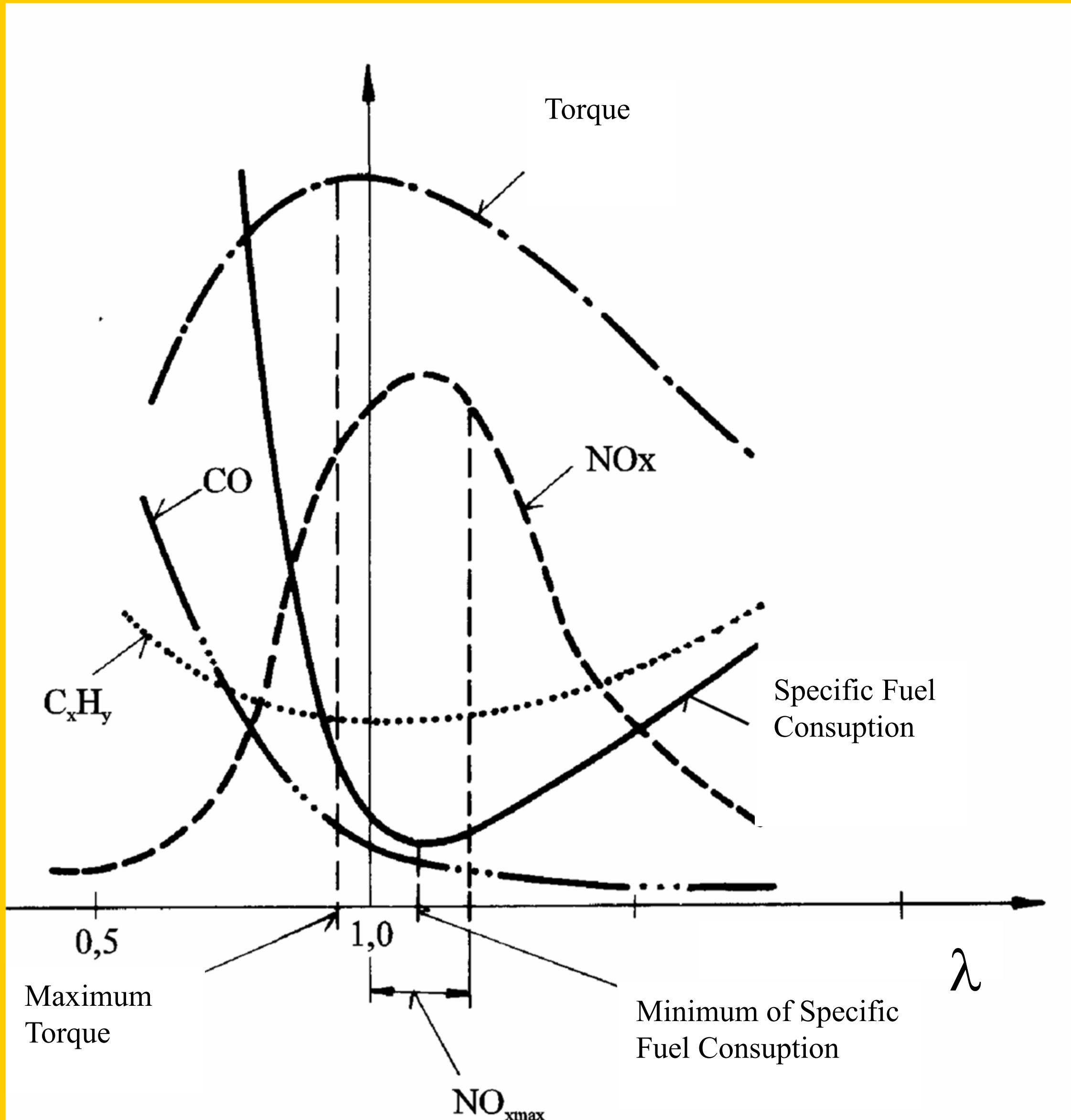




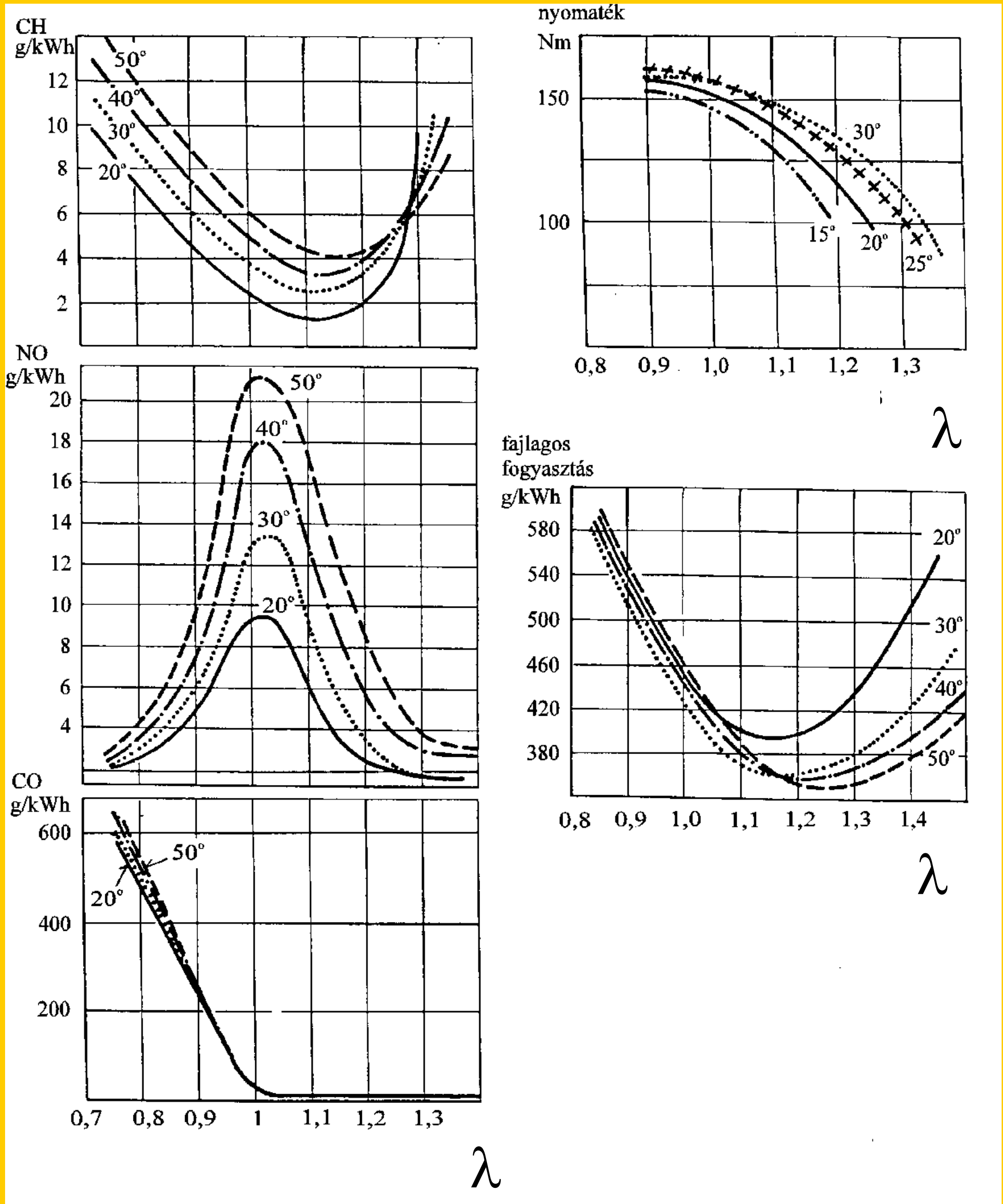


Possibilities of Emission Decrease

- Emission Decrease:
  - Before Engine
    - Fuel (S, Pb, Heavy metals)
  - In the Engine
    - Constuction
    - EGR
    - Air-to-Fuel ratio
