Power Flow and Efficiency

Efficiencies

When the engine converts fuel into power, the process is rather inefficient and only about a quarter of the potential energy in the fuel is released as power at the flywheel.

The rest is wasted as heat going down the exhaust and into the air or water. This ratio of actual to potential power is called the "THERMAL EFFICIENCY", of the engine.

How much energy reaches the flywheel (or dynamometer) compared to how much could theoretically be released is a function of three efficiencies, namely:

1. **Thermal**
2. **Mechanical**
3. **Volumetric**

Thermal Efficiency

Thermal efficiency can be quoted as either brake or indicated.

Indicated efficiency is derived from measurements taken at the flywheel.

The thermal efficiency is sometimes called the fuel conversion efficiency, defined as the ratio of the work produced per cycle to the amount of fuel energy supplied per cycle that can be released in the combustion process.

B.S. = Brake Specific

Brake Specific Fuel Consumption = mass flow rate of fuel ÷ power output

bsHC = Brake Specific Hydrocarbons = mass of hydrocarbons/power output.

e.g. 0.21 kg /kW hour

Thermal Efficiency

$$\eta_t = \frac{W_c}{m_f Q_{HV}} = \frac{P_s}{\mu_f Q_{HV}}$$

$W_c - work\_per\_cycle$

$P_s - power\_output$

$m_f - mass\_of\_fuel\_per\_cycle$

$Q_{HV} - heating\_value\_of\_fuel$

$\mu_f - fuel\_mass\_flow\_rate$

Specific fuel consumption

$$Sfc = \frac{\mu_f}{P_s}$$

Therefore

$$\eta_t = \frac{1}{Sfc \cdot Q_{HV}} = \frac{3600}{Sfc(g/kW \cdot hr) Q_{HV}(MJ/kg)} = \frac{82.76}{Sfc}$$

Since, Q_{HV} for petrol = 43.5 MJ/kg

Therefore, the brake thermal efficiency = 82.76/ sfc

The indicated thermal efficiency = 82.76 / isfc

Mechanical Efficiency

The mechanical efficiency compares the amount of energy imparted to the pistons as mechanical work in the expansion stroke to that which actually reaches the flywheel or dynamometer. Thus it is the ratio of the brake power delivered by an engine to the indicated power.

$$\eta_m = \frac{brake\_power}{gross\_indicated\_power}$$

Volumetric Efficiency

The parameter used to measure an engine's ability to breath air is the volumetric efficiency

$$\eta_v = \frac{mass\_of\_air\_indicated\_per\_cylinder\_per\_cycle}{mass\_of\_air\_to\_occupy\_swept\_volume\_per\_cylinder\_at\_ambient\_pressure\_and\_temperature}$$

Swept Volume

- At top dead centre, the volume remaining above the piston is termed the clearance volume. The swept volume is defined as the volume above the piston at bottom dead centre, less the clearance volume.

- Hence **SWEPT VOLUME = Total volume - clearance volume.**

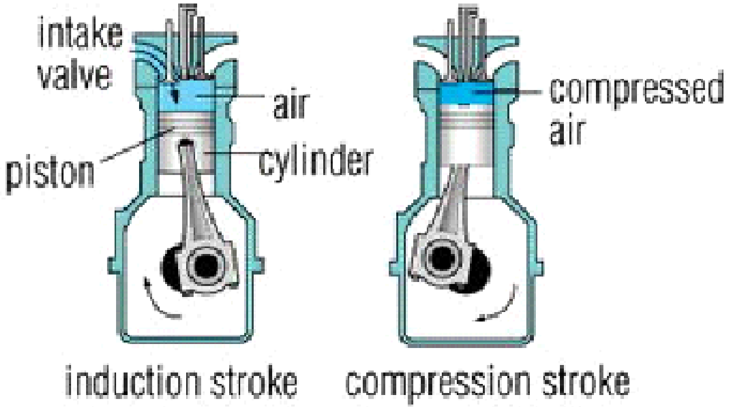
Air Fuel Ratio

- The air fuel ratio mixture induced into the cylinder will properly ignite and combust only if the air fuel ratio lies within a certain range. The normal operating range for a naturally aspirated spark ignition engine is between 12 and 18:1 AFR.

- Note: Combustion limits for petrol/air mixtures are theoretically 3:1 to 40:1, but practically are nearer 9:1 to 25:1

Compression ratio C R

$$\frac{V_1}{V_2} = \frac{\text{swept volume} + \text{clearance volume}}{\text{clearance volume}}$$



Typical values for compression ratio are 8 to 12:1 for spark ignition engines and 12 to 24:1 for compression ignition engines.

Brake Specific Air Consumption (bsac)

Similarly, specific air consumption is defined as the air mass flow rate per unit power output:

Bsac = Air flow ÷ Power

Factors which affect bsfc and bsac

- Compression ratio
- AFR and ignition settings
- Friction ~ rubbing in the engine and also accessories
- Pumping losses ~ intake system restrictive ness/cylinder head design and also exhaust system design
- Calorific value of the fuel
- Barrel swirl ratio of cylinder head
- Heat pick up through the induction system
- Heat transfer from the combustion chamber
- Mixture preparation
- Fuel air mixture distribution

IMEP Indicated Mean Effective Pressure

IMEP = Indicated work out put per cylinder per mechanical cycle/
Swept volume per cylinder

PMEP Pumping Mean Effective Pressure

This is a measure of the work done in drawing a fresh mixture through the induction system into the cylinder, and to expel the burnt gases out of the cylinder and through the exhaust system.

CEMEP Compression/Expansion Mean Effective Pressure

-the same as gross IMEP

FMEP Friction Mean Effective pressure

This is a measure of rubbing friction work and accessory work

Efficiency – comments 1

- The efficiency of an internal combustion engine depends on where and when in the cycle the heat is released. In an ideal Otto cycle (Four-stroke cycle) this means that heat release must occur at a constant volume. This however is not possible as combustion takes a finite time.

- As the piston approaches the top of it's stroke it slows, momentarily comes to rest and then moves back down. Ideally heat release (Combustion) should take place as close to this point as possible, as any heat release that occurs before this point causes a pressure increase that opposes the pistons last upward movement, and the opposition to the pistons movement represents a loss in efficiency.

Efficiency – comments 2

- To delay the heat release to overcome this means that part of the charge (Air/Fuel mixture) will be burnt too late for effective work extraction and this leads to the part of the energy of the charge being wasted as a hotter exhaust stream.

- It follows from this that to allow for the finite time of heat release the piston's downward stroke should be slowed down in its early stages. This would allow more time for heat release to take place before the expansion stroke progresses too far.

Efficiency – comments 3

- In theory this ideal can be achieved by offsetting the crankshaft axis from the piston axis.
- The additional effect of offsetting the crank is that the combination of the altered time/pressure history in the cylinder and the altered moment of the force acting through the connecting rod change the instantaneous torque at the crankshaft.
- An elegant solution to this is the Australian Scotch Yoke mechanism patented and manufactured with VW by the Collins Motor corporation of Australia
- The efficiency is fairly fixed for a given design of engine. It is generally based on the factors effecting heat loss, such as the design of the combustion chamber and materials used, and the compression ratio. The efficiency does not vary dramatically and tends to range from .34 to .38 .
- Due to there close relationship the indicated and combustion efficiencies are often evolved together as a fuel conversion efficiency,η_f.

Mechanical efficiency losses

- Friction losses in the engine come from friction between moving parts (such as the piston rings on the cylinder walls), the power required to run the valve gear and the pumps and the pumping losses involved in getting the gases in and out of the cylinder. A large part of the work appears as heat in the coolant and oil. The approximate breakdown of the contributions of each of the losses is as follows:

Friction losses:

- Crankshaft and seals – 12%
- Pistons, rings, pins and rods – 46%
- Valve train –23%
- Oil pump – 6%
- Water pump and Alternator – 13%

Mechanical efficiency losses

- Typical values of mechanical efficiency for modern automotive engines at WOT are 90%
- at speeds below 1800-2400 rpm decreasing to about 75% at maximum rated speeds.
- Lumley correlates a value of 60% around an average piston speed of 20 m/s. 20m/s represents a practical maximum speed for most engines irrespective of size due to the fact that the materials in contact are generally the same.

The mass flow rate of air

$$\dot{m}_a = \frac{N}{2} V_d \rho_i \eta_v$$

Control and optimisation of the three variables engine speed (N), volumetric efficiency (η_v) and Inlet density is very important and can potentially allow the tapping of extra energy from the bsfc.

Volumetric efficiency

If we think of the engine as an air pump then theoretically it should draw in and exhaust its own volume of air each time it cycles - that is, once every revolution if it's a two stroke and once every two revolutions if its a four stroke.
In fact, ordinary production engines don't achieve this and only manage to shift about 80% of their volume.
This ratio of possible air pumped to actual air pumped is called Volumetric Efficiency and this is what we have to improve to get more power.

Volumetric efficiency

Volumetric efficiency simply is a measure of an engine's ability to breath air.

$$\frac{\text{Mass of air inducted per cylinder per cycle}}{\text{Mass of air to occupy swept volume per cylinder at ambient pressure and temperature}}$$

Volumetric efficiency

- The volumetric efficiency is probably one of the most variable efficiencies governing the performance of engines. It is a measure of the effectiveness of an engine and exhaust system as air pumping devices. The volumetric efficiency is effected by a great many variables.

Volumetric efficiency

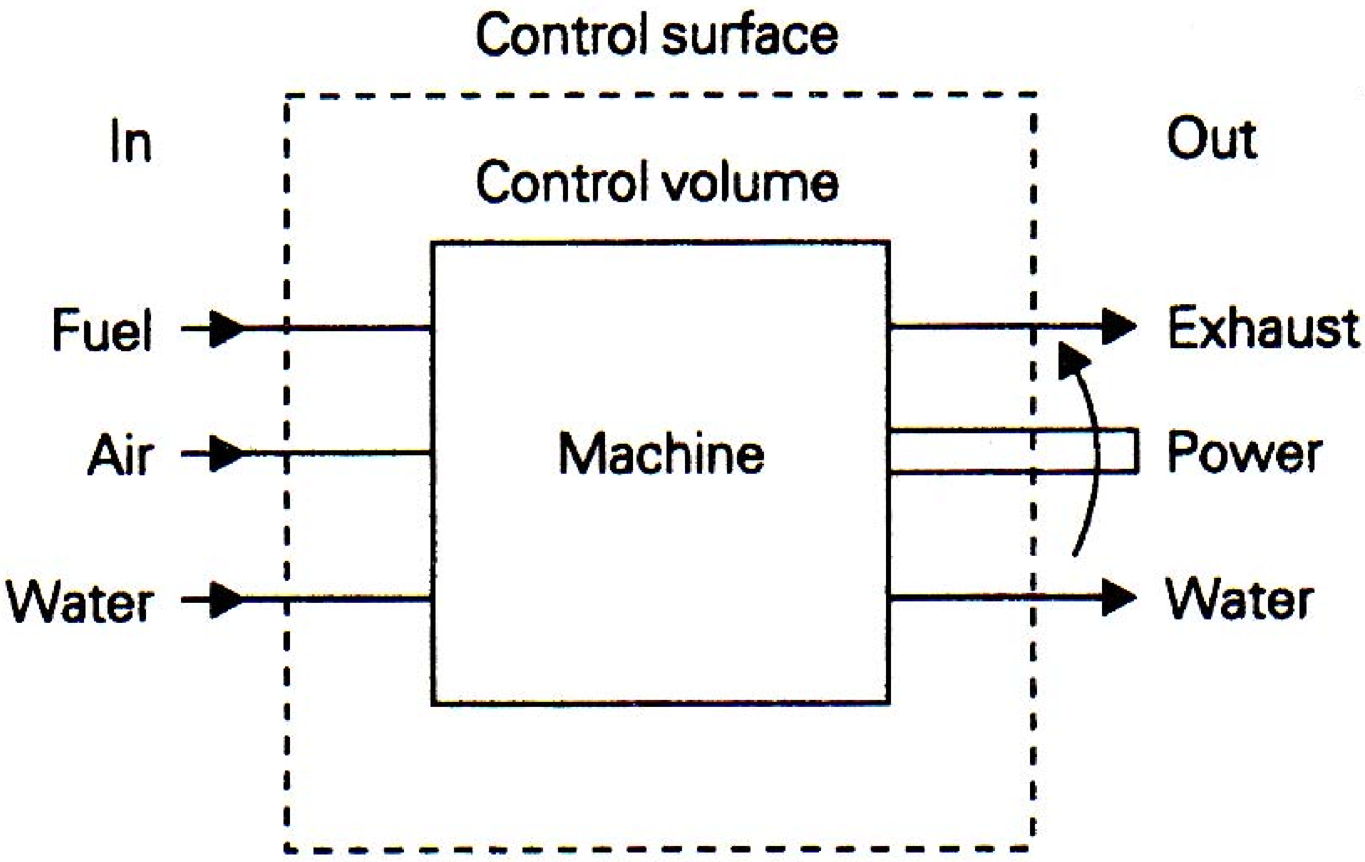
Fuel type, fuel air ratio, fraction of fuel vaporised in the intake system, and fuel heat of vaporisation. Mixture temperature as influenced by heat transfer Ratio of exhaust to inlet manifold pressures Compression ratio Engine speed Intake and exhaust manifold design Intake and exhaust valve geometry, size, lift and timing

- These variables can be very complicated to asses; some may be branded as quasi-static (their impact is independent of speed or can be evaluated in terms of a mean engine speed) where as others are completely dependent on time varying effects such as pressure waves

Methods of increasing Volumetric Efficiency

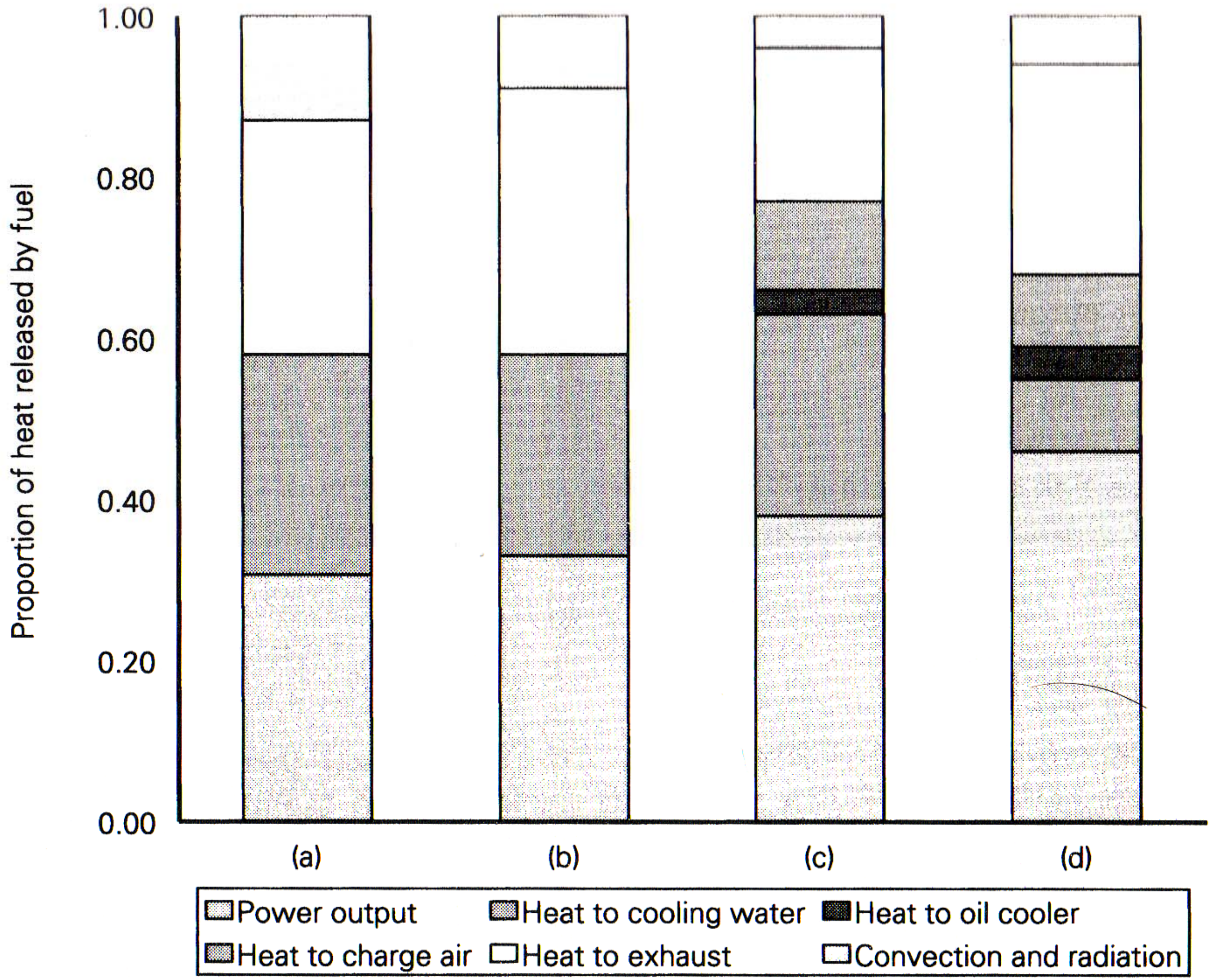
- Fortunately there are various things that can be done to improve the volumetric efficiency in the pursuit of power and torque. These are essentially:
- Optimal Intake design to ensure smooth unrestricted paths for the airflow
- Forced Induction- This comes down to supercharging or turbocharging
- Induction Ram - this only occurs at high speed and is due to the inertia of the high speed air. In part intake valves are left open after BDC to take advantage of this.
- Intake tuning- use of reflected pressure waves to increase the air density at the inlet valve just prior to closing - This method is key to the success of naturally aspirated engines.
- The implementation of these concepts must be done carefully as not to reduce throttle response and low speed performance too greatly.

Energy flows in engine testing room



Energy flows in engine testing room

In	*Out*
Fuel	Power
Air used by the engine	Exhaust
Cooling water	Cooling water
Cooling air	Cooling air
	Convection and radiation



Typical full power energy balances: (a) 1.71 gasolene engine; (b) 2.51 naturally aspirated diesel engine; (c) 200 kW medium speed turbo-charged marine diesel; (d) 7.6 MW combined heat and power unit

Power input-output balance of test cell

Test cell with hydraulic dynamometer and 100 kW gasolene engine

Energy Balance

In		*Out**	
Fuel	300 kW	Exhaust gas	60 kW
Ventilating fan power†	5 kW	Engine cooling water	90 kW
		Dynamometer cooling water	95 kW
		Ventilation air	70 kW
Electricity for cell services	25 kW	Heat loss, walls and ceiling	15 kW
	330 kW		330 kW

The energy balance for the engine, see Chapter 11, is as follows:

In		*Out*	
Fuel	300 kW	Power †	100 kW
		Exhaust gas	90 kW
		Engine cooling water	90 kW
		Convection and radiation	20 kW
	300 kW		300 kW

Test cell with regenerative electrical dynamometer and 100 kW diesel engine

Energy balance

In		*Out*	
Fuel	260 kW	Exhaust gas	45 kW
Ventilating fan power	5 kW	Engine cooling water	70 kW
Dynamometer excitation	5 kW	Dynamometer power	85 kW
		Ventilation air	80 kW
Electricity for cell services	25 kW	Heat loss, walls and ceiling	15 kW
	295 kW		295 kW

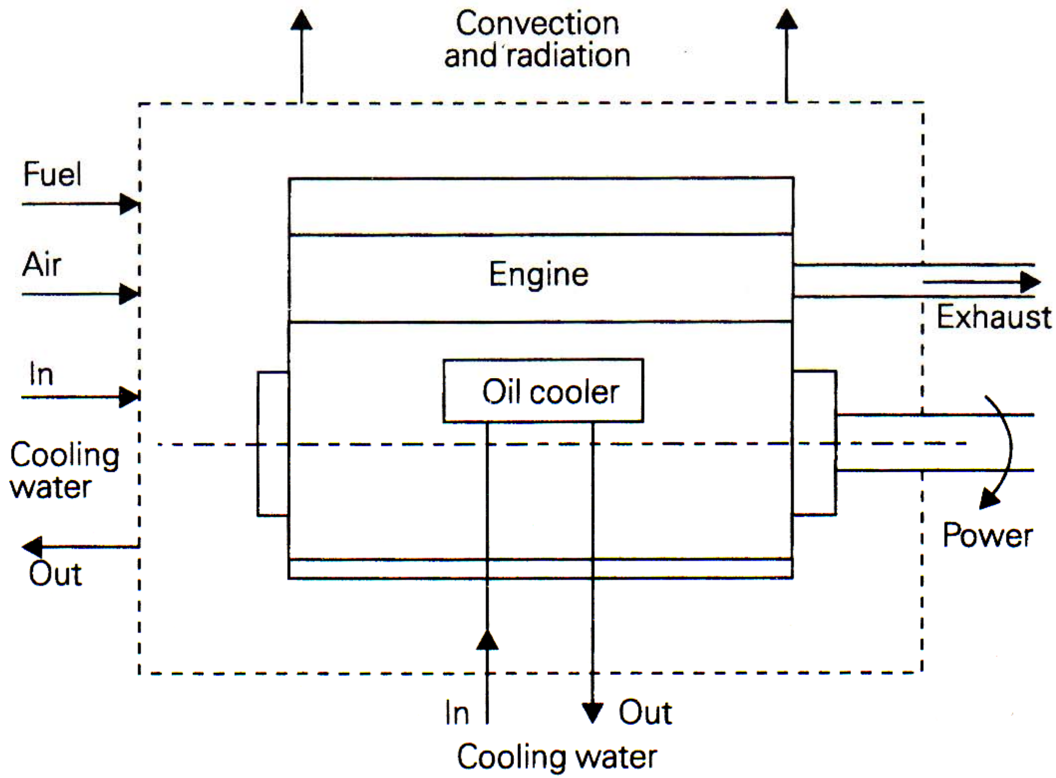
See Fig. 1.3 for the corresponding system.

In this case the energy balance for the engine is as follows:

In		*Out*	
Fuel	260 kW	Power	100 kW
		Exhaust gas	70 kW
		Engine cooling water	70 kW
		Convection and radiation	20 kW
	260 kW		260 kW

Notes

- Thermal efficiency of the engine = 0.385.
- Ventilation load is increased due to dynamometer air cooling load.



Input	Output
Fuel	Power
Air	Exhaust gas
	Heat to cooling water
	Heat via convection and radiation to surroundings

Engine heat equation

$$H_1 = P_s + (H_2 - H_3) + Q_1 + Q_2 \qquad (KW)$$

$$H_1 = fuel\_energy = \dot{m}_f C_L \times 10^3$$

$$P_s = power\_output$$

$$H_2 = heat\_in\_exhaust\_gas = (\dot{m}_f + \dot{m}_a)\, C_p T_e$$

$$H_3 = heat\_in\_inlet\_air = \dot{m}_a C_p T_a$$

$$Q_1 = heat\_passed\_to\_cooling\_water$$

$$Q_2 = heat\_losses\_by\_convection\_and\_radiation$$

An example:

Energy balance of a gasolene engine at full throttle (4 cylinder 4-stroke engine swept volume 1.7011)

Engine speed	3125 rev/min
Power output	$P_s = 36.8$ kW
Fuel consumption rate	$\dot{m}_f = 0.00287$ kg/s
Air consumption rate	$\dot{m}_a = 0.04176$ kg/s
Lower calorific value of fuel	$C_L = 41.87 \times 10^6$ J/kg
Exhaust temperature	$T_c = 1066$ K (793°C)
Cooling water flow	$\dot{m}_w = 0.123$ kg/s
Cooling water inlet temperature*	$T_{1w} = 9.2$°C
Cooling water outlet temperature*	$T_{2w} = 72.8$°C
Inlet air temperature	$T_a = 292$ K (19°C)

* The engine was fitted with a heat exchanger. These are the temperatures of the primary cooling water flow to the exchanger.

