Internal Combustion Engines

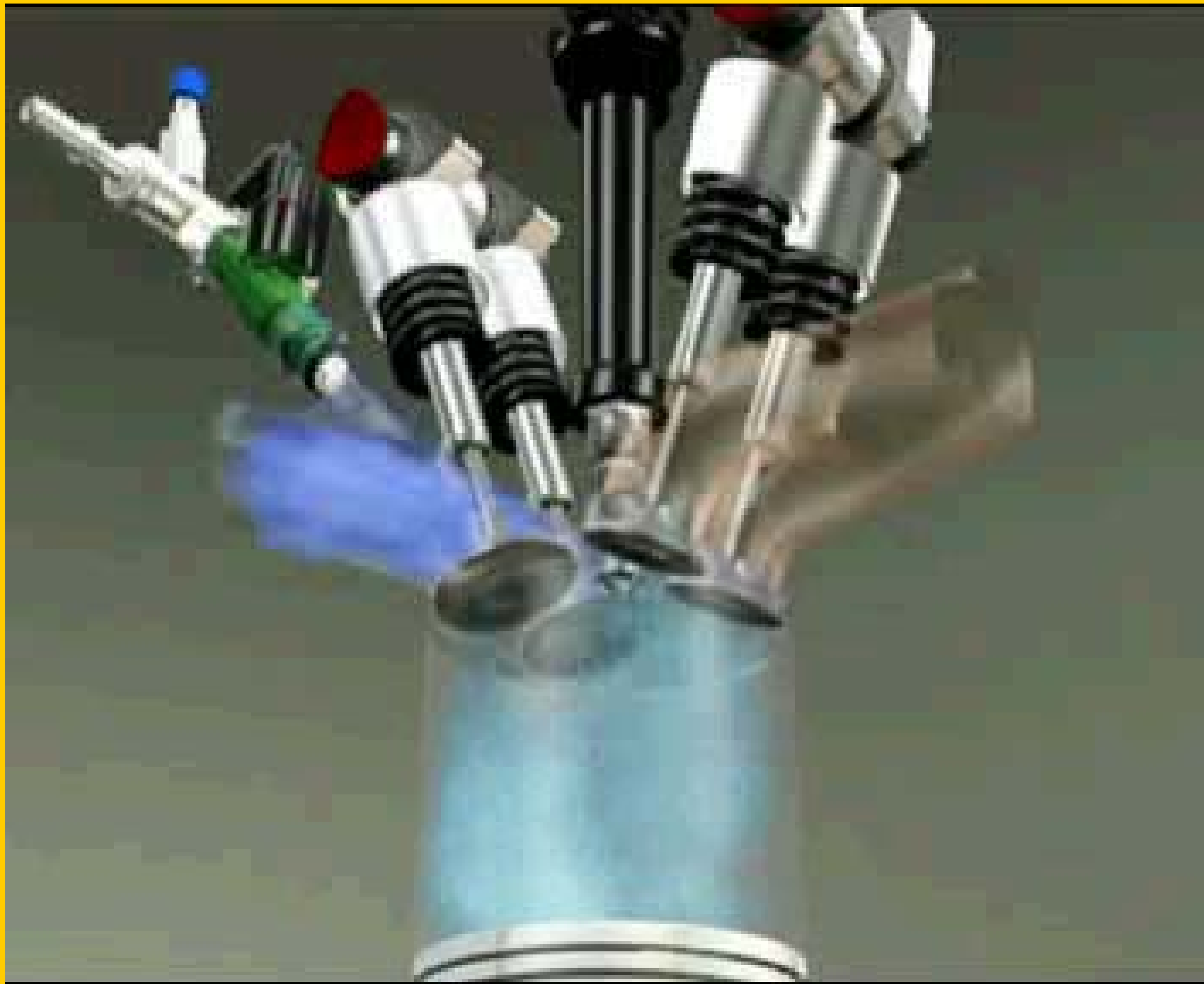
By Dr. Akos Bereczky, BME

Actual cycles of internal combustion engines

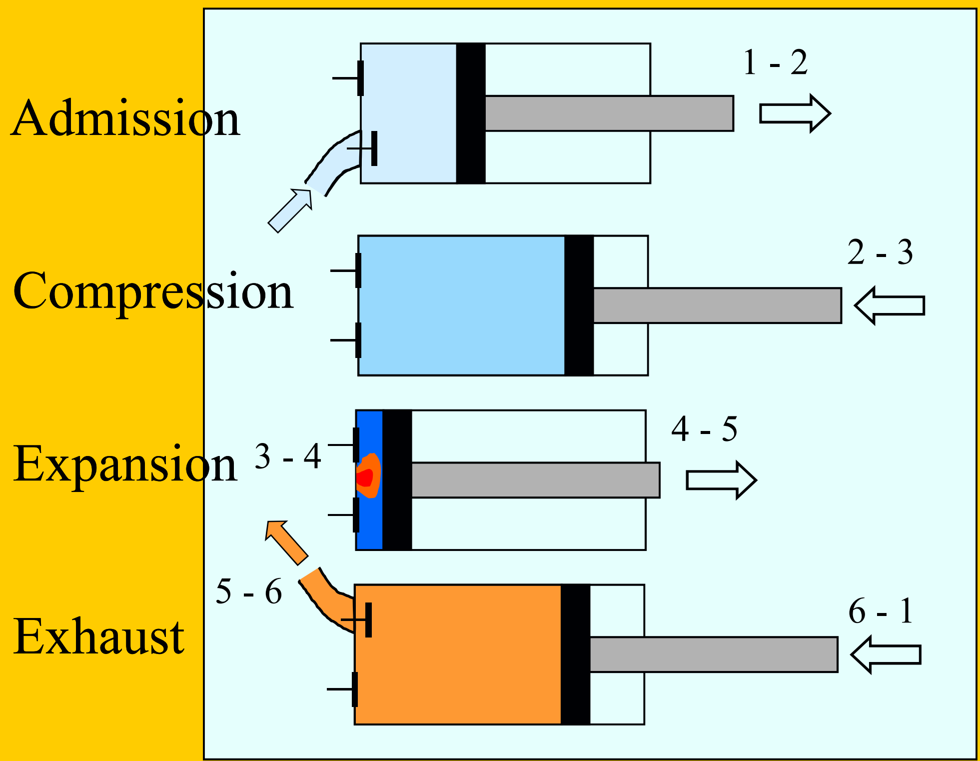
- Brake work: $W_B = Q_{in}$ - losses (Q, W)

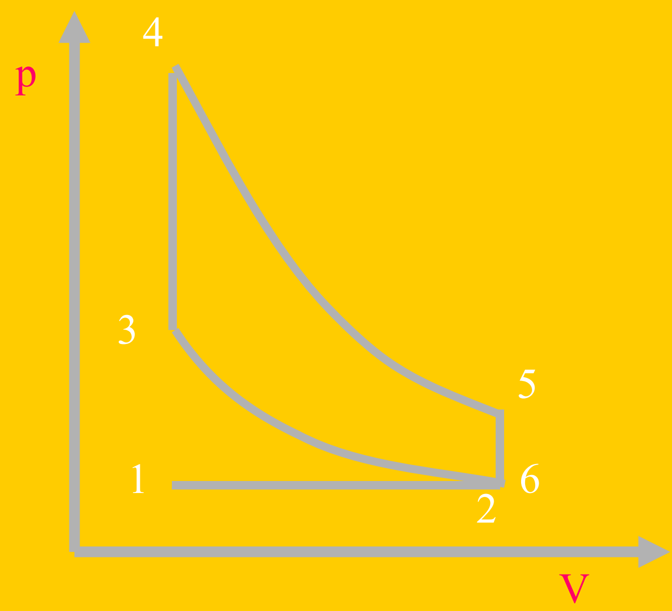
- Losses are divided into three main groups:
  - basic losses
  - internal losses
  - mechanical losses

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 basic loss

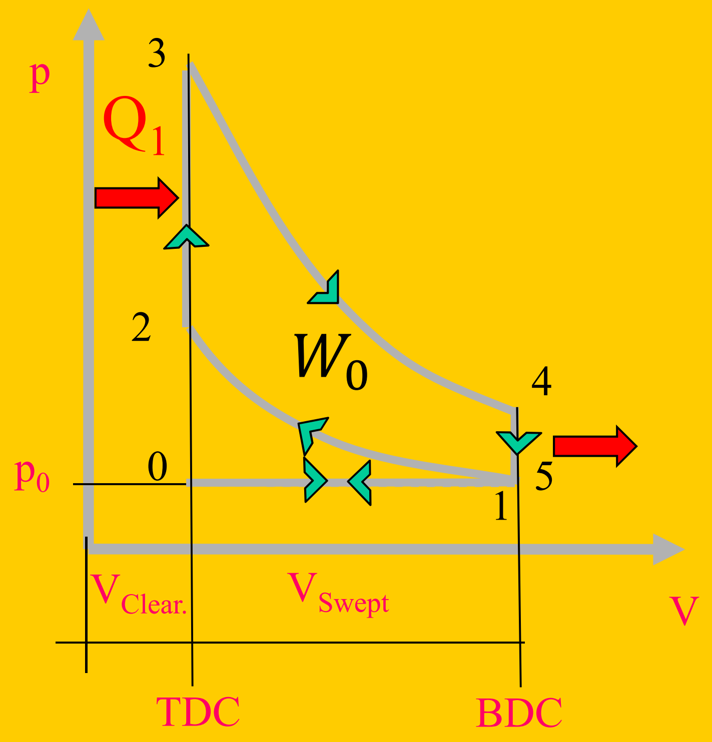
- The basic loss is described by the thermal efficiency: $\eta_0 = \frac{Q_{in} - Q_{out}}{Q_{in}} = \frac{W_0}{Q_{in}}$

- Theoretical S.I. (Otto) cycle:

$$W_0 = c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

$$\varepsilon = \frac{V_{Max}}{V_{Min..}} = \frac{V_{Clearence} + V_{Swept}}{V_{Clearence}}$$

$$\eta_0 = 1 - \frac{1}{\varepsilon^{\kappa - 1}}$$



COMPRESSION RATIO VALUES

1 **– atmospheric engine**

3 **– Otto engine**

4 **– side valves**

6 **– Ricardo (turbulance) head**

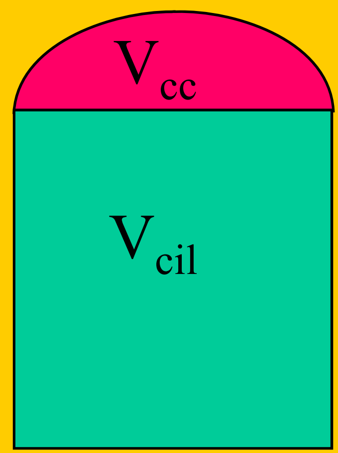
7 **– head valves**

9 **– leaded petrol (5 star)**

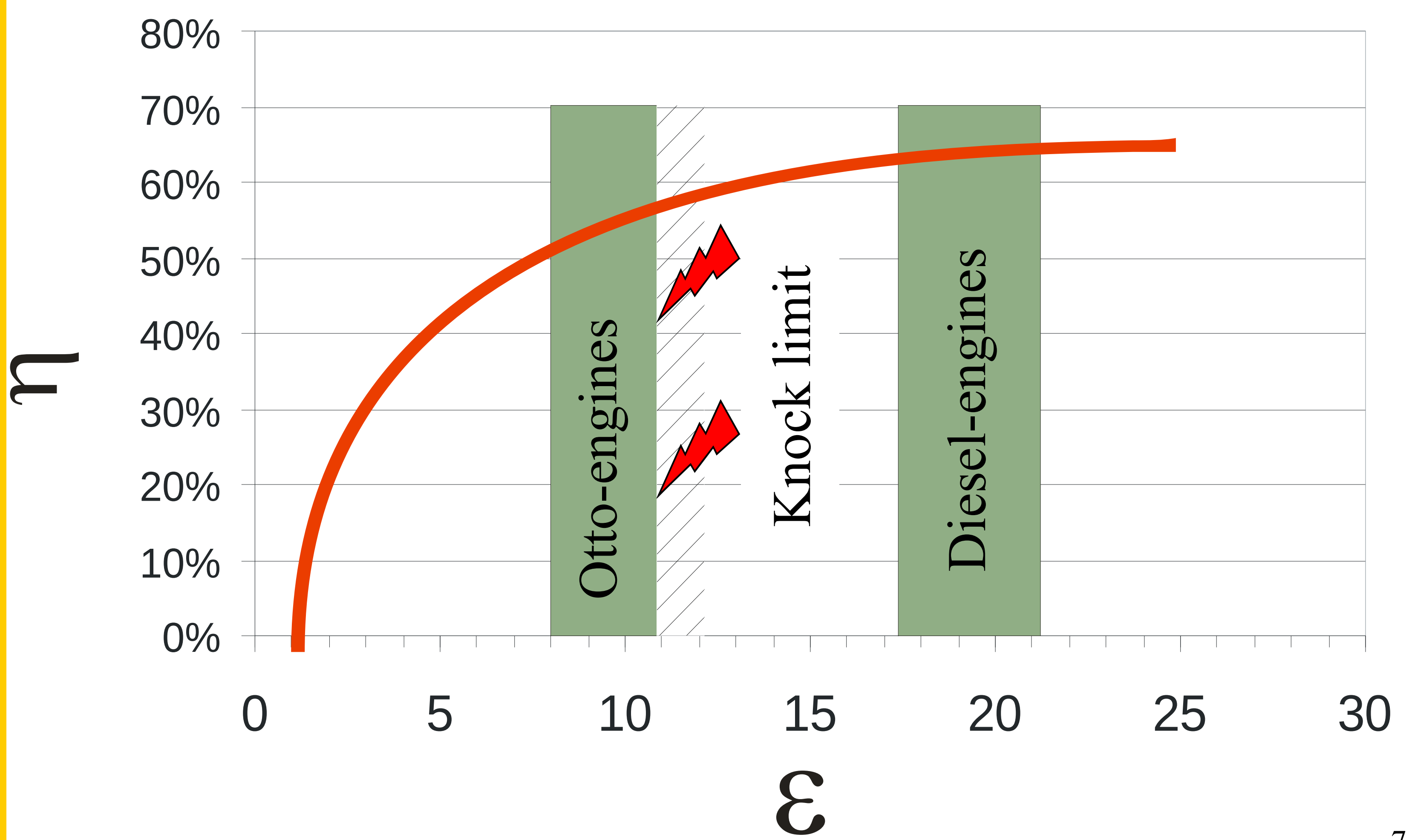
10 **– electronic injection MPI**

11 **– detonation control**

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

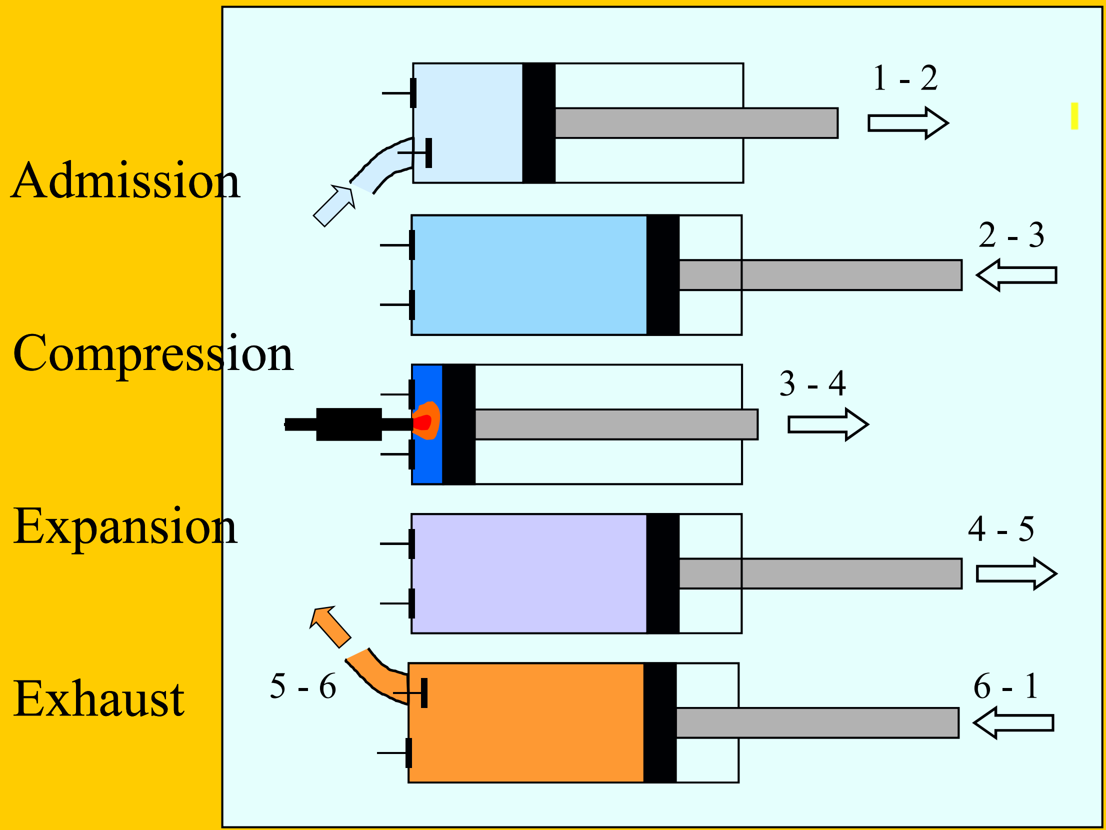


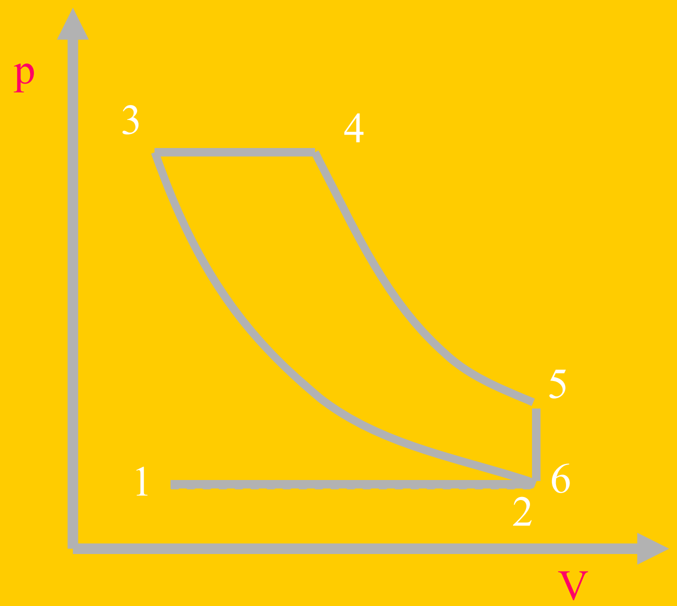
Efficiency in the function of the compression ratio



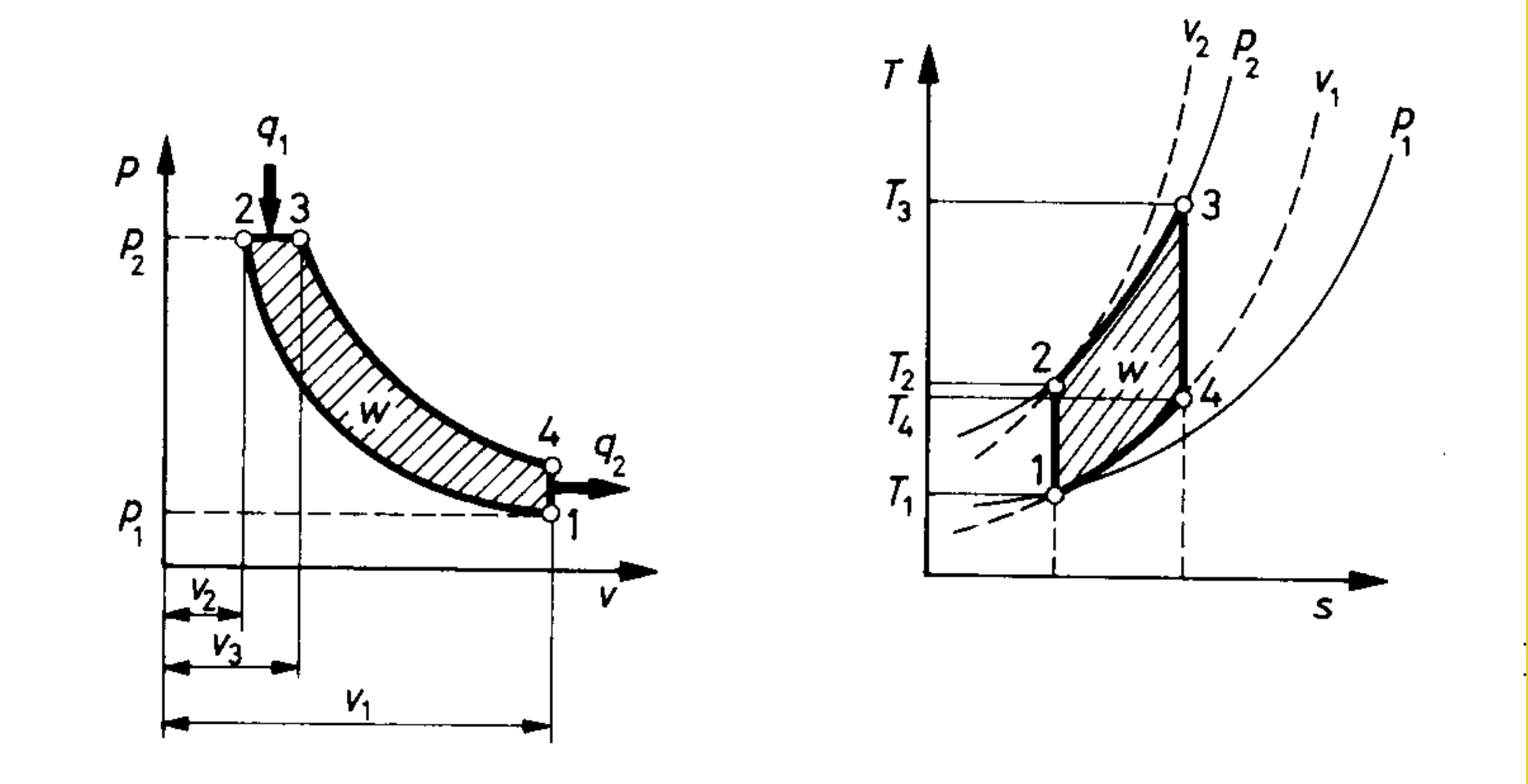
Direct Injection Combustion

DIESEL CYCLE





Diesel cycle



$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^\kappa - 1}{\kappa \cdot (\rho - 1)} \qquad \rho = \frac{v_3}{v_2}$$

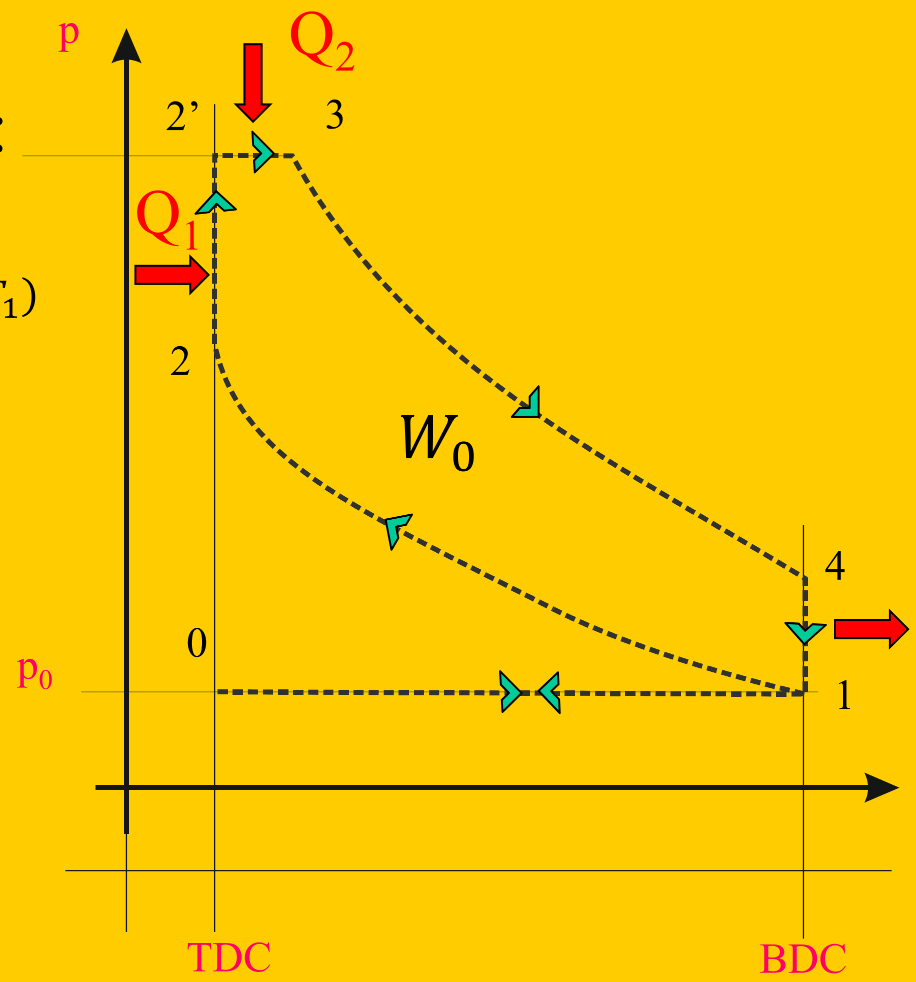
The basic loss

- The basic loss is described by the thermal efficiency: $\eta_0 = \frac{Q_{in} - Q_{out}}{Q_{in}} = \frac{W_0}{Q_{in}}$