  - After Engine (secunder methodes)
    - 3 way catalytic converter
    - oxidacion catalytic converter
    - ……

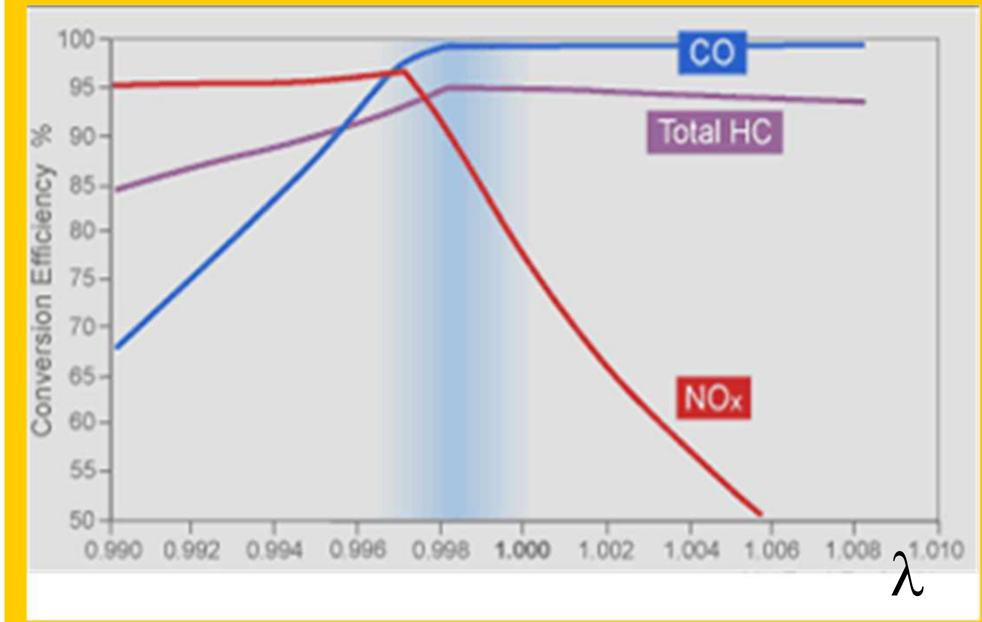


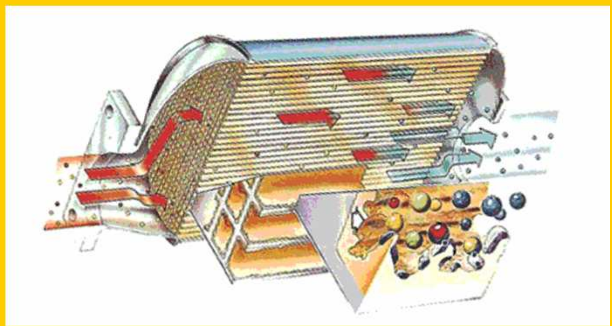
A káros anyagok emissziója a légfelesleg függvényében