Then noting that:

specific heat of air at constant pressure $C_p = 1.00\ \mathrm{kJ/kgK}$
specific heat of water $C_w = 4.18\ \mathrm{kJ/kgK}$

$$H_1 = 0.00287 \times 41.87 \times 10^3 = 120.2\ \mathrm{kW}$$

$$P_s = 36.8\ \mathrm{kW}$$

$$H_2 = (0.00287 + 0.04176) \times 1.00 \times 1066 = 47.6\ \mathrm{kW}$$

$$H_3 = 0.04176 \times 1.00 \times 292 = 12.2\ \mathrm{kW}$$

$$H_2 - H_3 = 35.4\ \mathrm{kW}$$

$$Q_1 = 0.123 \times 4.18\ (72.8 - 9.2) = 32.7\ \mathrm{kW}$$

$$Q_2 \quad \text{(by difference)} = 15.3\ \mathrm{kW}$$

We may now draw up an energy balance (quantities in kilowatts):

Heat of combustion H_1	120.2	Power output P_s	36.8 (30.6%)
		Exhaust $(H_2 - H_3)$	35.4 (29.5%)
		Cooling water Q_1	32.7 (27.2%)
		Other losses Q	15.3 (12.7%)
	120.2		120.2

The thermal efficiency of the engine

$$\eta_{th} = \frac{P_s}{H_1} = 0.306$$

Assume mechanical efficiency: 0.80

Then indicated thermal efficiency: 0.306/0.80=0.3825

Energy Balance for per 1 kW power output

Energy balance (kW per kW power output)

	Automotive gasolene	*Automotive diesel*	*Medium speed heavy diesel*
Power output	1.0	1.0	1.0
Heat to cooling water	0.9	0.7	0.4
Heat to oil cooler			0.05
Heat to exhaust	0.9	0.7	0.65
Convection and radiation	0.2	0.2	0.15
Total	3.0	2.6	2.2

Engine testing design example

Engine: 250kW turbo charged diesel engine at full power

Assume fuel 40.6MJ/kg, thermal efficiency 0.42:

Fuel input power = 250/0.422= 592 kW

Specific fuel consumption = 592(kJ/s)/40600(kJ/kg)$\times$3600(s/h)/250(kW)=0.21 kg/kWh

The power calculation using the last table is as below:

Energy balance, 250 kW turbocharged diesel engine

In		*Out*	
Fuel	592 kW	Power	250 kW (42.2%)
		Heat to cooling water	110 kW (18.6%)
		Heat to oil cooler	15 kW (2.5%)
		Heat to exhaust	177 kW (29.9%)
		Convection and radiation	40 kW (6.8%)
	592 kW		592 kW

Fuel flow calculation:

Assume density 0.9kg/litre, the fuel flow:

$$\frac{250 \times 0.21}{0.9} = 58.3 litre / h (52.5 kg / h)$$

Air flow

Assume full load air/fuel ratio=25:1 and air density: $1.2 kg / m^3$, air flow:

$$250 \times 0.21 x 25 = 1312.5 kg / h$$

$$1312.5 / 1.2 = 1094 m^3 / h$$

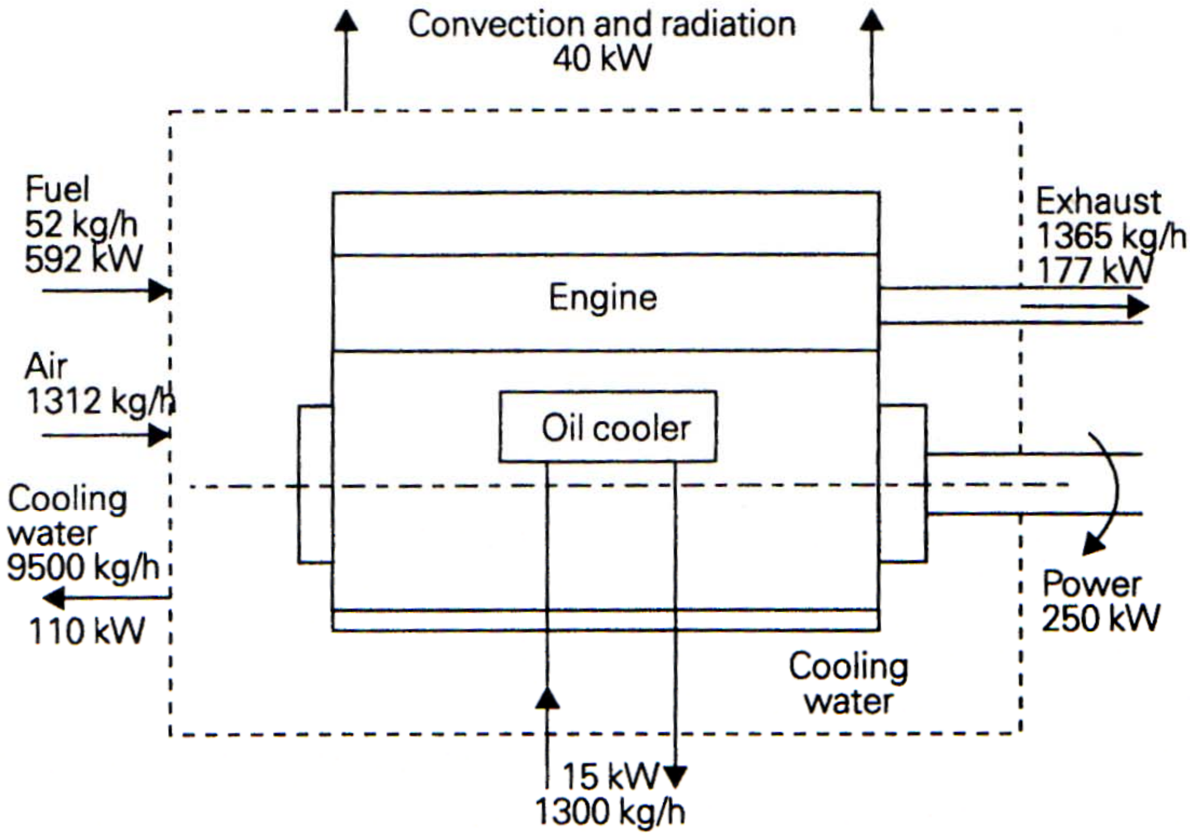
Cooling water flow

Assume temperature rise 10 degrees. Specific heat of water $4.18 kJ / kg {}^0C$, 1kWh=3600kJ

Water flow to jacket and oil cooler $\frac{125 \times 60}{4.18 \times 10} = 180 kg / \min$

Exhaust flow

Fuel flow + indicated air flow=1312.5+52.5=1365 kg/h



Control volume, 250 kW diesel engine, showing energy and fluid flows

Engine Size / Brake HP

Engine balance table

Fuel flow

Air flow

Cooling water flow

Exhaust flow

Engine size liters to horsepower

The horsepower output depends on more than just the engine size.
For example, a new 4.7 liter engine will have more horsepower than an old 4.7 liter engine. There is no direct conversion between them.

For new cars you may be able to find data sheets that lists both engine size and horsepower.

Example:
Rover 400 Engines

1.4 16v/4	BHP 102
2.0 16v/4	BHP 134
2.0 TD/4	BHP 104

BMW 3-series Engines

1.8/4	BHP 105
3.0D/4	BHP 150
3.0D/6	BHP 181

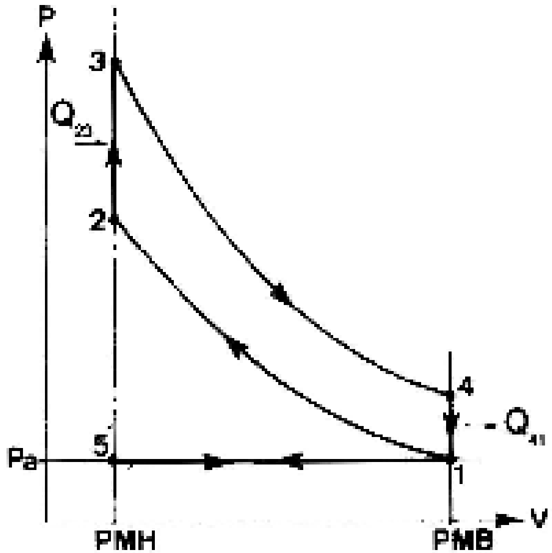
Ferrari F430 4.3 Liter V8 Engine: 490 BHP

5.9 Liter Aston Martin V12 DB9: 450 BHP.

(Brake Horse Power - power available at the flywheel of an engine)

THE THERMAL ENGINE: A THERMODYNAMIC MACHINE

The thermal engine is a thermodynamic machine, which converts the energy released by combustion Q_{23} into pressure work (cycle area). The piston's linear displacement is converted into a rotation by the crank system. This results in an engine torque at crankshaft outlet. The ability to develop this torque at various speeds w enables generating power (P= Cw)

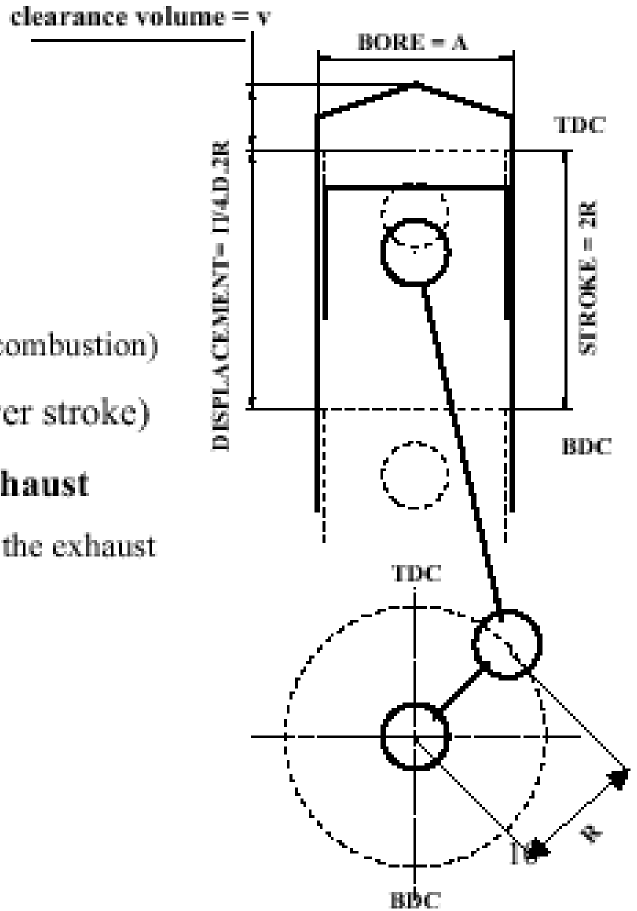


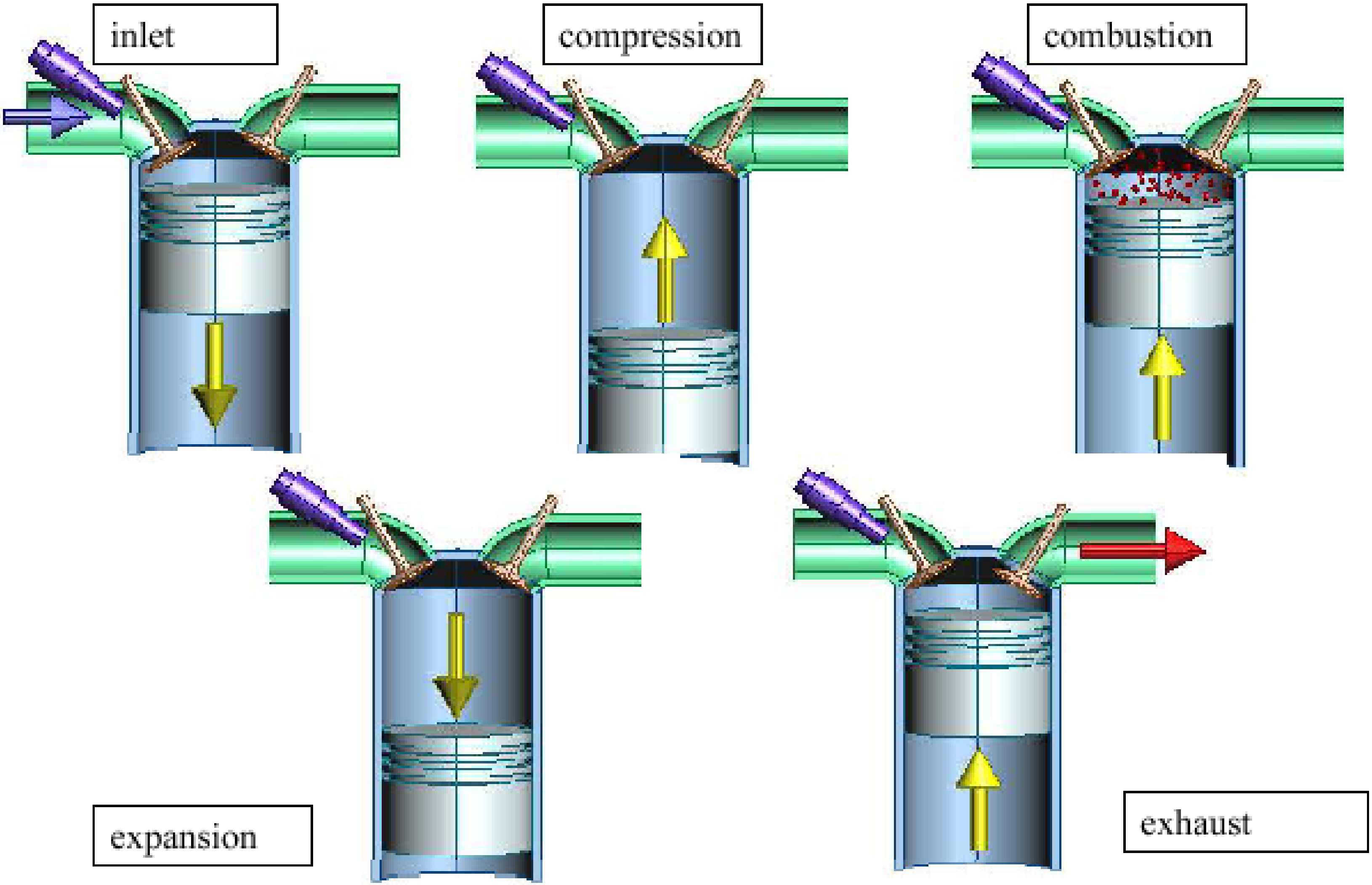
Four strokes:

- **5 to 1 = intake**
- **1 to 2 = compression**
- **2 to 3 = combustion**

  ($Q_{2\text{-}3}$ heat released by combustion)
- **3 to 4 = expansion** (power stroke)
- **4 to 1 = beginning of exhaust**

  ($Q_{4\text{-}1}$ heat contained in the exhaust gases)
- **1 to 5 = exhaust**




inlet
compression
combustion
expansion
exhaust

Thermodynamic system

• **Quantity of constant material isolated from the rest of the universe by real or fictitious boundaries**

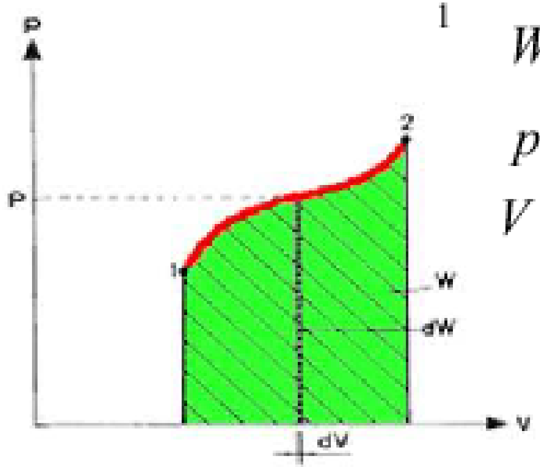
• The work received from a conversion is given by the expression:

$$W_2 - W_1 = -\int_1^2 p dV \quad \text{with} \quad :$$

$$W = \text{Work}$$

$$p = \text{Pressure}$$

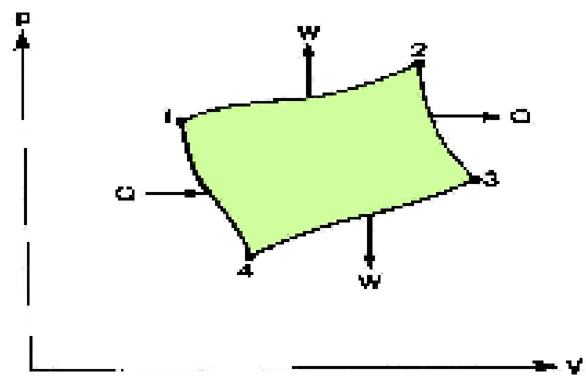
$$V = \text{Volume}$$



• The total energy during a closed cycle is given by the following equation:

$$W + Q = 0$$

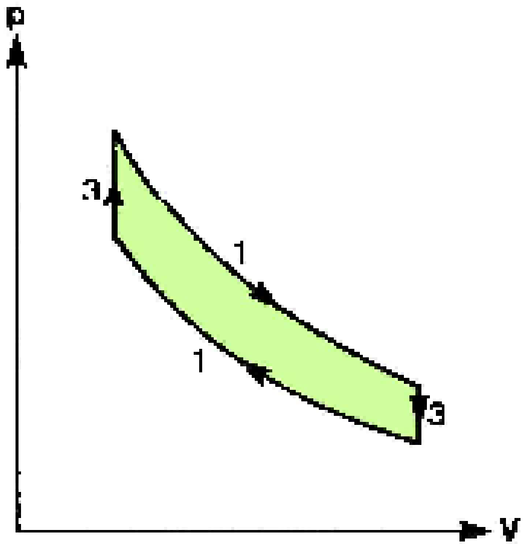
$$with \quad : Q = heat$$



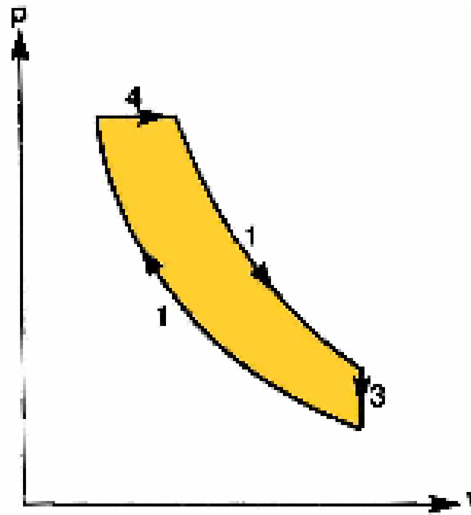
Main theoretical cycles

Spark ignition cycle | **Compression ignition cycle**

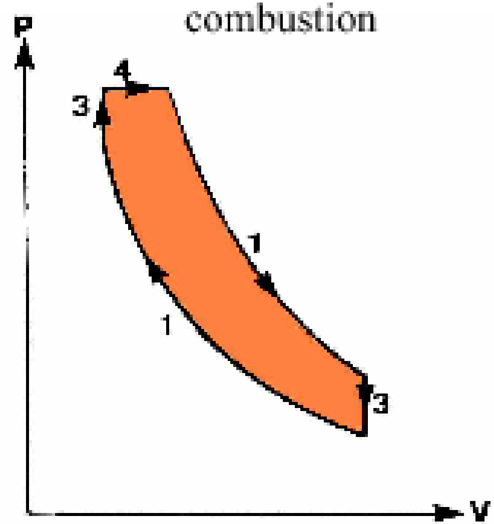
Constant volume combustion



Constant pressure combustion



Constant volume and constant pressure combustion



1: adiabatic conversion, 3: isochore conversion, 4: isobar conversion

Theoretical thermodynamic efficiency

- the theoretical thermodynamic efficiency η_t depends on two parameters, namely:
  - the compression ratio τ the formula for which is:

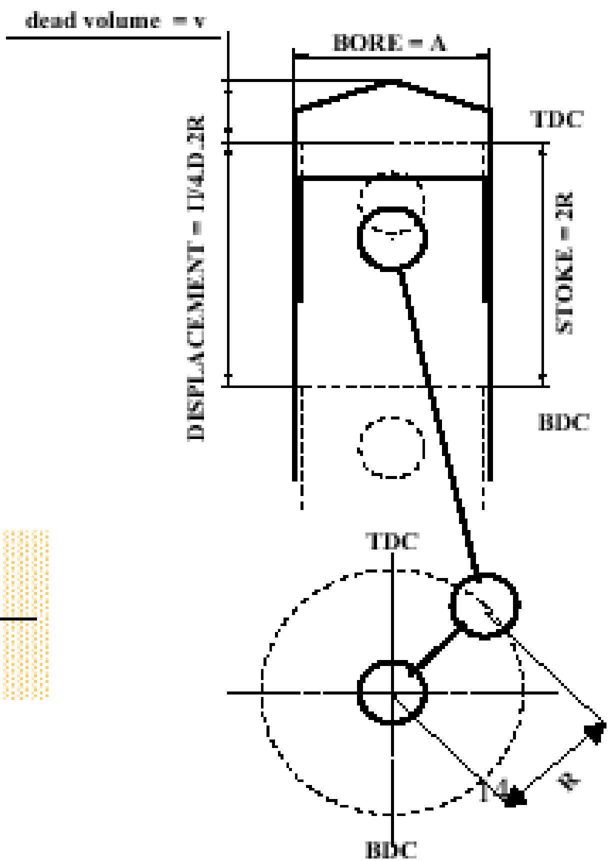
$$\tau = \frac{V + v}{v} \text{ with}$$

V = displacement $\qquad v$ = clearance volume

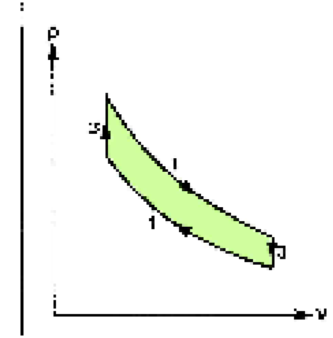
  - and the specific heat ratio γ

η_t has the following expression:

$$\eta_t = \frac{\text{Work}}{\text{Heat released}}$$



Theoretical thermodynamic efficiency



$$-W = (Q_3 - Q_2) + (Q_1 - Q_4)$$

$$-W = C_v(T_3 - T_2)\left(1 - \frac{T_1 - T_4}{T_3 - T_2}\right)$$ calculating T_4 and T_1 as a fonction of T_3 and T_2

$$-W = C_v(T_3 - T_2)\left(1 - \frac{(T_3 - T_2)\tau^{1-\gamma}}{T_3 - T_2}\right)$$ or

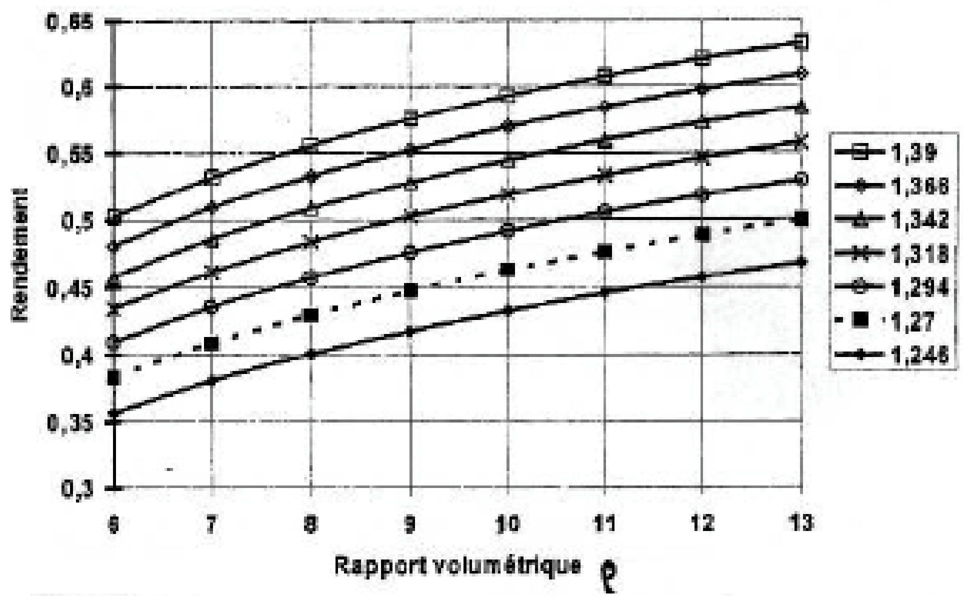
$$-W = (Q_3 - Q_2)(1 - \tau^{1-\gamma})$$ then

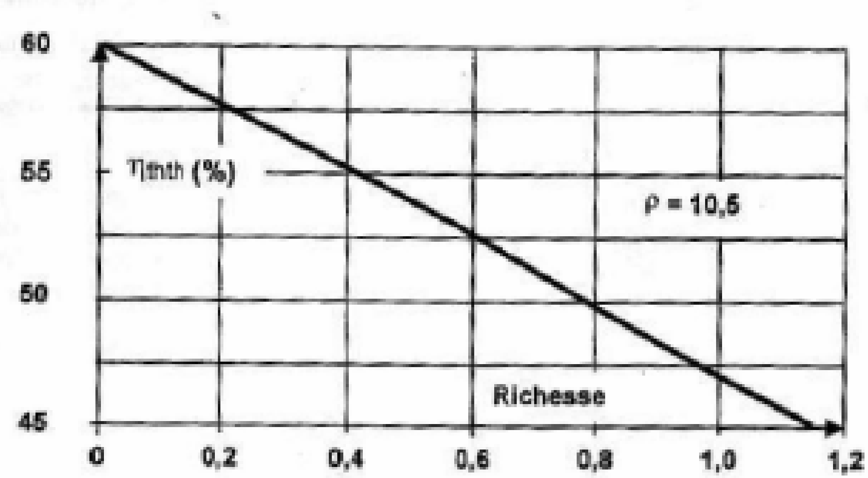
$$\boxed{\eta_t = \frac{-\mathbf{W}}{(\mathbf{Q_3} - \mathbf{Q_2})} = \mathbf{1} - \tau^{\mathbf{1}-\gamma} = \mathbf{1} - \frac{\mathbf{1}}{\tau^{\gamma-\mathbf{1}}}}$$

with $\tau = 10$ and $\gamma = 1{,}4$ the efficiency is $\eta_t = 0{,}60$

Theoretical thermodynamic efficiency: Consequences

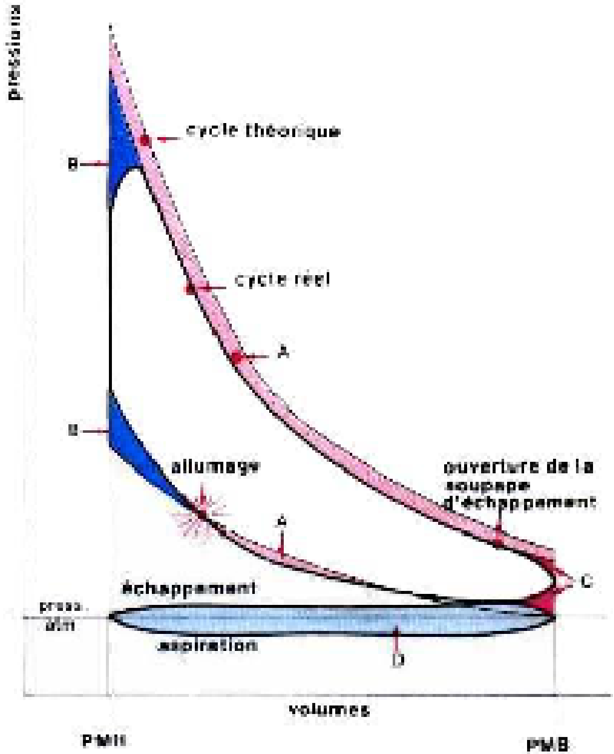
Considering its theoretical formulation, the thermodynamic efficiency can be improved by increasing the compression ratio. However, in a spark ignition engine, the compression ratio is limited by knocking. An increase in the efficiency can also be ensured by increasing γ, which is performed by lowering the fuel-air equivalence ratio.





Fuel-air equivalence ratio influence

Comparison Theoretical cycle – Real cycle

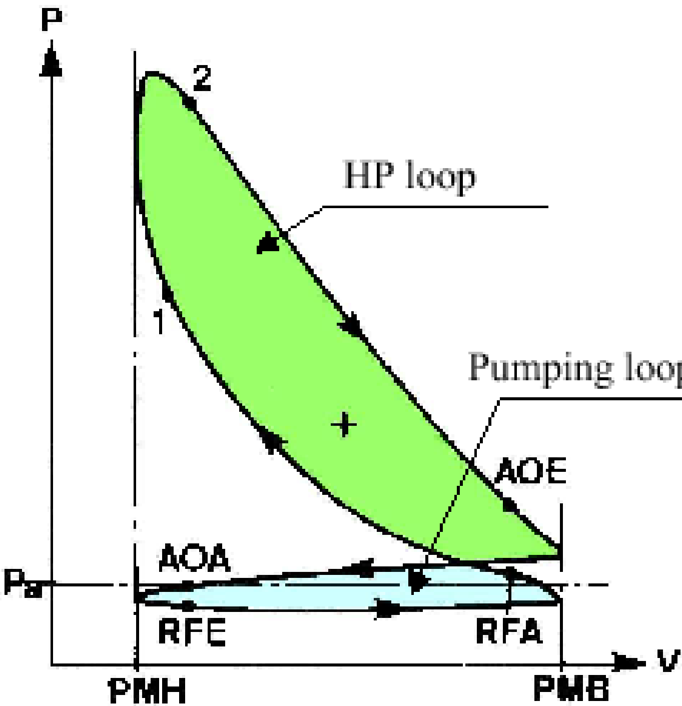


Actual evolutions do not follow exactly theoretical laws, hence the real cycle differs from the theoretical cycle

- the intake circuit and the exhaust circuit introduce pressure loss,
- compression and expansion are not completely adiabatic, as there are heat exchanges with the outside,
- combustion does not occur at constant volume.