- Seiliger - Sabathe cycle:

$$W_0 = c_v(T_{2'} - T_2) + c_p(T_{2'} - T_3) - c_v(T_4 - T_1)$$

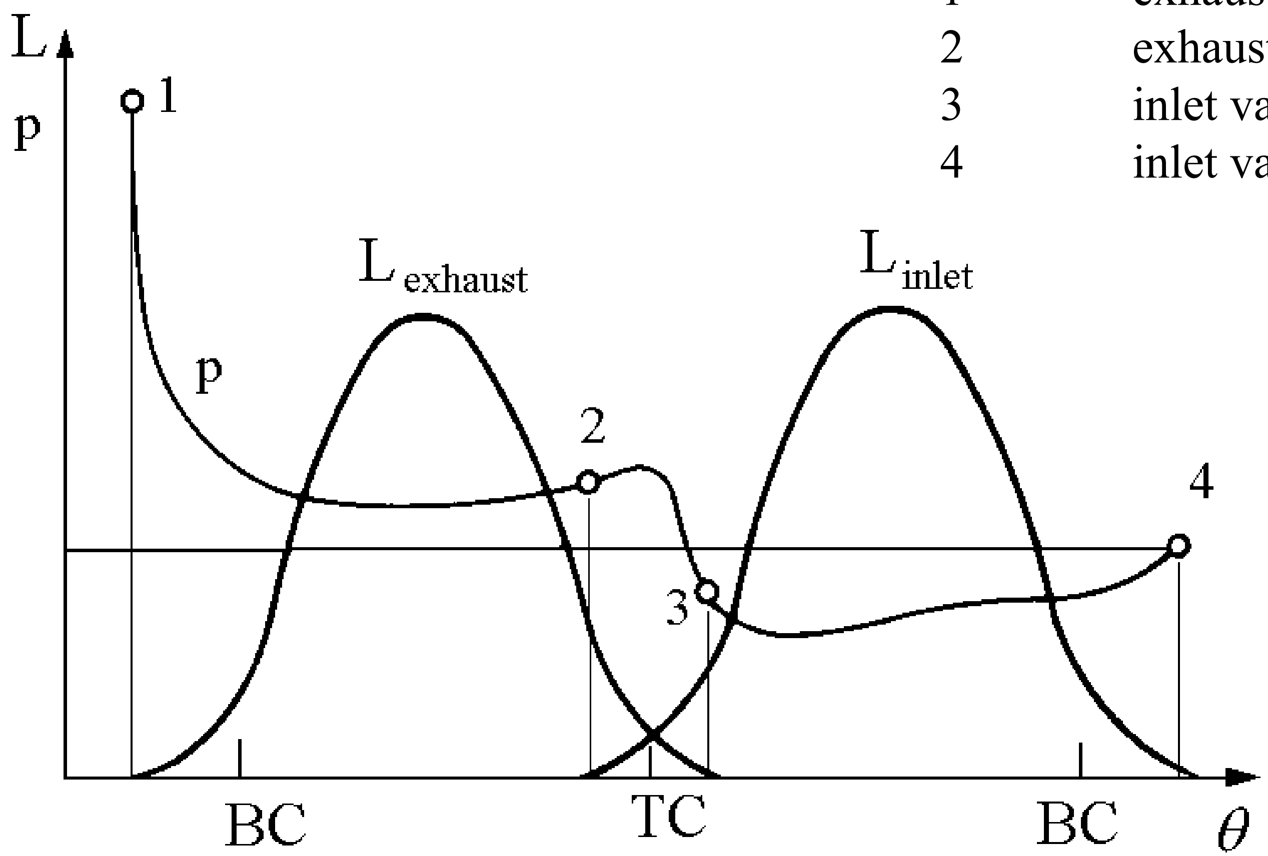
$$\eta_0 = \frac{W_0}{c_v(T_{2'} - T_2) + c_p(T_{2'} - T_3)}$$



The internal losses

- Intake and Exhaust Losses
- Heat transfer (non isentropic) Compression and Expansion;
- Limited Flame Propagation (Incomplete combustion)
- Gas losses (Blow-by)

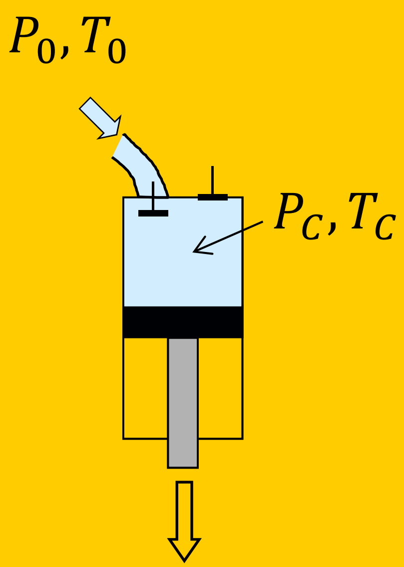
L	valve lift
BC	bottom dead center
TC	top dead center
1	exhaust valve opens
2	exhaust valve closes
3	inlet valve opens
4	inlet valve closes



2-3: overlap period

The internal losses

Intake and exhaust losses



$$m_{Charge,theo.} = \frac{p_0 V_l}{RT_0}$$

Mass of the real charge:

$$m_{Charge} = \frac{p_C V_l}{RT_C} = \frac{(p_0 - p_{Loss.})V_l}{R(T_0 + T_{Loss.})}$$

Delivery ratio $$\lambda_D = \frac{m_{C,real}}{m_{C,theo.}} = \frac{p_0 T_C}{p_C T_0}$$

$$\lambda_t = \frac{\dot{m}_{C,real}}{V_{S,C} * z * i * n * \rho_0}$$

Possibilities to increase the Delivery ratio:

- low valve resistance, use of multiple valves;
- small intake manifold resistance;
- reduce heating:
- use dynamic charging (valve overlaping);
- charging

Charging systems

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

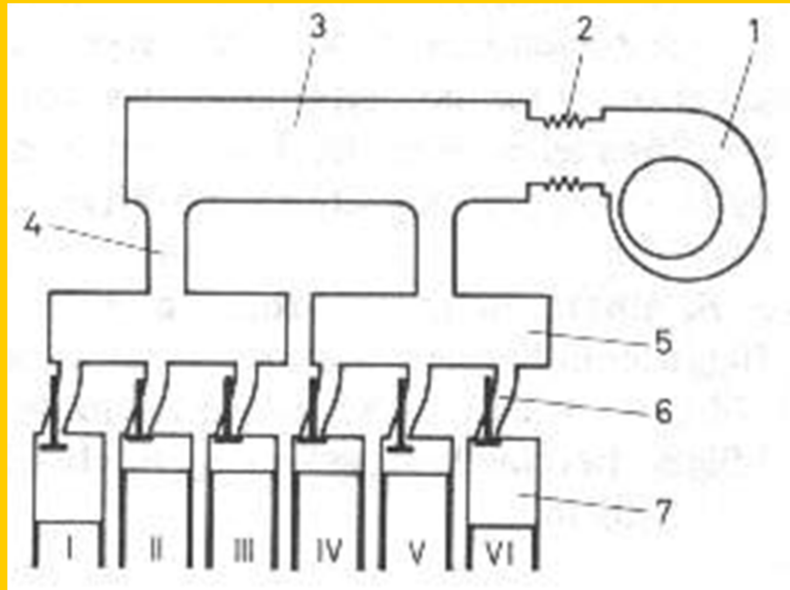
Acoustical charge

Helmholtz rezonátor:

$$f = \frac{a}{2\pi}\sqrt{\frac{A}{V_0 L}}$$

Orgonasíp effektus (negyed hullám)

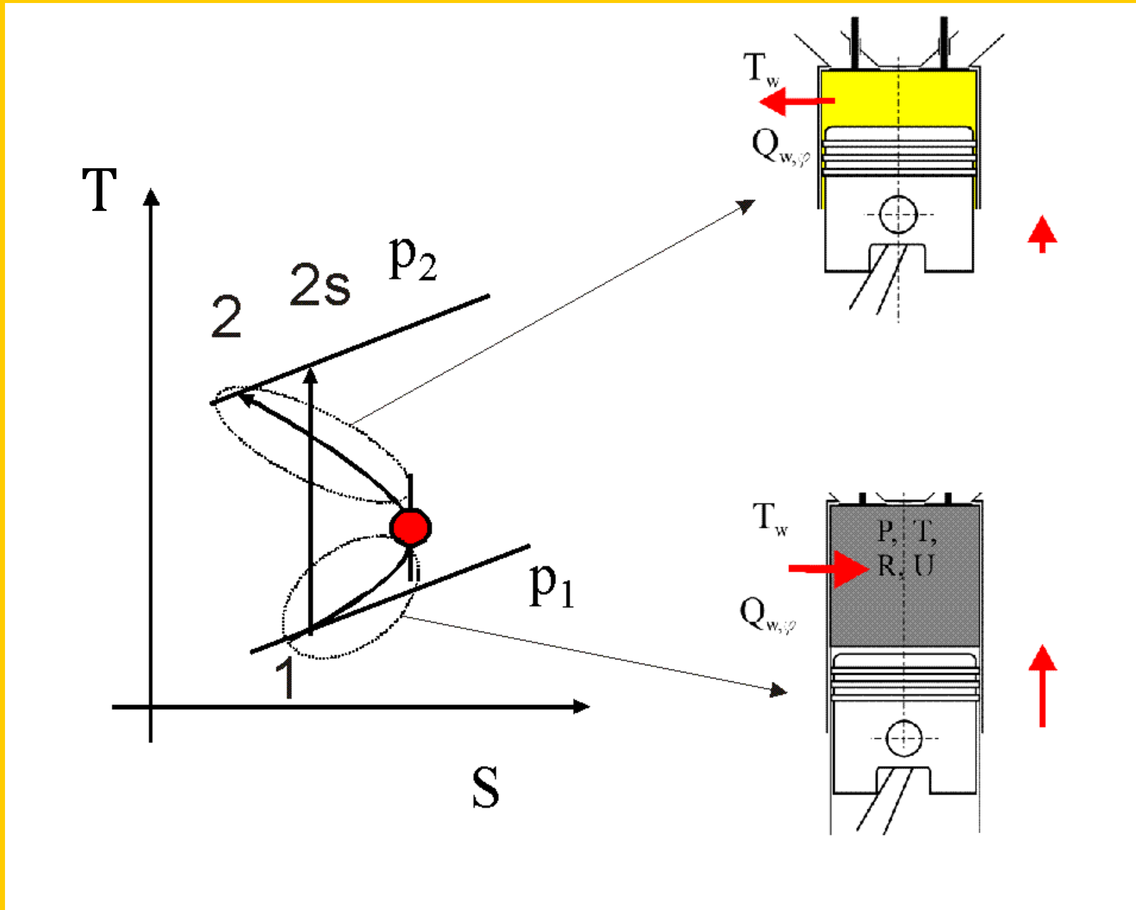
$$f = \frac{a}{4L}$$



1. Turbótöltő kompresszora; *2.* Levegő–levegő visszahűtő; *3.* Kiegyenlítőtartály; *4.* Rezonátor; *5.* Rezonátortartály; *6.* Szívócsatorna; *7.* Motorhenger

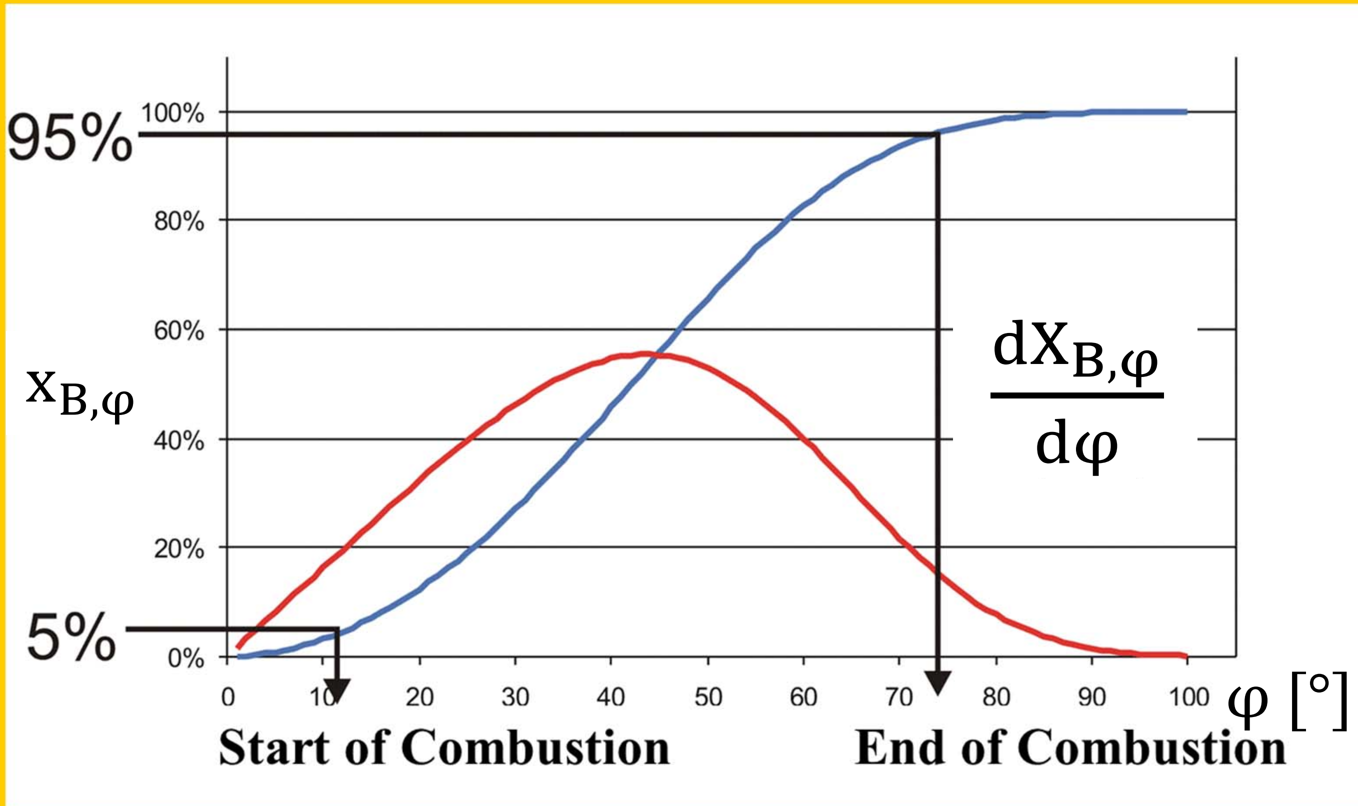
The internal losses

Heat transfer



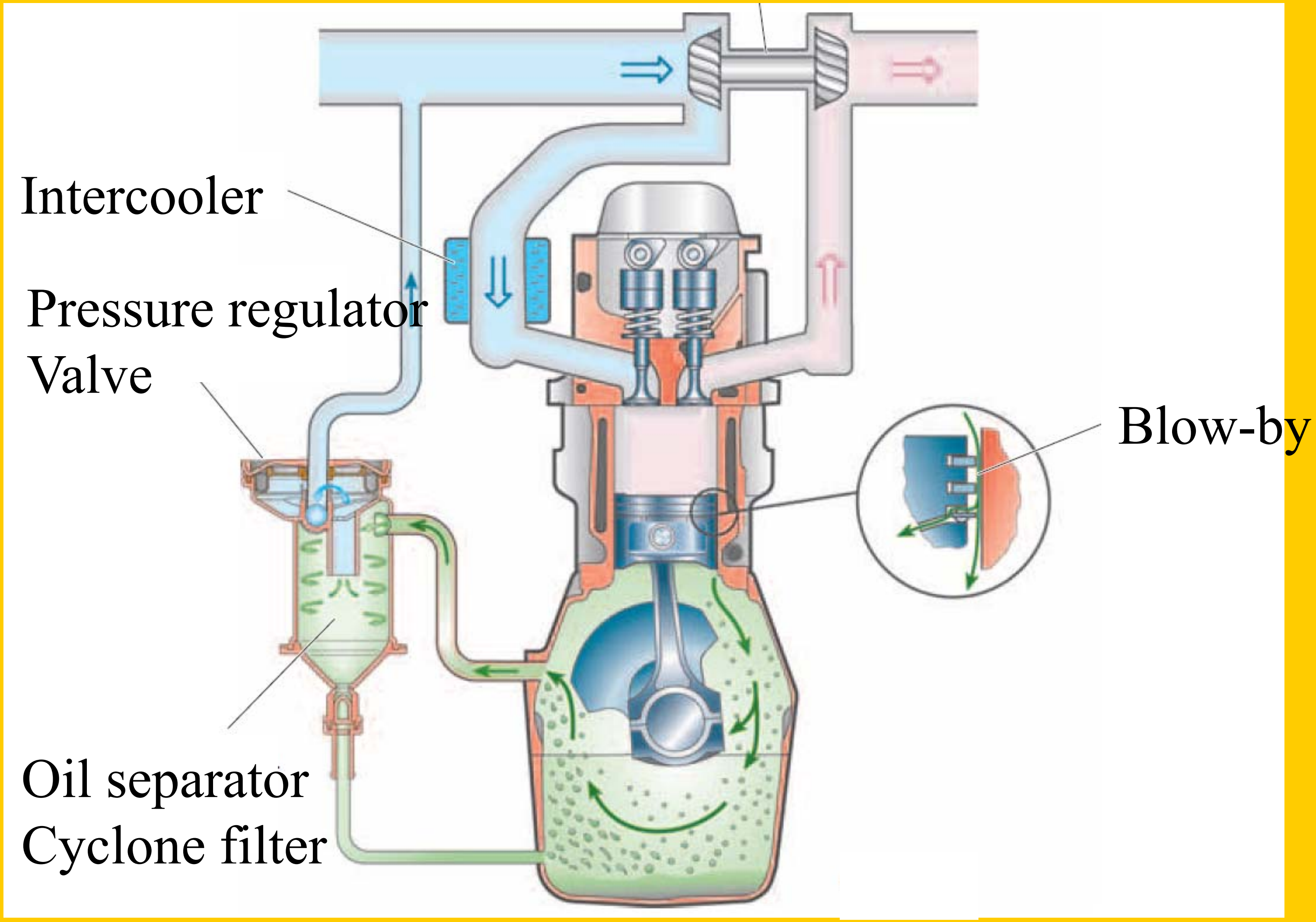
The internal losses
Limited Flame Propagation

Burned Fuel Ratio: $x_{B,\varphi} = \frac{m_{Fuel}}{m_{\varphi,Burned\,Fuel}}$

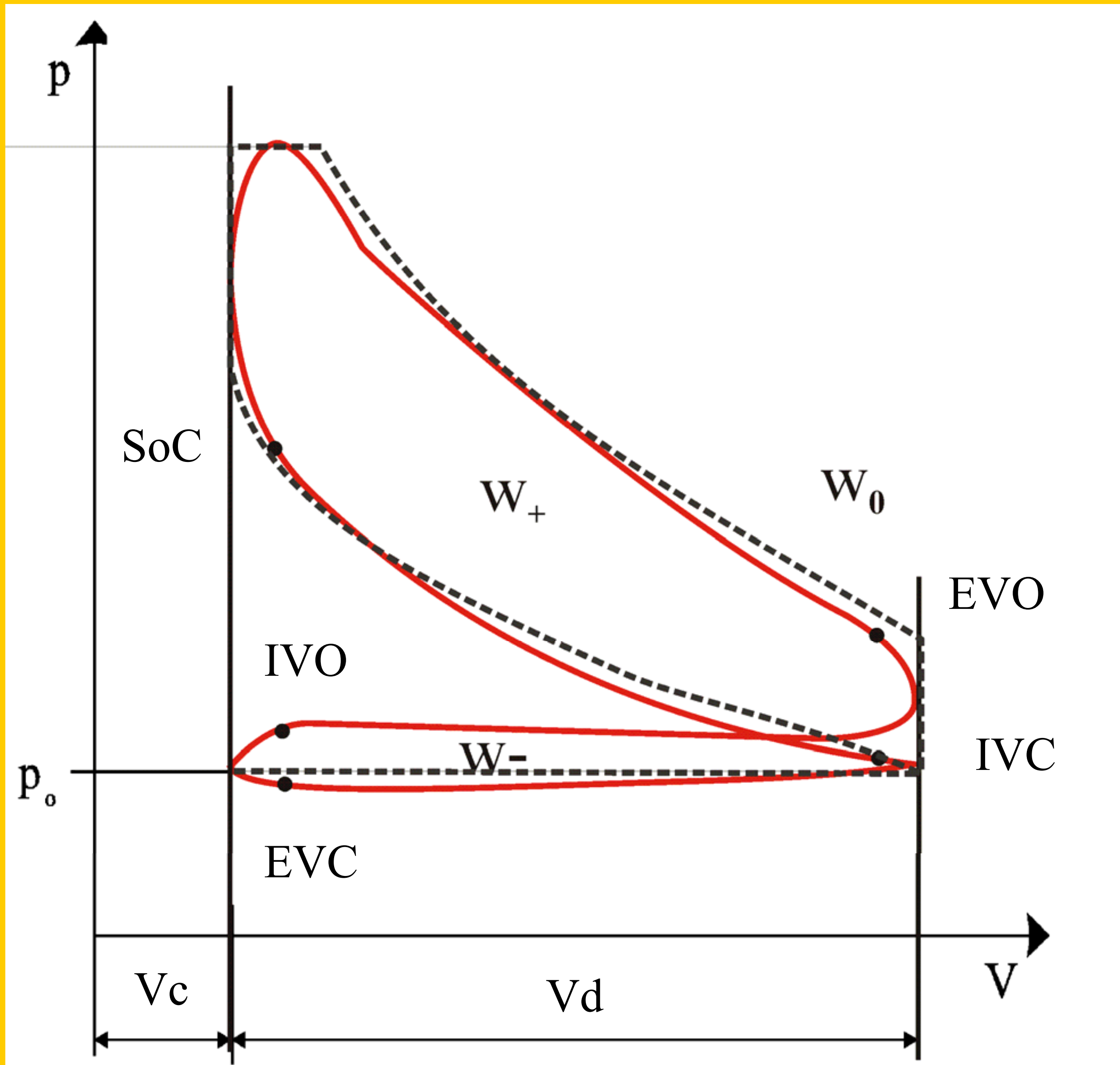


The internal losses

Gas losses (Blow-by)



Actual cycles of internal combustion engines



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

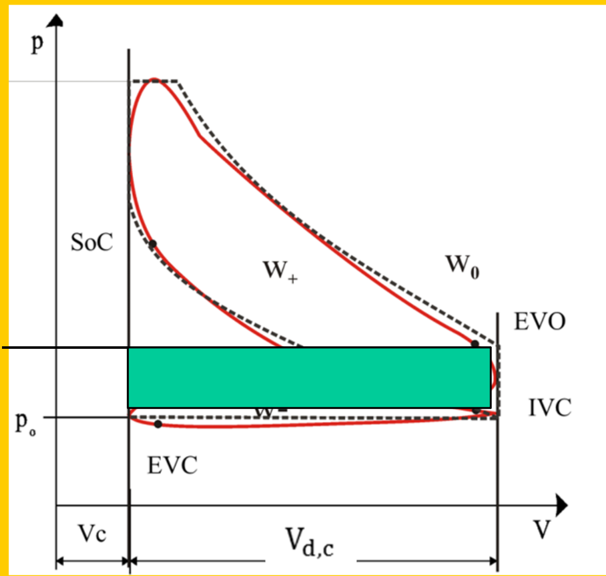
The Indicated Parameters

$$\mathbf{W_i = W_+ + W_- = IMEP * V_{d,C}}$$

$$\eta_i = \frac{W_+ + W_-}{q_{in}} = \frac{W_i}{q_{in}}$$

$$P_i = W_i * z * n * i$$

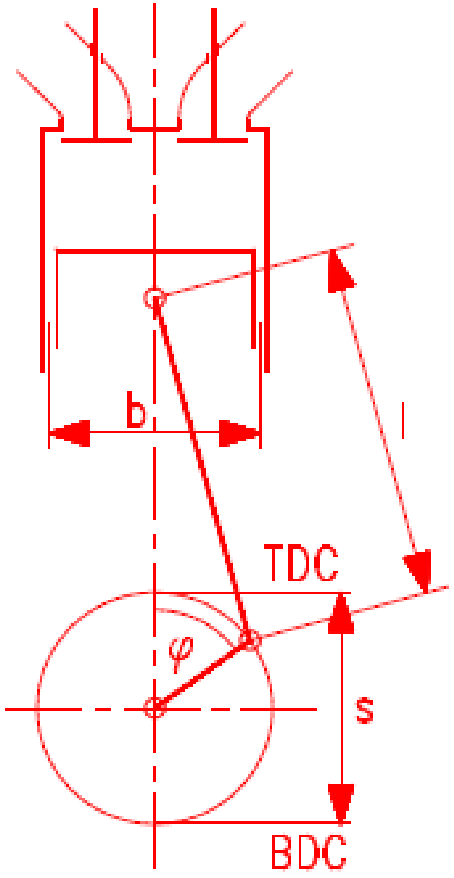
IMEP



IMEP=Indicated Mean Effective Pressure
z – Number of the Cylinders
n – Speed
i – work number (4 stroke -> 0,5)

$$P_i = IMEP * V_{d,C} * z * n * i$$

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)

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

Clearance volume V_c $[cm^3]$

Cylinder volume $V = V_d + V_c$

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 :

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

Volumetric efficiency :

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

ρ_i : fuel-air mixture density in the intake manifold

Indicated efficiency :

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

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

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

Brake thermal efficiency :

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

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 :

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

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

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

Excess air factor :