Effects of the pre-ignition settings

Catalytic Converters





3-way (NSCR) Catalysts (λ=1)

$NO_x \rightarrow N_2+O_2$

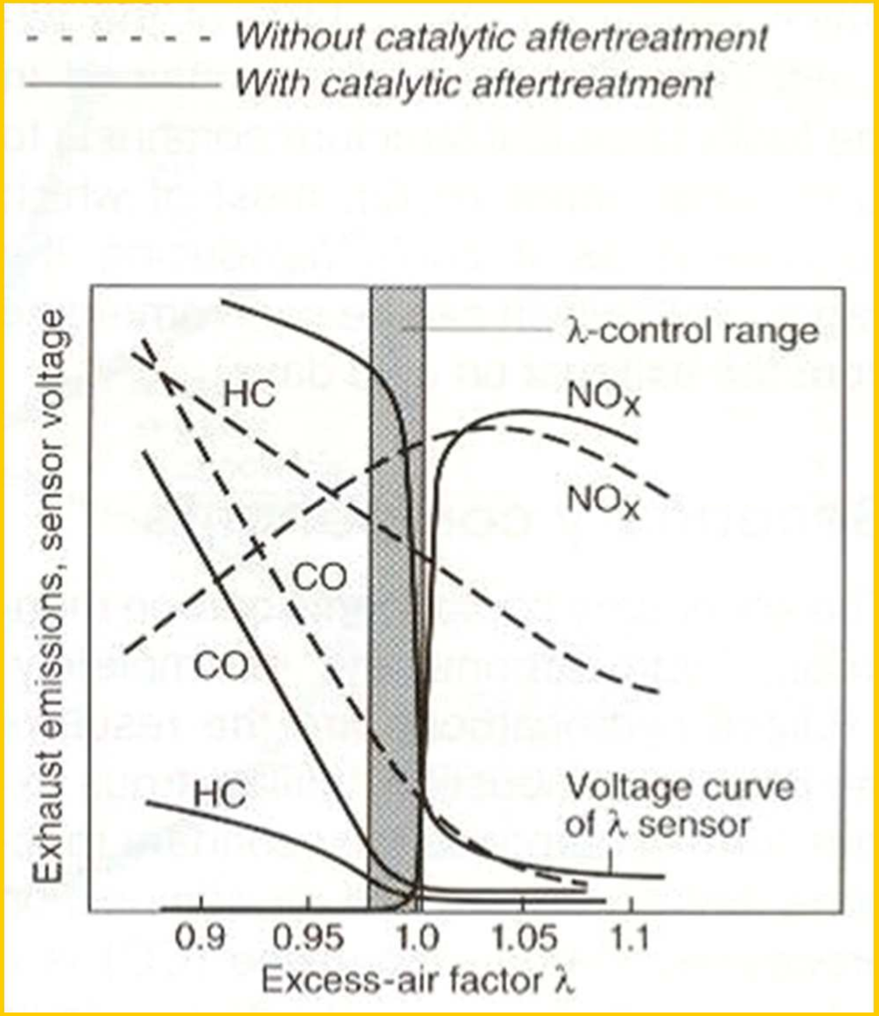
$CO \rightarrow CO_2$

$C_xH_y \rightarrow H_2O+CO_2$

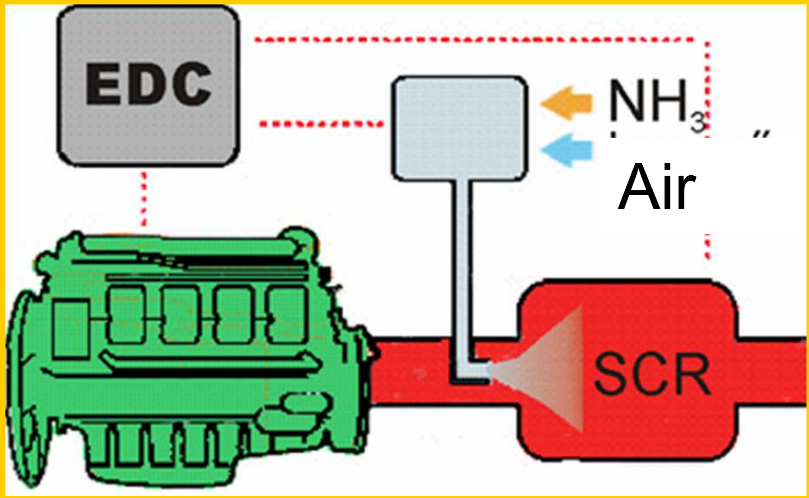
2-way (oxidation) Catalysts

$CO \rightarrow CO_2$

$C_xH_y \rightarrow H_2O+CO_2$


Without catalytic aftertreatment
With catalytic aftertreatment
λ-control range
HC
NOx
NOx
CO
CO
HC
Voltage curve
of λ sensor
0.9
0.95
1.0
1.05
1.1
Excess-air factor λ
Exhaust emissions, sensor voltage