We can see the « low pressure » loop, i.e. the area between the exhaust and intake strokes, and the « high pressure » loop, i.e. the compression, combustion and expansion strokes.

The real cycle: indicated work, Imep, cycle efficiency



Indicated work W_i:

The indicated work resulting from the pressure of the gas on the piston is determined by:

$$W_i = \text{surface HP loop - surface pumping loop}$$

indicated mean effective pressure imep:

It is the value for a constant pressure on the piston, which would result for the sweeping of a displacement (engine time) in the same indicated work value:

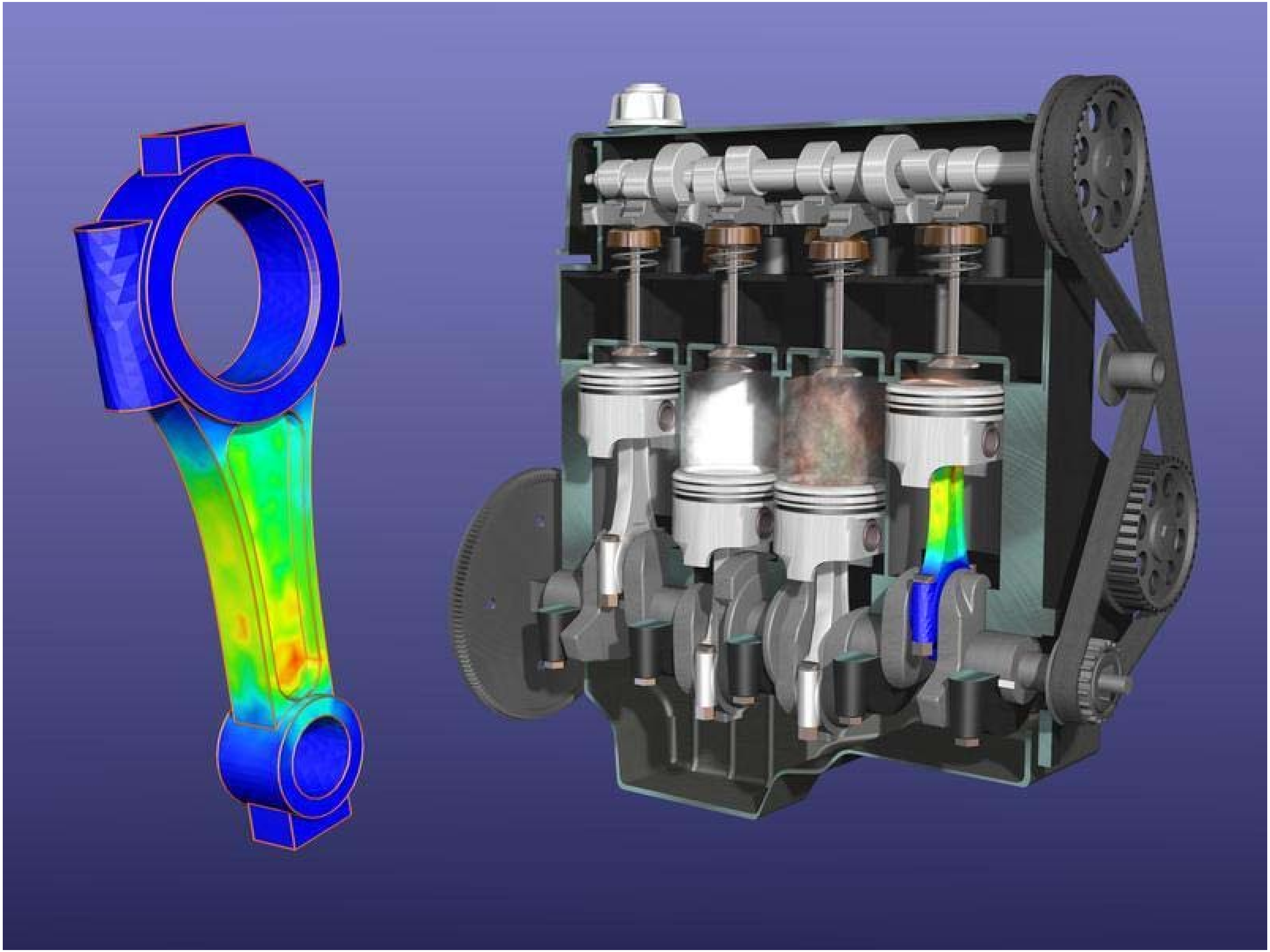
$$\mathrm{Im}ep = \frac{W_i}{V}$$

Cycle efficiency or shape efficiency η_c :

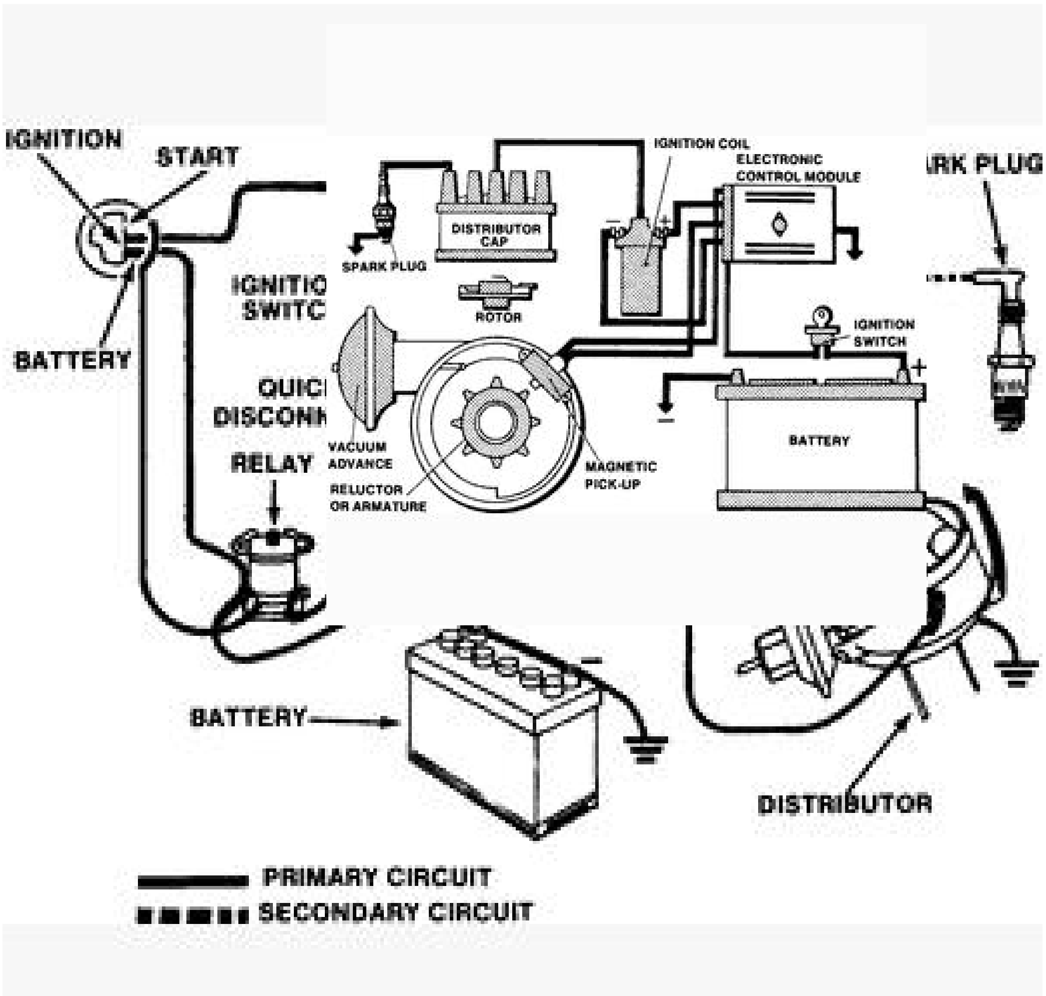
It is the ratio between the Imep measured and the Imep calculated from the theoretical cycle. It changes according to the volumetric efficiency. Its maximum value is 0.8.

18

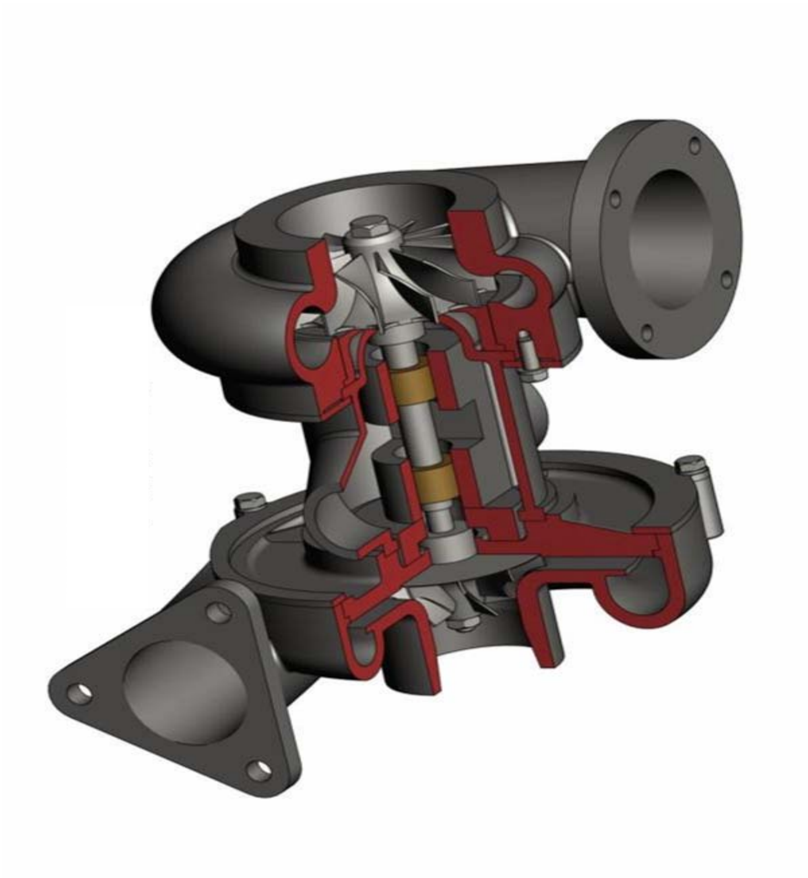
What is happening inside the engine?