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

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

μ : stoichiometric air-fuel ratio

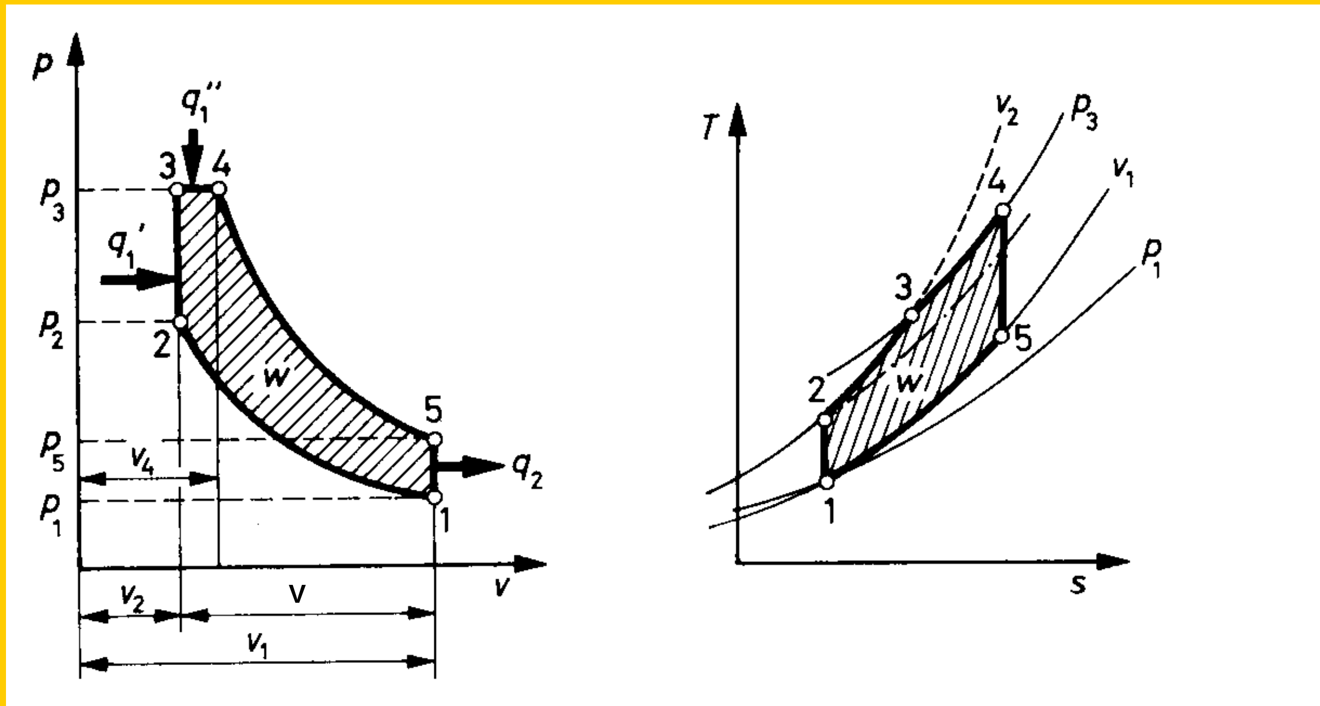
Brake specific fuel consumption (bsfc):

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

Mean piston speed :

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

The Dual-combustion cycle

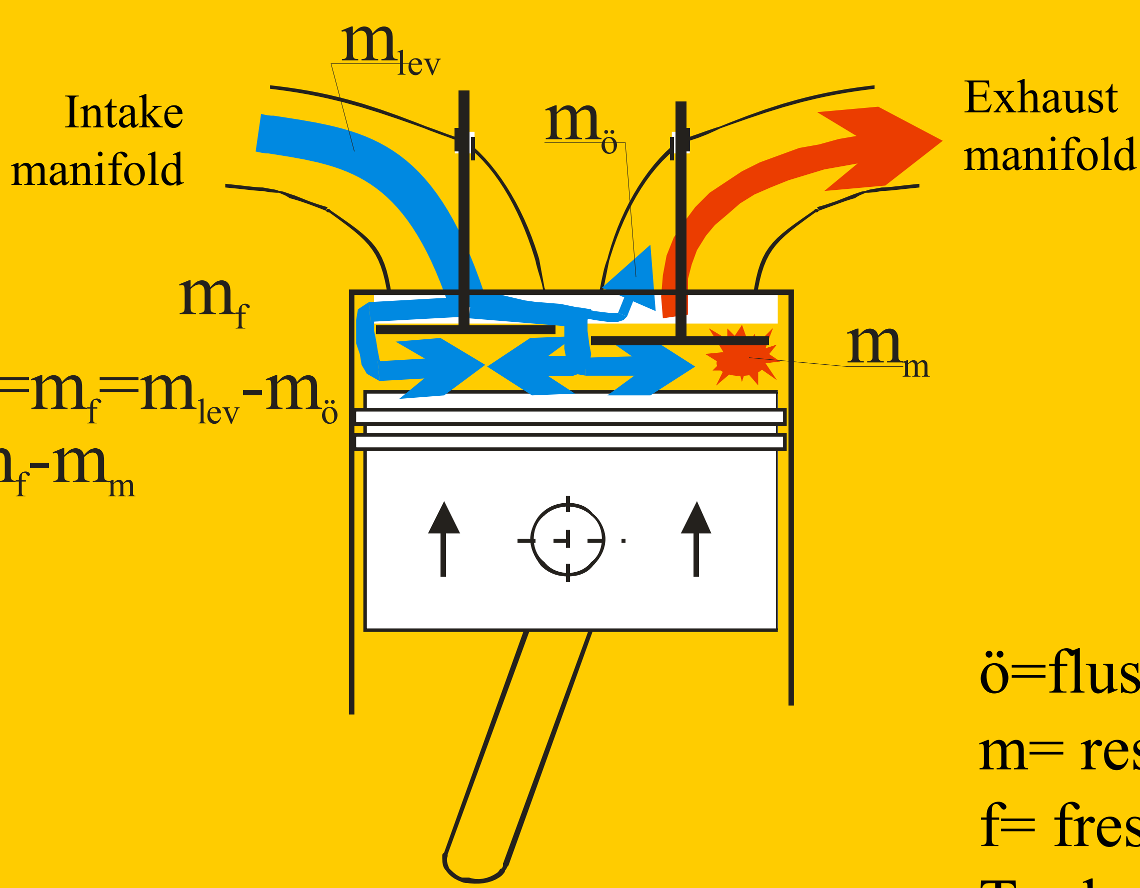


$$\eta = \frac{-\sum W}{Q_1} = \frac{\sum Q}{Q_1} = \frac{c_p(T_3 - T_2) + c_v(T_4 - T_3) - c_v(T_5 - T_1)}{c_p(T_3 - T_2) + c_v(T_4 - T_3)} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \cdot \frac{\rho^{\kappa} \cdot \lambda - 1}{(\lambda - 1) + \kappa \cdot \lambda \cdot (\rho - 1)}$$

$$\rho = \frac{\boldsymbol{v}_4}{\boldsymbol{v}_3} \qquad \lambda = \frac{\boldsymbol{p}_3}{\boldsymbol{p}_2}$$

- **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

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
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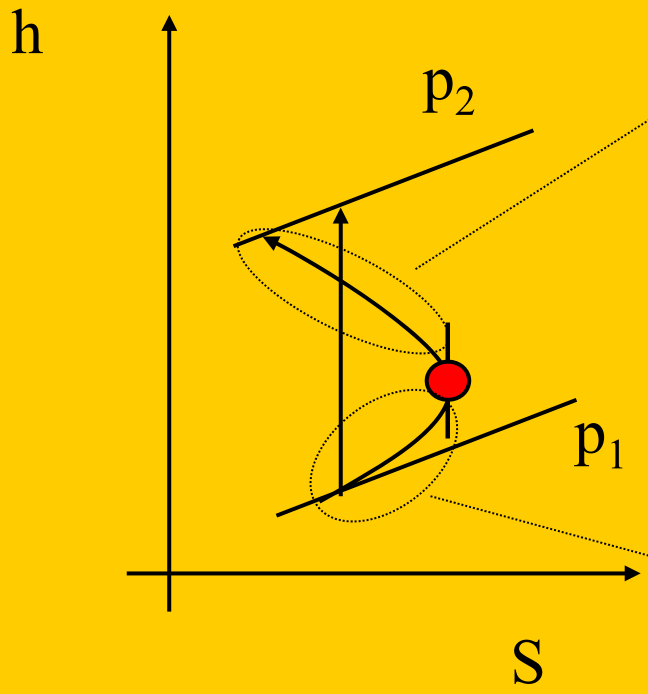
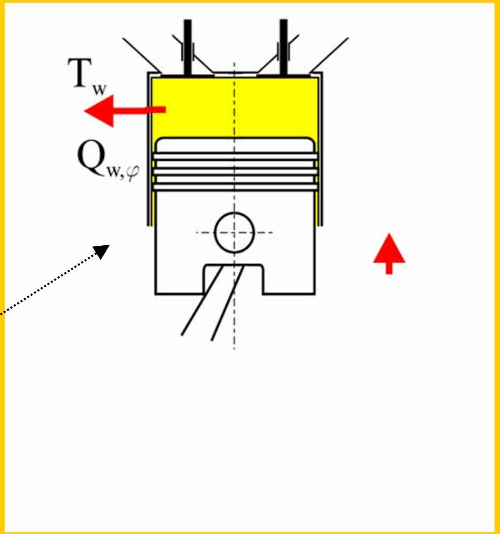
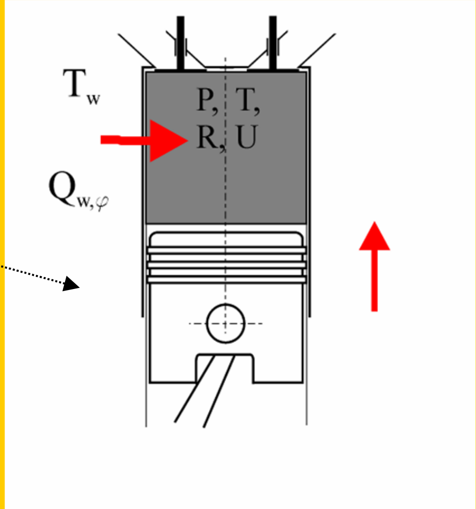
  - Gas losses (Blow-by)
  - Friction

Compression

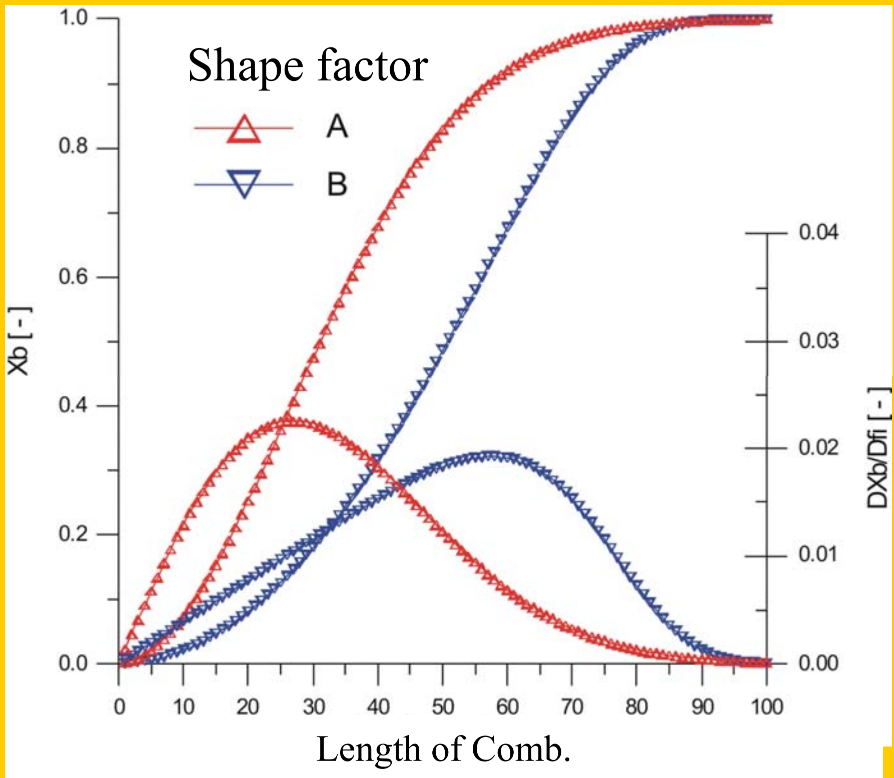


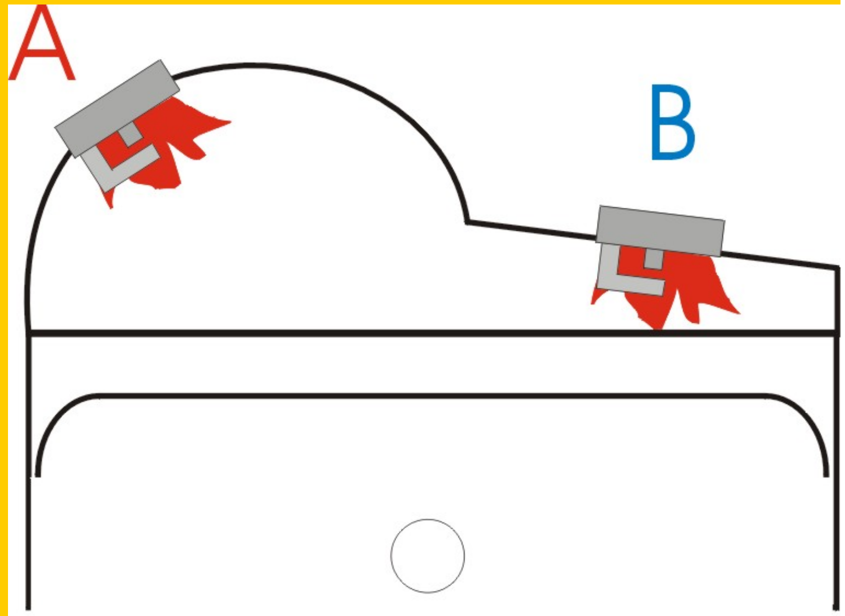




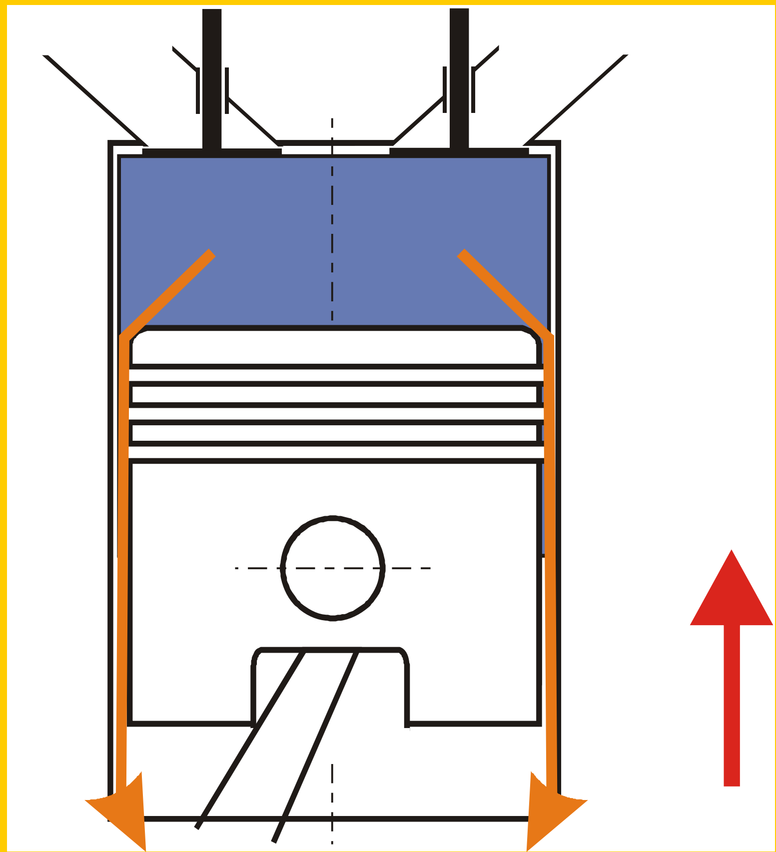
- **LOSSES IN INTERNAL COMBUSTION ENGINES**:
  - Intake and exhaust losses (Fresh mixture (air) loss, Rest gases, valve loss, ...)
  - Heat transfer (non isentropic) compression
  - **Incomplete combustion**
  - **Limited combustion speed (+ Heat loss of Combustion)**
  - Gas losses (Blow-by)
  - Friction

Influence of Geometry on the Combustion process





Gas losses (Blow-by)



- <u>Mechanical losses</u> is made up partly of the friction losses of moving parts and partly of the energy needed to drive auxiliary equipment (oil pump, water pump, cooling fan, metering pump, ignition system, etc.).
- Mechanical losses can be characterised by the mechanical efficiency:

$$\eta_m = \frac{W_e}{W_i}$$

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 (H2, 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**

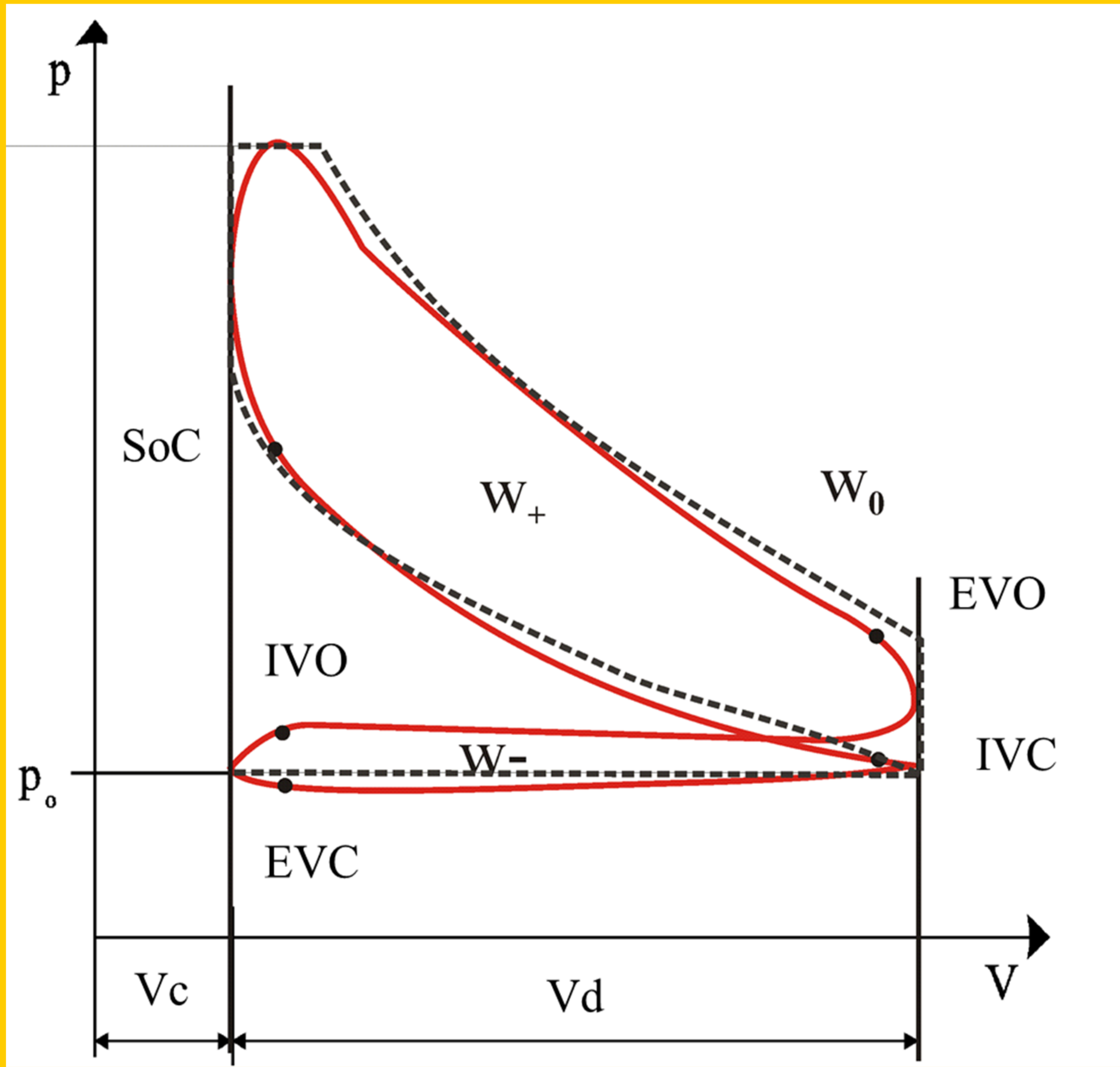
Considering that the blow down process takes time the exhaust valve must be opened before the BC thus the expansion stroke will be uncompleted and work will be lost.

- **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

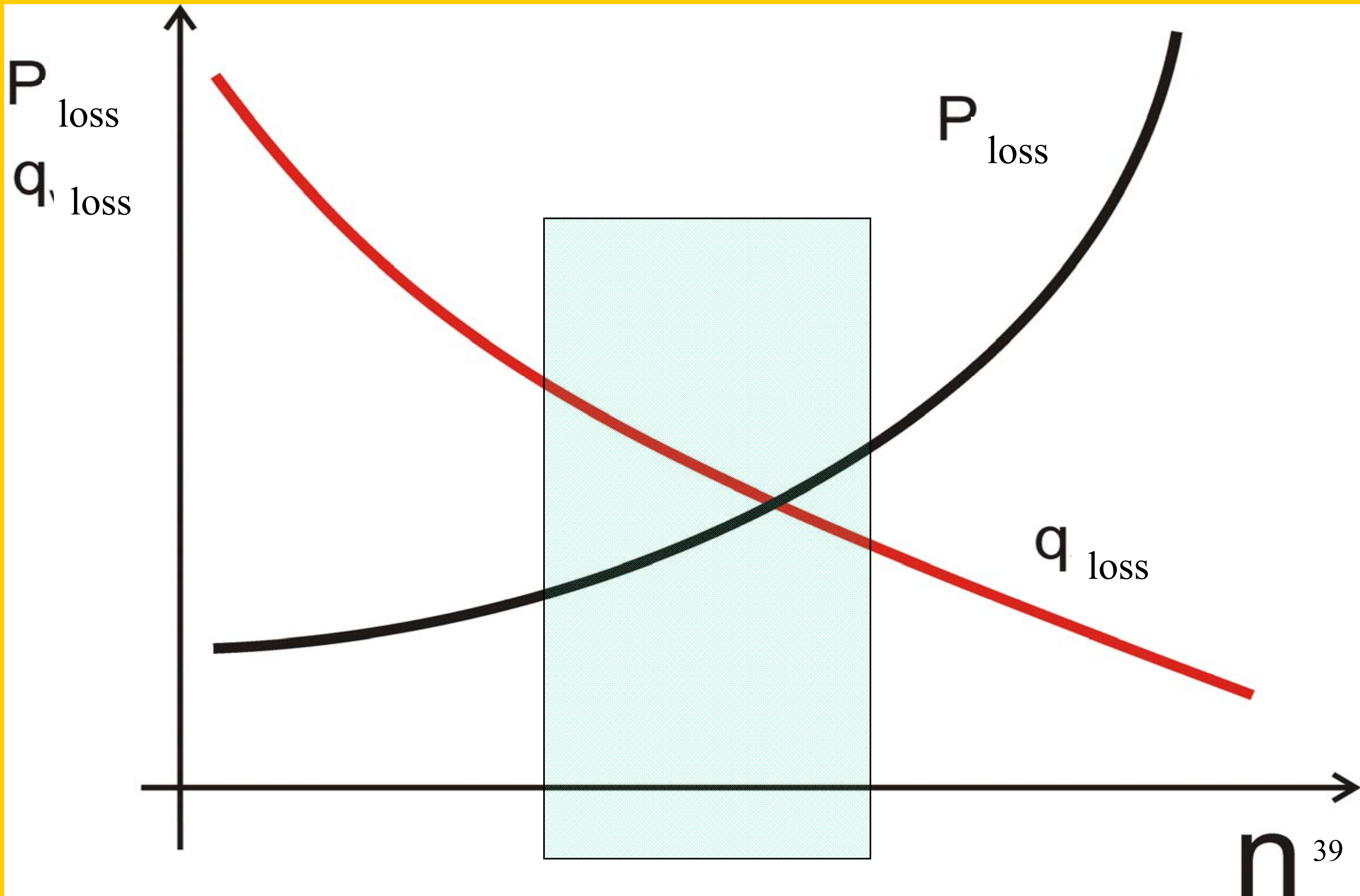
The most significant source of this loss is the friction between the piston skirt, rings and the cylinder (about 60-80% of the total frictional work). Usually it is higher in diesel engines, because of the stronger piston rings. The other sources of frictional losses are the crankshaft, camshaft, valve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



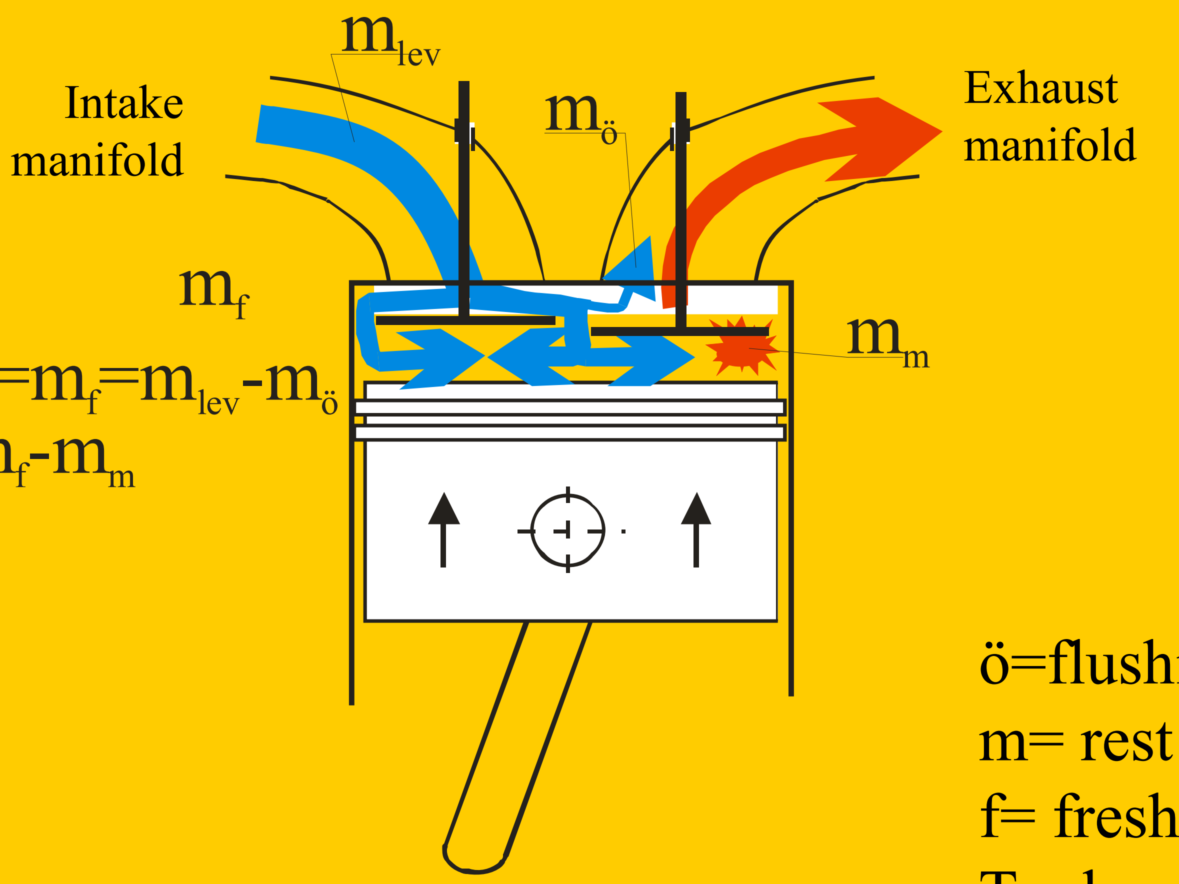
Theoretical (dot line) and real indicator diagramm (cont. line)