Katalizátorok

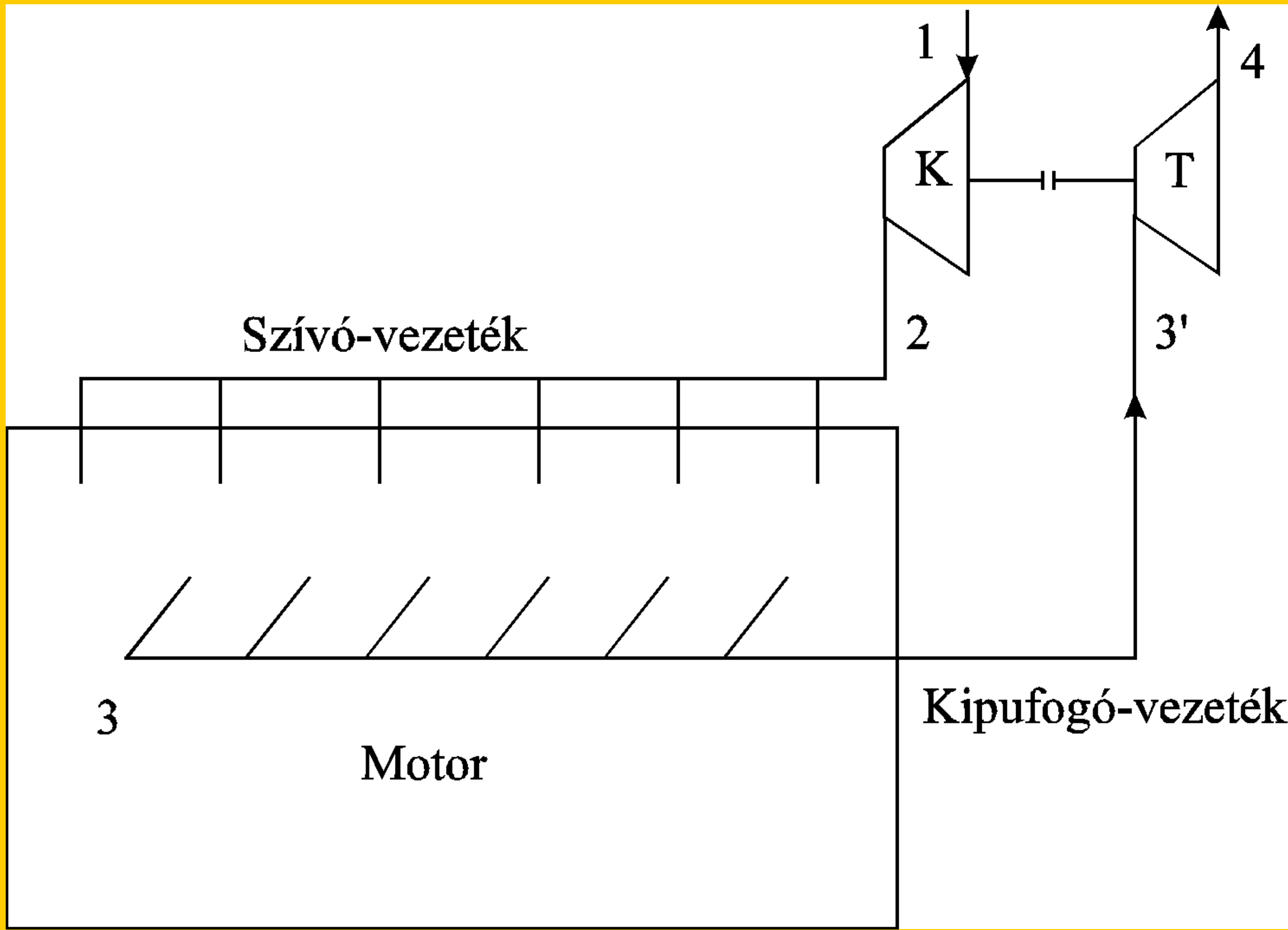


Selective Catalytic Reduction (SCR)