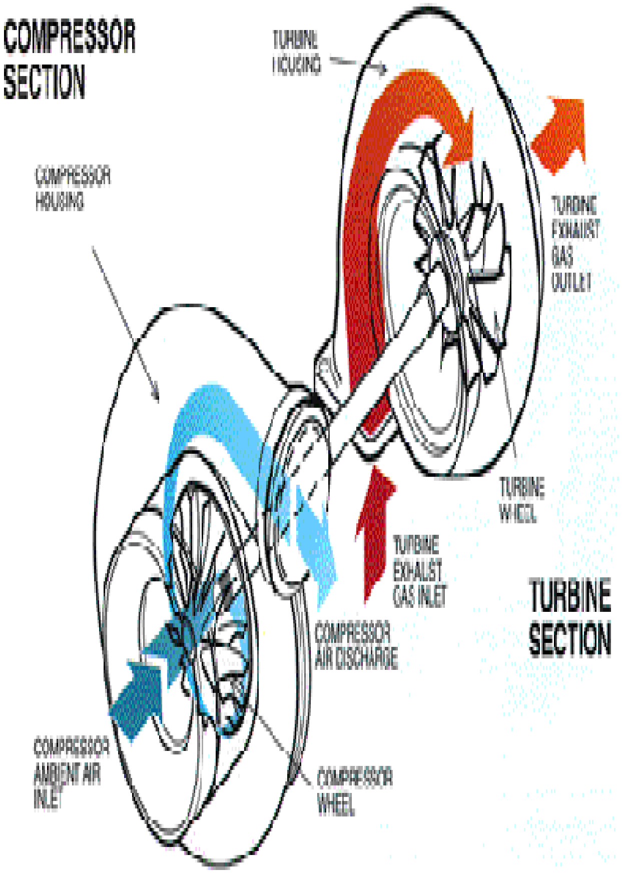


Early ignition system post points

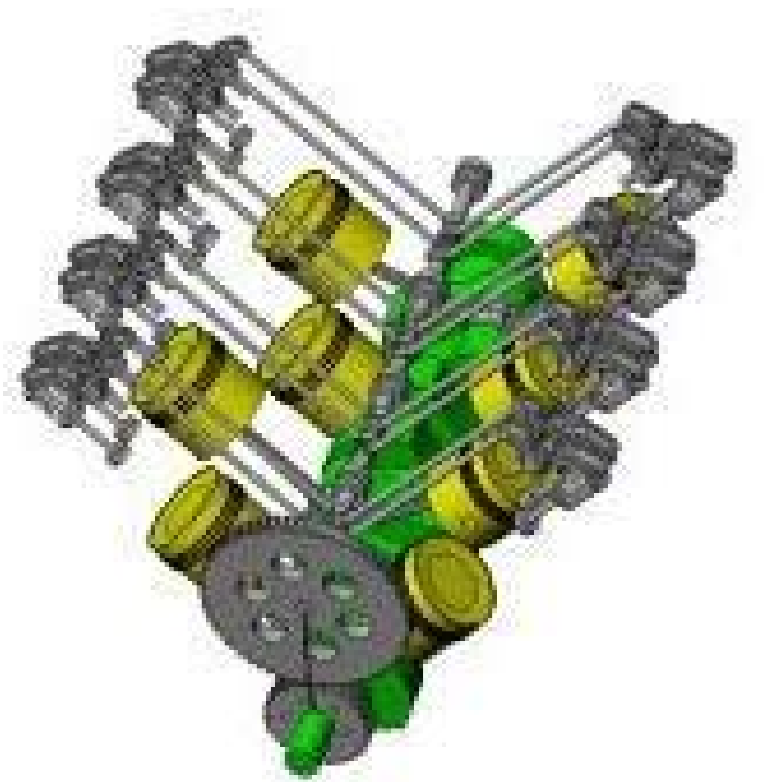


Turbo charger

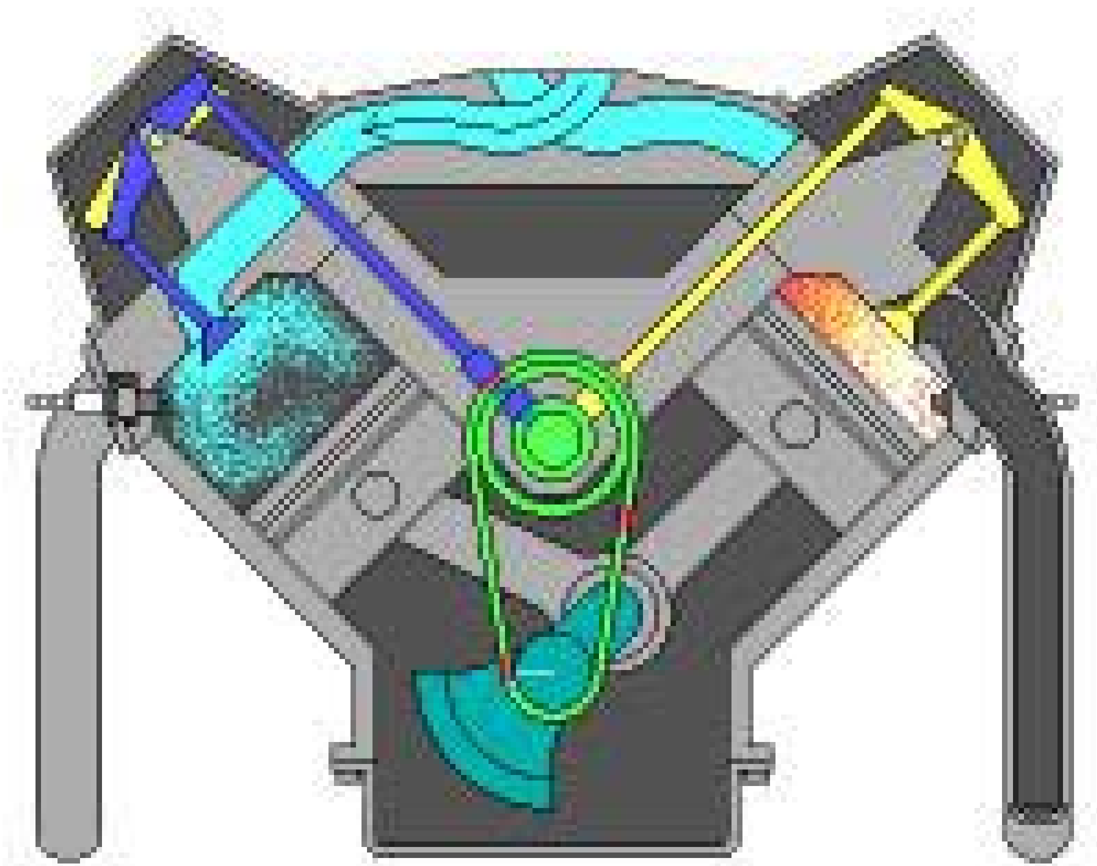


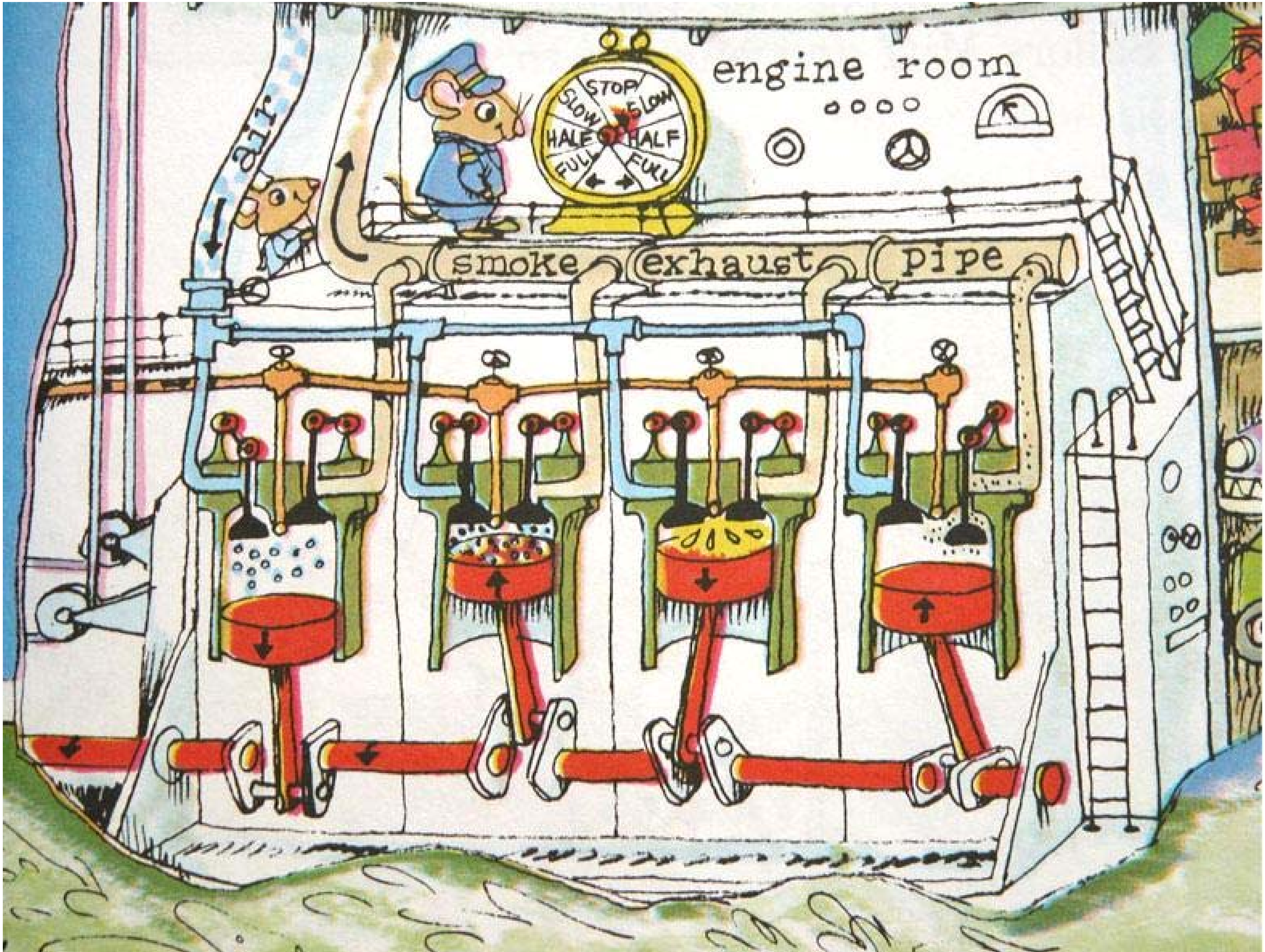


Push rod V8 single cam shaft

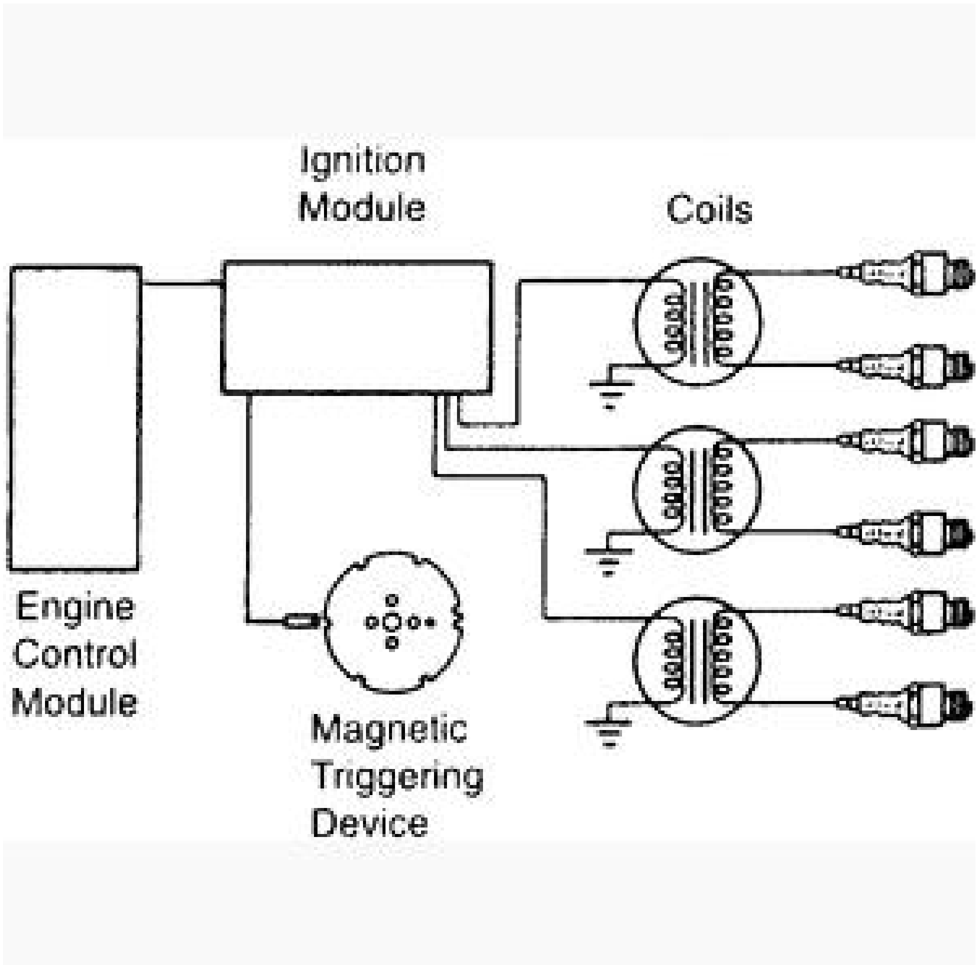


V formation

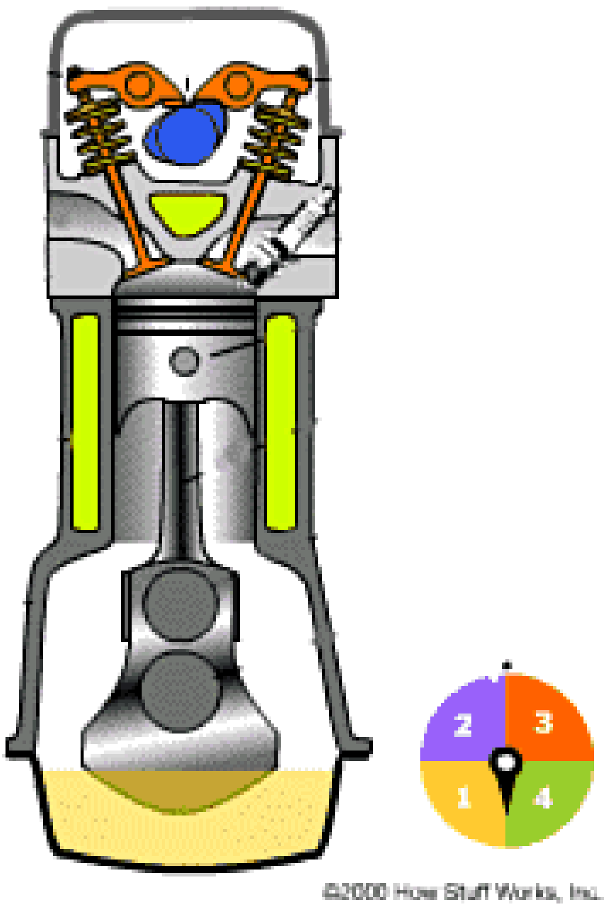



engine room
air
smoke exhaust pipe
STOP
SLOW
SLOW
HALF
HALF
FULL
FULL

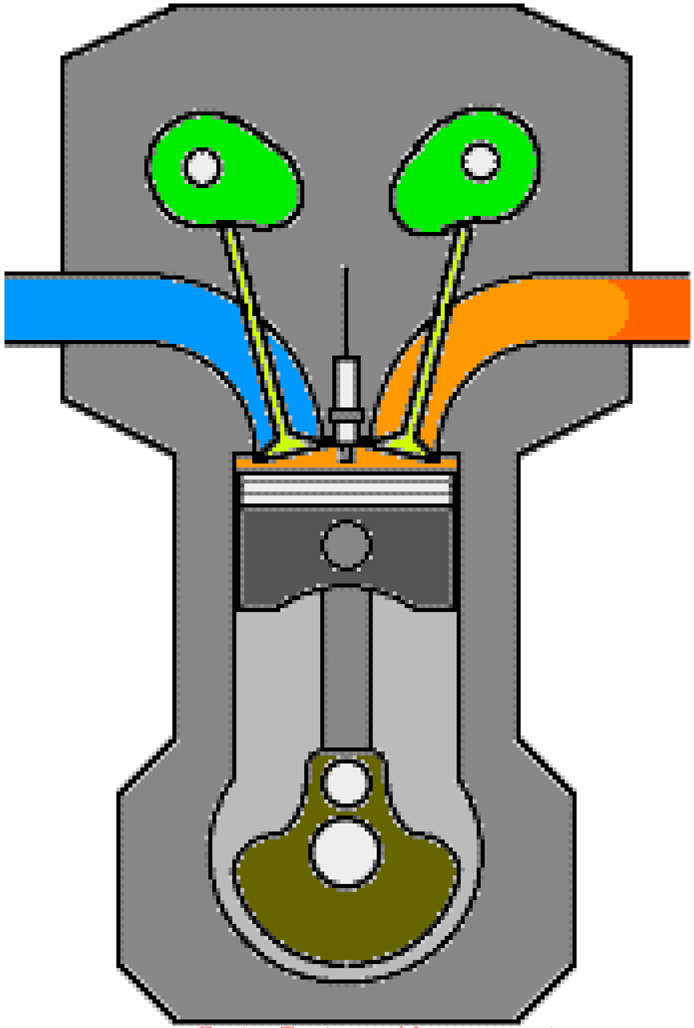
Current ignition system



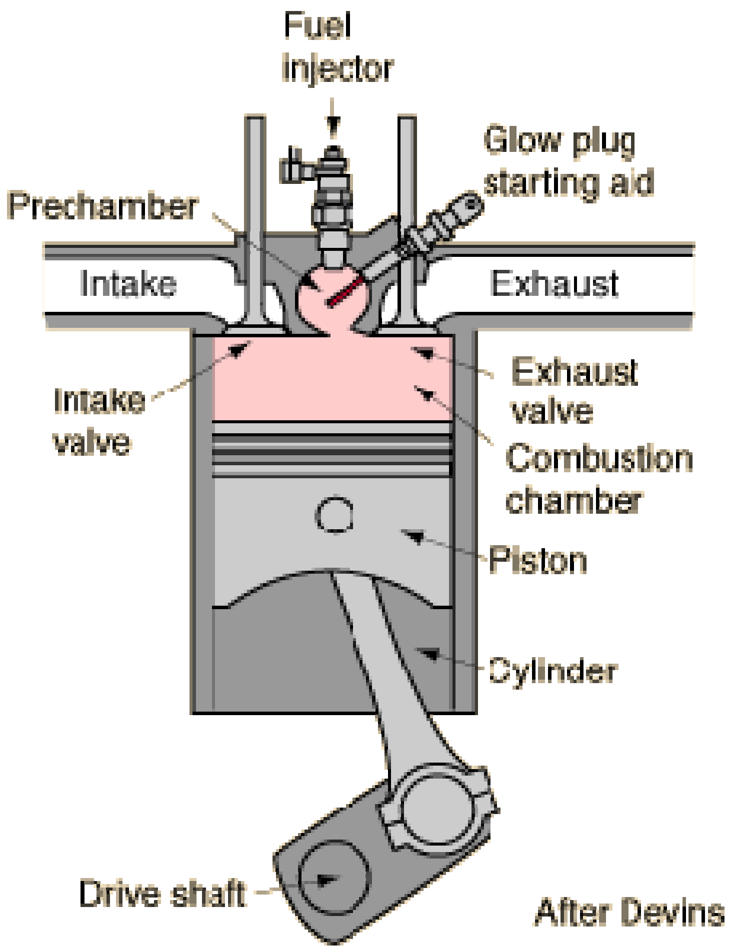
Cylinder block, V 12 (12 cylinders)


2
3
1
4
©2000 How Stuff Works, Inc.

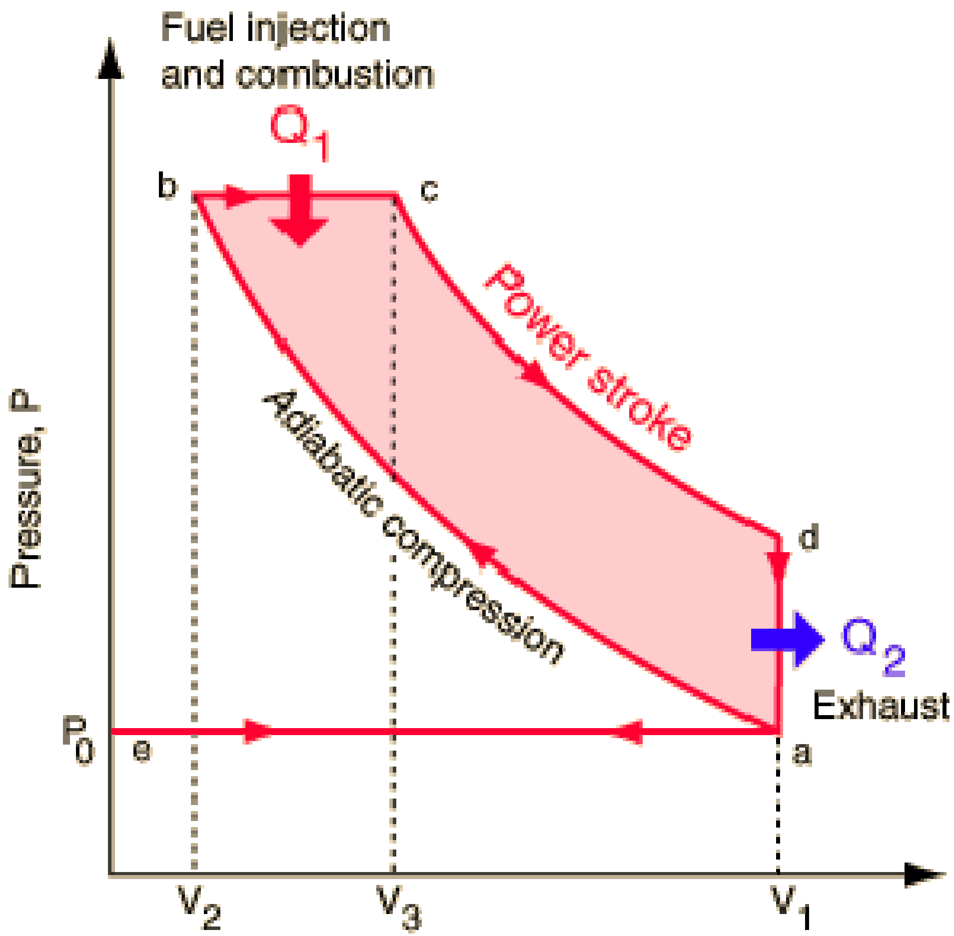
Valve actuation



A Diesel system, note the pre-combustion chamber

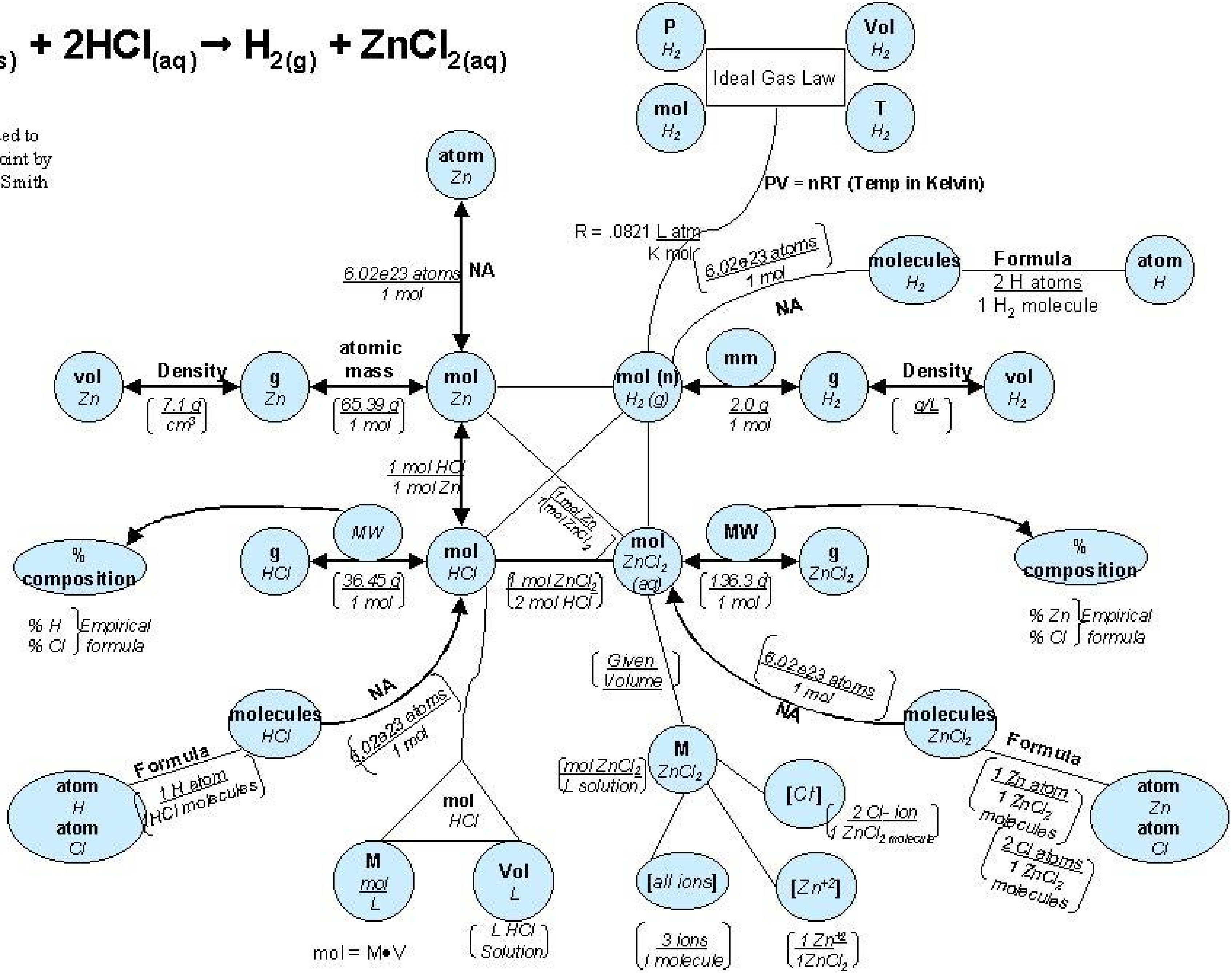


Theoretical pressure volume diagram

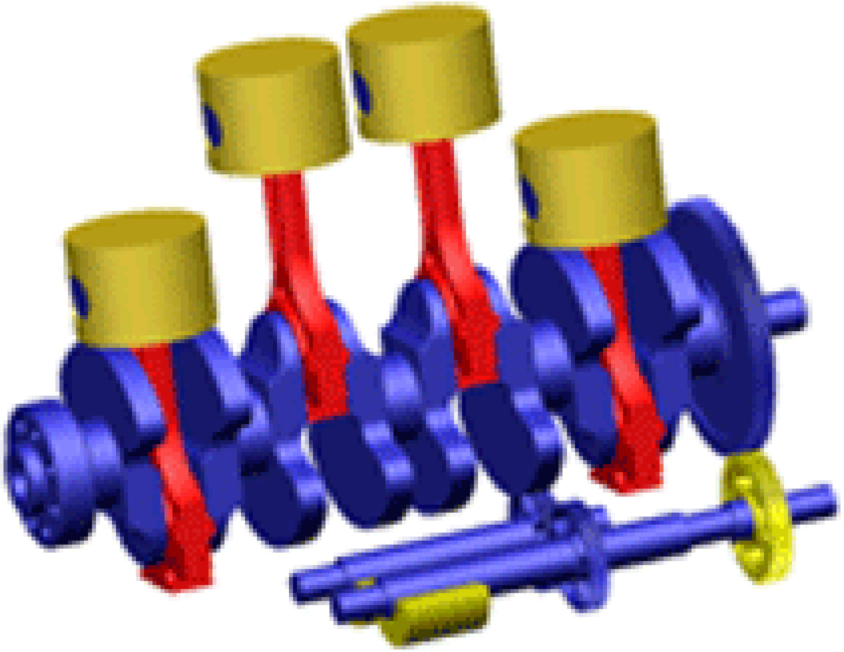


$Zn_{(s)} + 2HCl_{(aq)} \rightarrow H_{2(g)} + ZnCl_{2(aq)}$

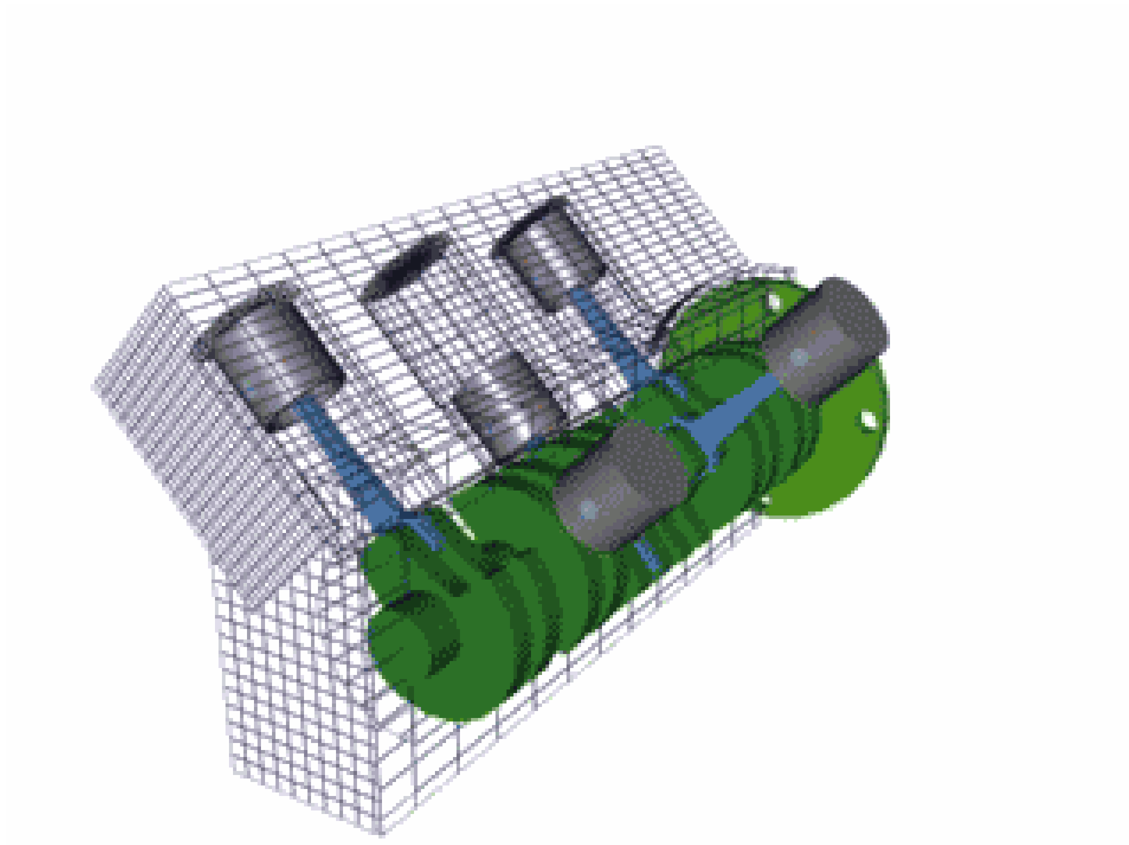
Converted to PowerPoint by Maddie Smith



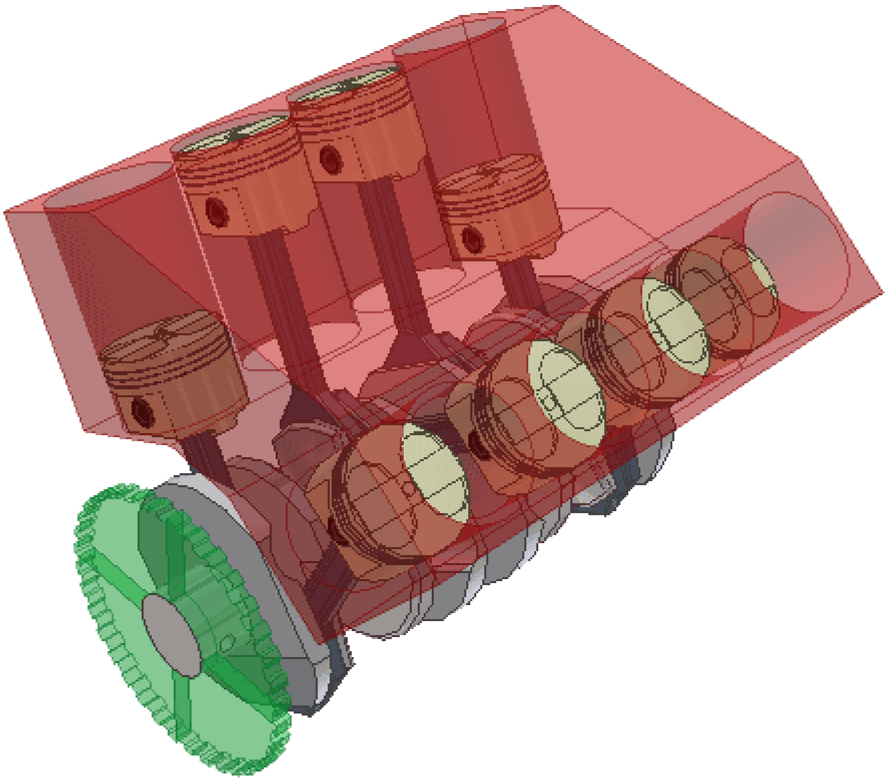
This is a attempt to balance the forces from the crankshaft throws and piston inertia forces.



V6

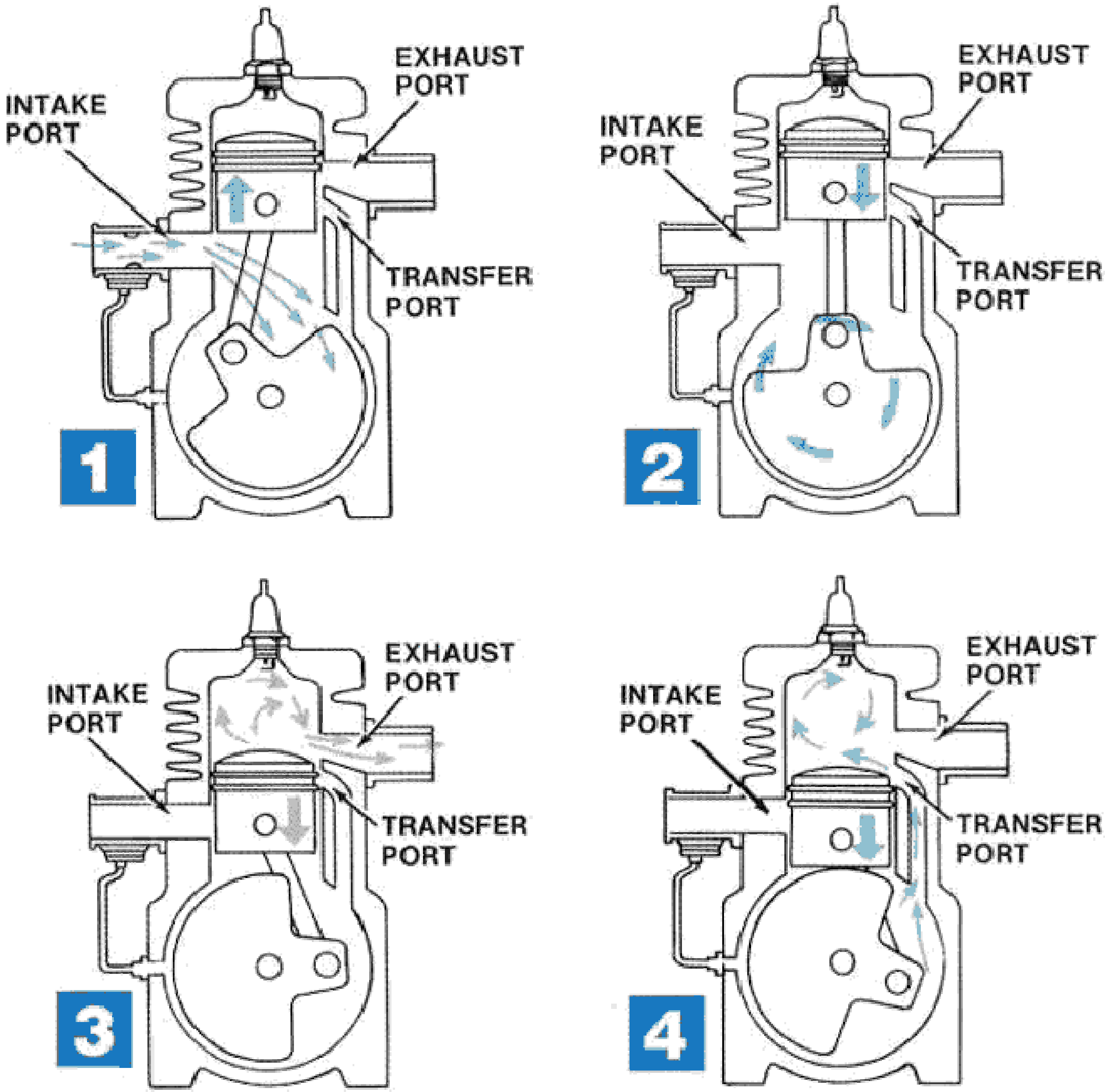


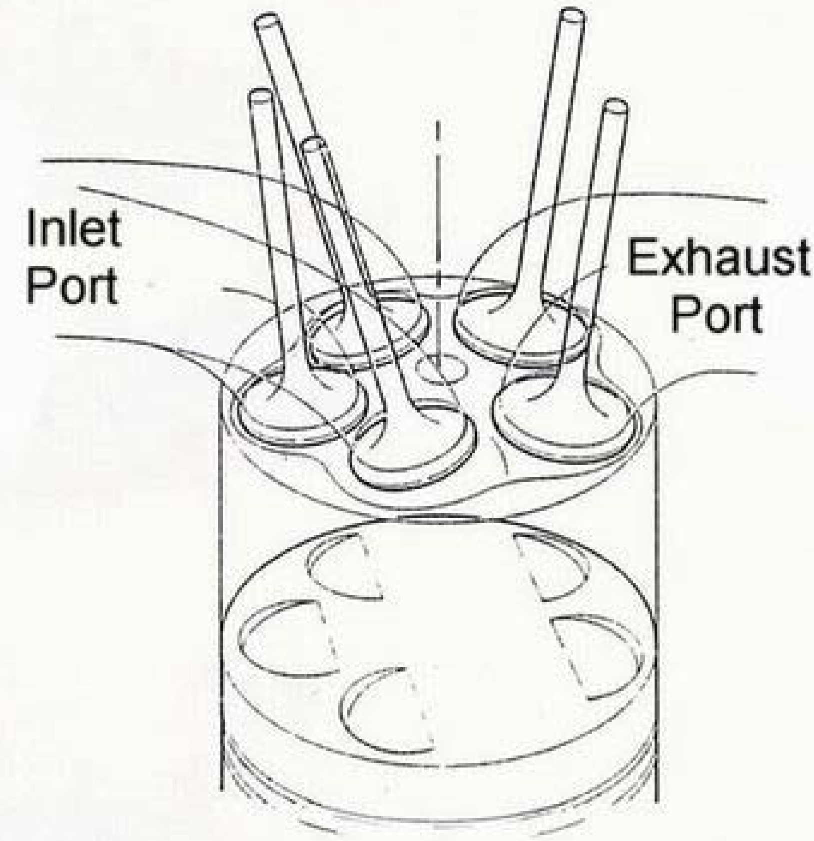
V8 Cylinder Block



Crankshaft

How many cylinders and how many main bearings


INTAKE PORT
EXHAUST PORT
TRANSFER PORT
1
INTAKE PORT
EXHAUST PORT
TRANSFER PORT
2
INTAKE PORT
EXHAUST PORT
TRANSFER PORT
3
INTAKE PORT
EXHAUST PORT
TRANSFER PORT
4