Characteristic of ICE Engines

Losses in the Function of the Speed



aerodynamic losses during intake



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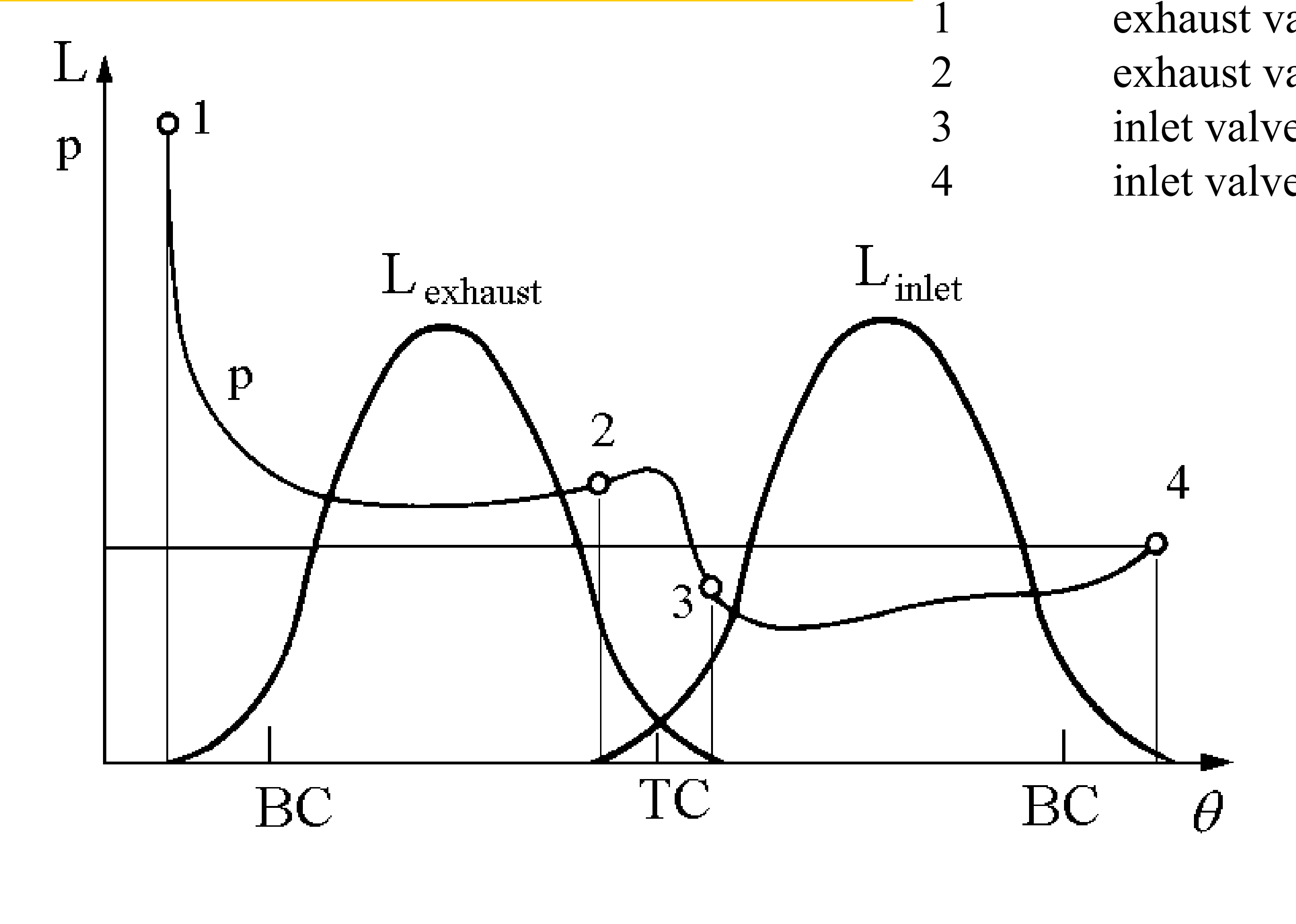
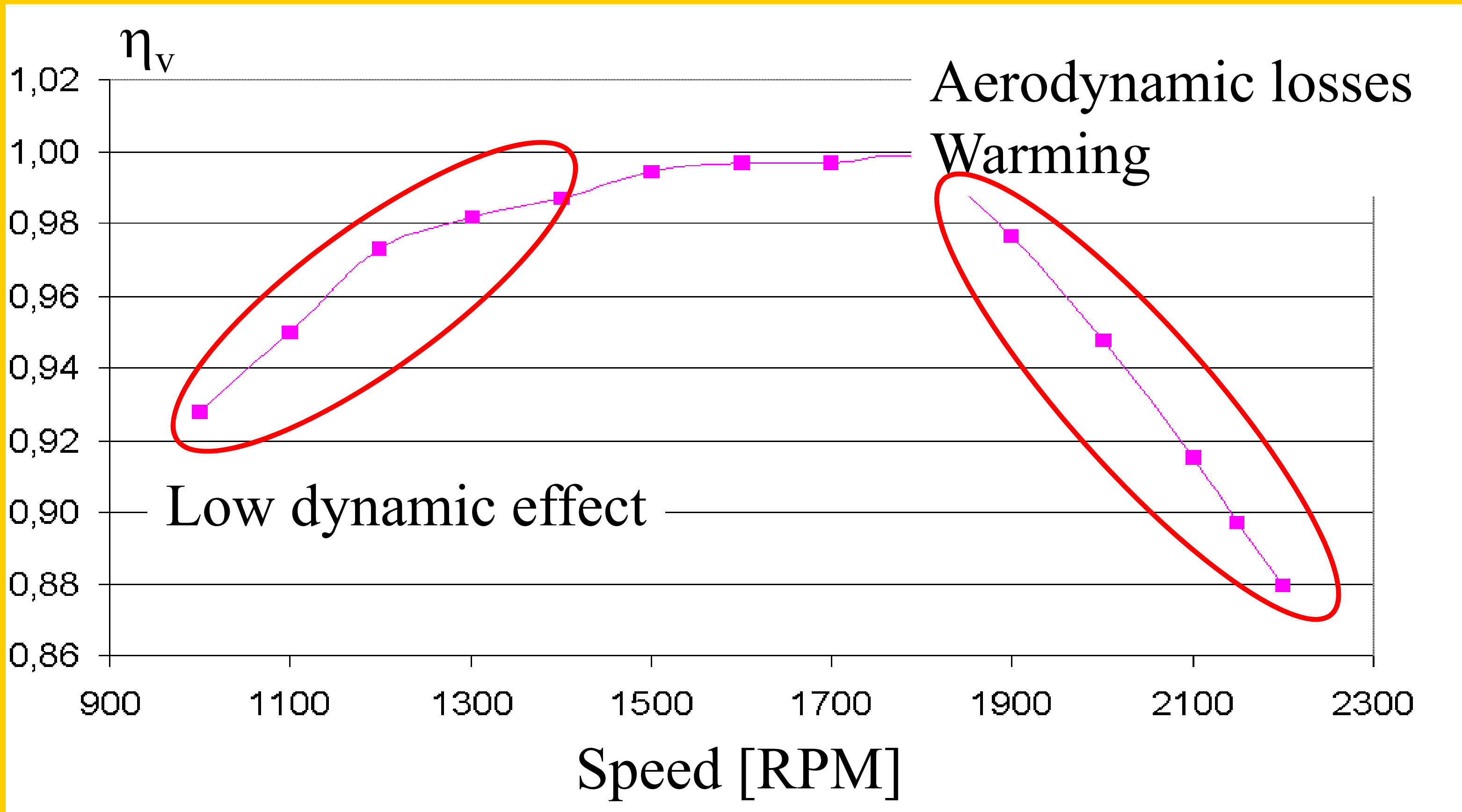
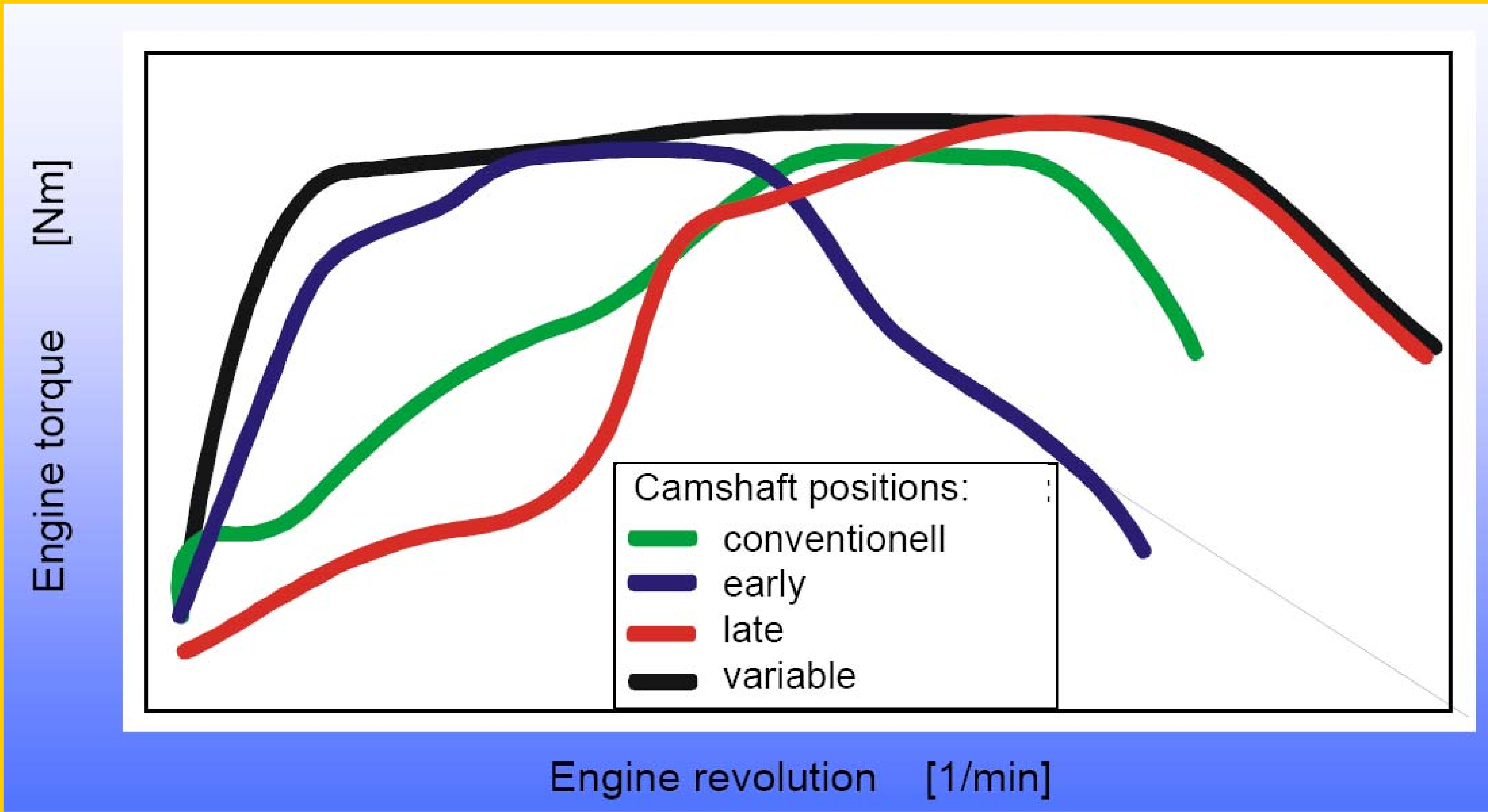


L	valve lift
BC	bottom dead center
TC	top dead center
1	exhaust valve opens
2	exhaust valve closes
3	inlet valve opens
4	inlet valve closes

2-3: overlap period

Volumetric efficiency in the Function of the Speed



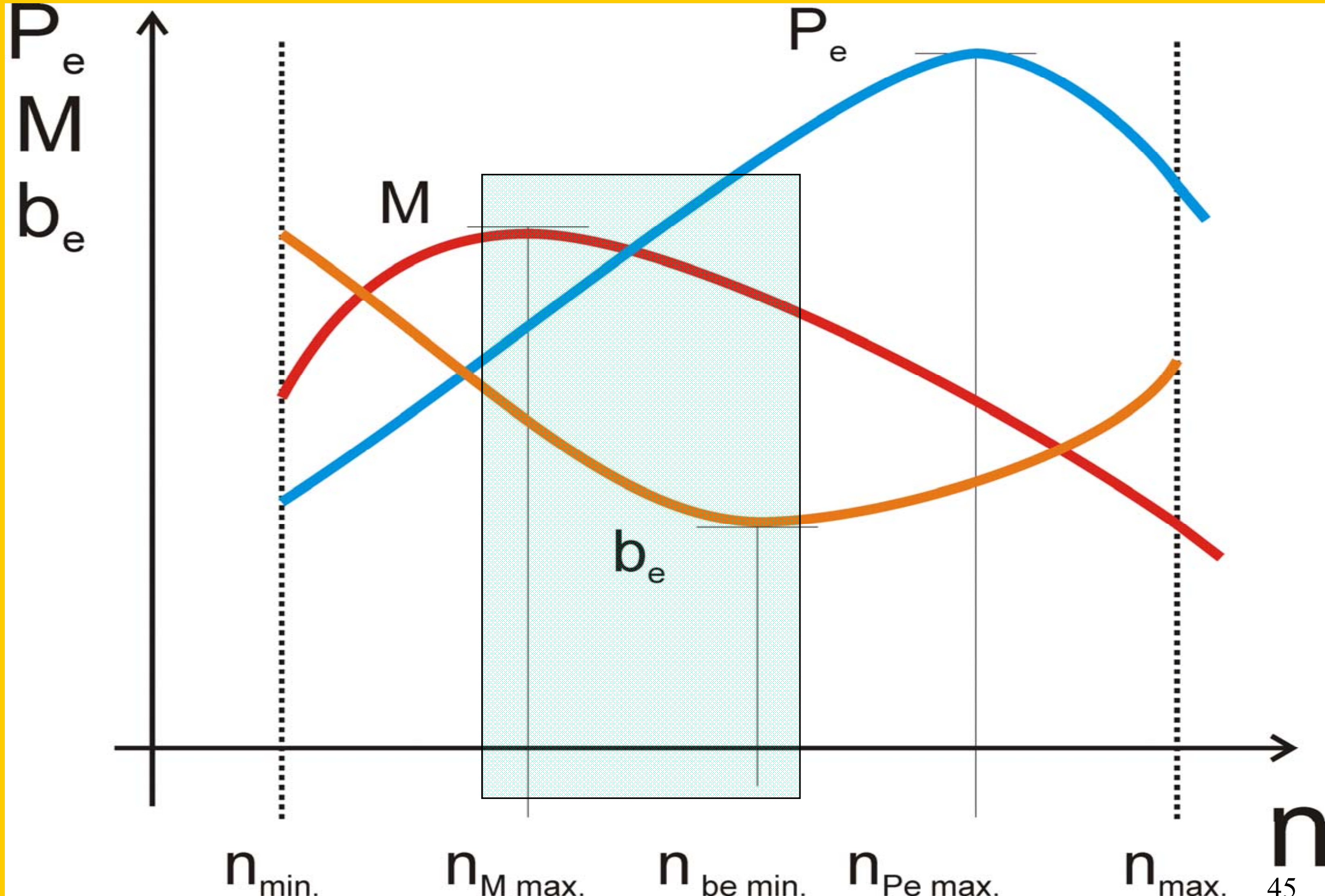

Engine torque [Nm]
Camshaft positions:
conventionell
early
late
variable
Engine revolution [1/min]

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

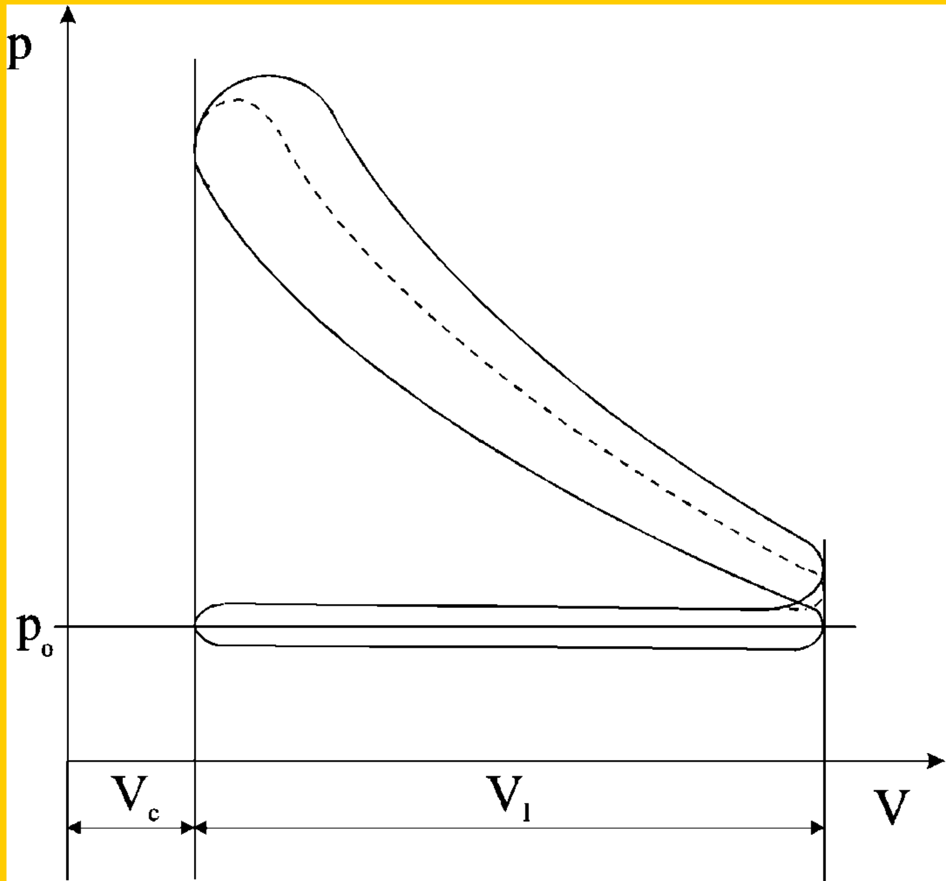


- $0\text{-}n_{min.}$:
  - Flywheel does not store enough energy,
  - Wrong mixing,
  - Big heat losses,
- $n_{min.}\text{-}n_{M\ max.}$:
  - Better mixing,
  - Growing volumetric eff.,
  - Decreasing Heat Losses,
- $n_{M\ max} - n_{be\ min.}$:
  - Decreasing volumetric eff.,
- $n_{be\ min.} - n_{pe\ max.}$:
  - 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)

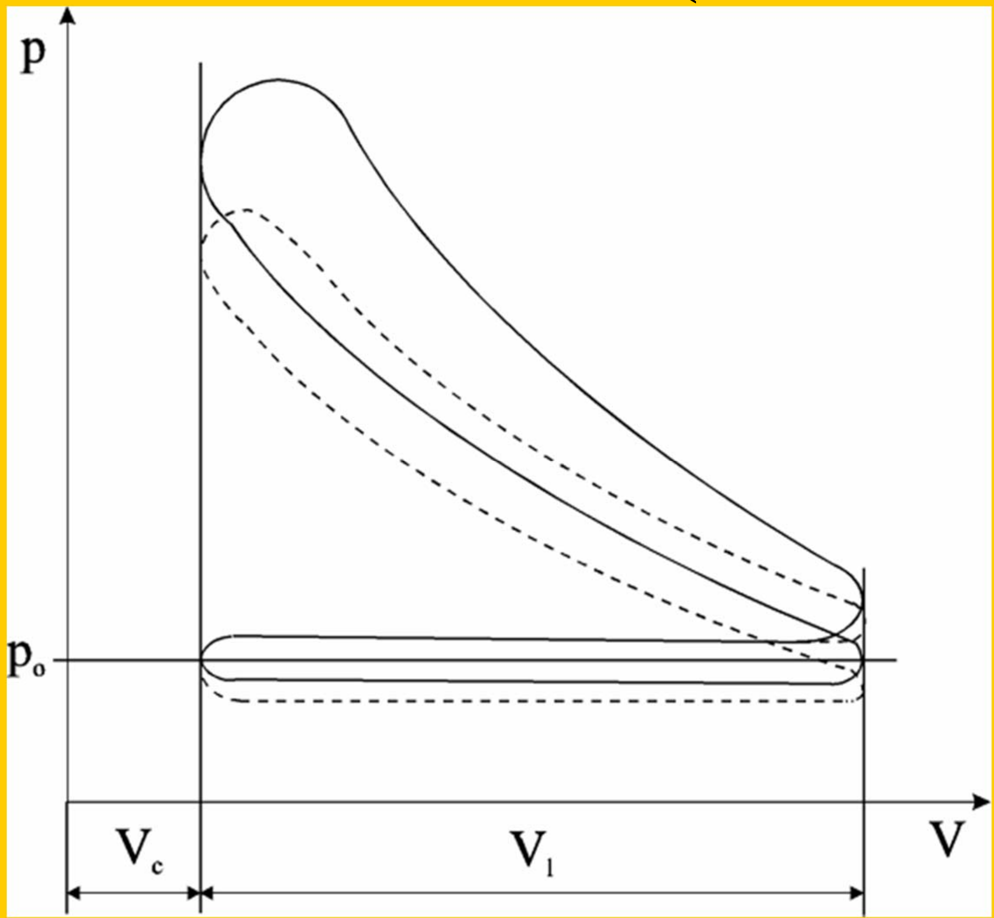
$\lambda \neq$ constant



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

Quantitative Control (SI ICE)

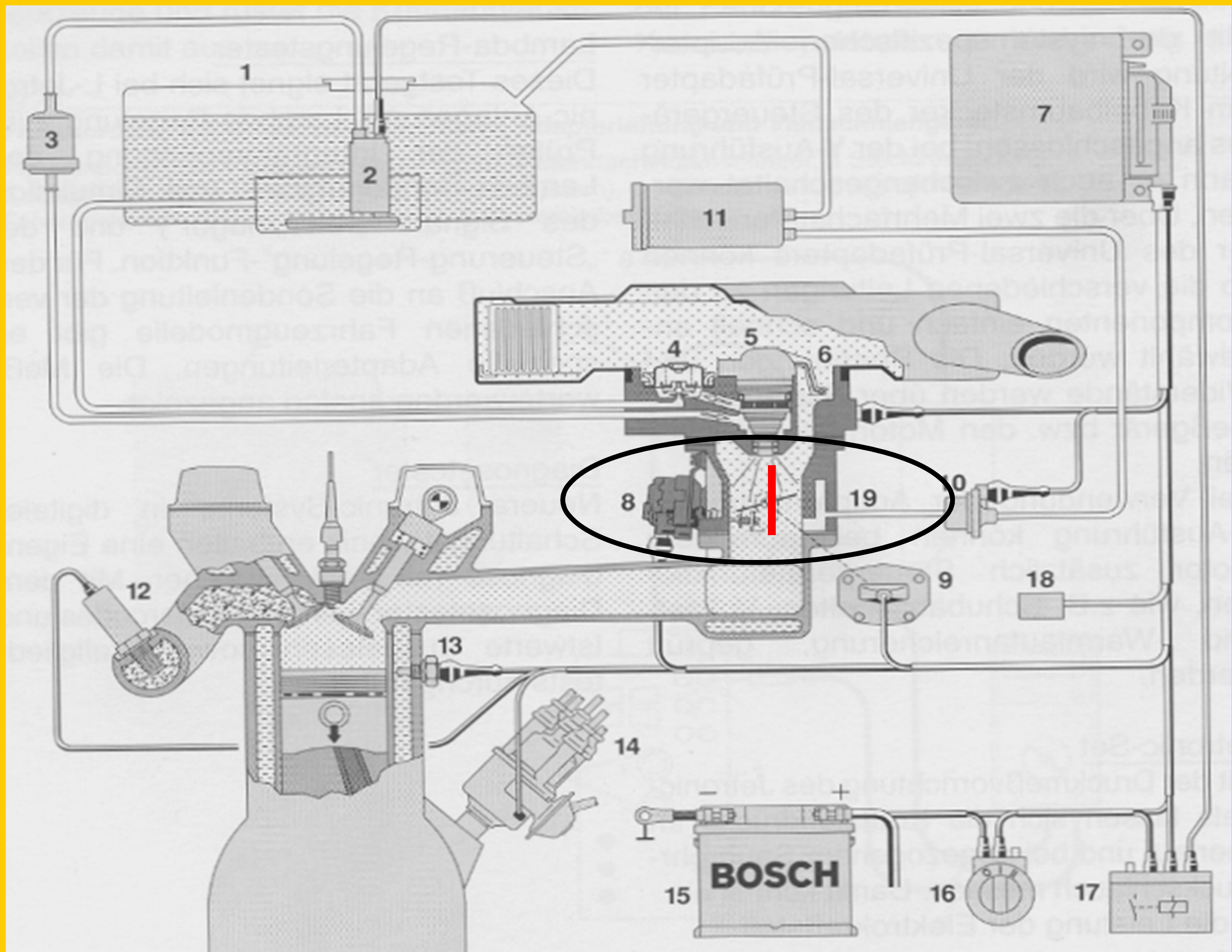
$\lambda \approx$ constant



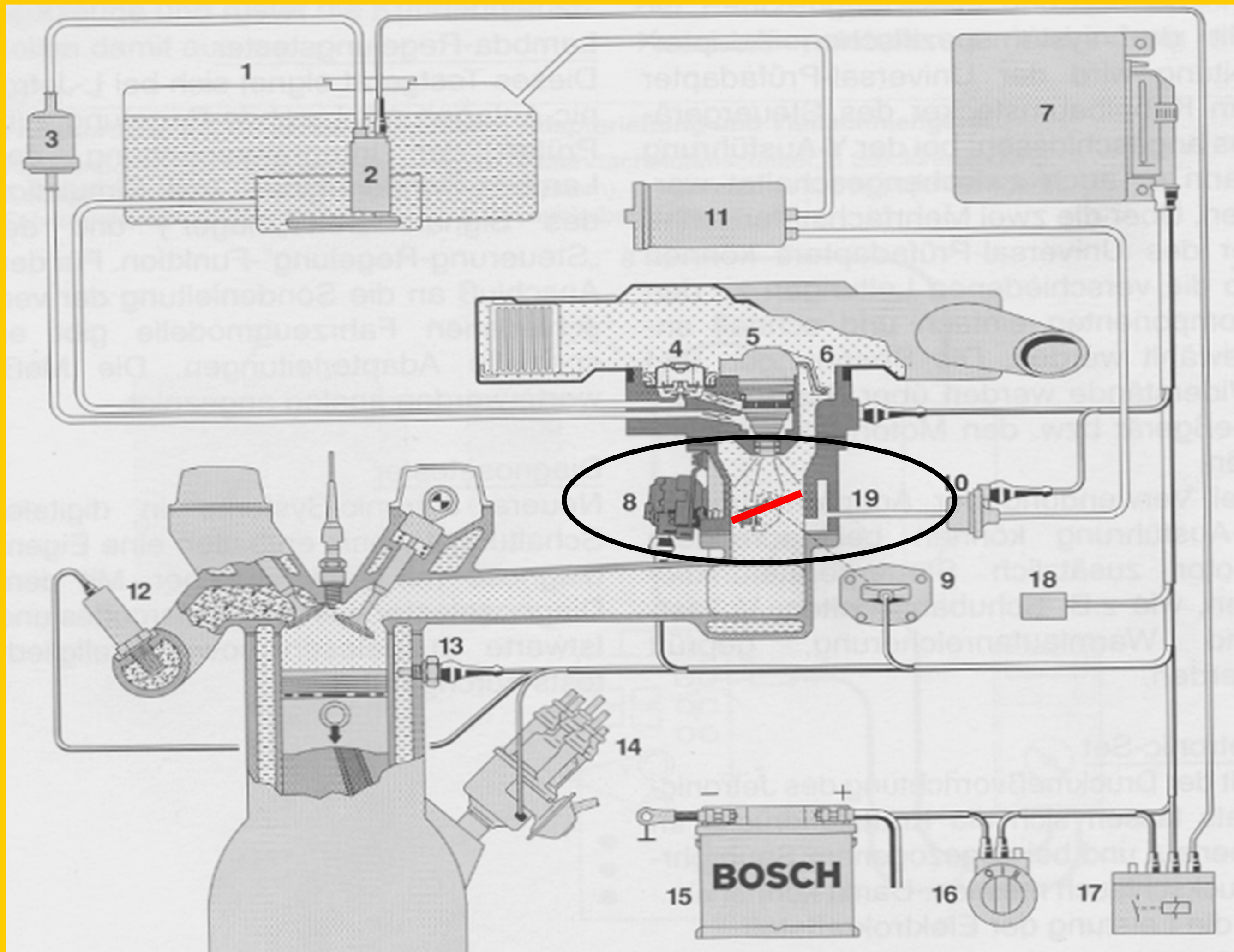
Control possess of spark ignition engines
(— full load (throttle is open) , - - - part load (throttle is partially closed)