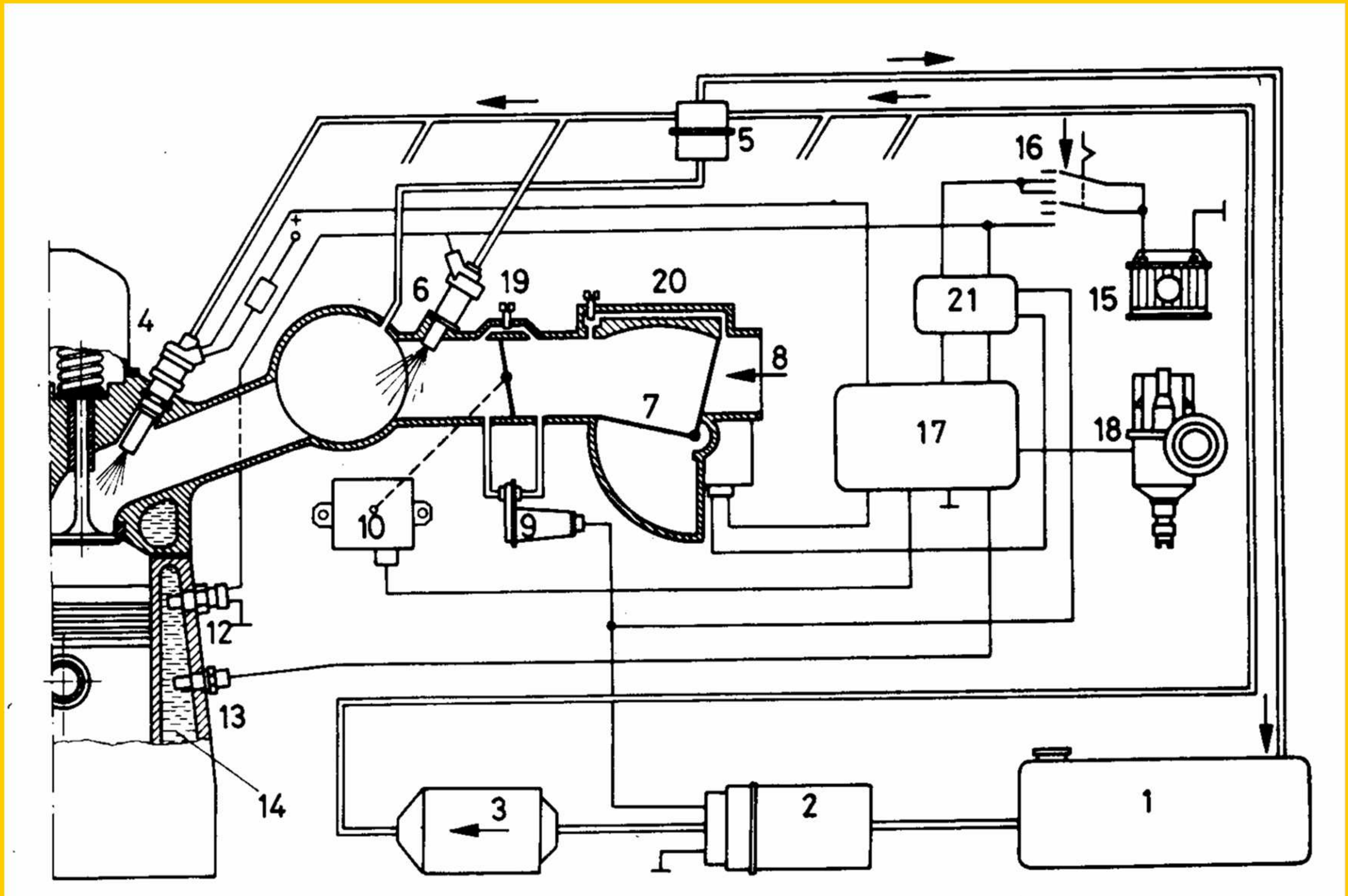
$NO_x + NH_3 \rightarrow N_2 + H_2O$

Particulate Filters (PF)