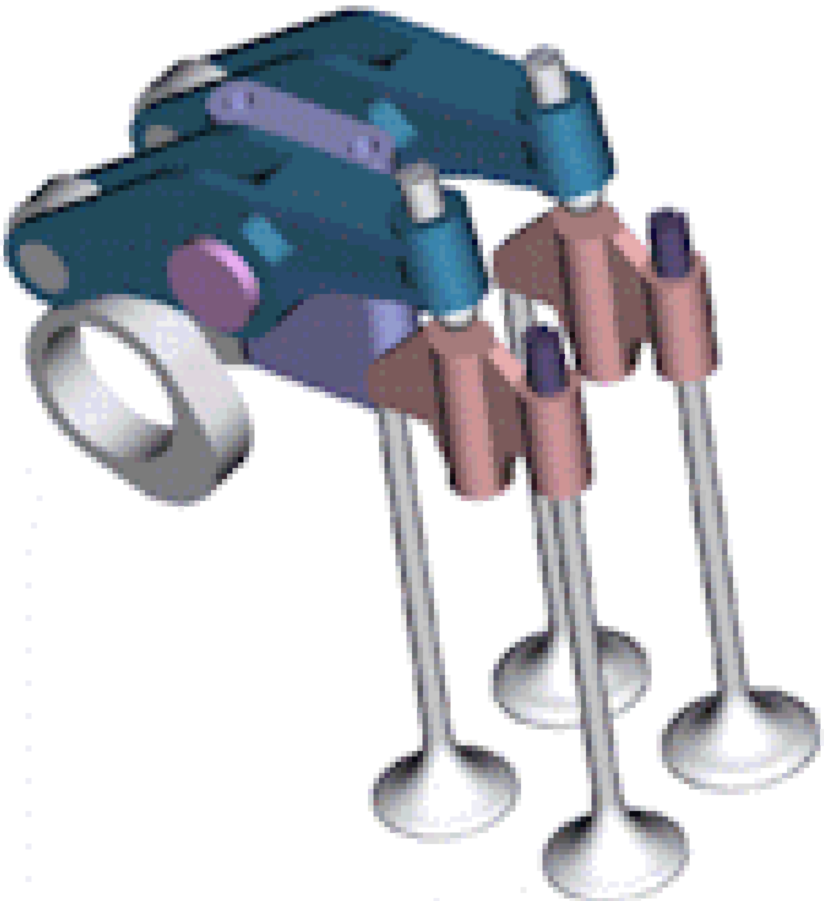


5 Valve Cylinder Head Layout

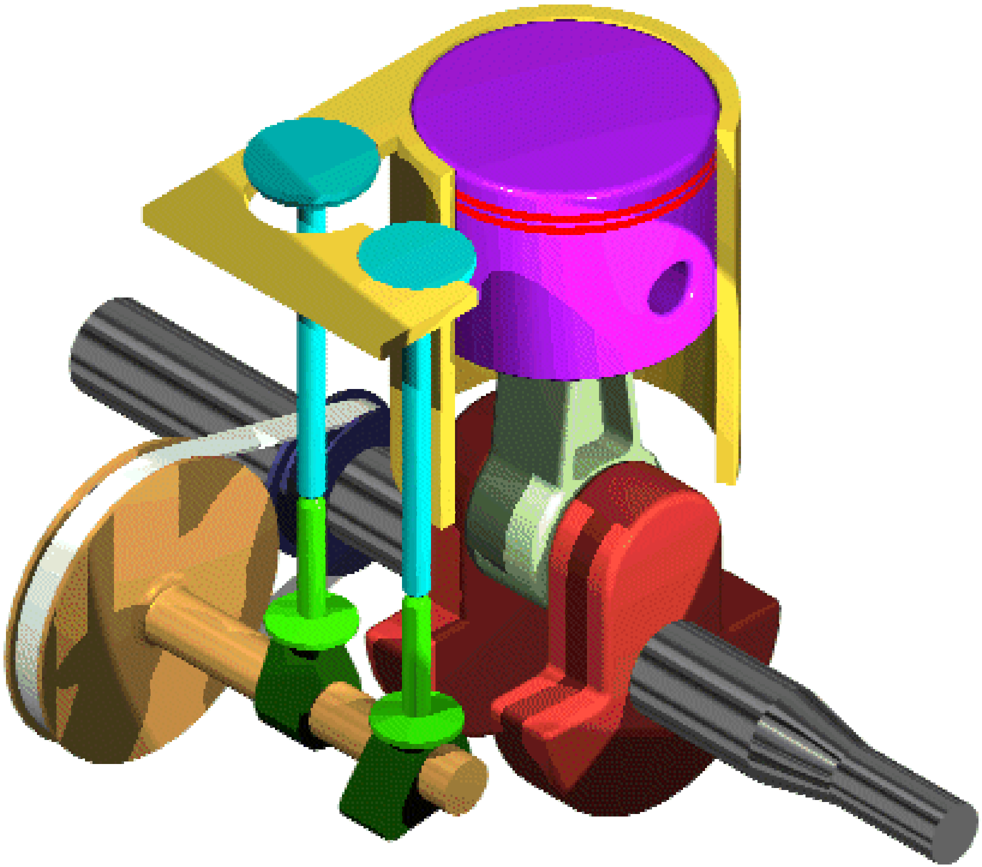
Tested and rejected for Jaguar V8

A cylinder head, 4 valves per cylinder, central sparking plug location

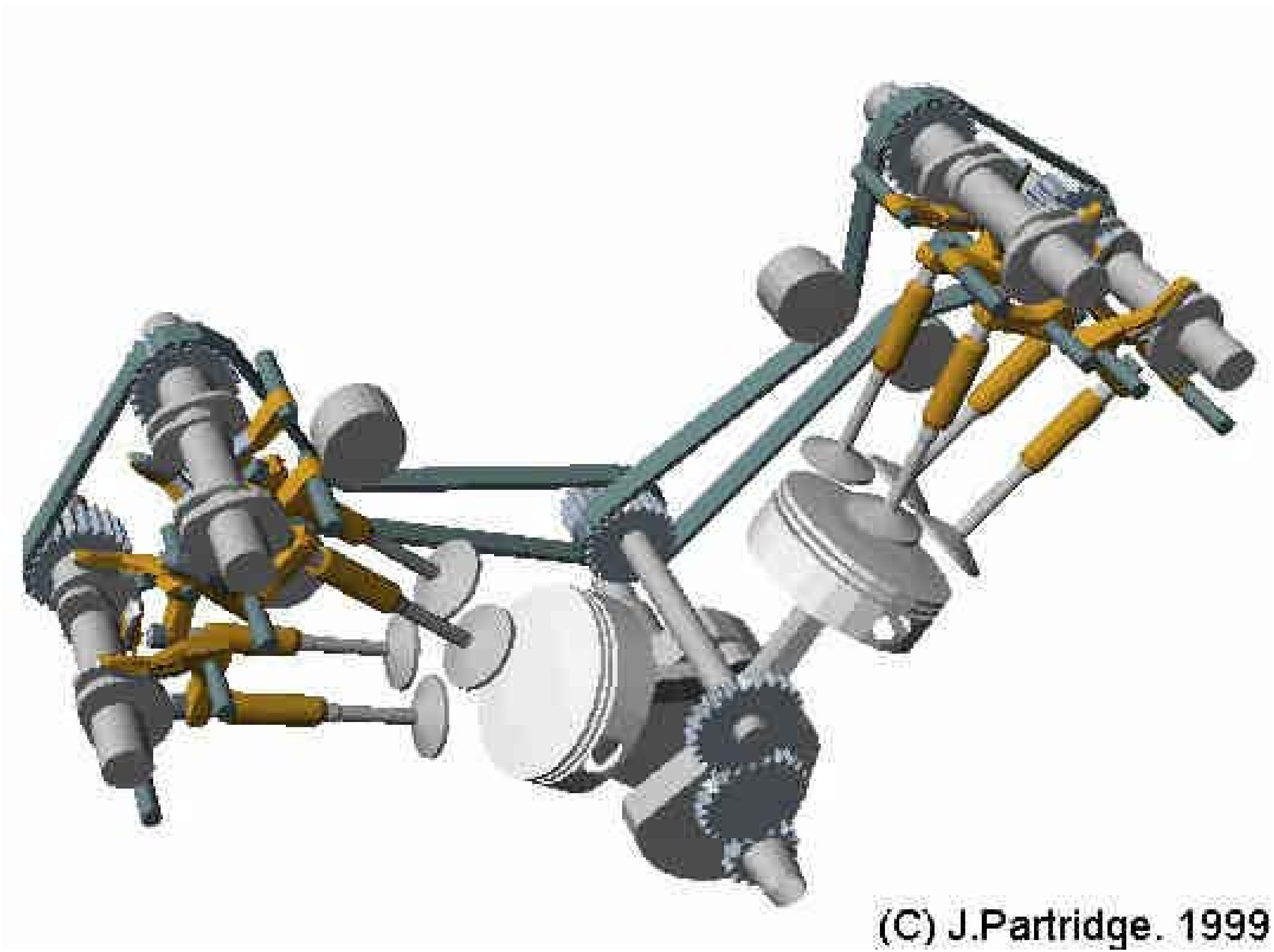
Unusual design of valve actuation



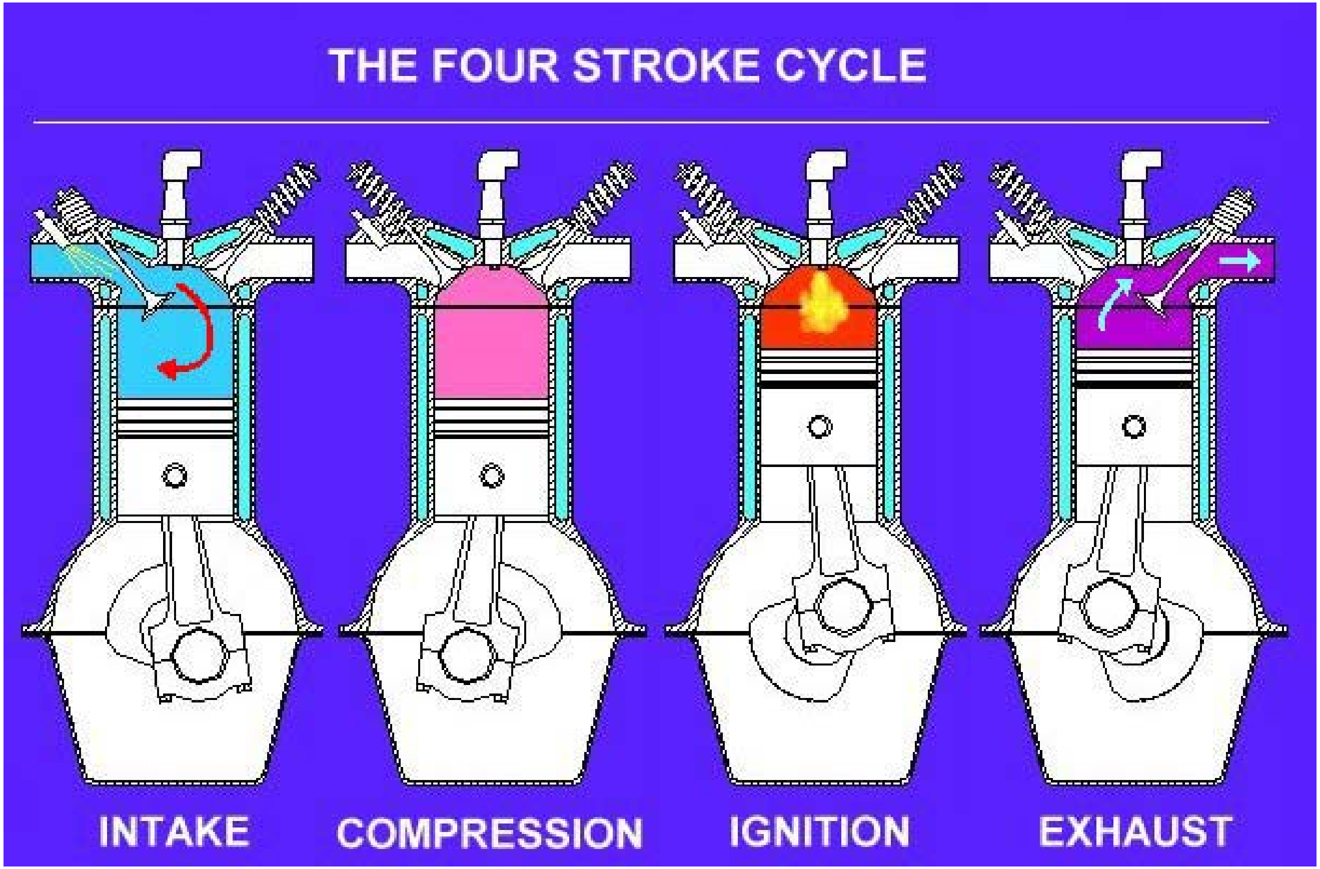
Valve actuation

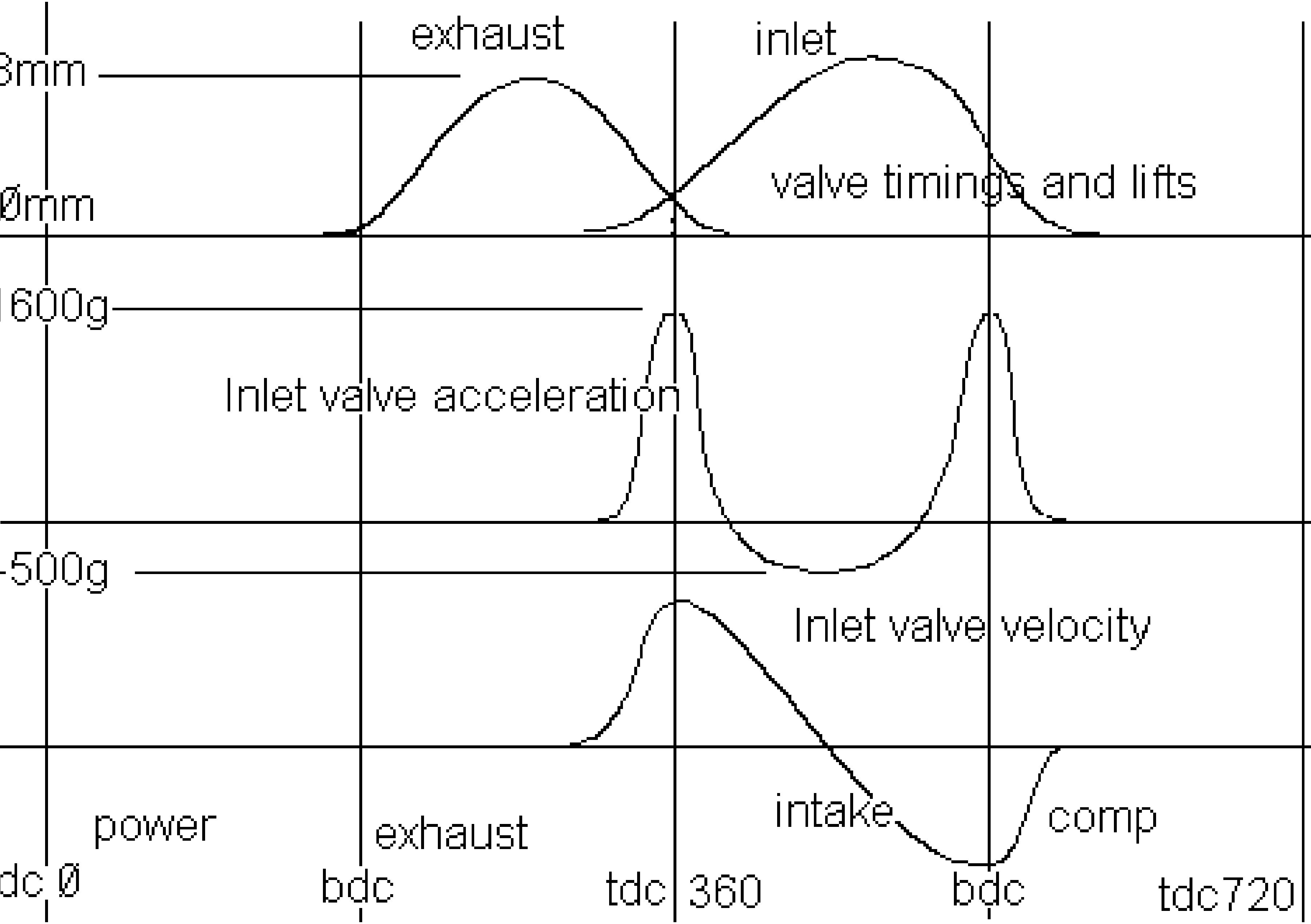


Desmidromic valve actuation



Re-cap of the four stroke cycle

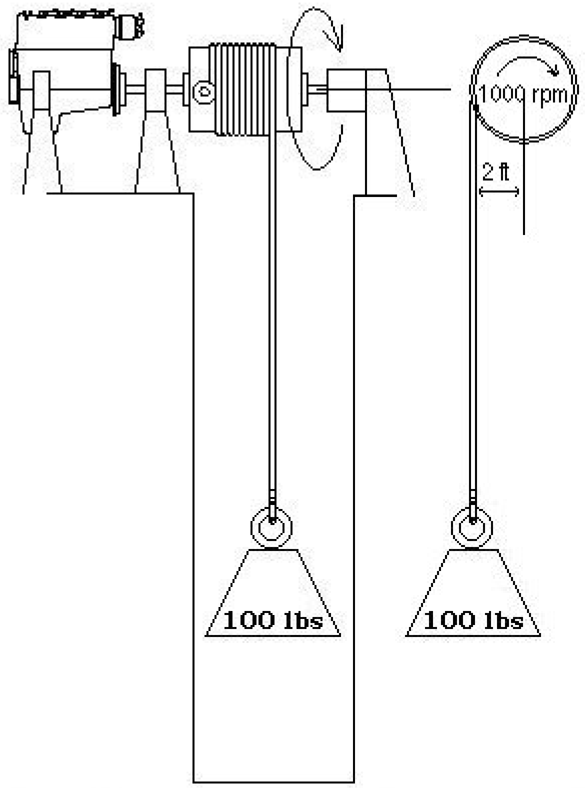



exhaust
inlet
mm
Ømm
valve timings and lifts
600g
Inlet valve acceleration
-500g
Inlet valve velocity
intake
comp
power
exhaust
dc Ø
bdc
tdc 360
bdc
tdc720

Abbreviations

WOT - Wide Open Throttle
SCI - Stoichiometric Compression Ignition
Duratec HE - The Duratec HE is the name used by Ford of Europe for its family of small straight-4 and V6 gasoline engines.

Measurement of Power



Imagine an engine sited at the top of a deep well turning a drum, which is four feet in diameter, i.e. 2 feet radius. A rope attached to the drum is hanging down the well with a weight of 100 lbs. on the end. As the engine turns the drum it will lift the weight. The drum is four foot in diameter and the rope is being pulled in at two foot from the centre of rotation; therefore the work being done or torque is measured as 2ft x 100lbs = 200 foot pounds.

The speed at which the drum is rotating is measured as Revolutions Per Minute (R.P.M).

B.H.P is calculated as follows

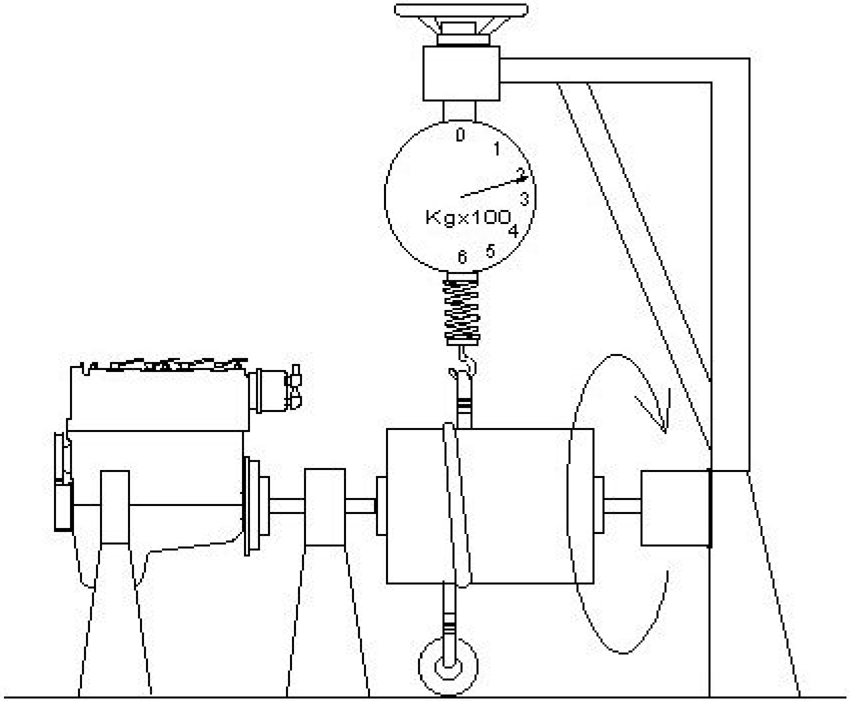
$$\frac{\text{TORQUE X R.P.M}}{\text{CONSTANT}} = \text{B.H.P}$$

The constant depends on the units of torque, which are being measured. As we are using ft.lbs, it will be 5250, so if we say that the engine is turning at 1000 R.P.M then:

$$\frac{200 \text{ X } 1000}{5250} = 38 \quad \text{B.H.P}$$

To better understand the way in which the dynamometer works, imagine anchoring a spring balance to the ground, with a rope attached to the top eye and wrapped around a drum with a slipknot is tightened as the drum is rotating, the rope will be tensioned and the balance will extend to indicate this tension as a 'weight'.

friction between rope and drum will slow the drum and its driving engine until, at 1000 R.P.M, the spring balance reads 100 lbs. The weight being lifted is 100 lbs and the speed of the drum or engine will then be used in the formula to calculate the horsepower.



If the speed were 1500 R.P.M this would mean the engine was lifting the weight faster and exerting more power to do so.
The calculation would then be:

$$\frac{200 \times 1500}{5250} = 57$$

Power and Torque

- The powertrain is essentially involved in the production of power and torque in a convenient usable manner. An important equation involved with this is shown below.
- Brake Power is a product of the mass flow rate of air and the inverse of the brake specific fuel consumption, for a given fuel air ratio. The specific fuel consumption is effectively a measure of the engines ability to convert the chemical energy to mechanical energy.

$$P_b = \frac{\dot{m}_a}{bsfc}(F/A)$$

SFC

- The specific fuel consumption is the inverse of the product of the efficiencies that describe the energy conversion process and the specific calorific value of the fuel.
- These efficiencies take on the form of thermodynamic and mechanical variables.
- Combustion efficiency η_c

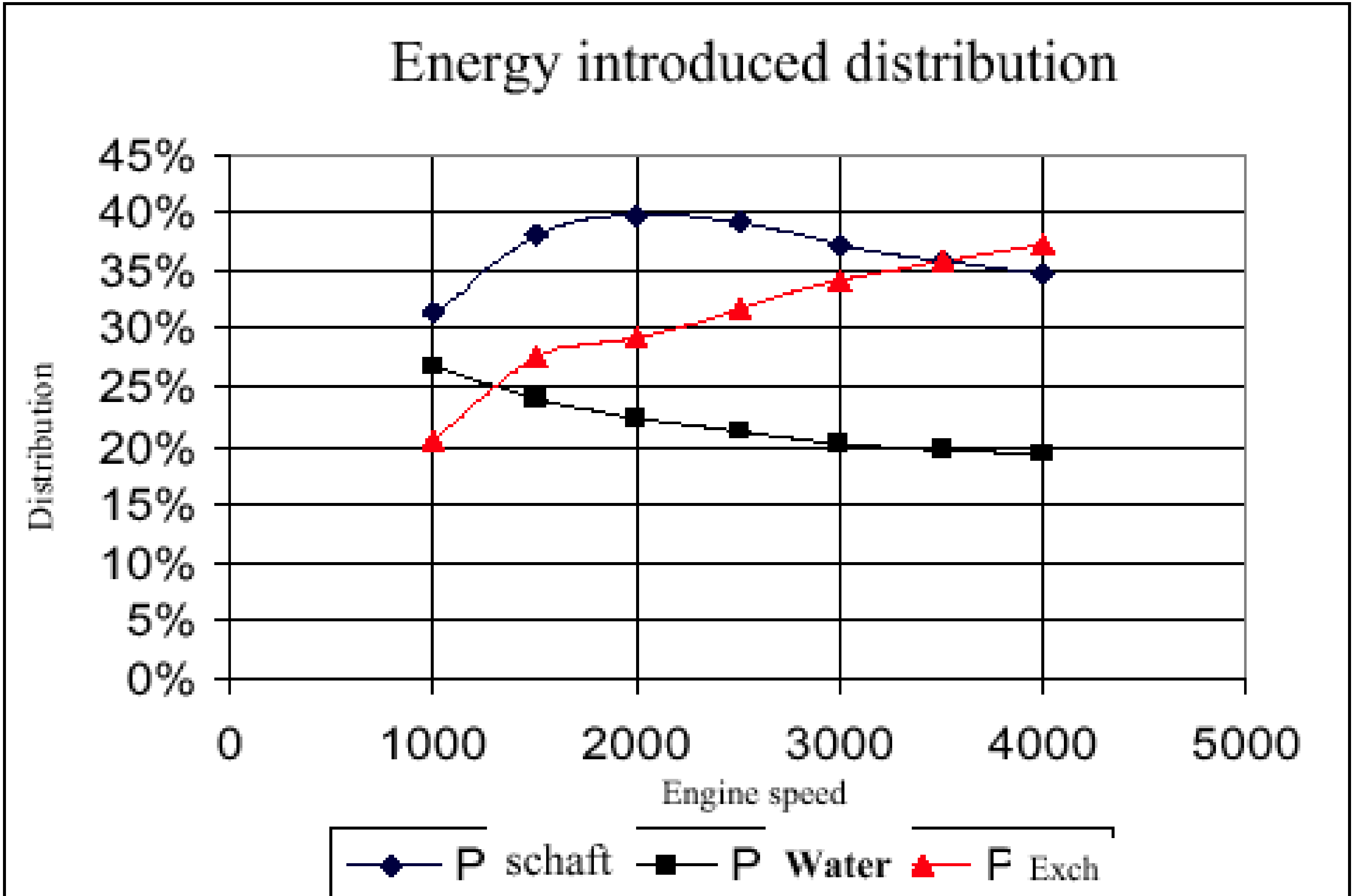
$$bsfc = \frac{1}{\eta_m \eta_i \eta_c Q}$$

THERMAL EXCHANGE IN AN ENGINE

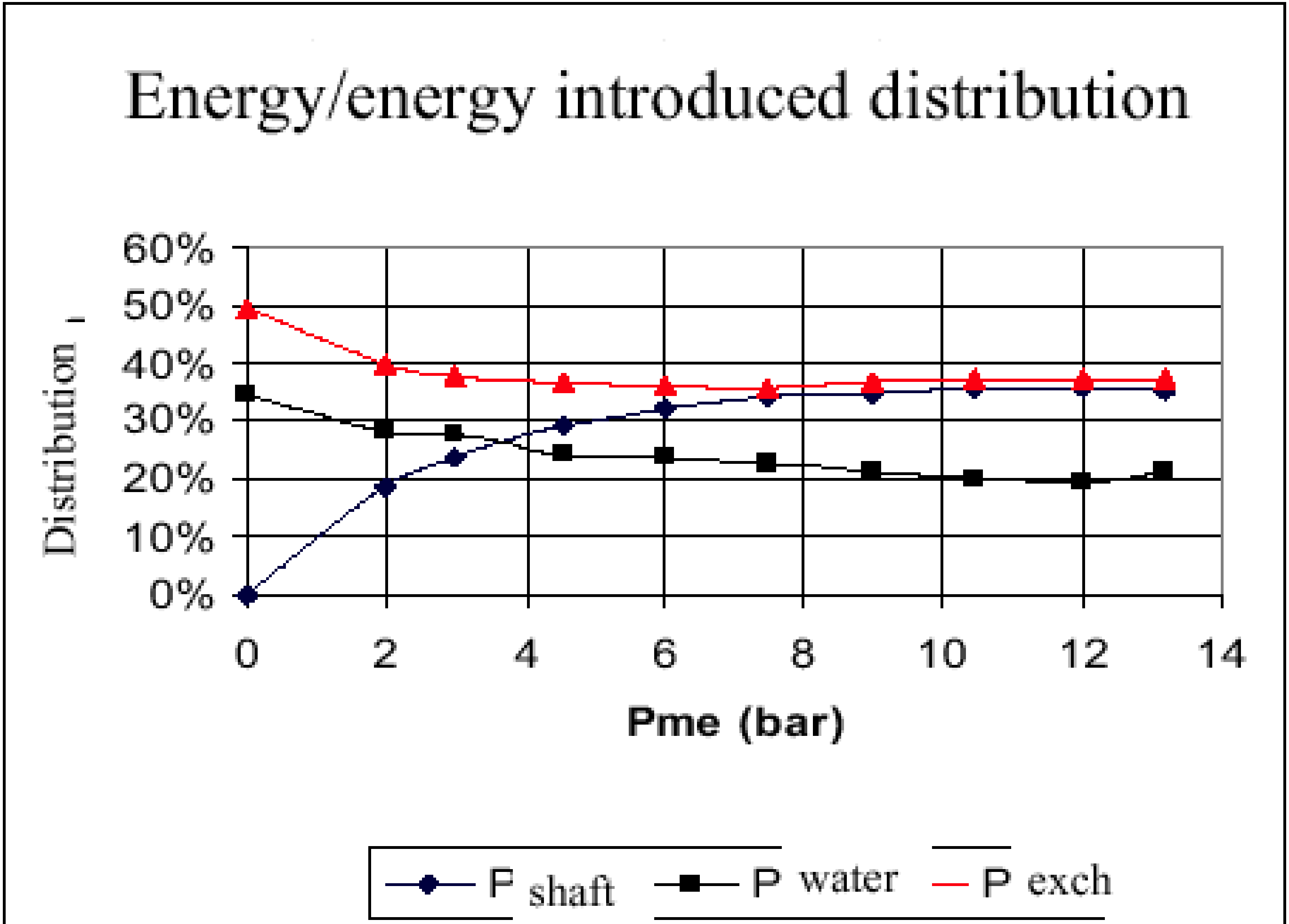
The main source for heating the engine parts is the heat exchange between the combustion gases and the walls of this chamber (friction between the parts being of the second order).