Engine Maps
(part load characteristic)

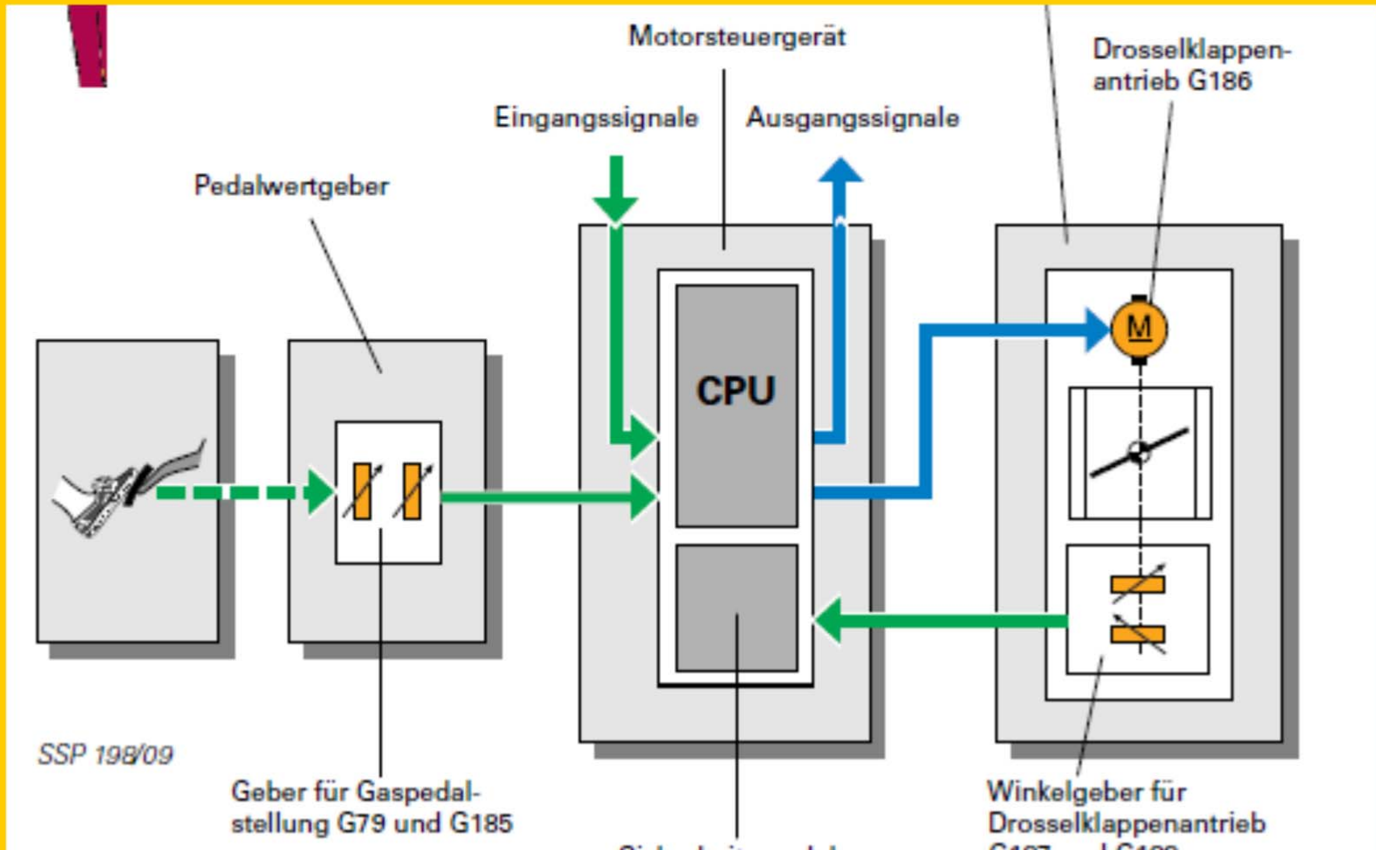
Quantitative control with throttle-valve, full-load



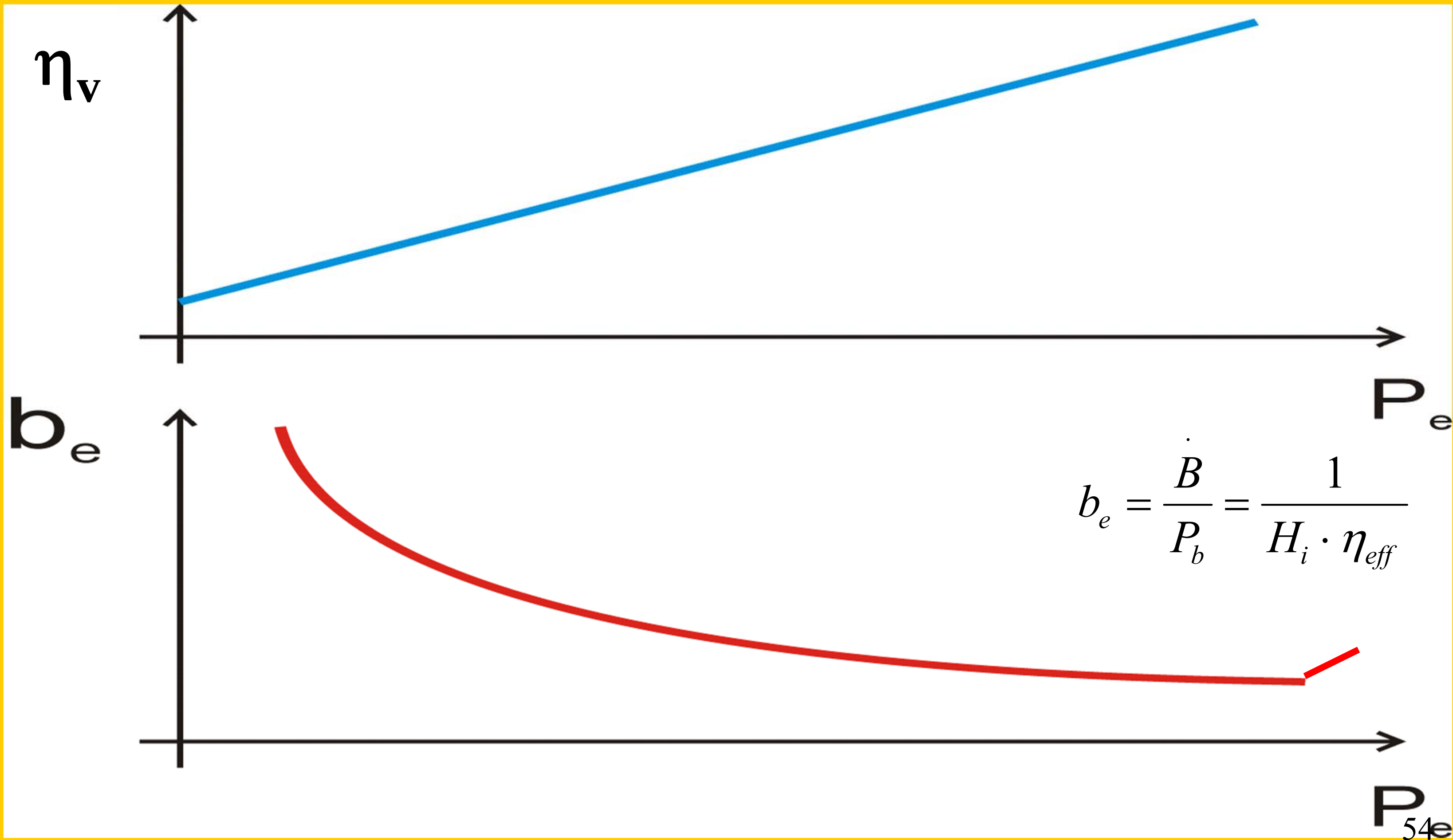
Quantitative control with throttle-valve, partial-load (05/11/21)



Electronic Throttle Control (ETC)

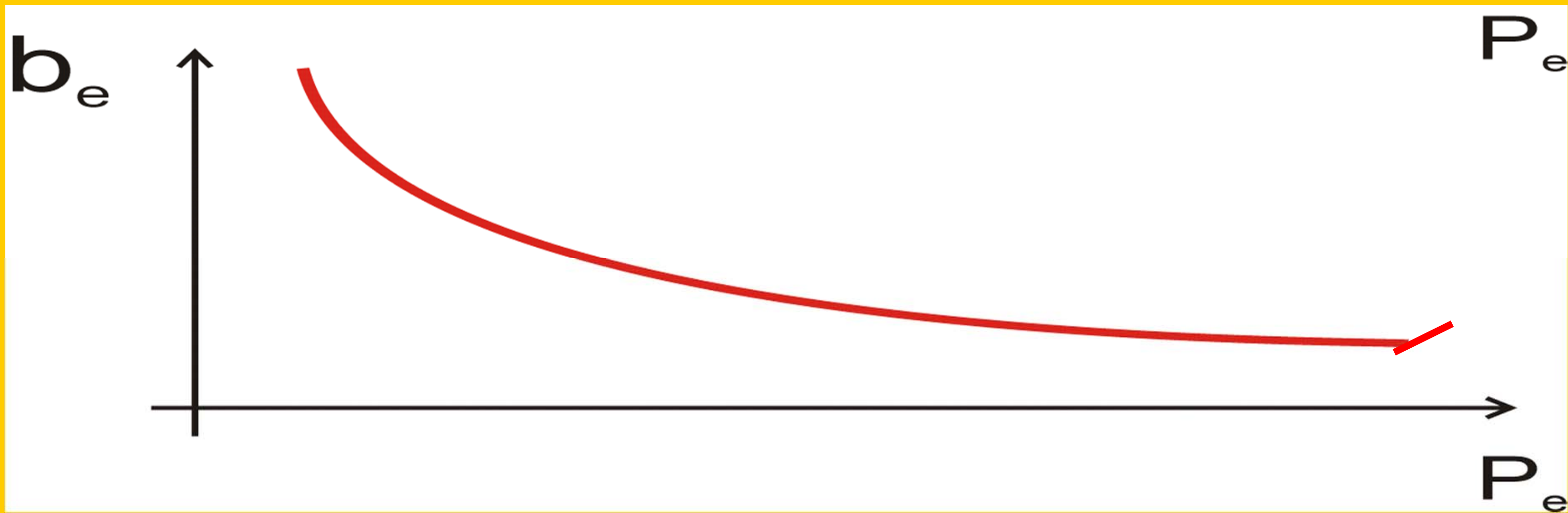


Volumetric efficiency and the Brake Specific Fuel Consumption at different loads, constant speed (S.I. ICE)

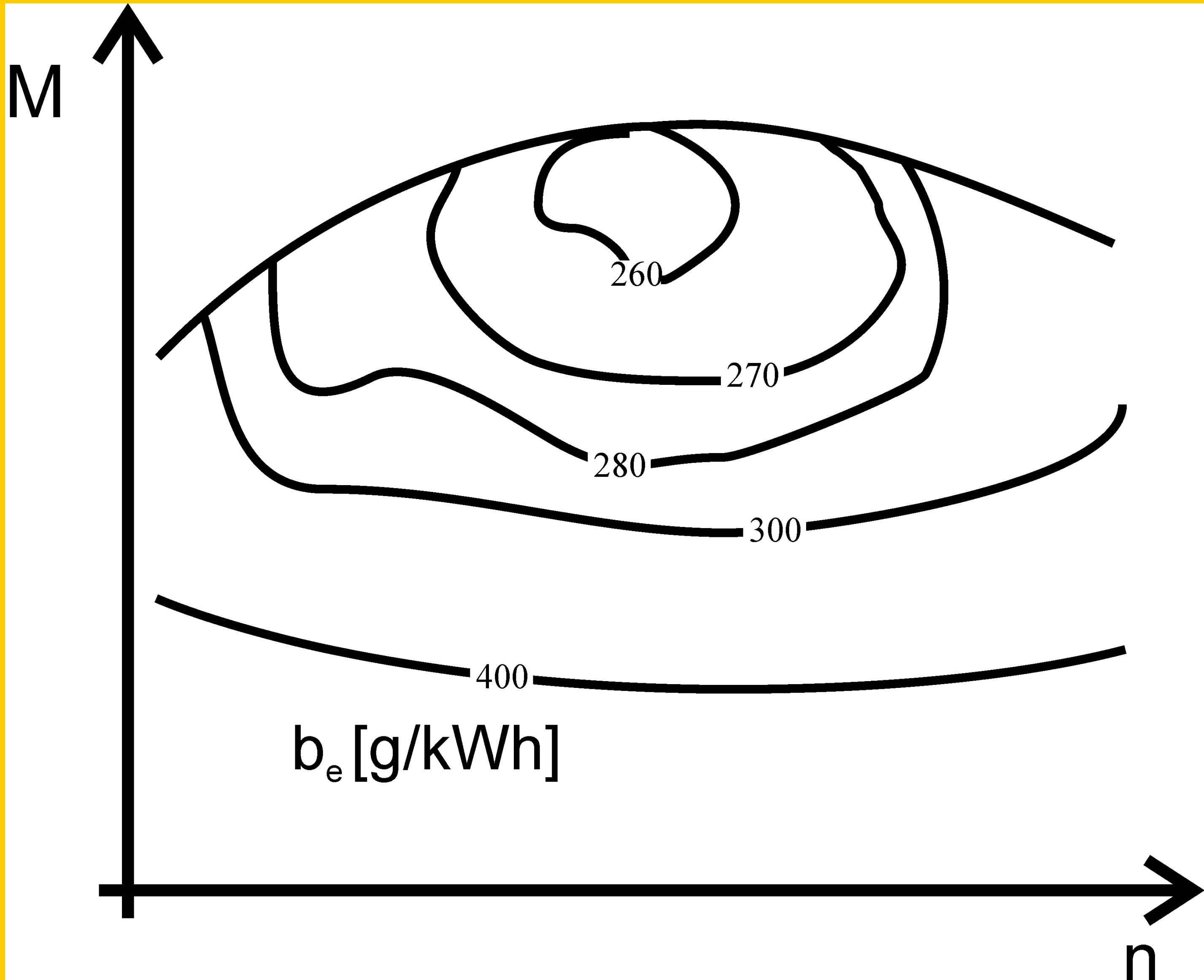


$$b_e = \frac{\dot{B}}{P_b} = \frac{1}{H_i \cdot \eta_{eff}}$$

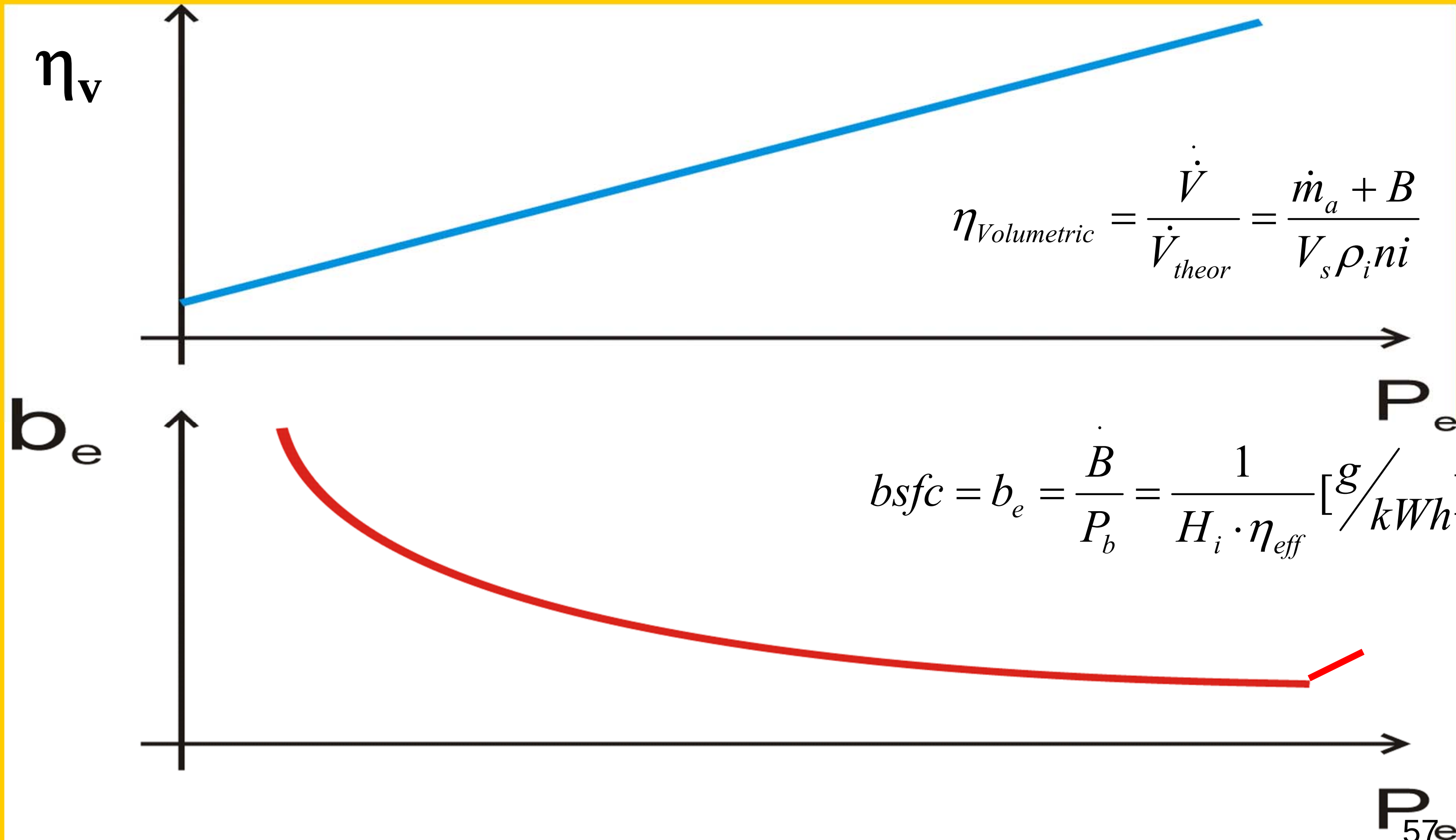
Brake Specific fuel consumption at different loads, constant speed



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



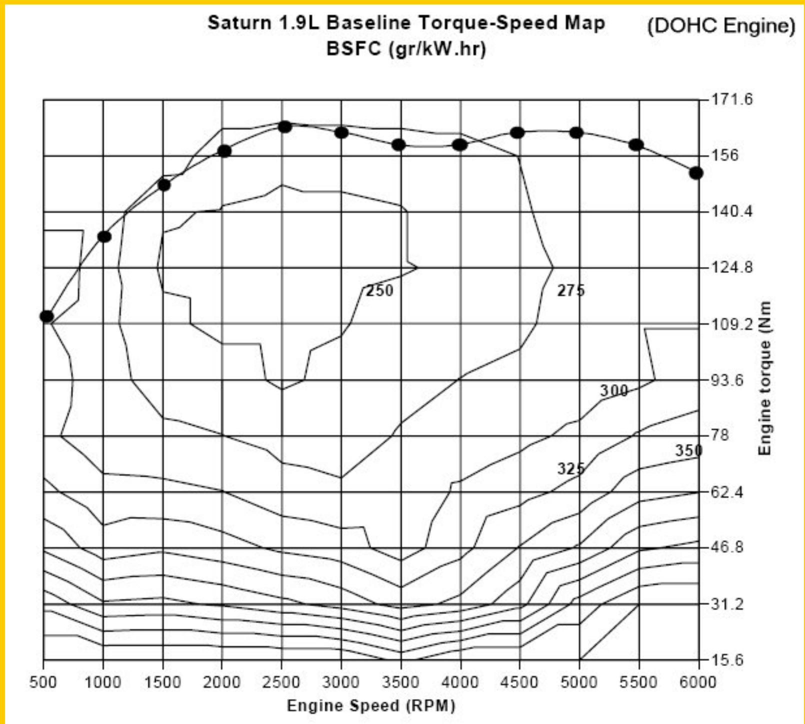
Volumetric efficiency and the Brake Specific Fuel Consumption at different loads, constant speed

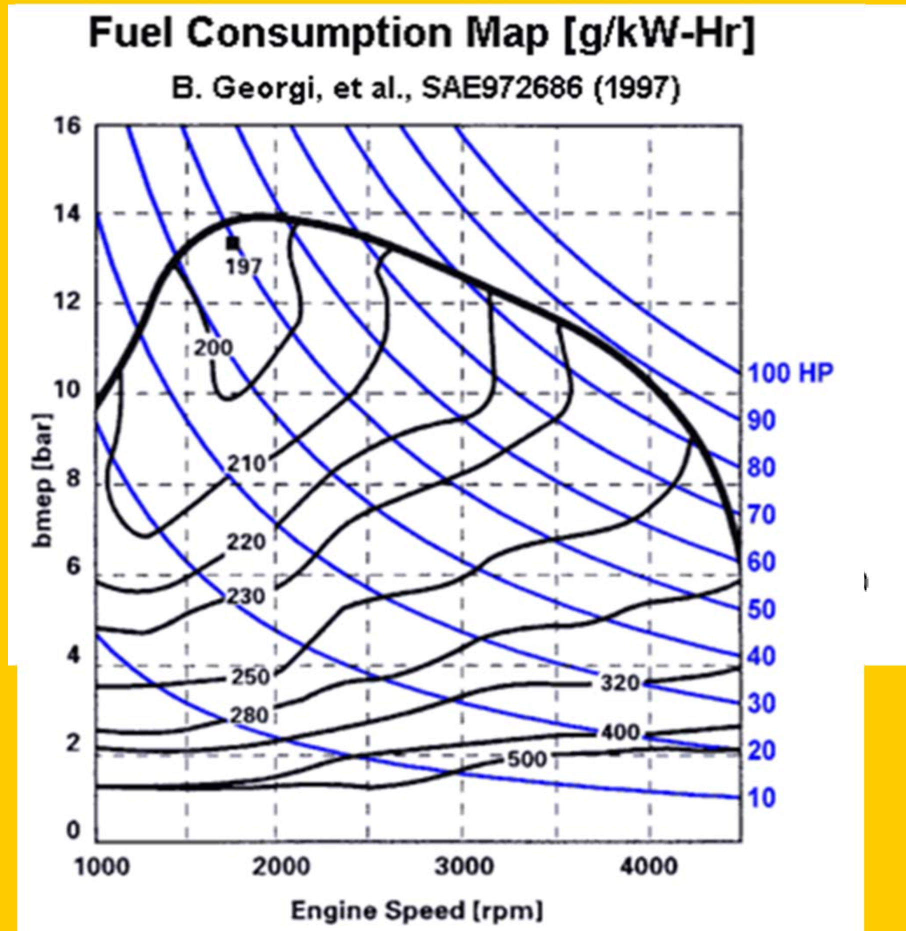


$$\eta_{Volumetric} = \frac{\dot{V}}{\dot{V}_{theor}} = \frac{\dot{m}_a + B}{V_s \rho_i n i}$$