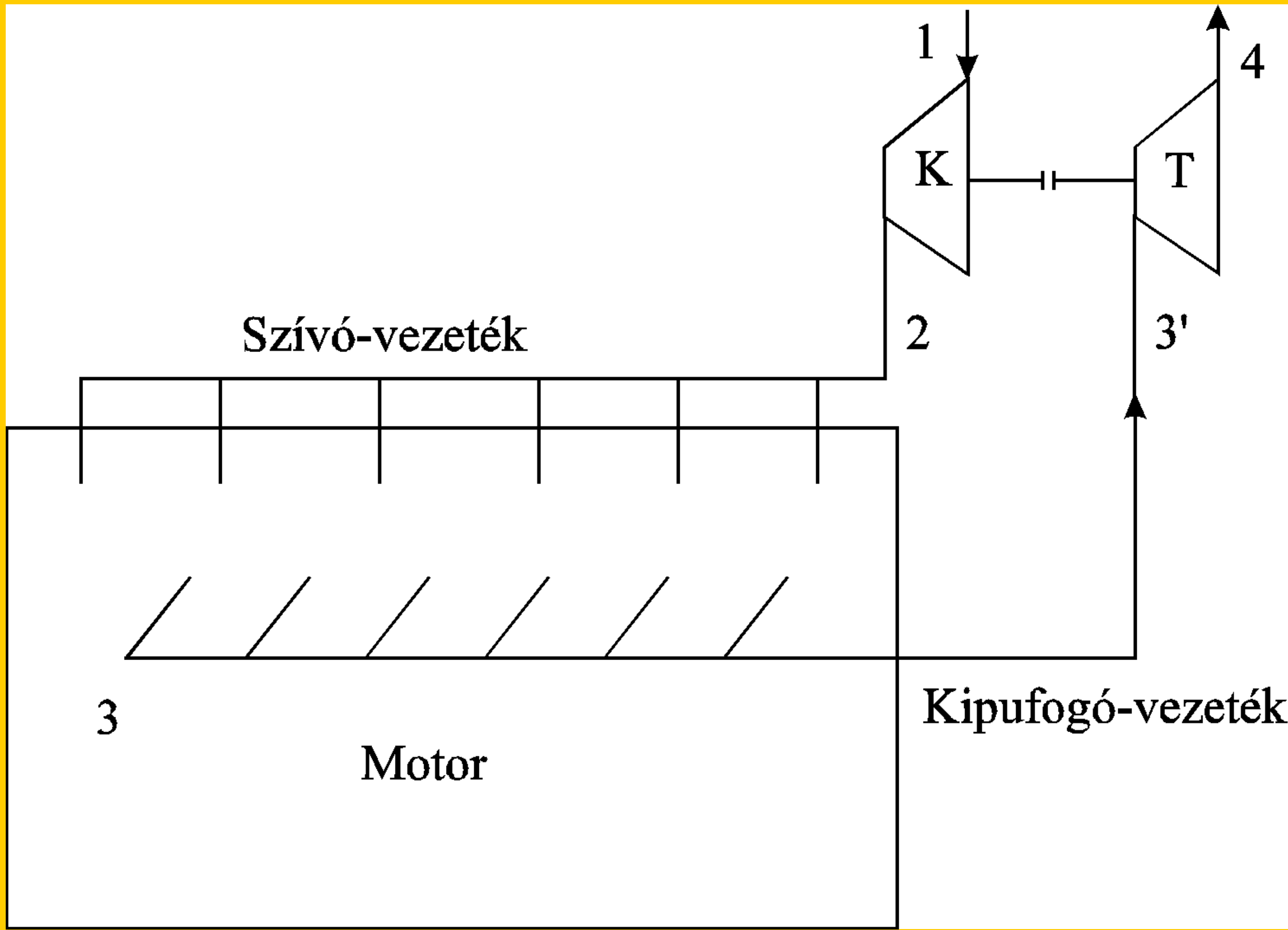
END



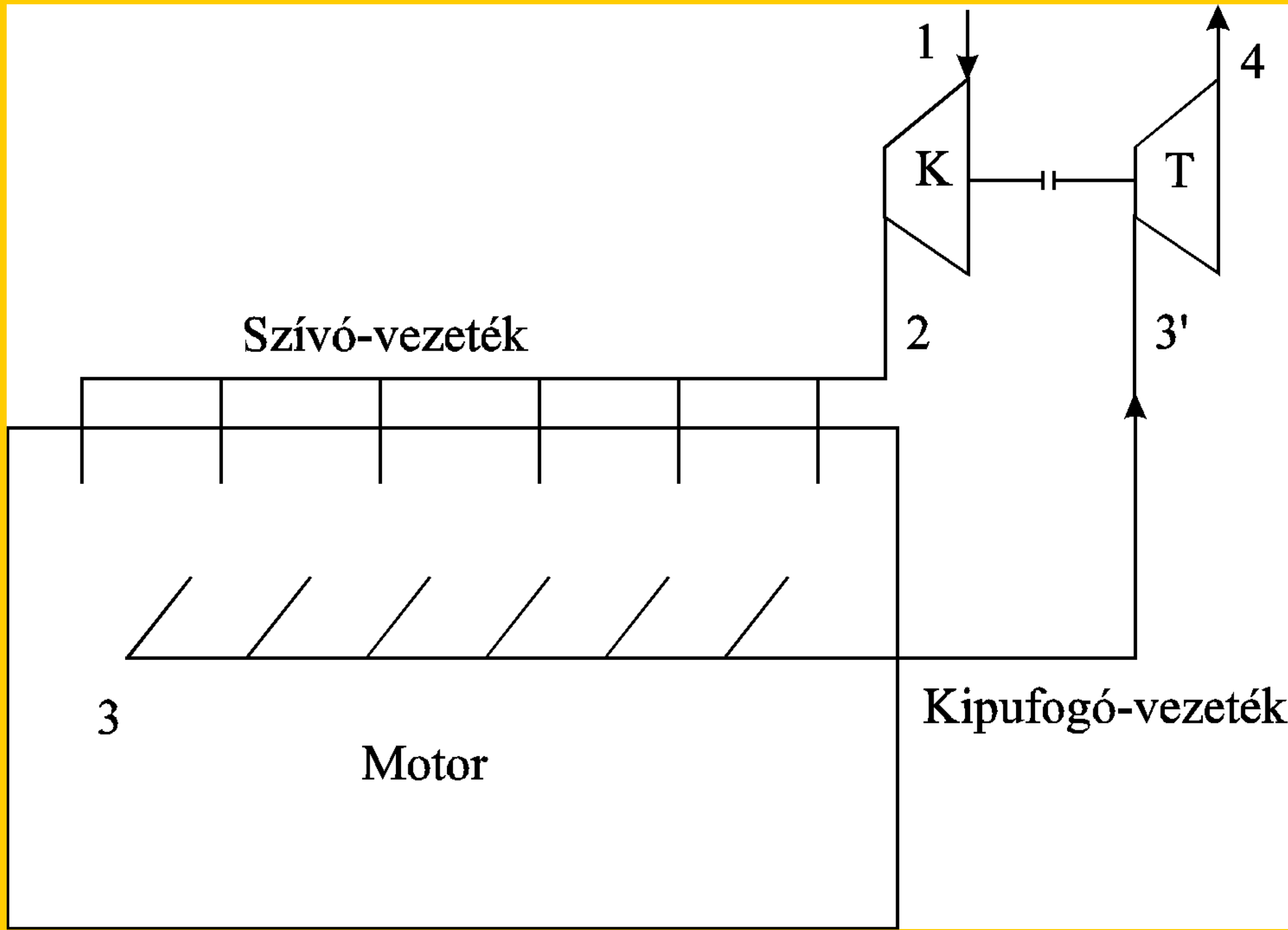
Turbótöltéses motor elvi elrendezése



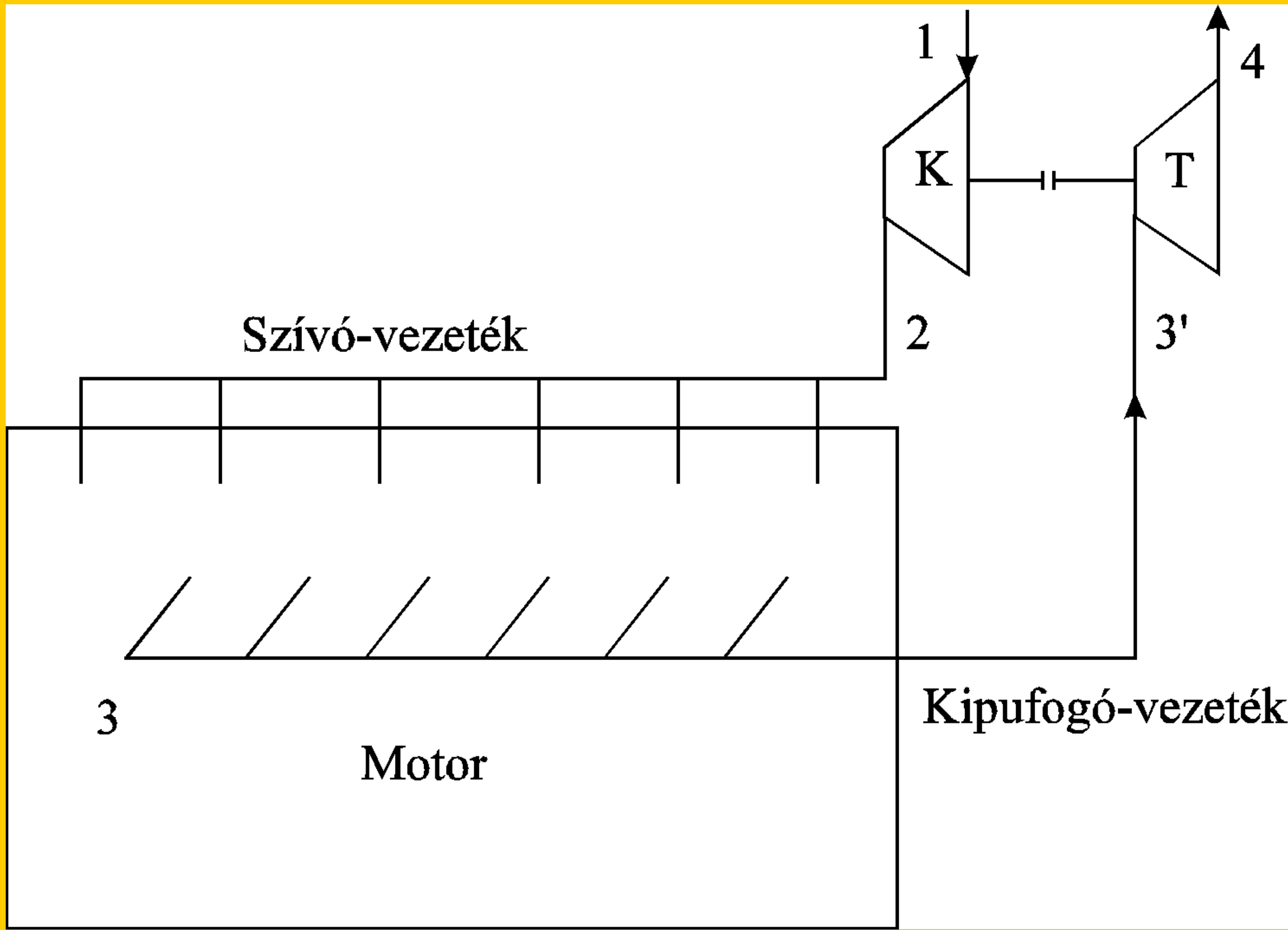
Bosch L-Jetornic system



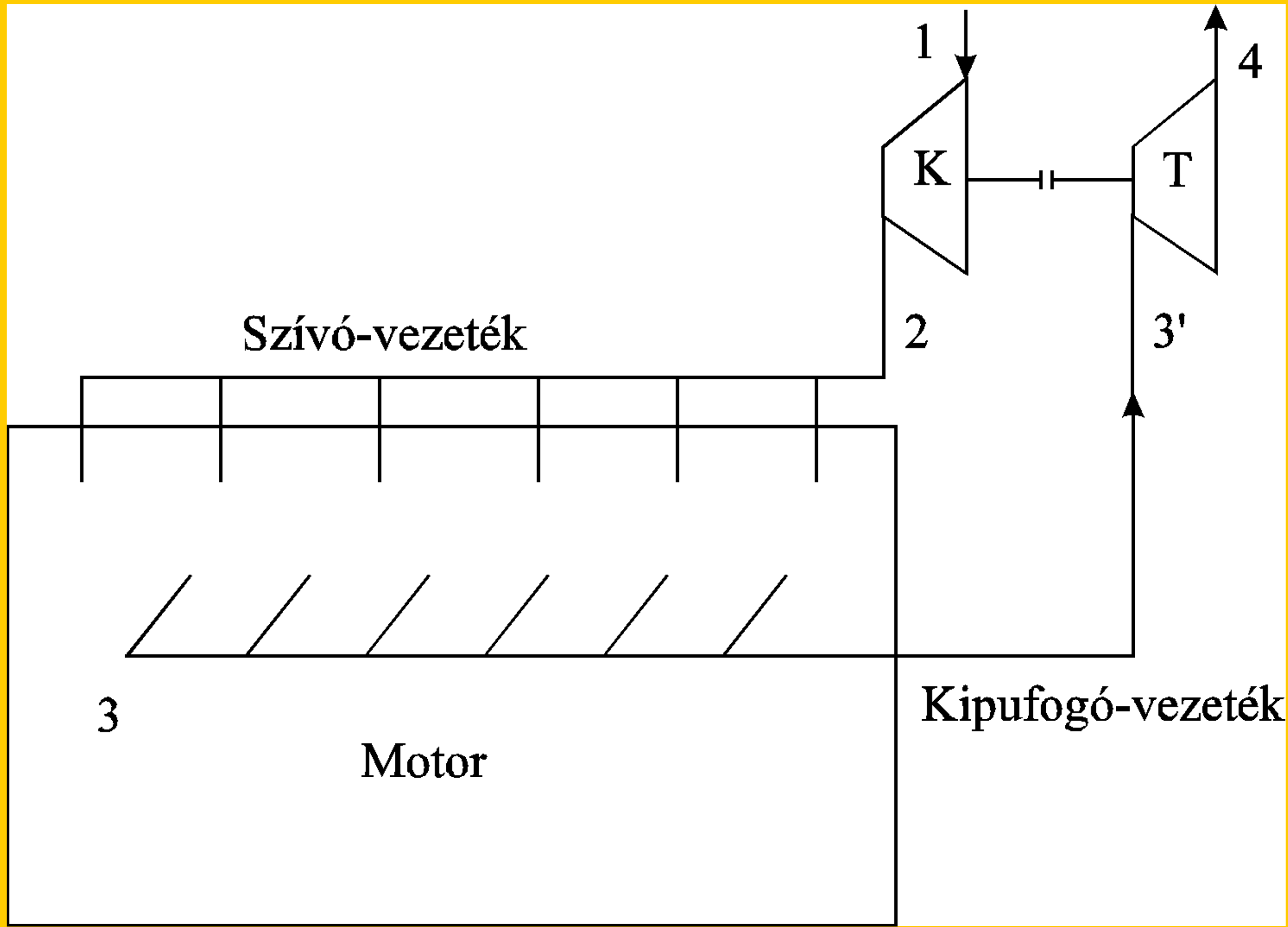
Turbótöltéses motor elvi elrendezése



Turbótöltéses motor elvi elrendezése



Turbótöltéses motor elvi elrendezése



Turbótöltéses motor elvi elrendezése

Nyomáshullámmal történő feltöltés

- Négyütemű motor jó hengertöltése akkor érhető el, ha:
  - a kiömlő szelep nyitási szakaszának a vége felé a hengerben kicsi nyomás uralkodik, hogy lehetőleg kevés maradék gáz maradjon vissza, és jó legyen az öblítés
  - a szívószelep zárásakor a hengerben nagy nyomás található, hogy a töltet minél nagyobb legyen.
- A vezetékhosszak és keresztmetszetek összehangolásával az előző feltételek teljesíthetők. Összehangolás csak szűk fordulatszám-tartományban lehetséges, mert a gázoszlopok sajátfrekvenciája a vezetékhosszaktól függ. Ha a sajátfrekvenciákat a motor különböző fordulatszámaihoz akarjuk illeszteni, akkor a vezetékhosszakat a motor fordulatszáma függvényében változtatni kell.

Motorokban keletkező főbb káros anyagok

- A motorokban lejátszódó égés során a következő káros anyagok keletkeznek:
  - szén-monoxid (CO)
  - nitrogén-oxidok (NO_x)
  - szénhidrogének (C_xH_y)
  - részecskék (korom)

Basic principals of mechanical construction

Arrangement of cylinders

- in line arrangement
- V arrangement
- opposed cylinder engine
- radial type engine
- other

Fluid inlet-outlet control

- side valve (SV) arrangement
- overhead valve (OHV) arrangement
- overhead camshaft (OHC) arrangement

cooling system

- air cooling
- water cooling