Hence, the temperature of the assemblies, such as the cylinder head, results from the balance between the heat provided by the gases (compression, then combustion) and:
- That evacuated by the cooling circuit (main),
- That evacuated by the oil,
- That radiated towards the outside.

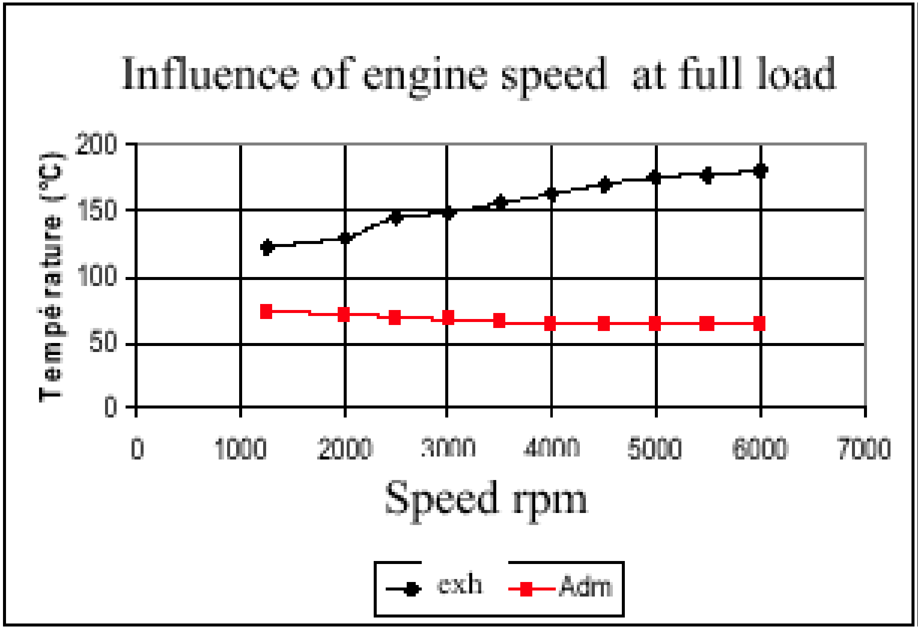
Thermal balance of a direct injection Diesel engine over a full load curve

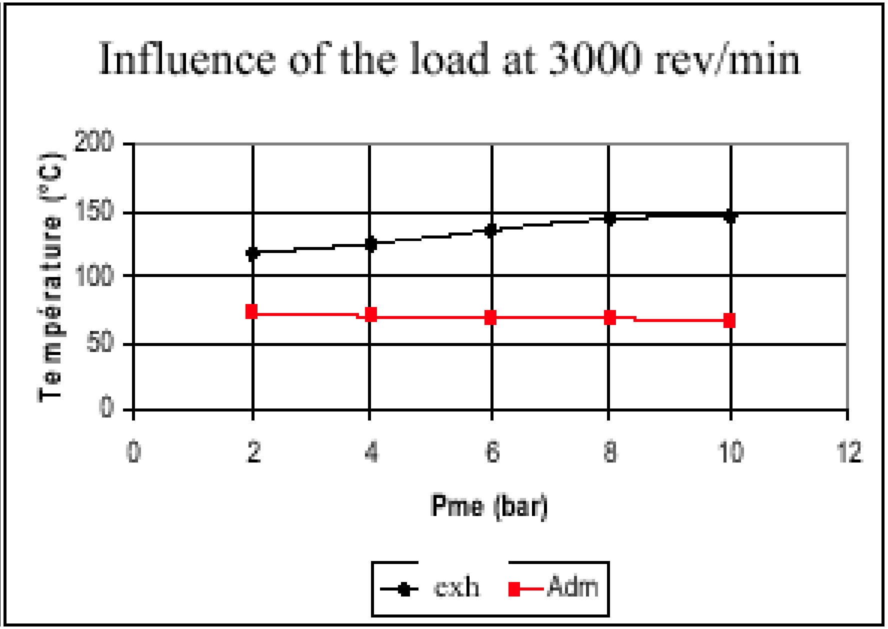


Thermal balance of a direct injection Diesel engine over the isospeed 4000 rev/min



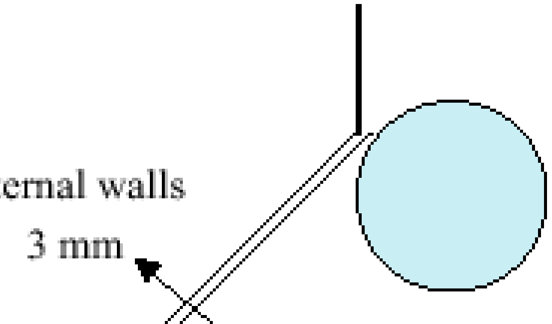
INFLUENCE OF THE RUNNING PARAMETERS ON THE THERMAL LOAD FOR A C.I ENGINE



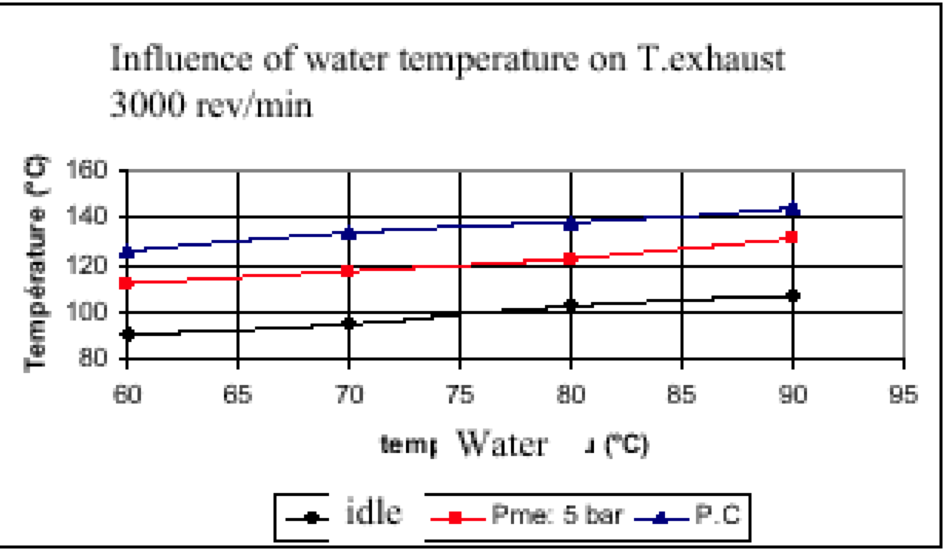


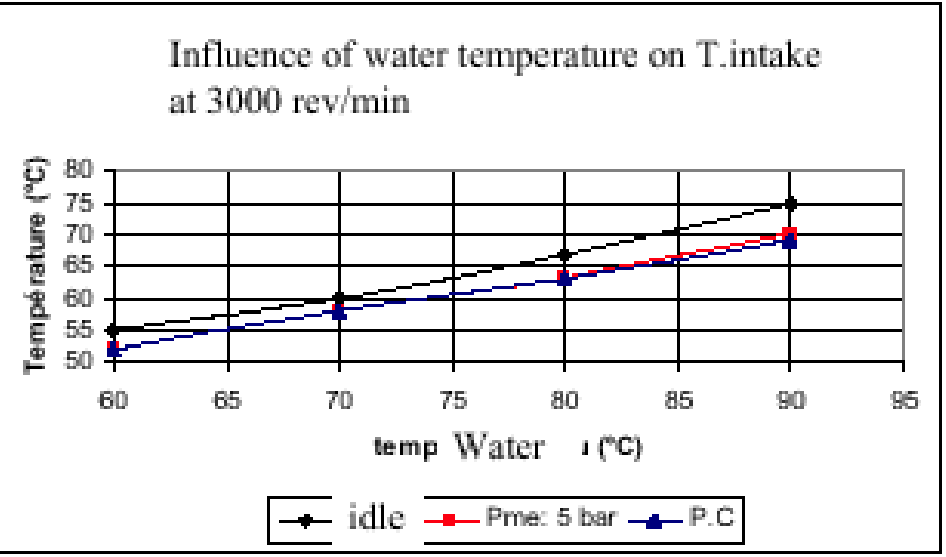
The influence of the engine's running parameters on the thermal load is locally very variable

Measurements made in the material, at 3mm from the duct internal walls

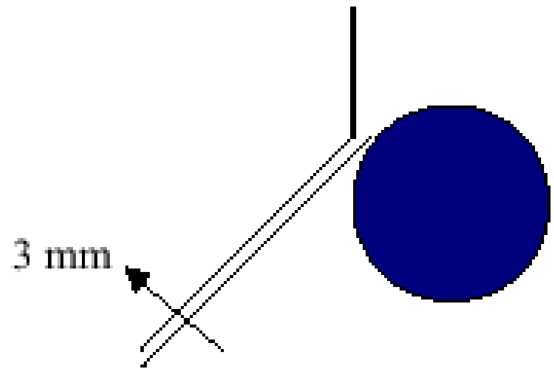


INFLUENCE OF THE RUNNING PARAMETERS ON THE THERMAL LOAD OF A C.I. ENGINE





Measurements made at 3mm from the internal duct wall



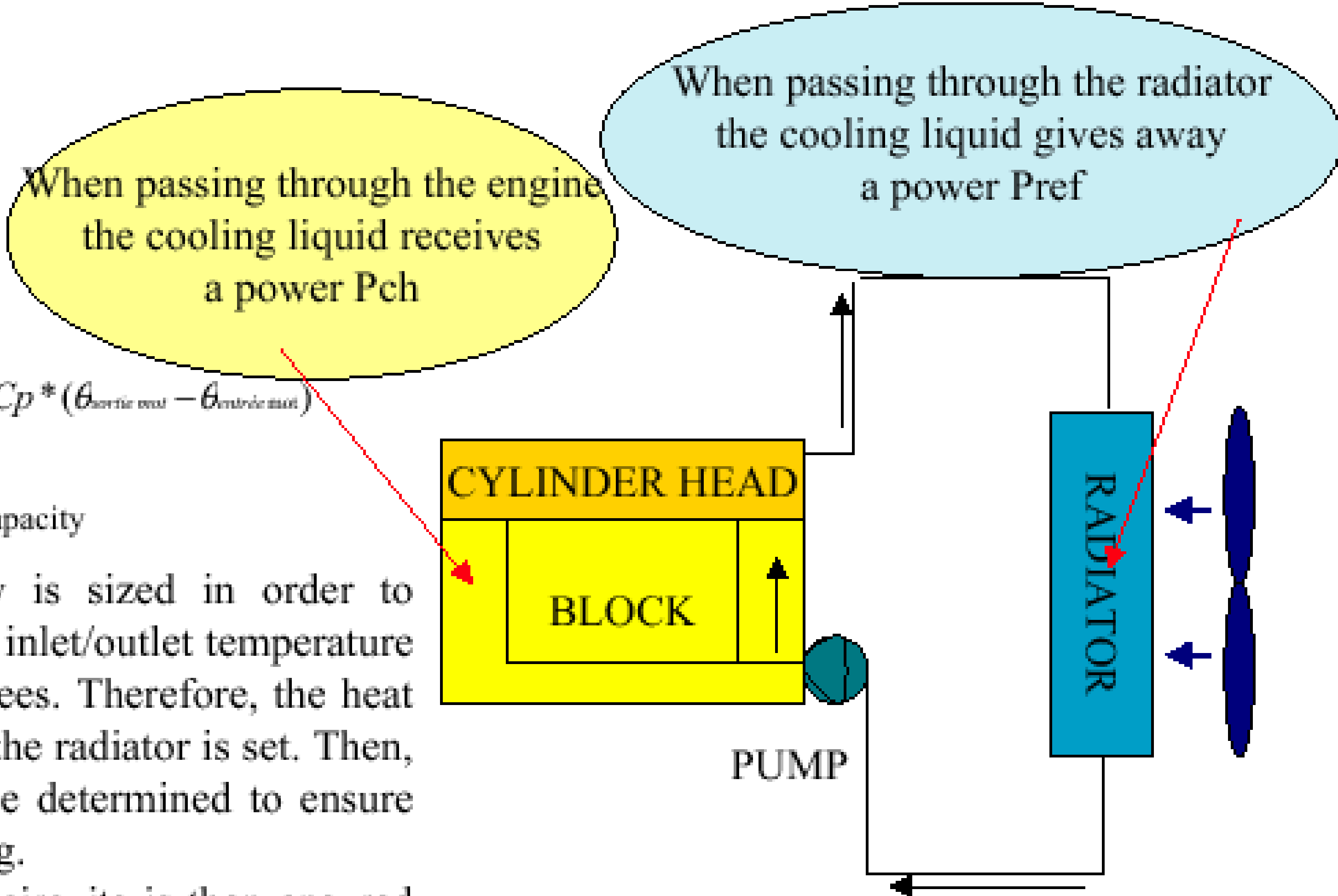
Thermal exchange in an engine

The heat exchanges between the gases in the combustion chamber and the walls are mainly convective (in Diesel, the radiated part is not totally negligible). The phenomena are hence:

Heating power = h*Surface* (Tgas-Twall)
h: thermal exchange ratio
Surface: gas and wall exchange surface
Tgas: temperature of the gases; Twall: wall temperature

Heat evacuation is performed thanks to the cooling liquid flow. Hence, we have:
Cooling power = dm/dt*Cp* (Ts-Te)
dm/dt: liquid flow
Cp: massic heat
Ts: outlet temperature
Te: intake temperature

PROBLEM



$pech = Débit * Cp * (\theta_{sortie\ mot} - \theta_{entrée\ mot})$

θ : *température*

Cp : Calorific capacity

The pump's flow is sized in order to restrict the engine inlet/outlet temperature delta to 8-10 degrees. Therefore, the heat exchange ratio in the radiator is set. Then, its surface must be determined to ensure the liquid's cooling.

The sizing of the circuits is then ensured as to reduce load loss.

$$pech = \textbf{Flow} * Cp * (\theta_{sortie\ rad} - \theta_{entrée\ rad})$$

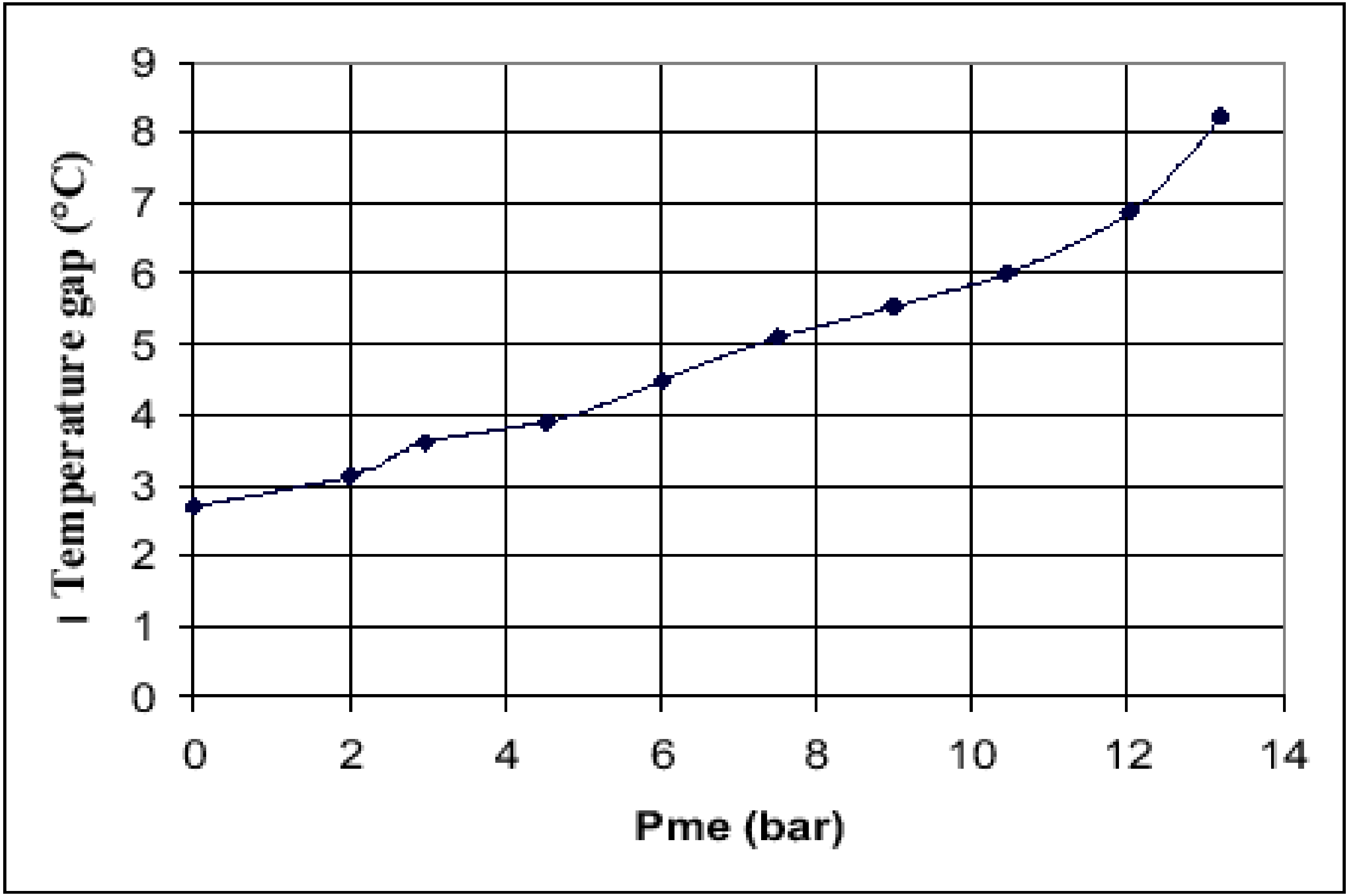
Characterisation of thermal exchanges over a full load curve

Over a full load curve, the engine globally runs at constant fuel-air ratio, hence leading to a constant cycle temperature. The gas-wall thermal exchanges being mainly convective at cycle iso temperature, the power exchanged is virtually proportional with the engine speed (power 0.8).

As the pump's flow rate is proportional with the engine speed (with mechanically driven pumps) and the power to evacuate is virtually proportional to engine speed, the cylinder head inlet-outlet temperature delta is globally constant.

CYMINDER HEAD INLET-OUTLET TEMPERATURE DELTA OVER AN ISO-SPEED

Unlike over an iso-speed curve, the (mechanical) pump flow rate is constant, hence as the power transmitted to the cooling liquid is linear with the bmep, the cylinder head inlet-outle temperature delta is also linear with the bmep



HEAT TRANSFER IN THE ENGINE: N= 6000 rev/min

Influence of the water flow

V= 2m/s

$h_{eau} = 5893 w/m^2/K$

$hg = 700\ w/m^2/K$

$\lambda/e = 53333\ w/m^2/K$

$h_{global} = 618 w/m^2/K$

T gaz = 1000 °C

T water = 100 °C

T int wall = 205 °C

T ext wall = 193°C

V= 4m/s

$h_{eau} = 10261 w/m^2/K$

$hg = 700\ w/m^2/K$

$\lambda/e = 53333\ w/m^2/K$

$h_{global} = 647 w/m^2/K$

T gaz = 1000 °C

T water = 100 °C

T int wall = 168 °C

T ext wall = 157°C

PRESSURE LOSS
PHYSICAL PHENOMENA:
FLUID FLOW ENERGY

The energy in a fluid flow is 'stored' either in the form of speed (kinetic energy), or in the form of a static pressure (in our applications, we do not consider gravity because of the altitude discrepancies on the circuits).

Therefore, in a path where the fluid is subject to no energy loss (no friction, no singularities) nor receives any energy (no pump), energy conservation translates into the following equation:

$$p + 1/2 * Ro * V^2 = \text{constant}$$

with : p static pressure

Ro density of the liquid

V flow rate speed

PRESSURE LOSS
PHYSICAL PHENOMENA: ENERGY LOSS

In real life, there is no situation for which there is no energy loss along the circuit.

This « pressure loss: Dp » energy loss results from:

- the friction against the walls of the organs and within the fluid,
- the passing through particular « singularity » areas, such as: elbows, junctions, flarings, restrictions.

In these conditions, along a path from a point 1 to a point 2, the previous equation becomes:

$$p_1 + 1/2\ Ro_1 V_1^2 = p_2 + 1/2^*\ Ro_2 V_2^2 + Dp$$

Generally, pressure losses are determined in respect of the kinetic energy contained in the fluid through the pressure loss coefficient f.

$$Dp = f * 1/2 * RoV^2$$

PRESSURE LOSS: THE f COEFFICIENT

The following curve shows the variation of the pressure loss ratio according to the Number of Reynolds for the flow (ref: Mémento des pertes de charge Idel Cik).