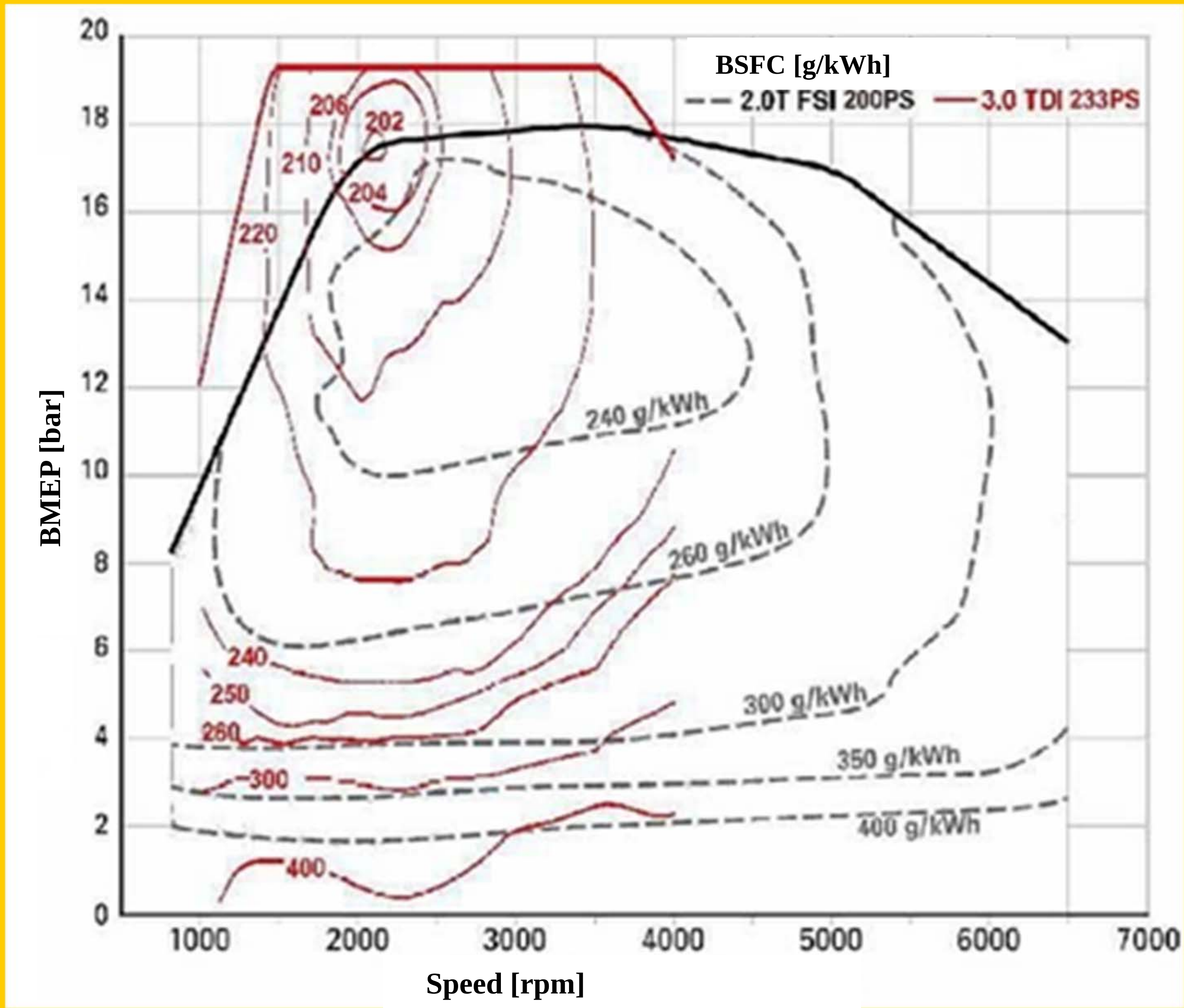
$$bsfc = b_e = \frac{\dot{B}}{P_b} = \frac{1}{H_i \cdot \eta_{eff}} [g/kWh]$$

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



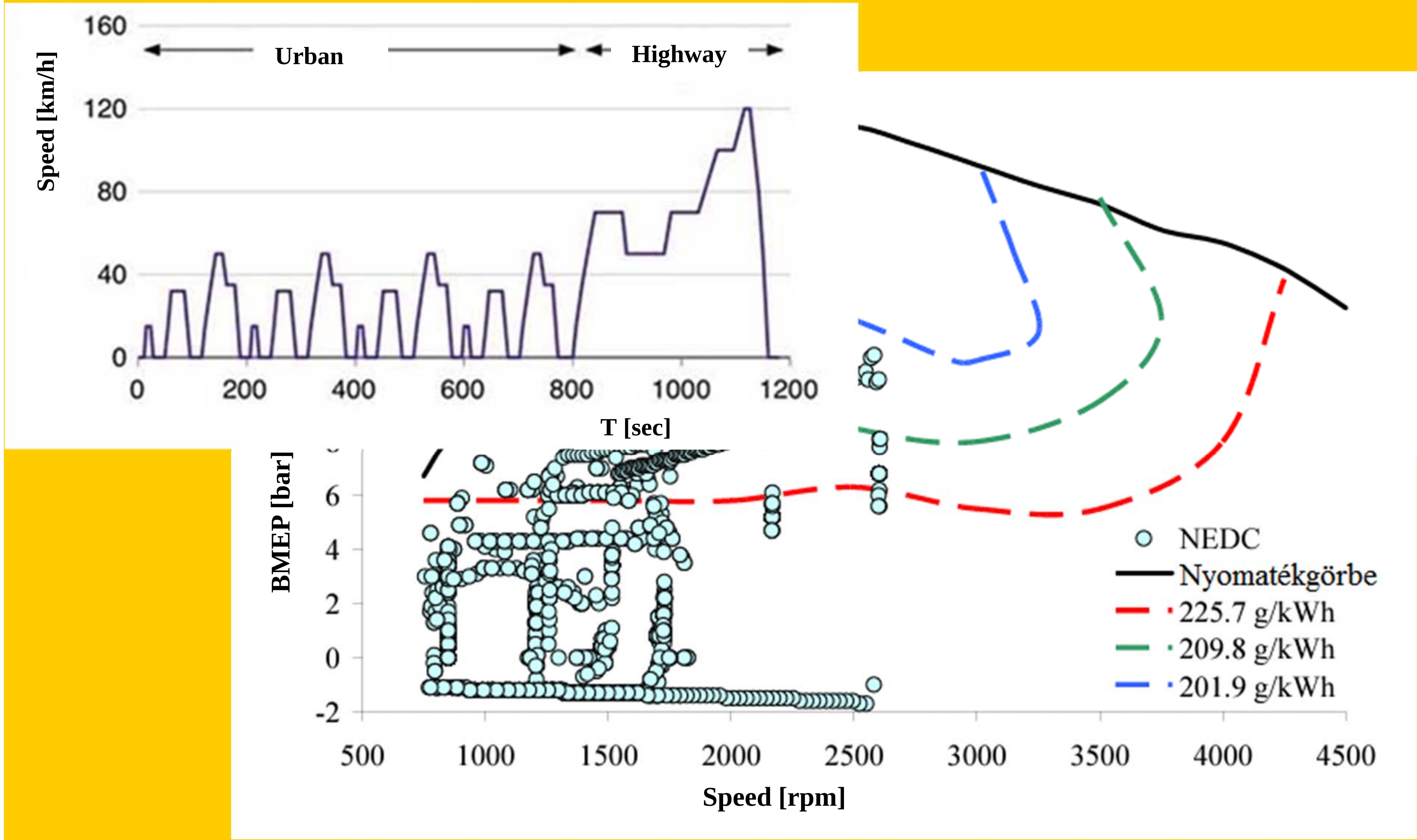


TDI 1.9L ALH 1999.5-2003



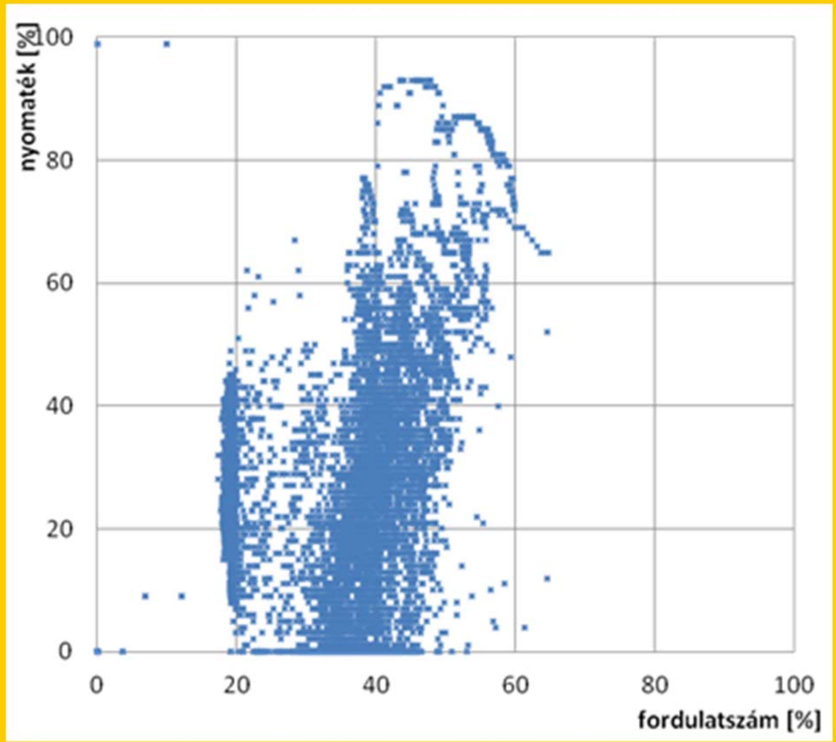
BSFC [g/kWh]
2.0T FSI 200PS
3.0 TDI 233PS
BMEP [bar]
Speed [rpm]
20
18
16
14
12
10
8
6
4
2
0
1000
2000
3000
4000
5000
6000
7000
206
202
210
204
220
240
250
260
300
400
240 g/kWh
260 g/kWh
300 g/kWh
350 g/kWh
400 g/kWh

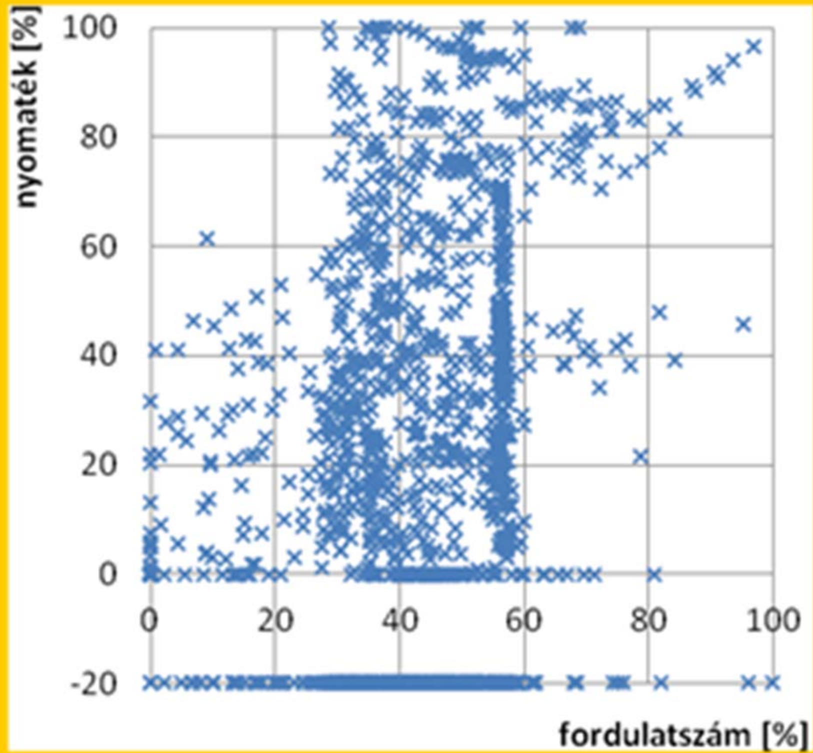
EcoModder



NEDC cycle operating conditions over the operating range of the 1.6 TDI engine

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





Real Loads

WHTC Loads

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

Hybridisation

Vehicles with conventional internal combustion engines provide good performance and long range by exploiting the high energy density of their fuels. However, they also have the disadvantage of being fuel inefficient and polluting. The main reasons for their high consumption are

the engine efficiency is low because its operating range is not in the best efficiency range

kinetic energy generated during braking is not recovered, which is particularly important in urban environments,

gearboxes operate at low part load efficiency

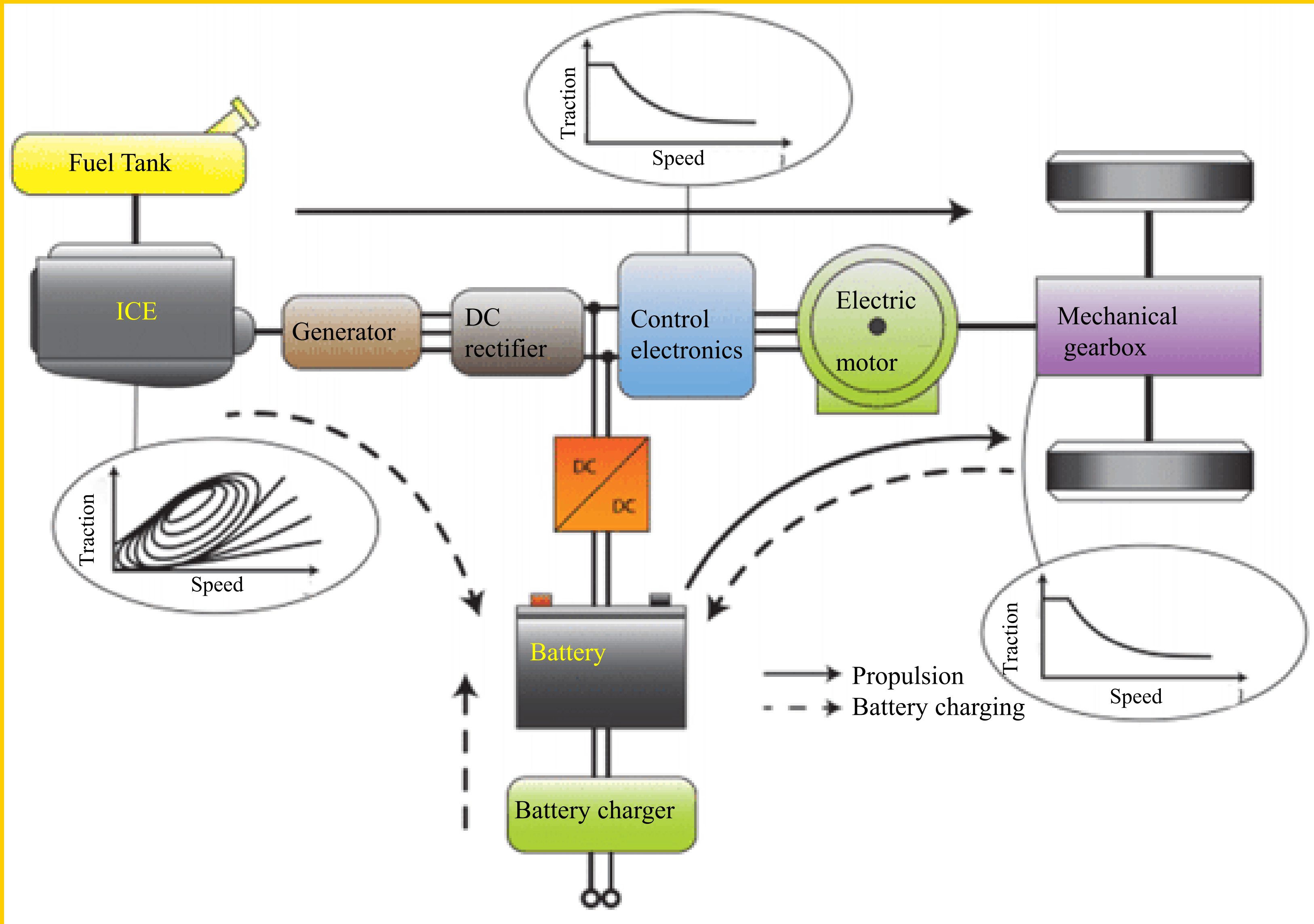
Hybrid electric vehicles (HEVs), which use two sources of power - a primary and a secondary source - are supposed to combine the advantages of internal combustion and electric drive and eliminate their disadvantages.

Steps of hybridisation

- **Start/Stop system**
  - automatically stop the internal combustion engine when the vehicle is stationary.
  - The stop is triggered when:
    - the gearbox is in neutral,
    - the ABS indicator signals zero,
    - Operating parameters (e.g. cooling water)
    - battery charge is at the correct level.

- **Mild hybrids**
  - Start/stop
  - recuperative braking by electric machine
  - with the possibility of electrical assistance.
- **Full hybrids**
  - The full hybrid can also be driven purely by electricity and can drive longer distances. In this case, only the electric motor drives, the combustion engine is not running.
- **Plug-in hybrids**
  - Plug-in, i.e. the chargeable full hybrid. The battery pack can be charged not only from the internal mains by back-charging, but also from an external source (e.g. mains socket). External charging is done via a special connector

Serial hybrid



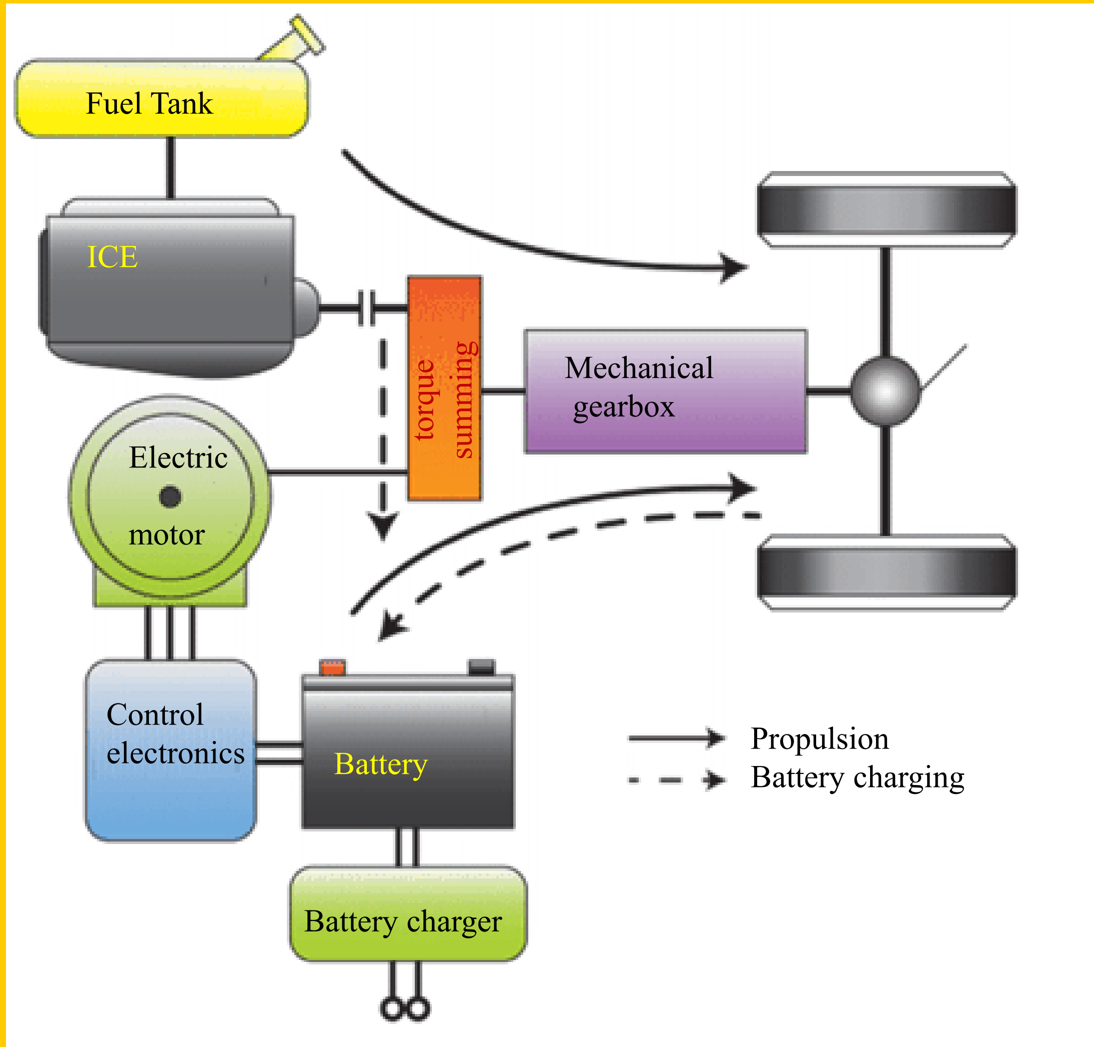
Serial hybrid modes of operation

1. Pure electric drive (electric motor (E.M.)->Drive)
2. Pure motor mode (ICE-> Drive)
3. Hybrid mode (ICE+ E.M. (battery)-> Drive)
4. Motor and battery charging mode (ICE-> Drive + E.M (Generator))
5. "Regenerative" braking mode (E.M.(G)->Battery)
6. Battery charging mode (ICE-> E.M.(G))
7. Hybrid battery charging mode (E.M.(G) + ICE also charges)

Advantages and disadvantages:

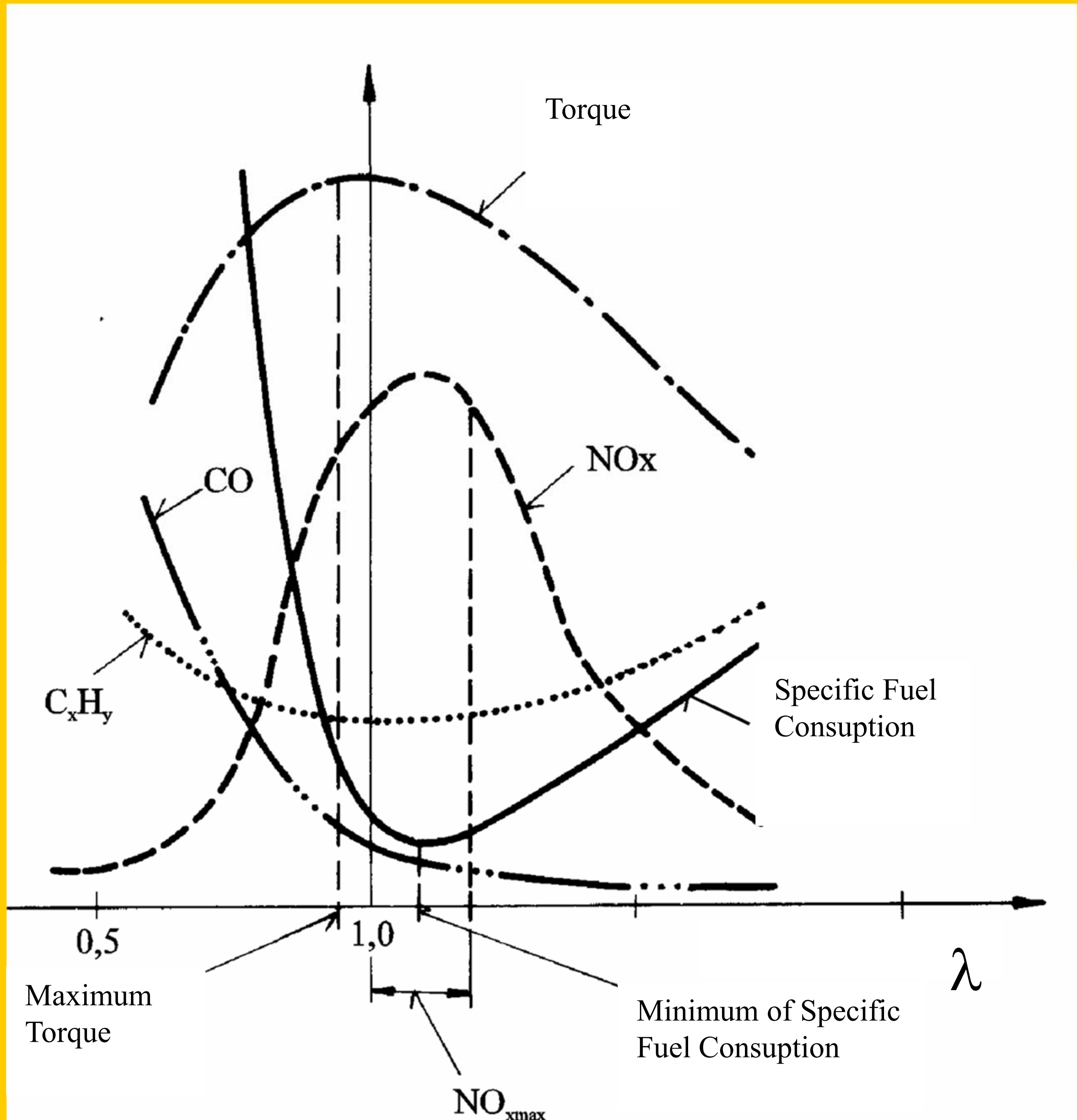
- The combustion engine operates independently of the vehicle's propulsion needs. This allows it to operate over any range of its speed-torque characteristics and even to be maintained exclusively near maximum efficiency.
- As electric motors have a favourable torque-speed characteristic for vehicle propulsion, there is no need for multiple gearboxes.
- Simple control can be used, due to the mechanical decoupling by the electric gearbox
- The energy coming from the engine is converted twice (from mechanical to electrical in the generator and from electrical to mechanical in the traction motor). The efficiency of the generator and the traction motor is multiplied and the loss can be significant.
- The traction motor must be sized to meet the maximum requirements, as it is the only source of power that drives the vehicle

Parallel hybrid-electric drive system



Mixing systems of SI ICE

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



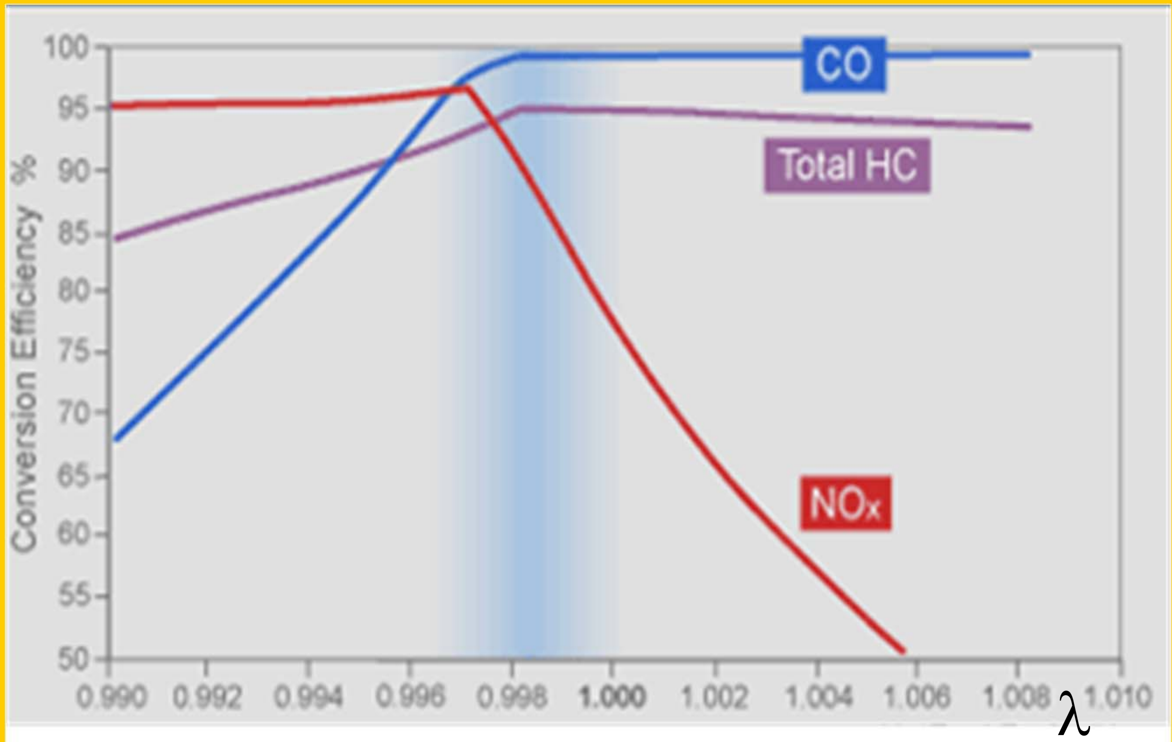
Catalytic Converters

3-way (NSCR) Catalysts (λ=1)

$NO_x \rightarrow N_2+O_2$

$CO \rightarrow CO_2$

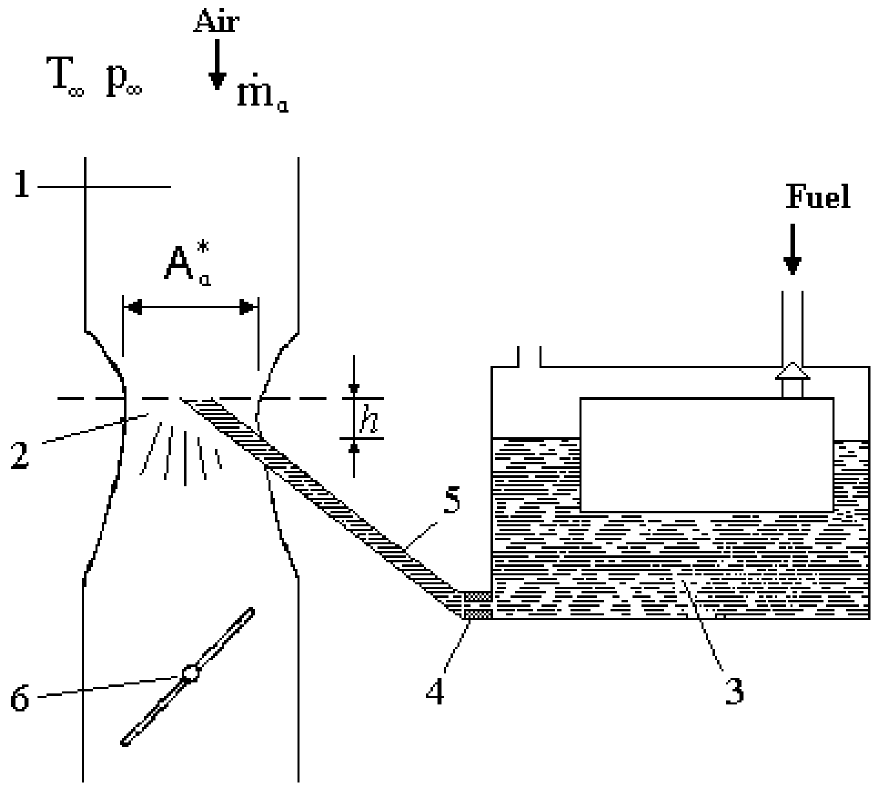
$C_xH_y \rightarrow H_2O+CO_2$



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

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

After simplifications:

$$m_a = C_{DT} A_T \sqrt{2 \rho_a \Delta p_a} \Phi \qquad [2.\ 2]$$

Where:

$$\Delta p_a = p_0 - p_T \qquad [2.\ 3]$$

$$\Phi = \sqrt{\frac{\kappa - 1}{\kappa} \; \frac{p_T / p_0^{\;2/\kappa} - p_T / p_0^{\;\kappa+1/\kappa}}{1 - p_T / p_0}} \qquad [2.\ 4]$$

Massflow of Fuel

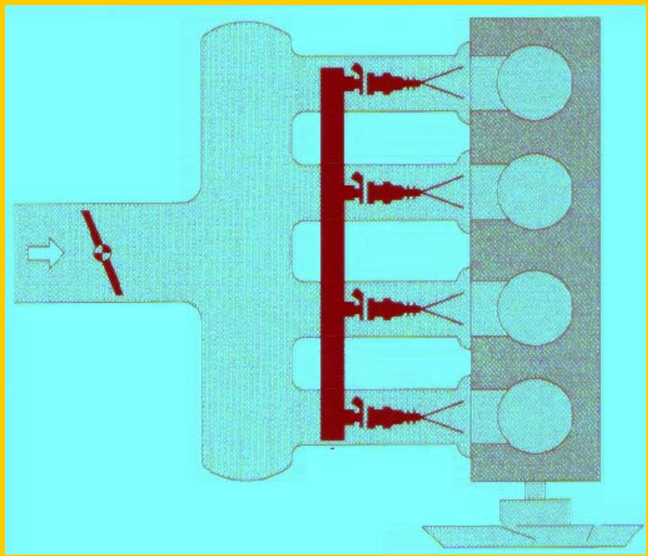
$$m_f = C_{DO} A_o \sqrt{2 \rho_f \Delta p_f} \qquad [2.\ 5]$$

Access air factor:

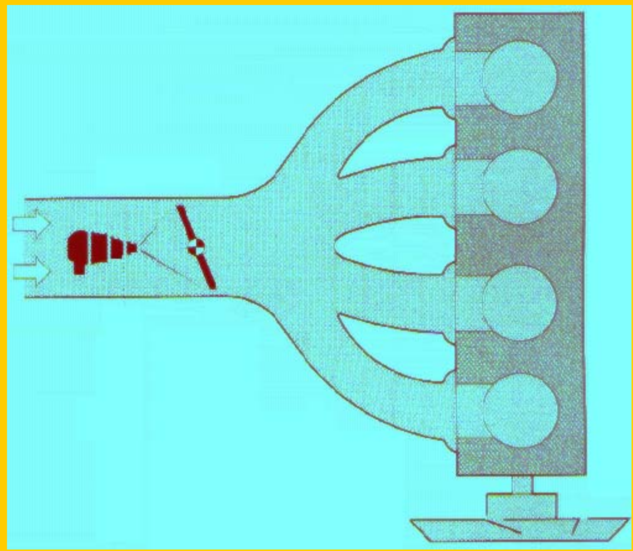
$$\lambda = \frac{m_a}{m_f L_0} = \frac{\Phi}{L_0} \frac{C_{DT}}{C_{DO}} \frac{A_T}{A_O} \sqrt{\frac{\rho_a}{\rho_f} \frac{\Delta p_a}{\Delta p_a - \rho_f g h}} \approx C \frac{1}{\sqrt{\Delta p_a}} \qquad [2.\ 6]$$

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

Injector Types



MPI



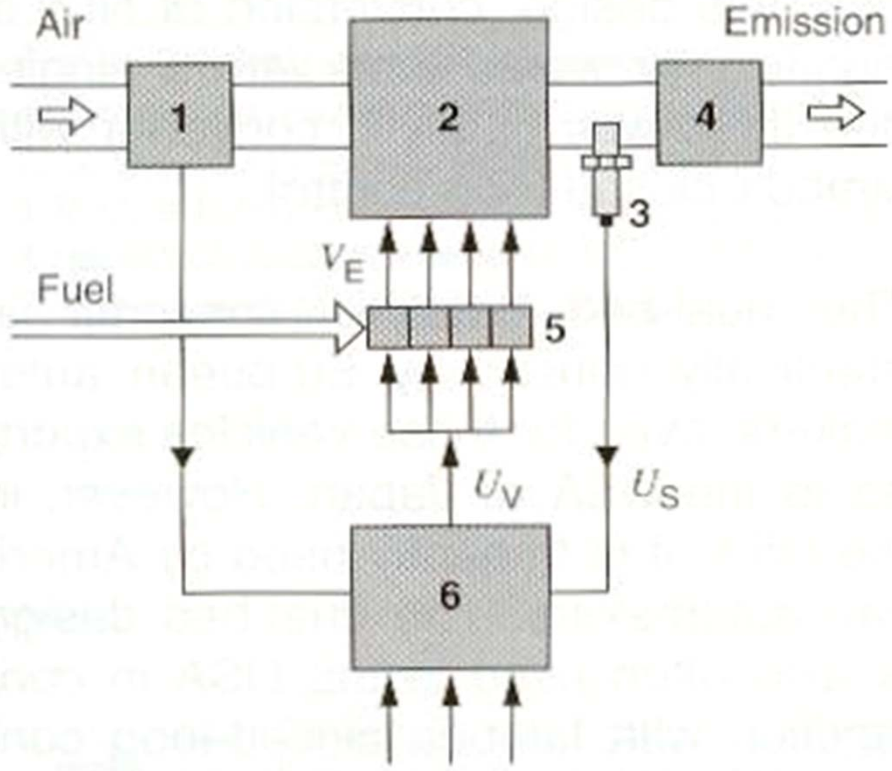
SPI

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.



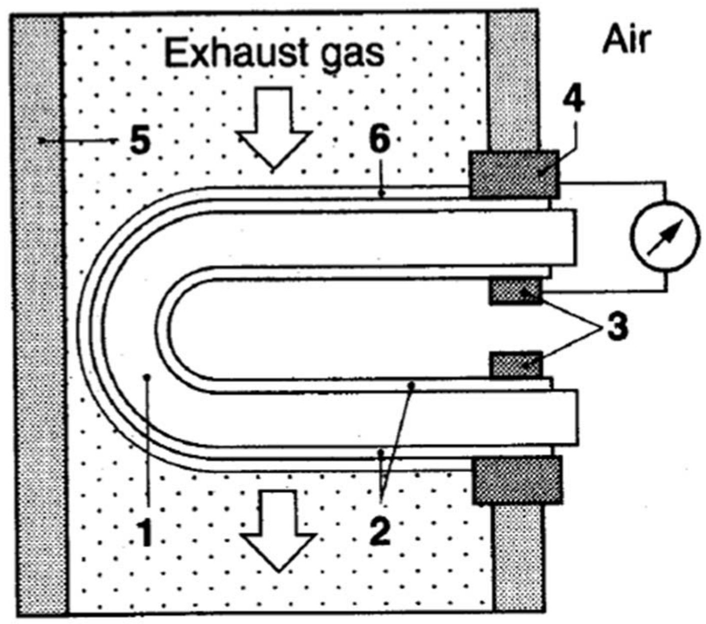
Source: BOSCH

Lambda Oxygen Sensor

- Solid-state electrolyte made of ZrO ceramic material.

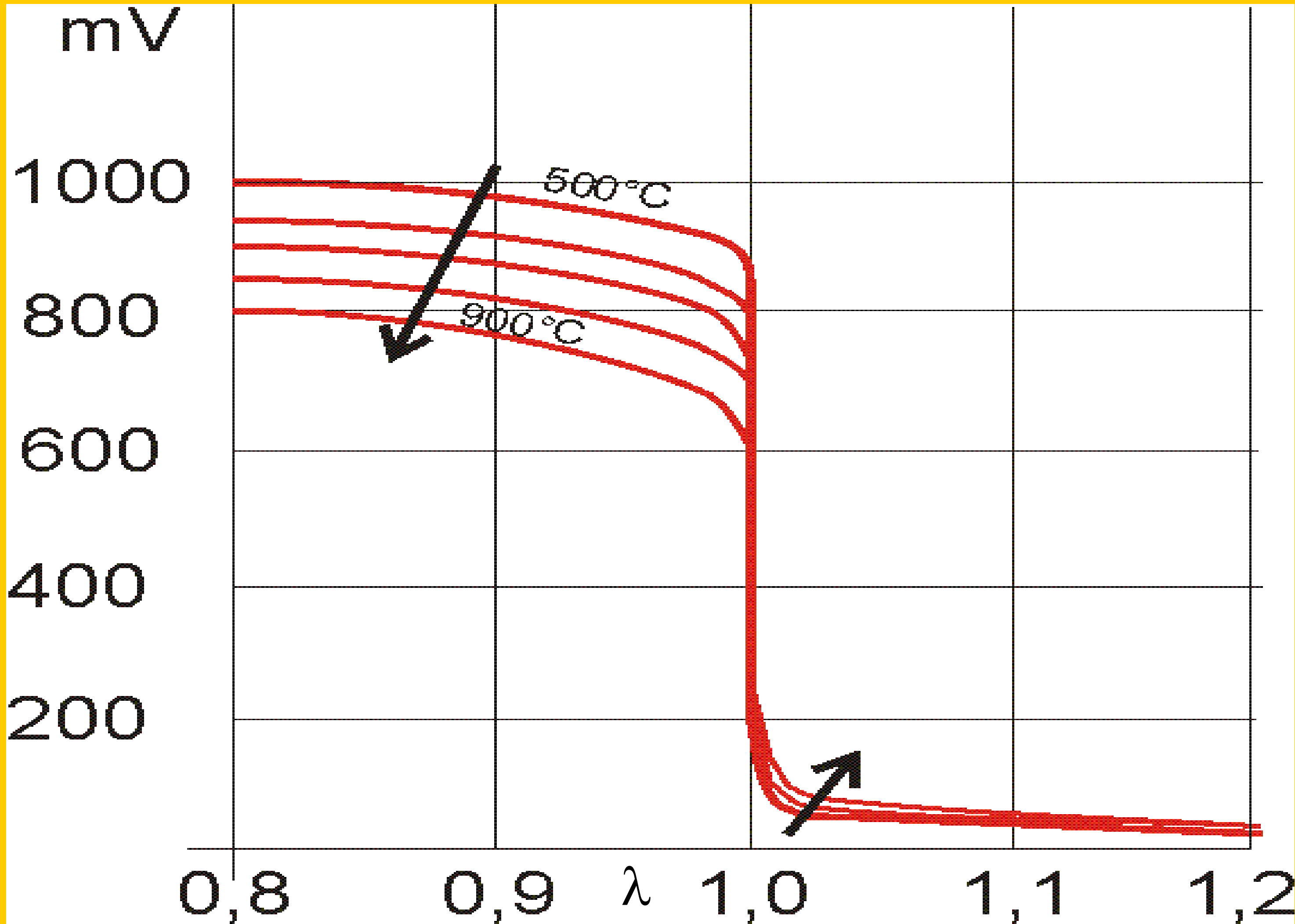
  At high temperatures, the electrolyte becomes conductive and generates a characteristic galvanic charge at the sensor connections this voltage is an index of exhaust gas oxygen content

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

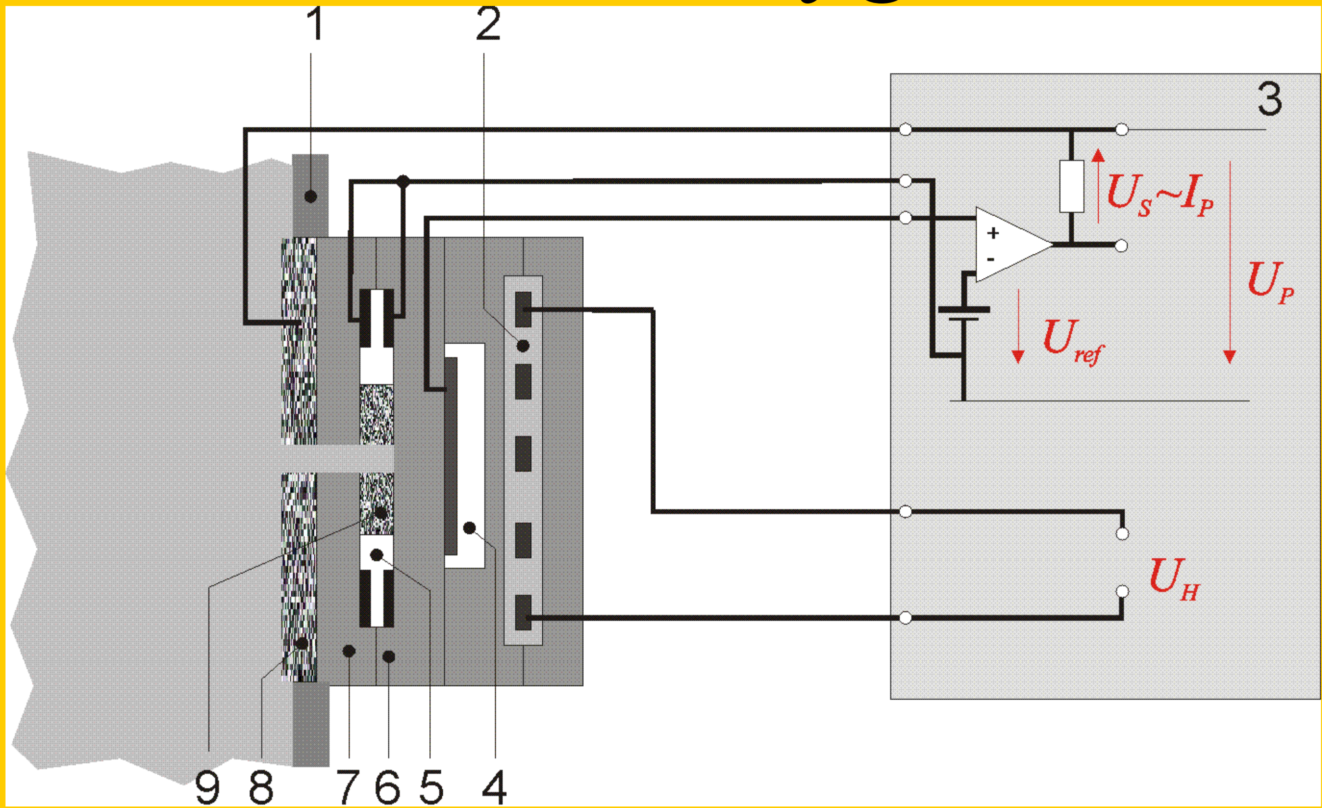


Source: BOSCH

Lambda Oxygen Sensor

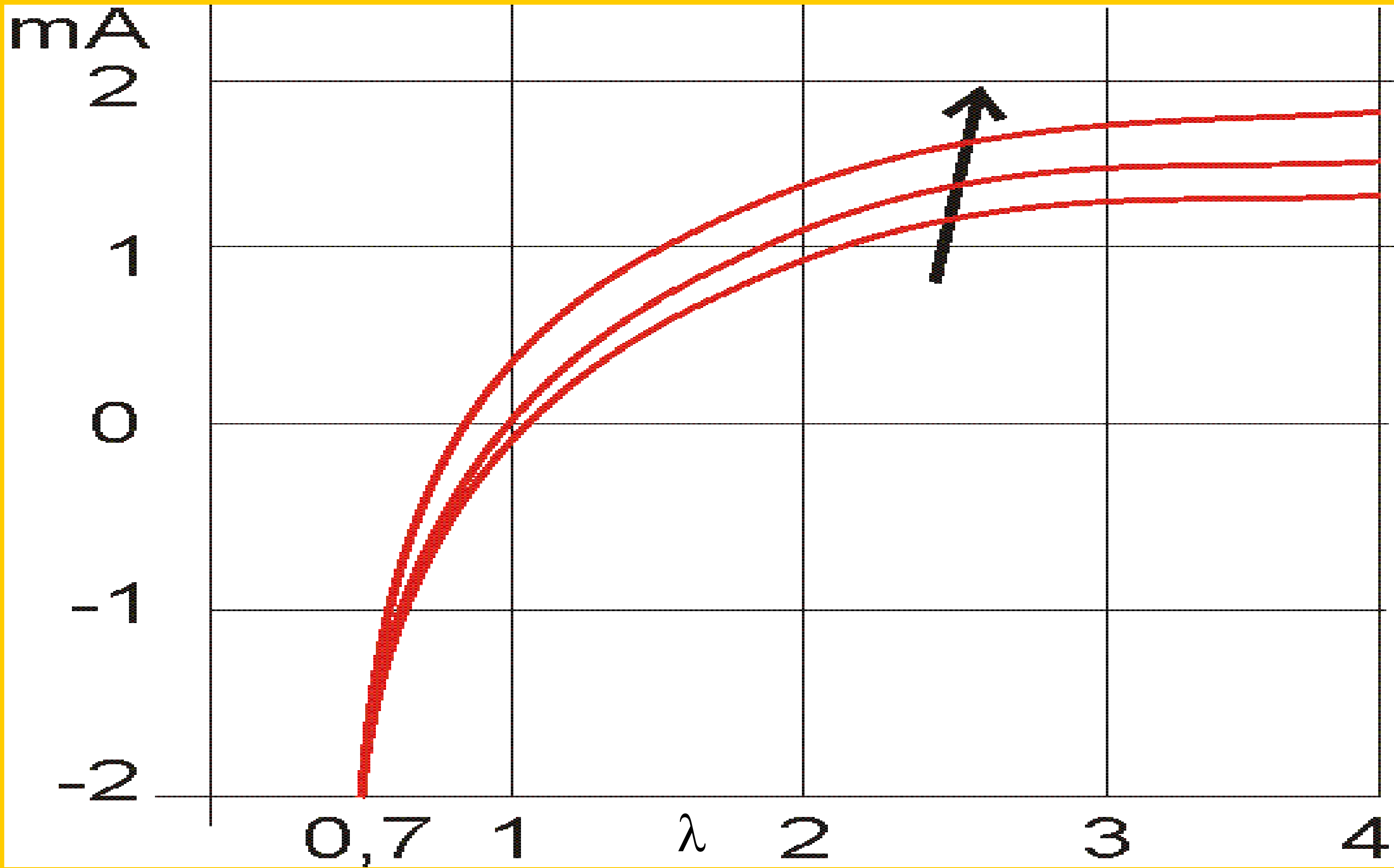


Broadband lambda oxygen sensor

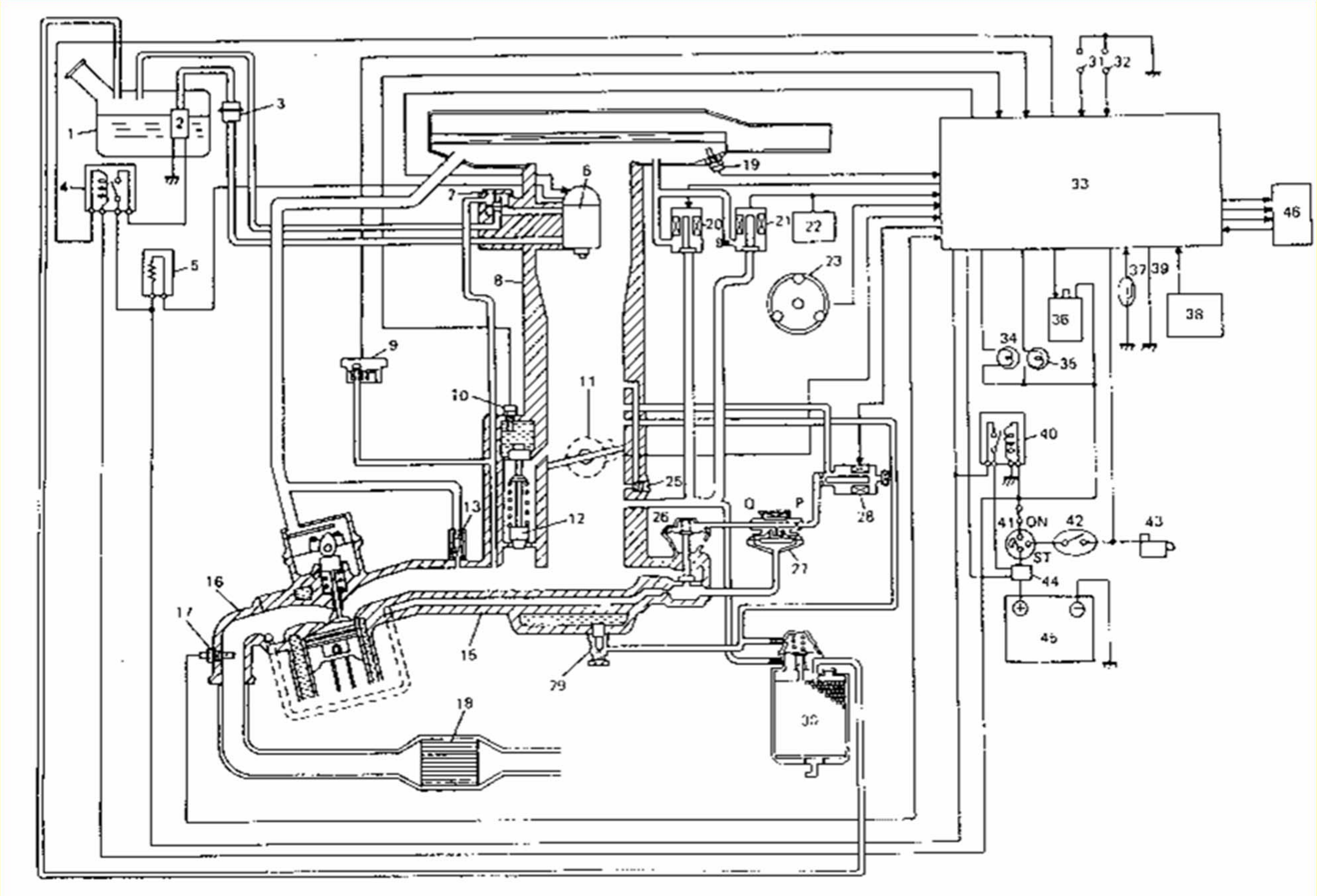


1: Exhaust gases, 2: Heater, 3: Control loop (electronics), 4: Reference air channel, 5: Diffusion gap, 6: Nernst cell, 7:Oxygen pump, 8: Protector, 9: Diffusion gap