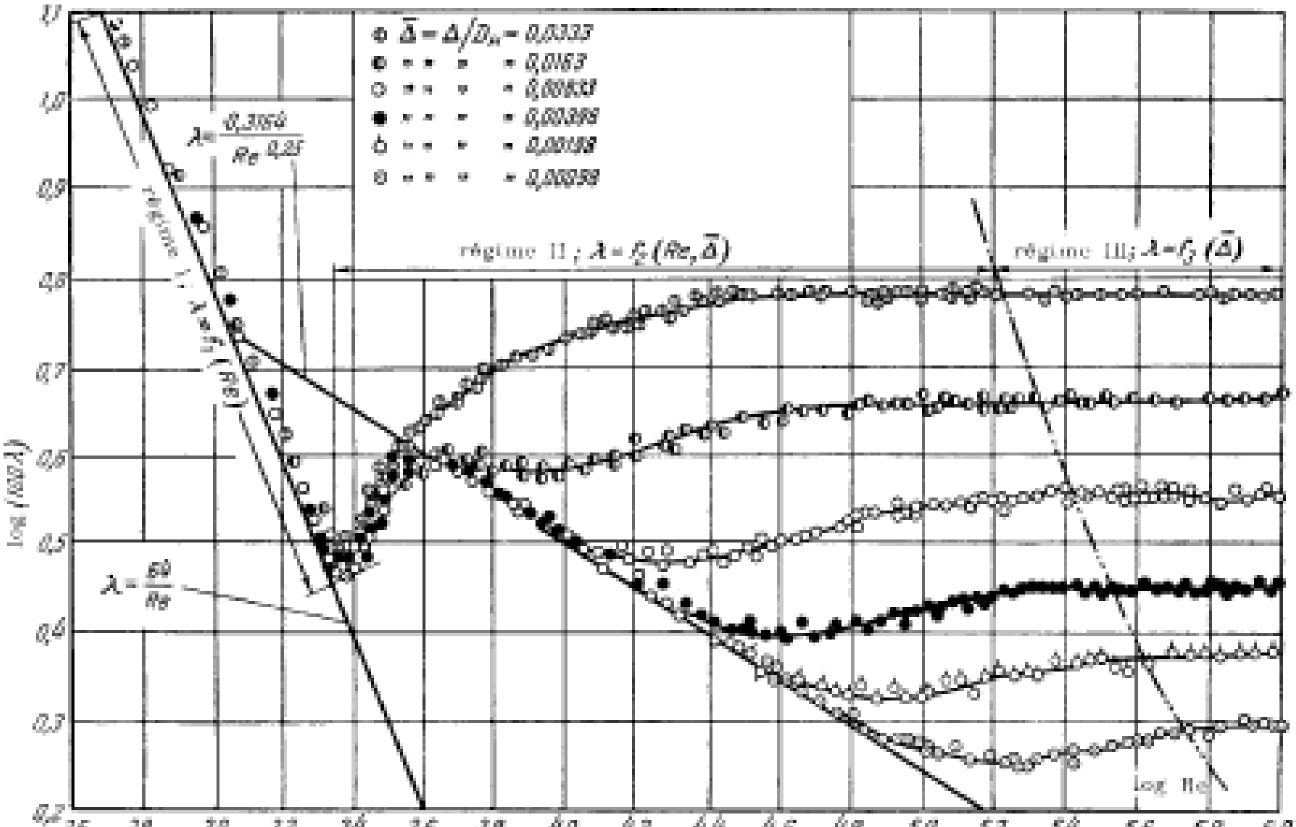
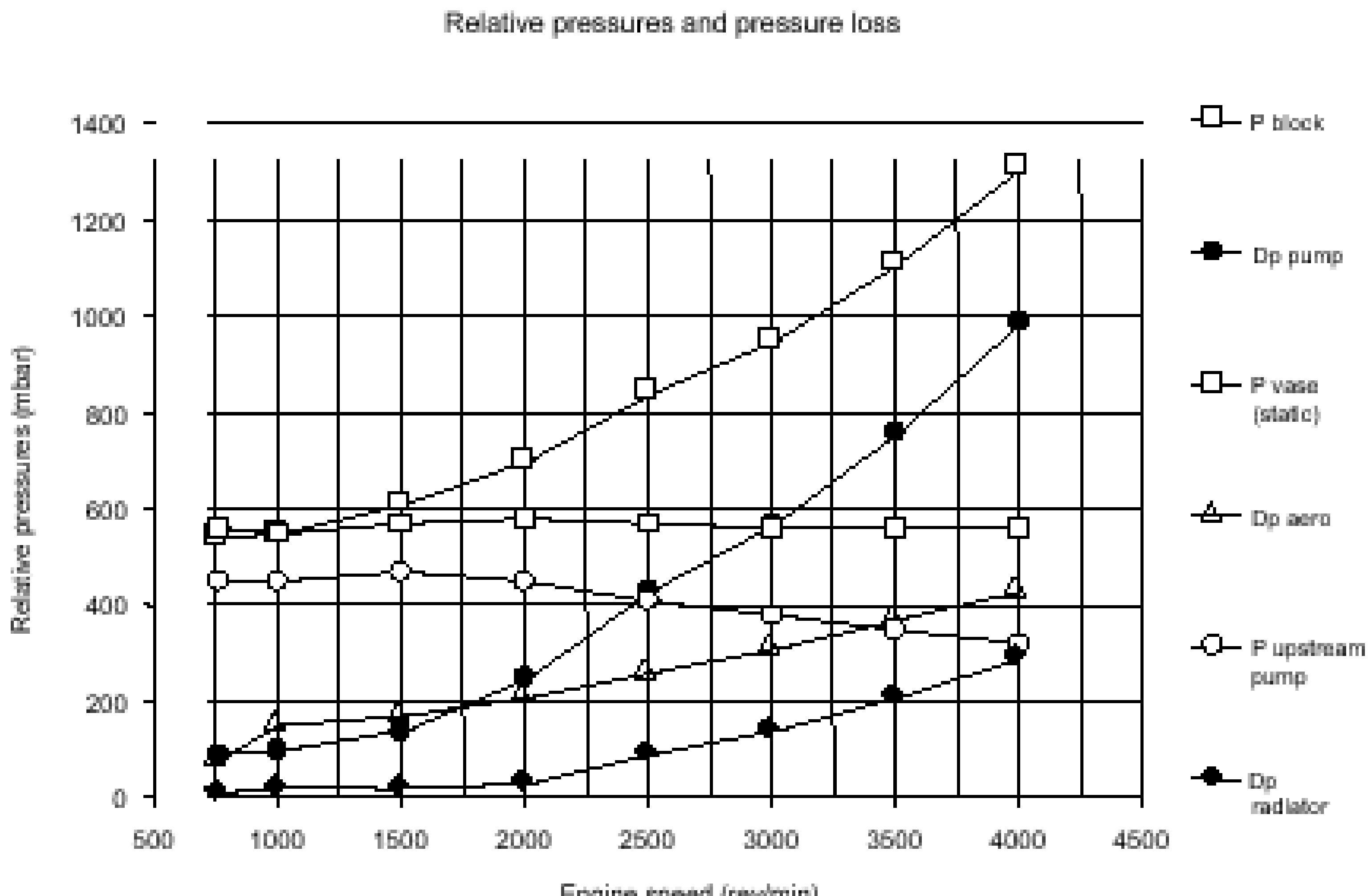


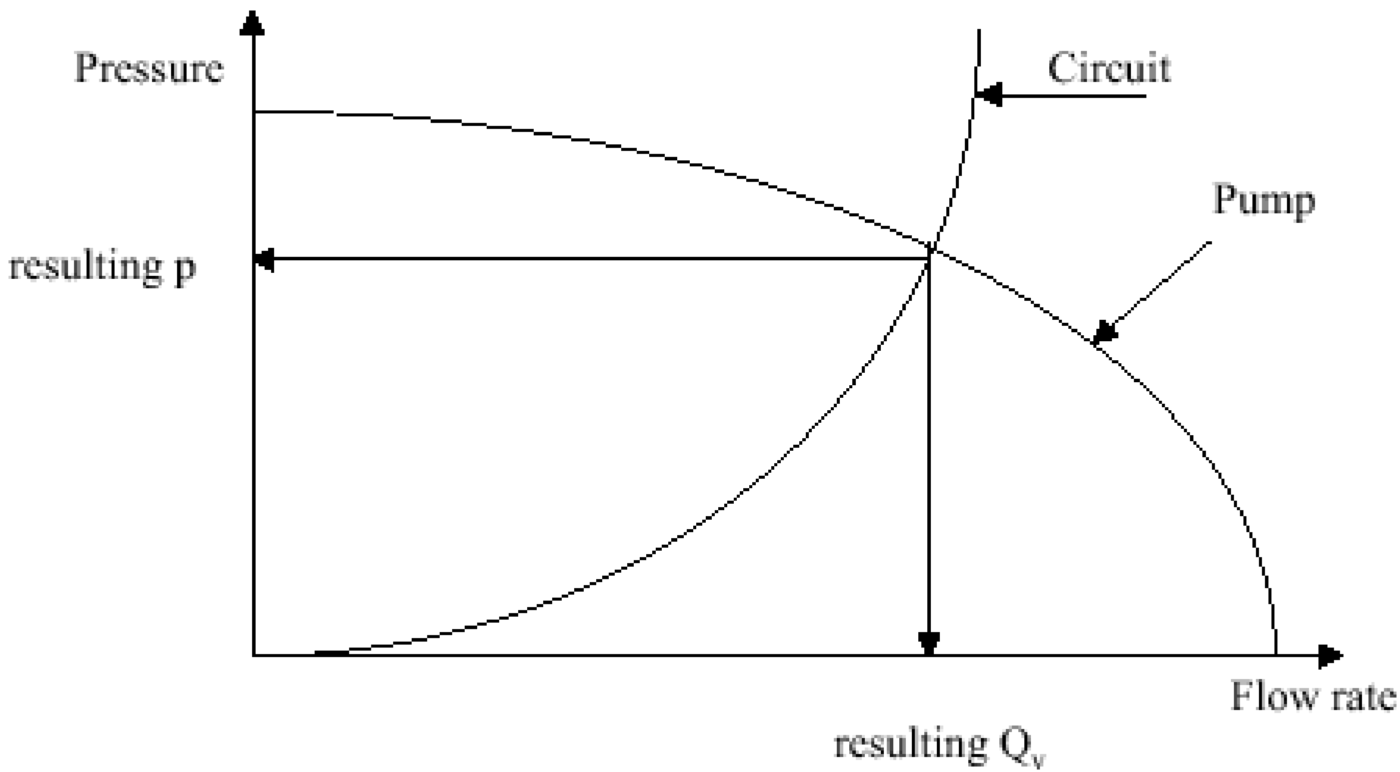
FIG. 2.1. — Variation on the lambda load loss ratios according to Re for homogeneous roughness ducts

EXAMPLES OF PRESSURE LOSS
Hot loop



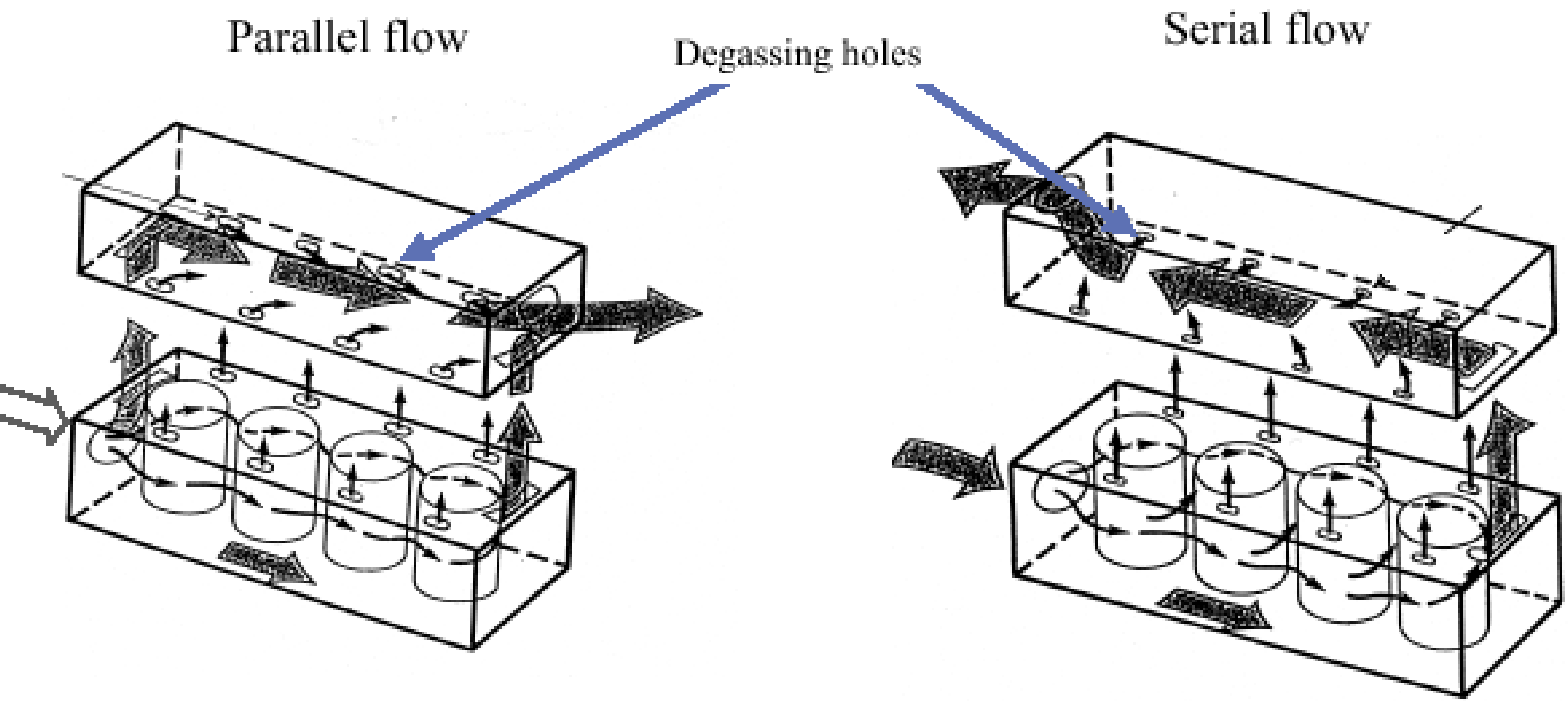
ADAPTATION OF A PUMP TO A CIRCUIT

The term **adaptation** designates the pressure-flow rate couple: pressure resulting from the flow rate delivered by a pump (considering its characteristic curve) in a given circuit (considering its pressure loss characteristic).



INTERNAL COOLING

Two types of cooling liquids are usually implemented. The first, called parallel flow, consists in supplying the engine block and cylinder head concurrently.
The second, called serial flow, consists in first supplying the engine block, then the cylinder head.
Each of these flows can be established in the longitudinal direction, as the following diagrams show, or transversely as in the following diagram.



INTERNAL COOLING

The drawback of serial flow is that it generates thermal gradients in the engine's longitudinal directio
Therefore, the cylinder located opposite the water inlet is less cooled. The same problem occurs on the
level of the cylinder head in the return direction.
Parallel flow requires dividing the cooling flow between the cylinder head and the block, then using calibrated holes to spray each area with an equivalent flow.
Intermediate flow consists in the serial supply of each area comprising the cylinder and the cylinder head part. In this case, the difficulty is cooling the lower part of the block, as the liquid tends to rise.

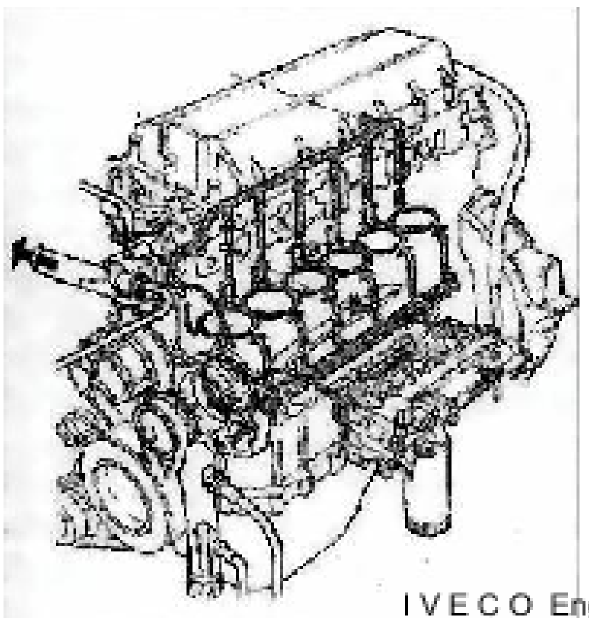
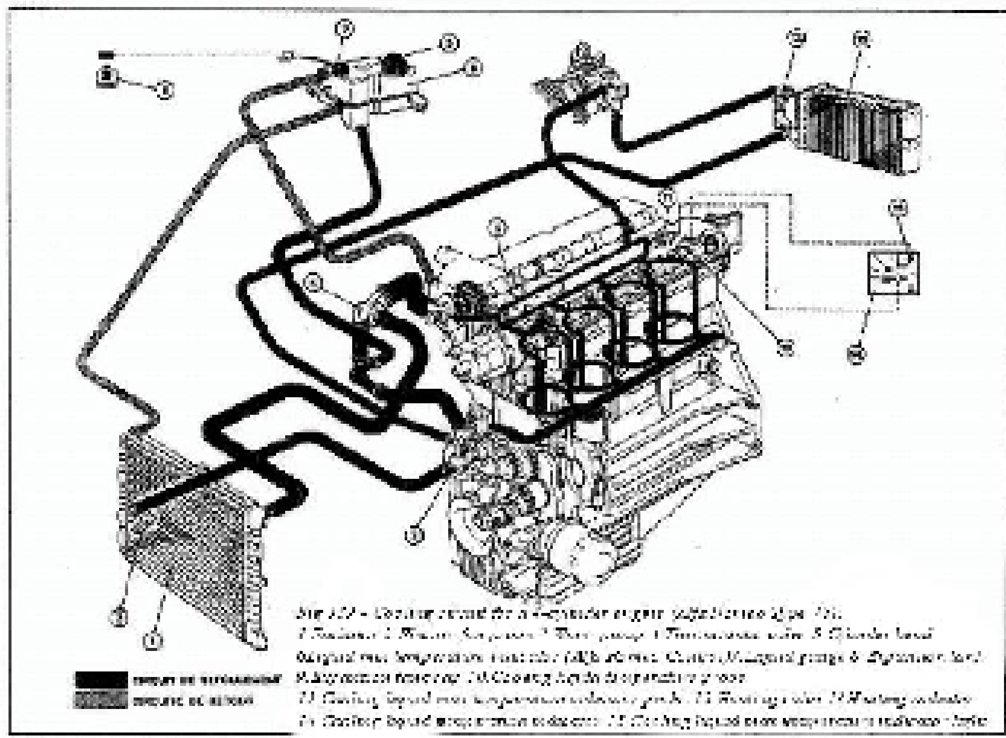


Bild 4: Kühlwasserkreislauf
Figure 4: Cooling water circuit

I V E C O Engine



INTERNAL COOLING

Cylinder head supplied through an internal channel the distribution of the liquid between the exhaust valves.
No forced flow in the block.

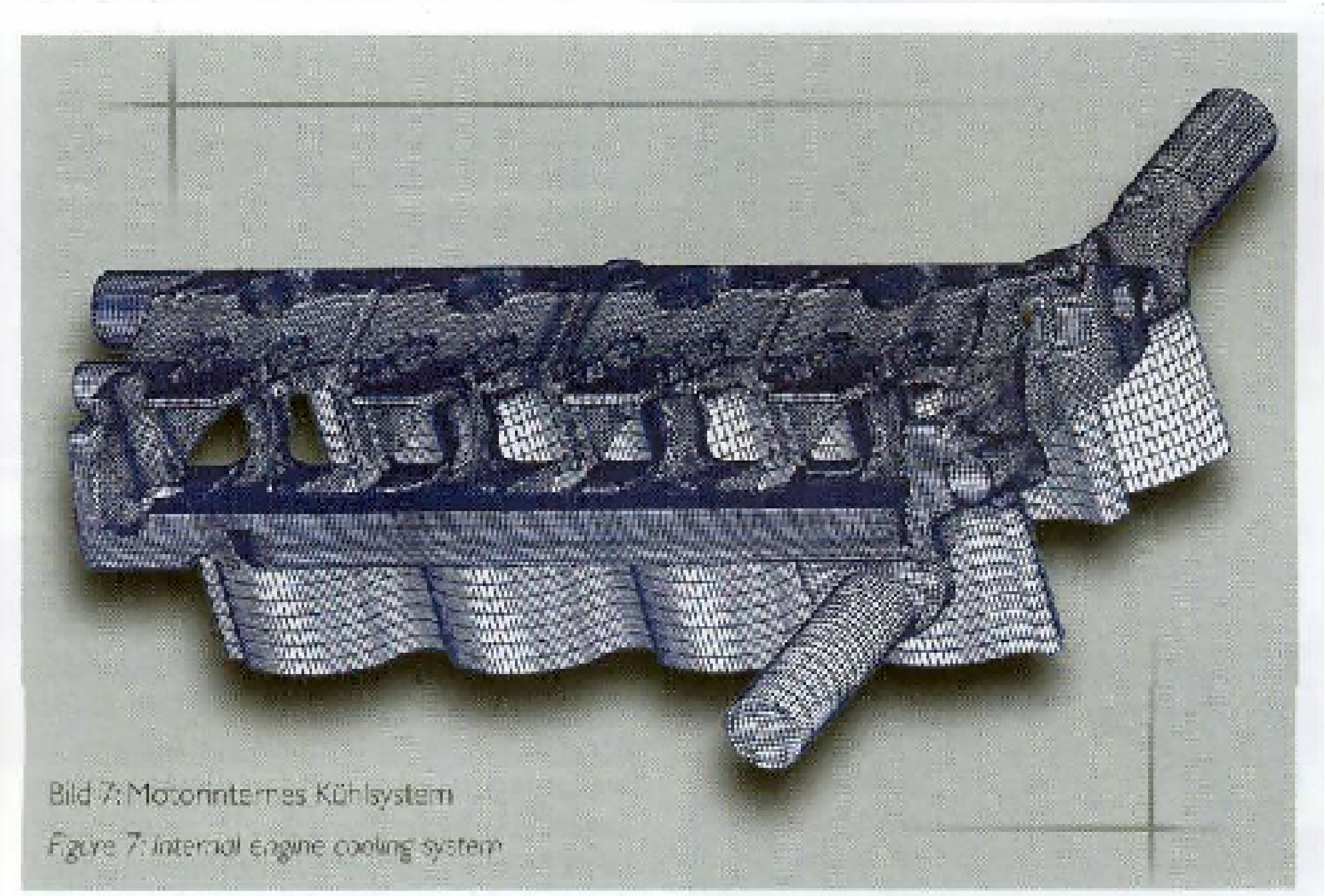


Bild 7: Motorinternes Kühlsystem

Figure 7: Internal engine cooling system

Water flow in the cylinder head and in the engine block
BMW, 4 cylinders, valvetronic - Ref MTZ June 2001

eduction of the
mperature of the
haust intervalve
idges by 30°C
eduction of
essure loss by
0% Reduction of
e driving power of
e pump, 40 %

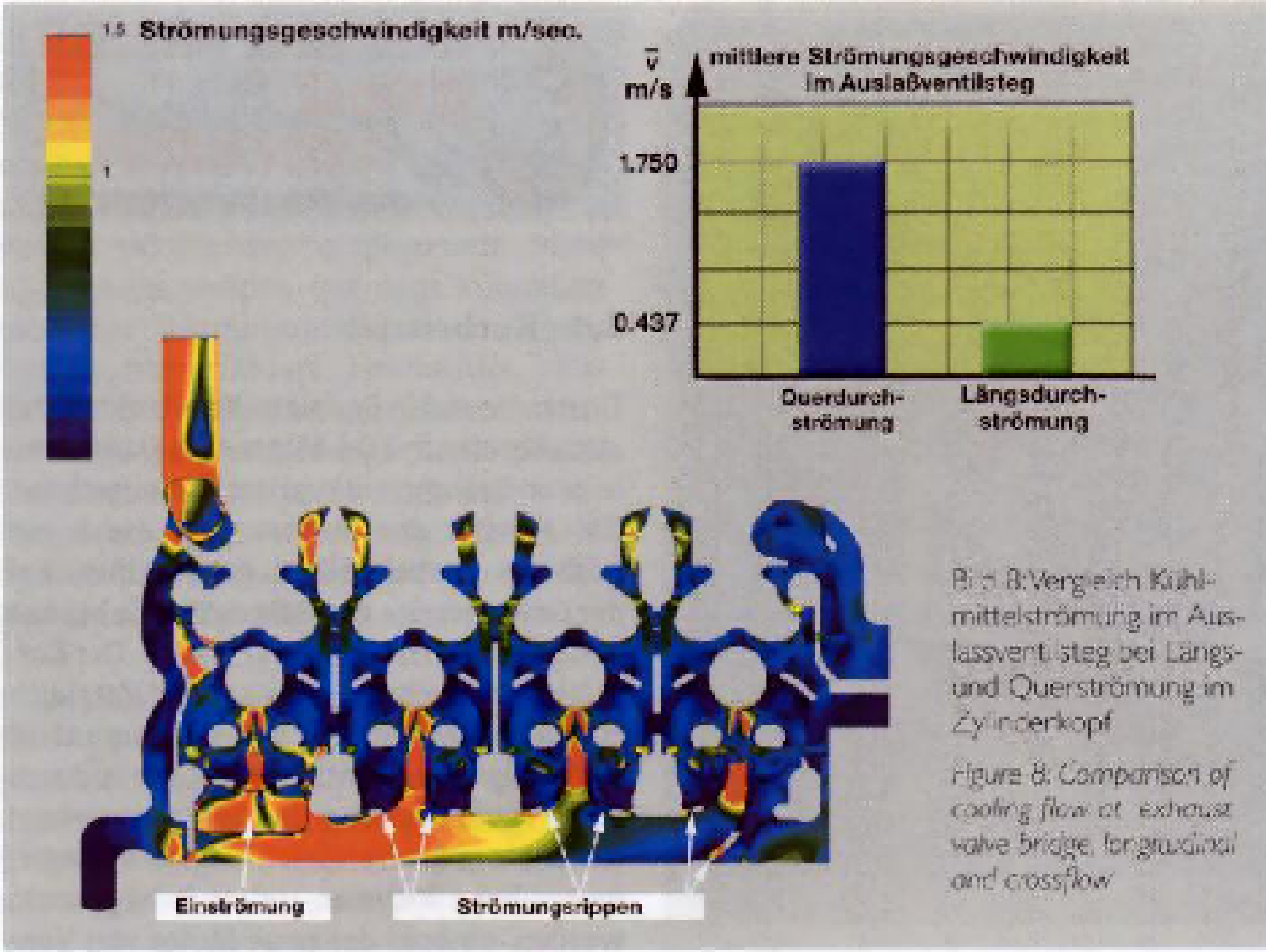


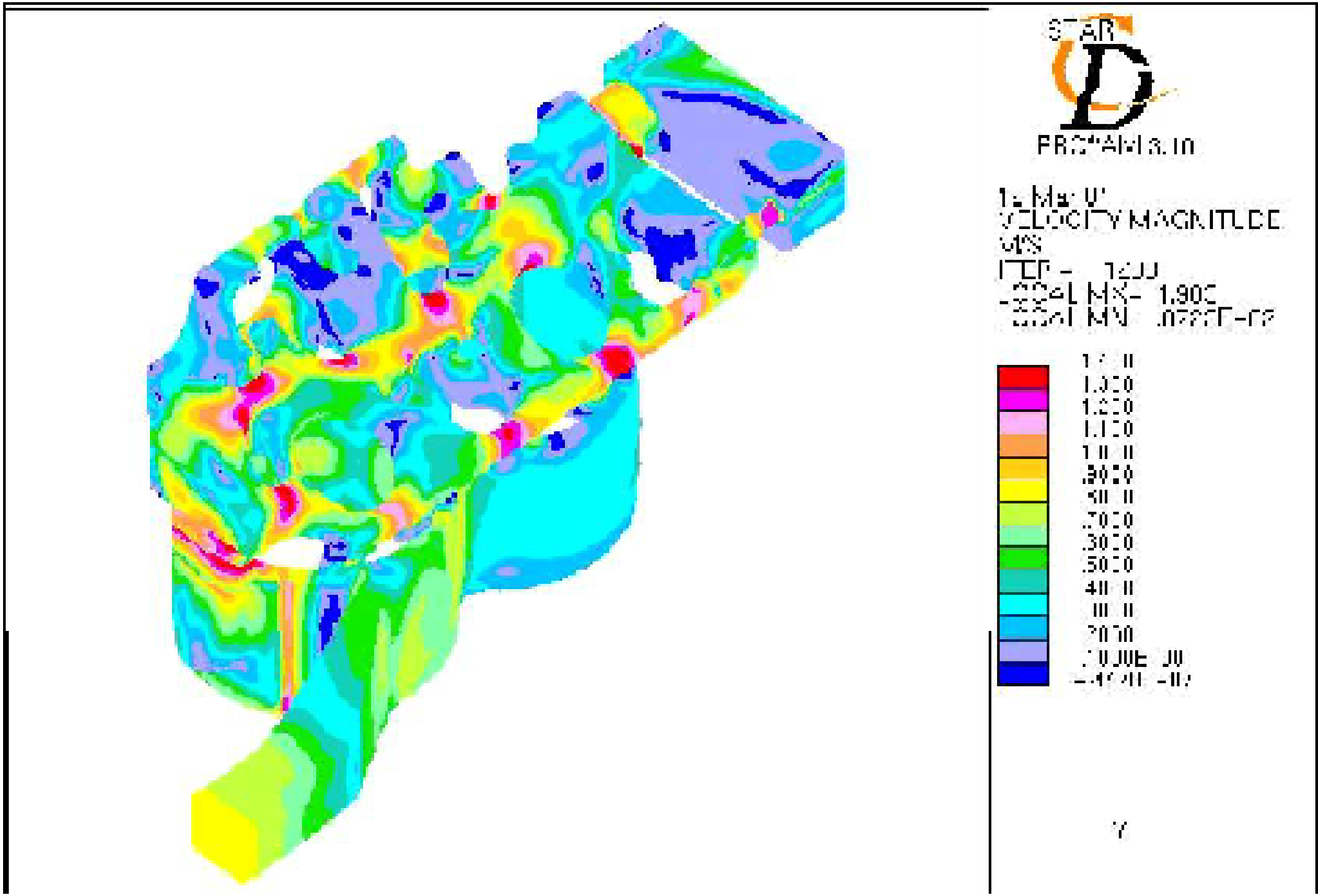
Bild 8: Vergleich Kühlmittelströmung im Auslassventilsteg bei Längs- und Querströmung im Zylinderkopf