Broadband lambda oxygen sensor

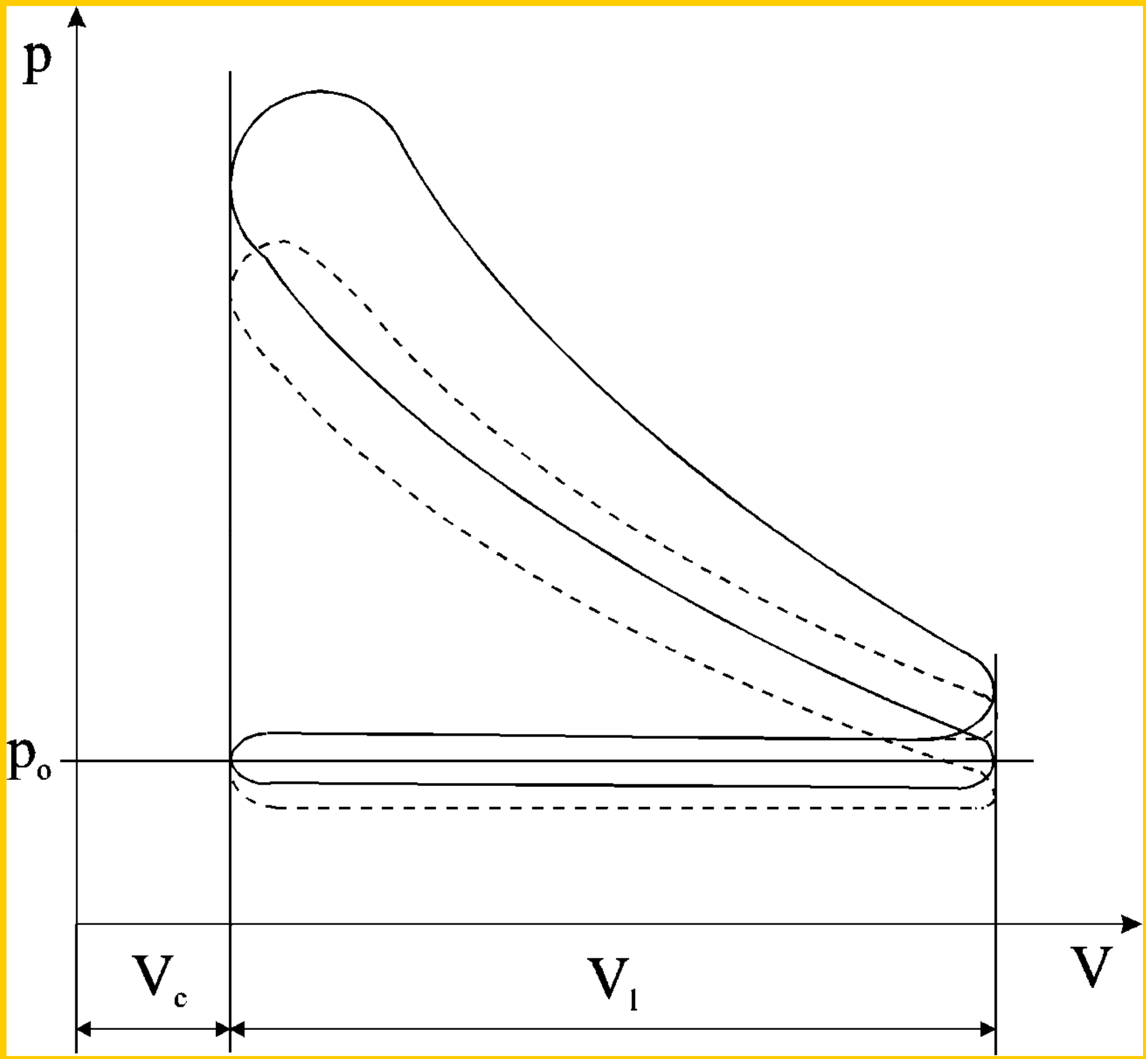


MPI (Ford)

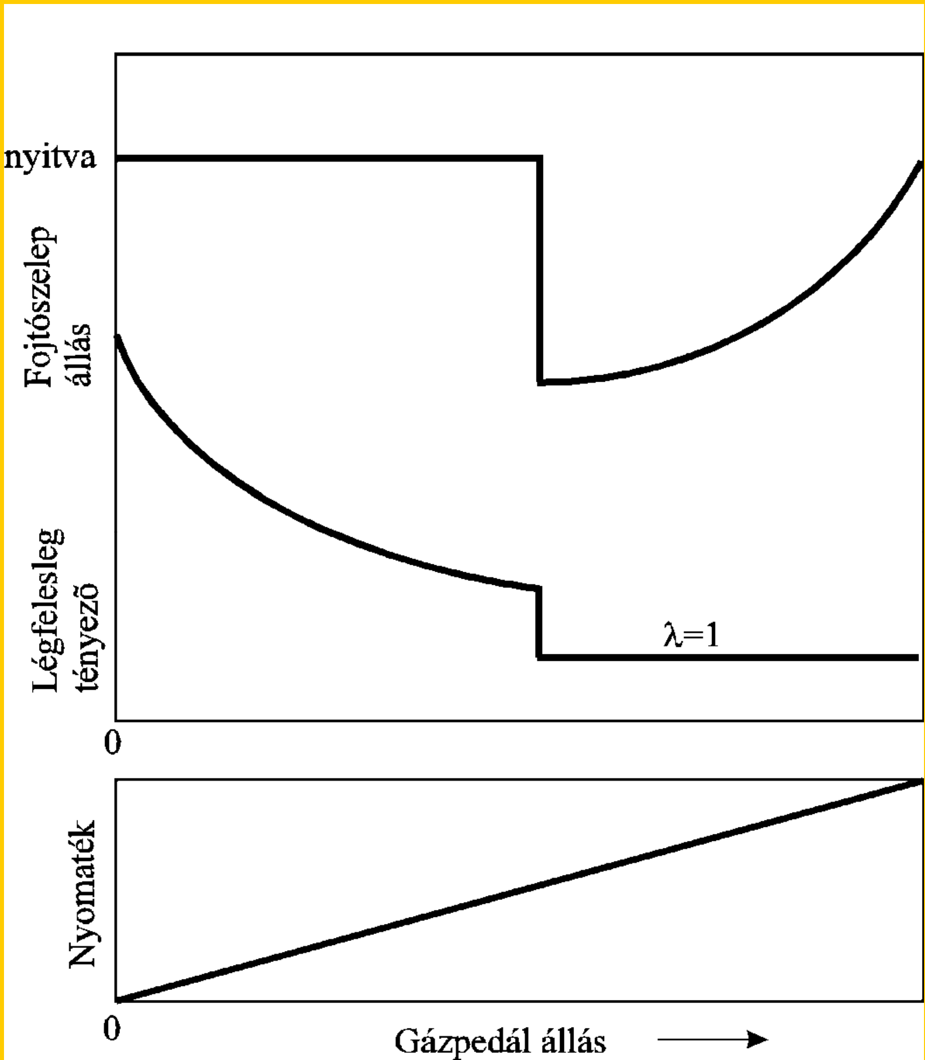


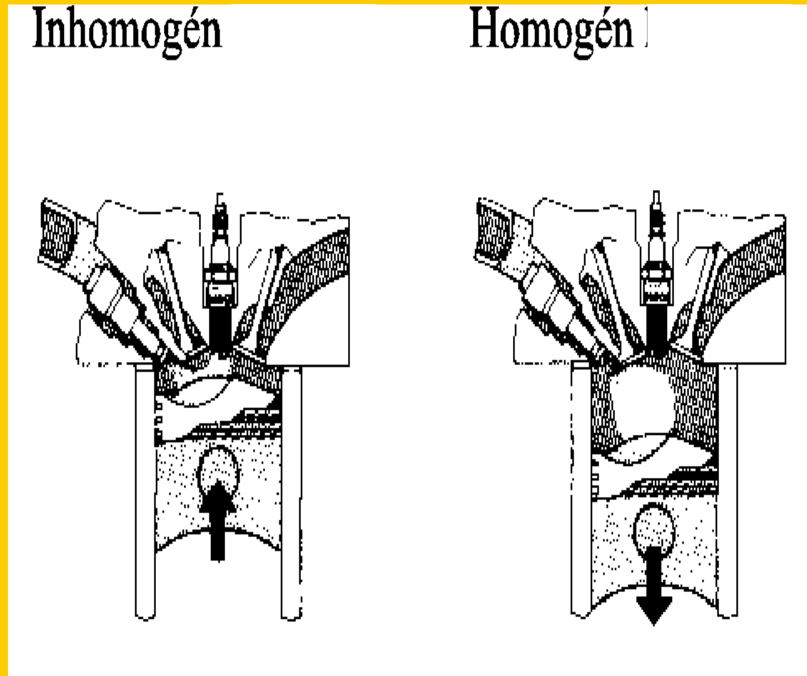
Monotronic system

Direct injection



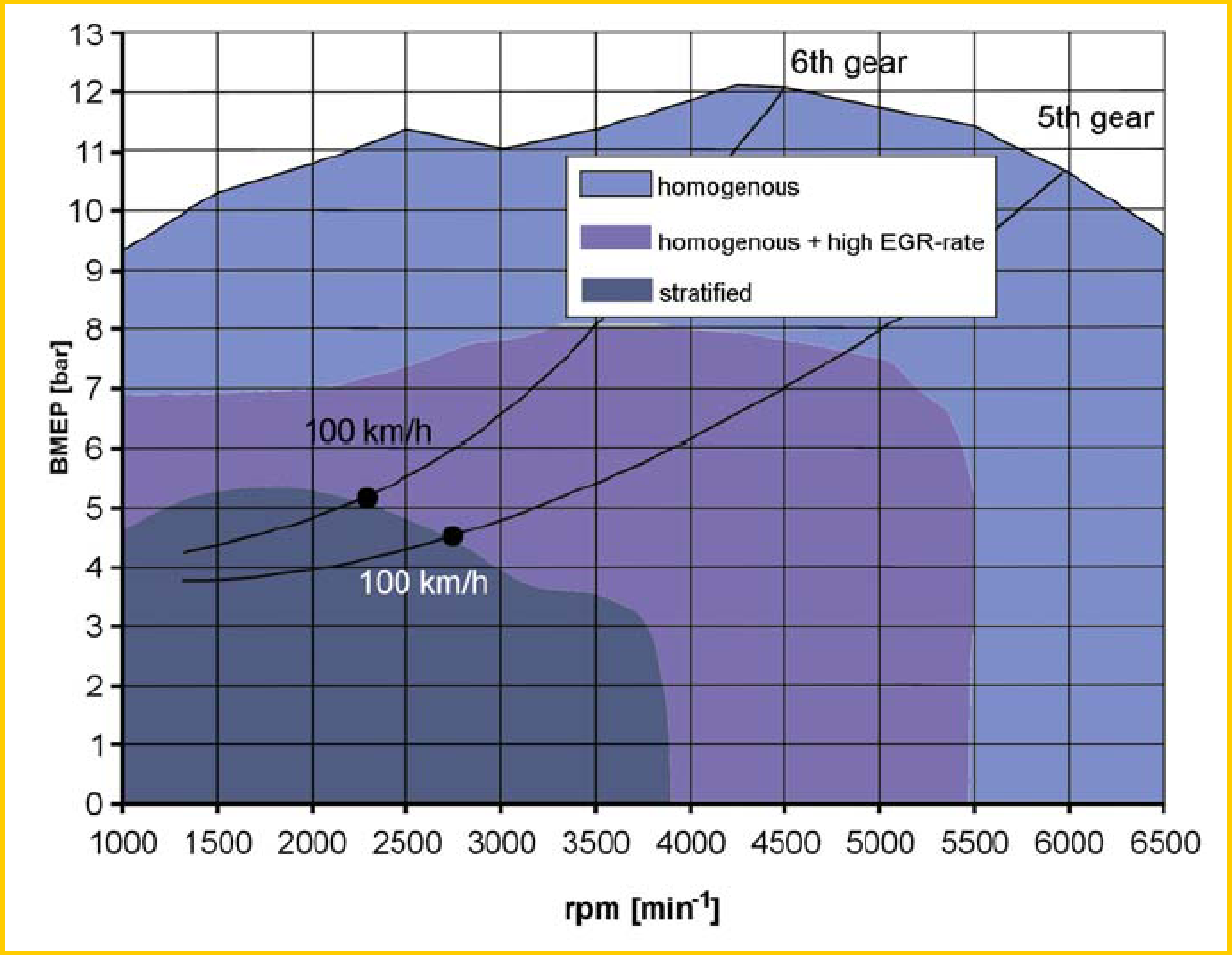
Control of the Otto cycle (- full load, --- partial load)





Bosch direct injection system

FSI (GDI) Engines mixtures

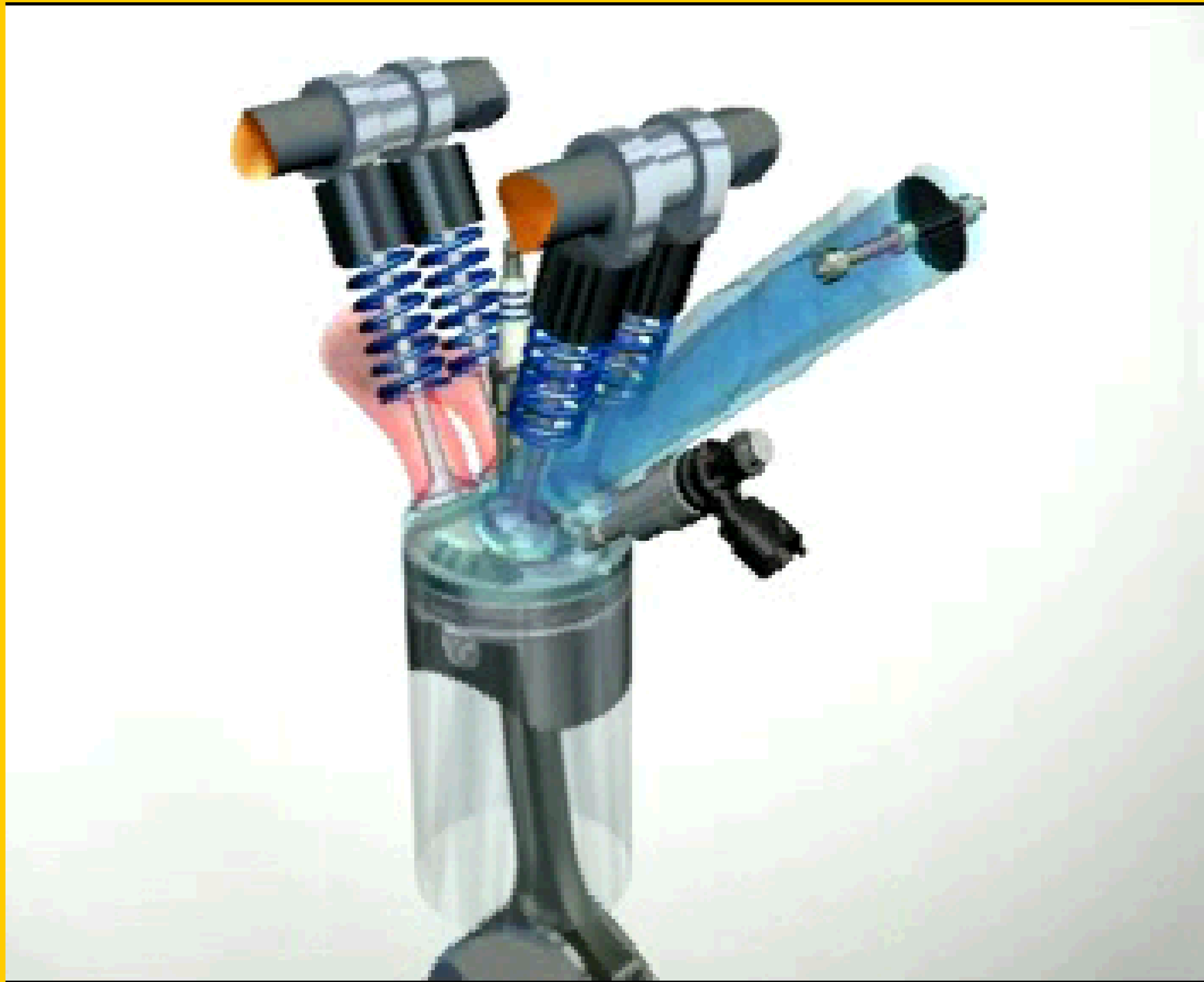


FSI (GDI) Engines Piston

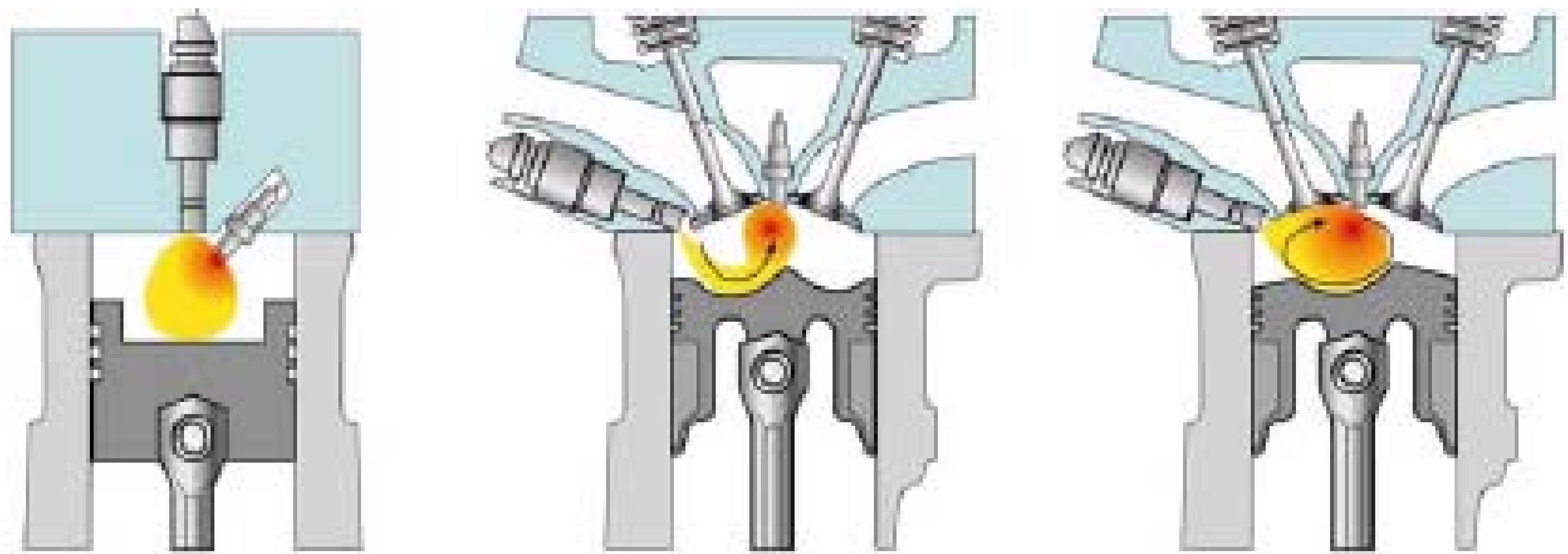
- Wall control type

FSI (GDI) Engines Piston

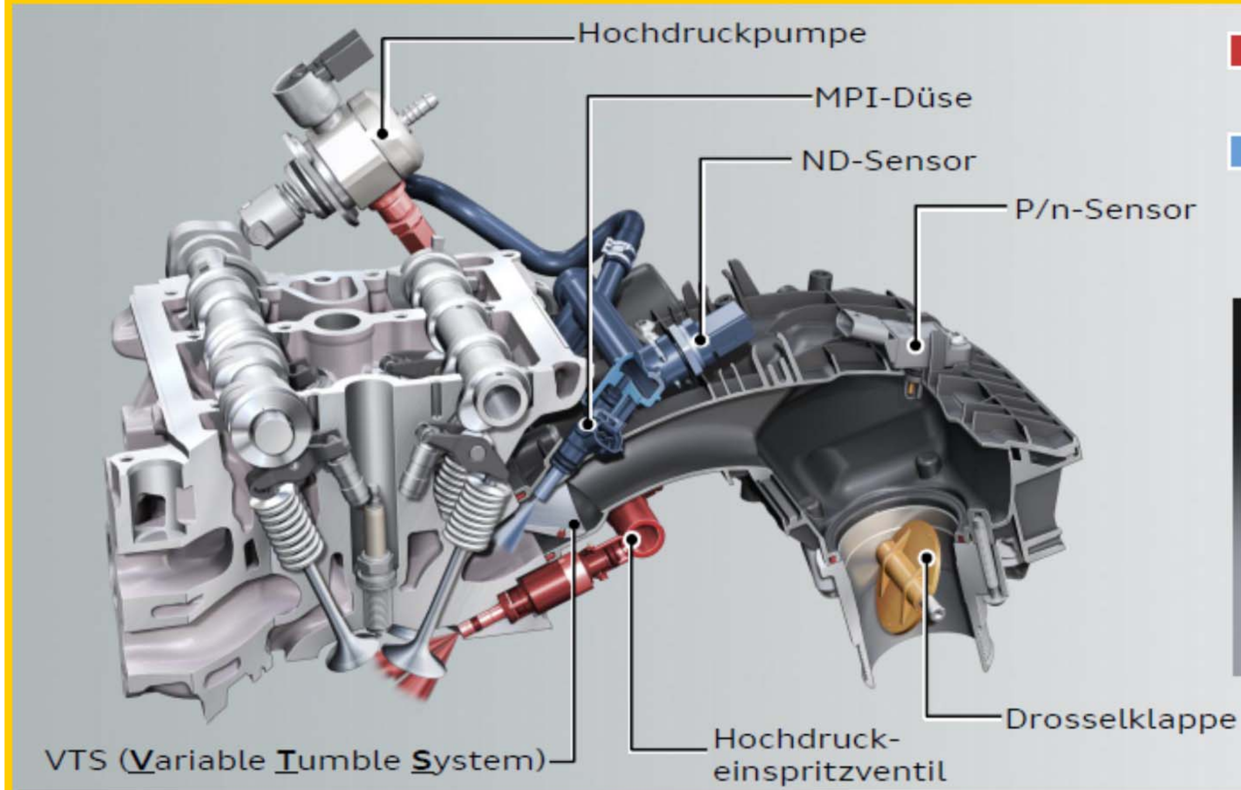
- Wall control type



Control of Direct Injection Systems

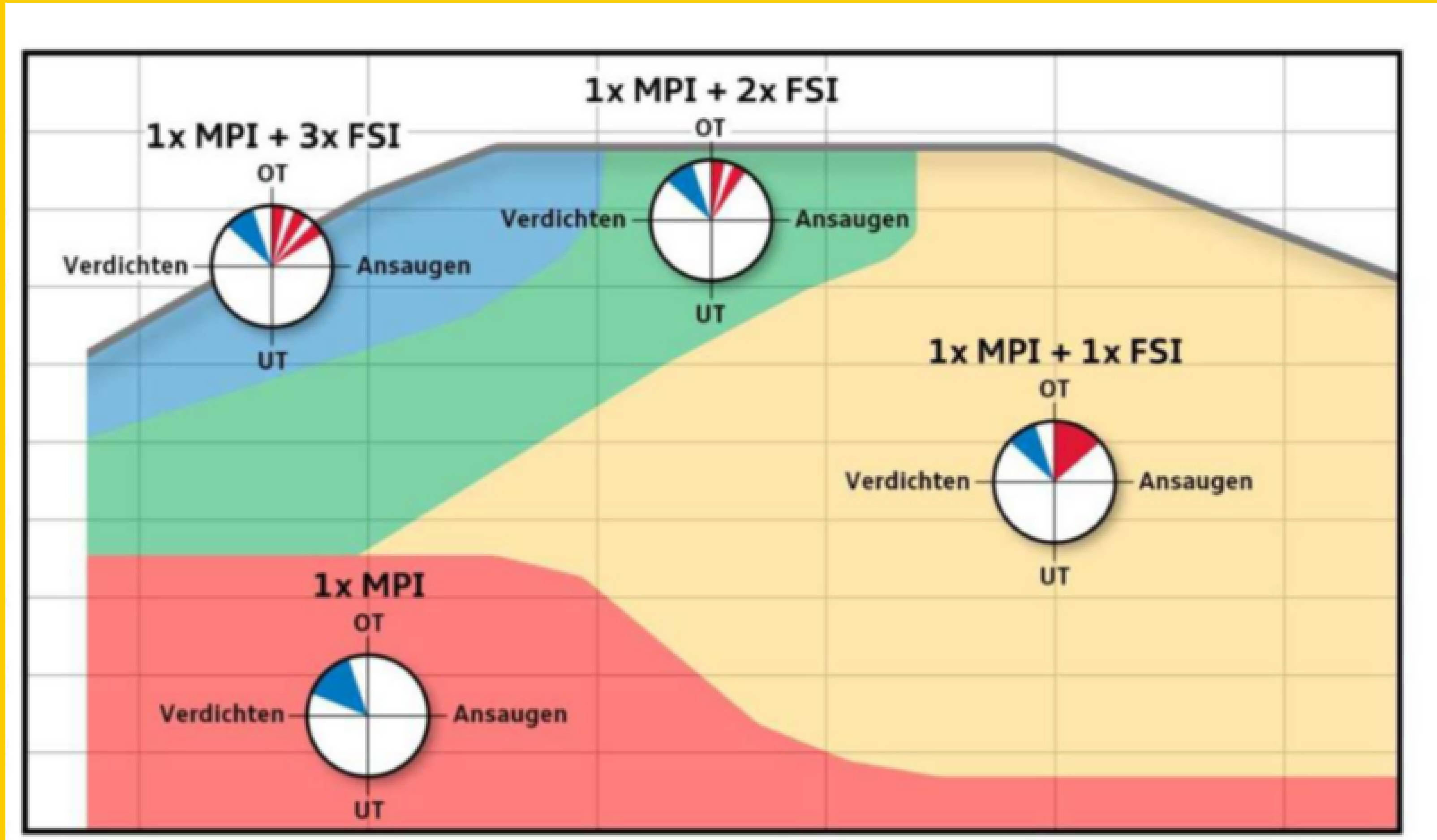


Spark Cont. Wall Cont. Swirl Cont.



Hochdrucksystem / ***Magas nyomású rendszer***

Niederdrucksystem / ***Alacsony nyomású rendszer***



14/10/20; **05/11/22**

2016, 2022