Figure 8: Comparison of cooling flow at exhaust valve bridge, longitudinal and crossflow

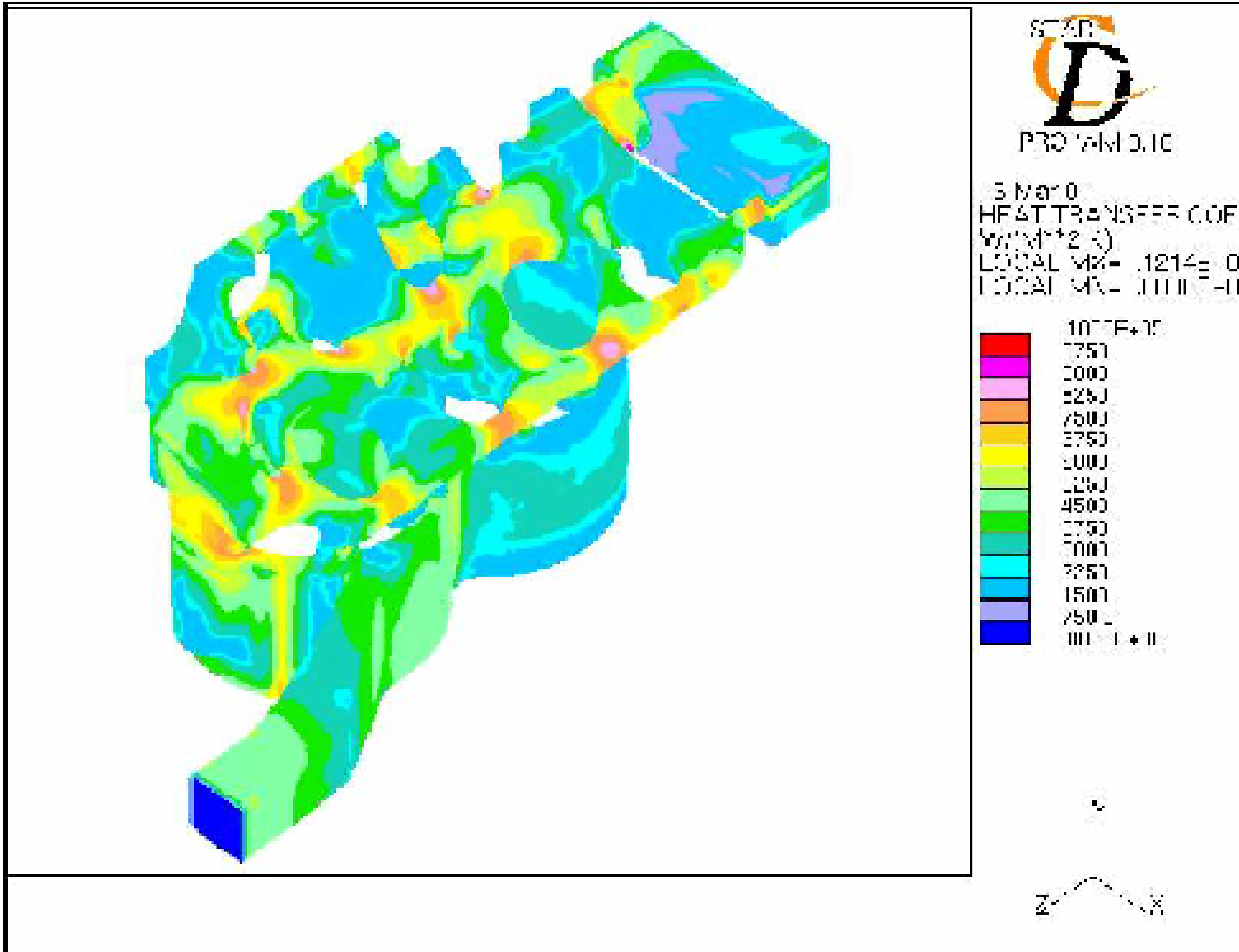
Water flow in the cylinder head and the engine block
BMW, 4 cylinders, valvetronic - Ref MTZ June 2001

NTERNAL COOLING: SIZING METHOD

Considering current computation means, the flows and heat exchanges in the water cores are calculated using 3D code (Fluent,StarCD,Vectis,Fire…). Experimental confirmation by means of laser tomography can be performed.



INTERNAL COOLING: SIZING METHOD



SIZING POINTS AND CRITICAL POINT

The pump is sized to ensure sufficient flow, hence sufficient cooling at the maximum power point.

The fan/radiator assembly is sized to enable sufficient cooling of the liquid on a full load low engine speed running. The radiator's sizing is a compromise between the engine requirement, the possible exchange surface with the air (influence on the vehicle's SCx) and the load loss on the cooling circuit.

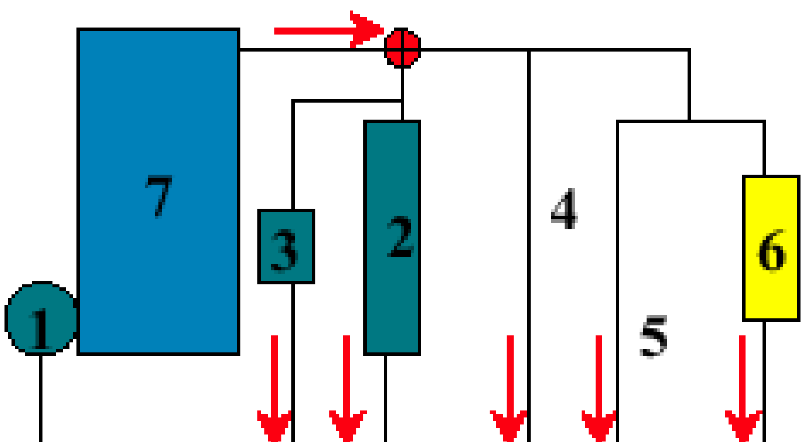
Particular running point: Idle running after an extended full load running (stopping at a toll booth). This running point imposes having sufficient minimum flow rate at idle running, often leading to having a flow rate exceeding Nmax. To achieve an excess flow rate in idle running, the engine does not return to the nominal idle running speed, but to a higher speed, called accelerated idle running, for a few minutes.

ENGINE A: CIRCUITS IMPLEMENTED

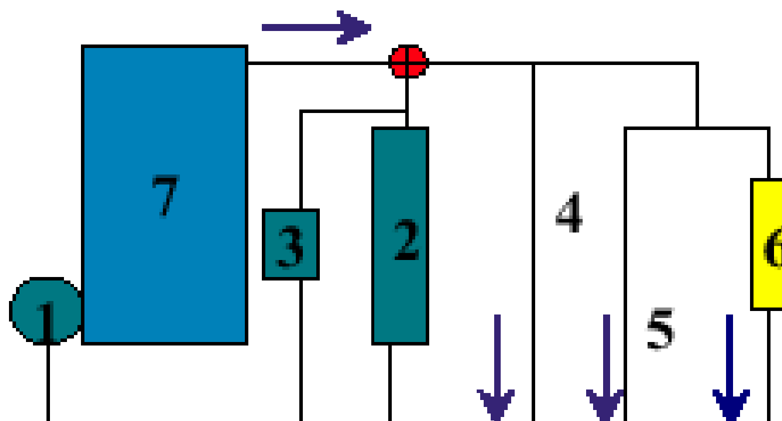
1: water pump; 2: radiator; 3: expansion tank; 4: external bypass

5: aerotherm bypass; 6: aerotherm

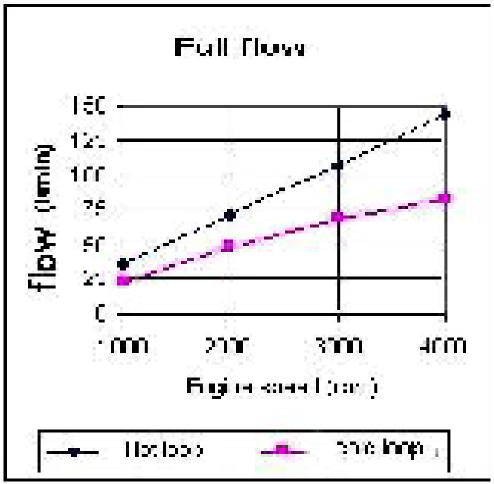
HOT LOOP

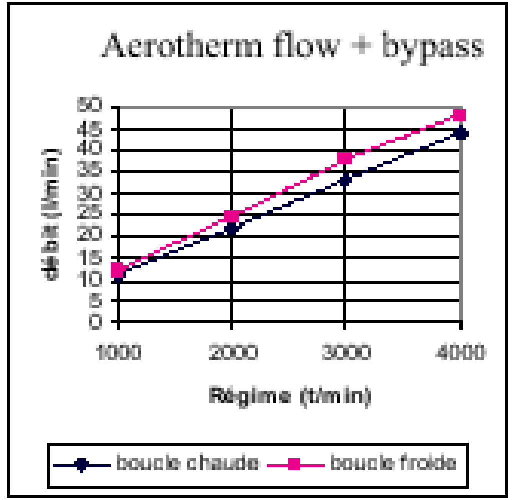


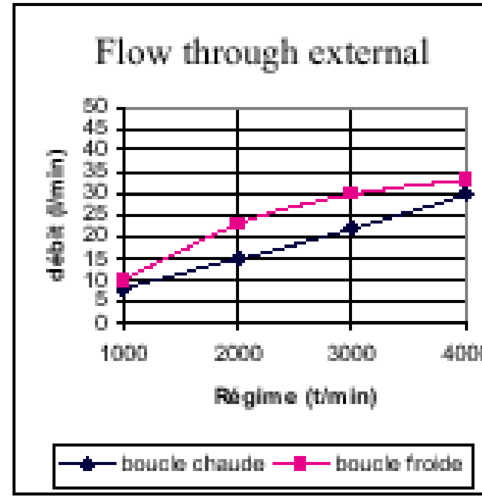
COLD LOOP



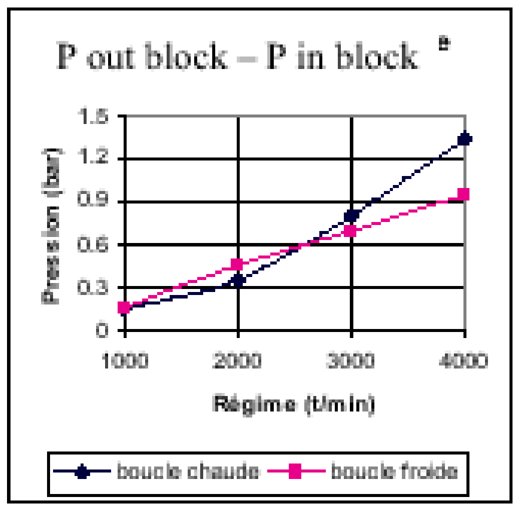
ENGINE A

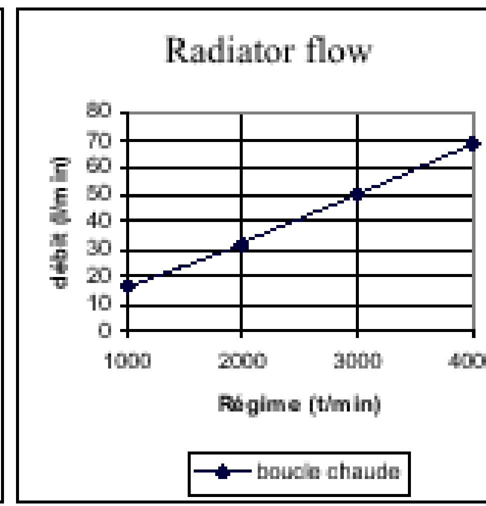






In a cold loop, the radiator is not « replaced » in terms of cross-section by an equivalent sized bypass. The pump cannot generate a pressure enabling to shift from the missing flow with the aerotherm + aerotherm bypass + external bypass.

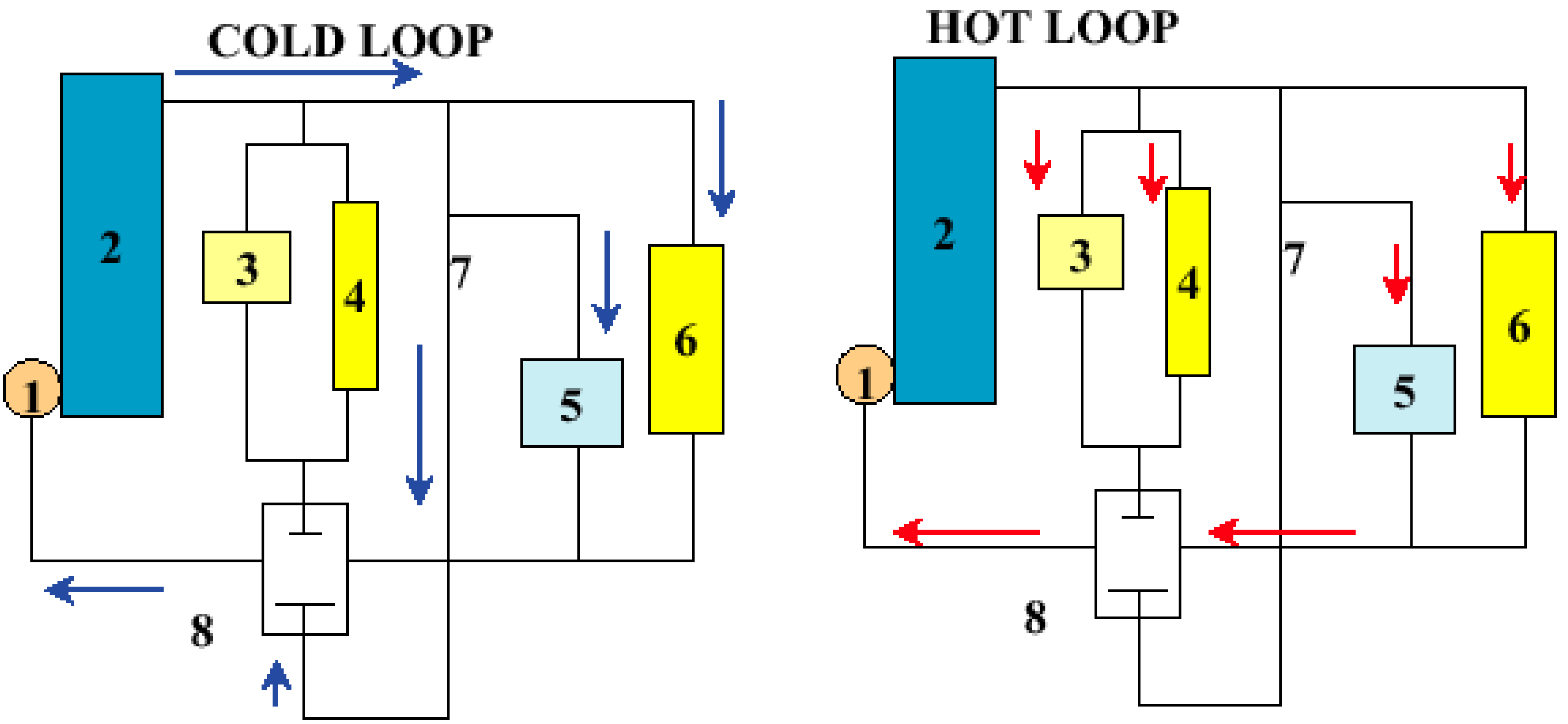




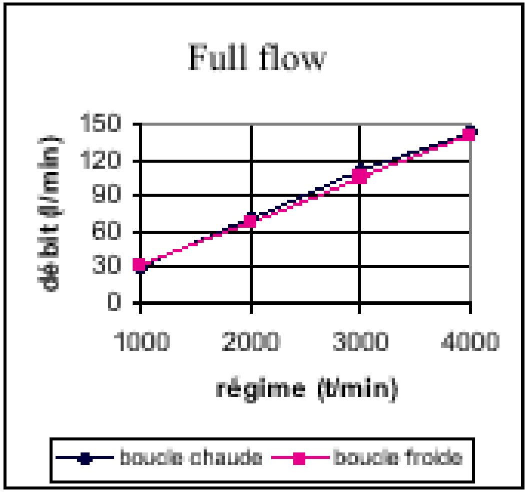
ENGINE B: CIRCUITS IMPLEMENTED

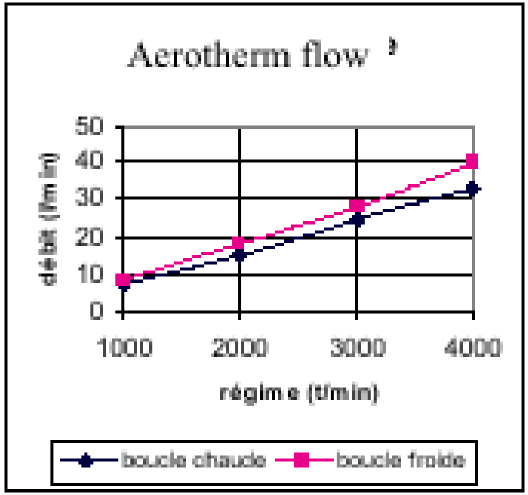
1: water pump ; 2: engine ; 3: expansion tank ; 4: radiator ; 5: water-oil exchanger

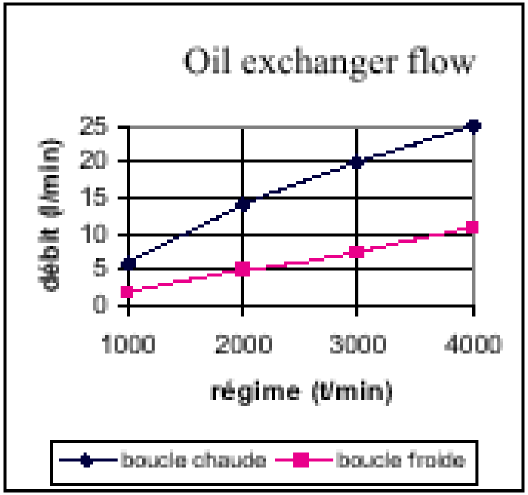
6: aerotherm ; 7: bypass ; 8: mixer



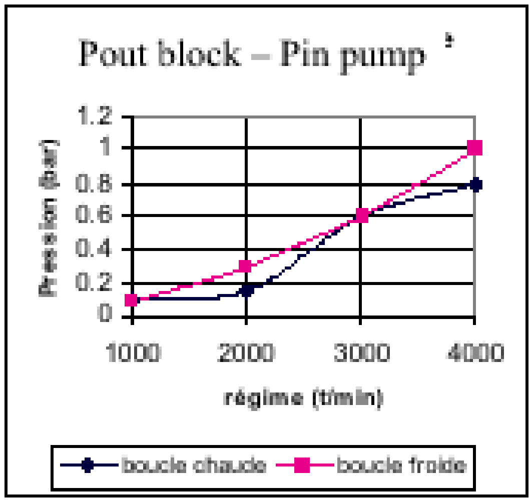
PROBLEM BALANCING HOT LOOP – COLD LOOP CIRCUITS

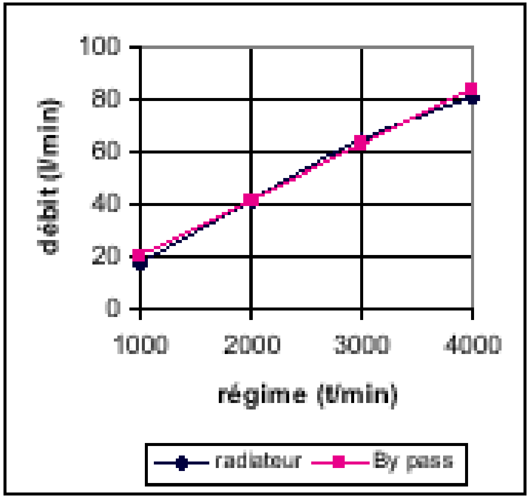






this case, the mixer installed ables shifting part of the flow her through the radiator (hot op) or through a bypass (cold op), sized in order to obtain e same flow cross-section, abling to maintain the flow.





ENGINE B

COMPLEMENTARY STRATEGIES

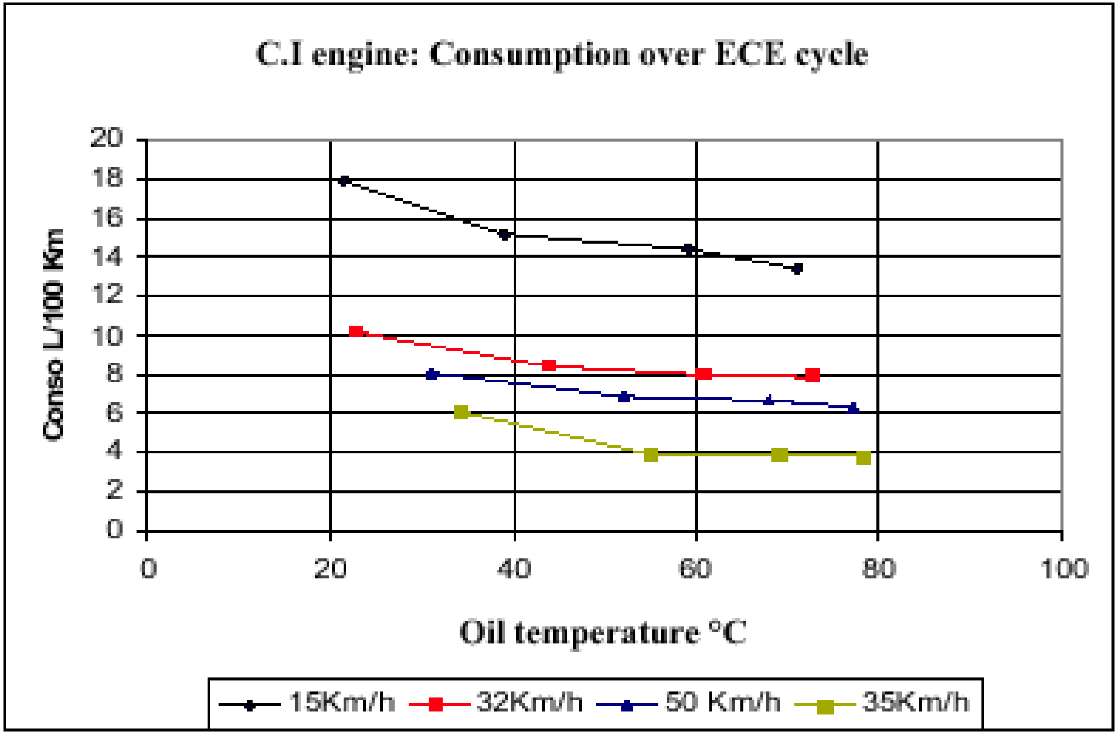
As the water pump flow is proportional to engine speed, the water flow is too high at partial loads. In this case, reducing the flow enables increasing the temperature, resulting in a gain in consumption. The rise from 90°C to 110°C enables saving 3% consumption over an urban cycle.

Conversely, after full load running, the pump's flow is insufficient in idle running to evacuate all the calories (hence the accelerated idle running).

In both cases, dissociating the pump flow and engine speed is interesting, hence the electric pump.

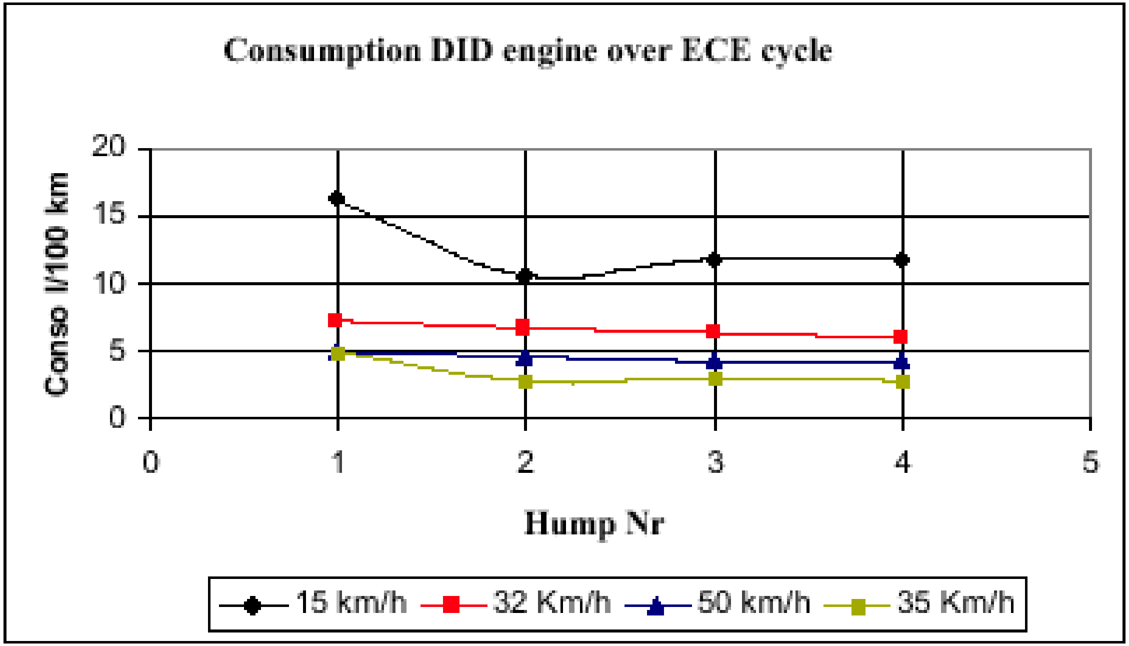
The use of mixers provides enhanced temperature regulation, by preventing the flow of volumes of cold water followed with volumes of hot water. Today, piloted thermostats enable obtaining a temperature of 110° at partial load (pollution - consumption) and 90° at full load (engine protection).

INFLUENCE OF ENGINE TEMPERATURE ON CONSUMPTION OVER AN ECE CYCLE



For a controlled ignition engine, increasing the oil temperature from 20 to 80° reduces the engine's consumption by 20 to 30 % over the four peaks of the ECE cycle

INFLUENCE OF ENGINE TEMPERATURE OVER CONSUMPTION FOR AN ECE CYLCE



For a direct injection Diesel engine, fuel consumption also drops by 20 to 25% when repeating the peaks during the ECE cycle.

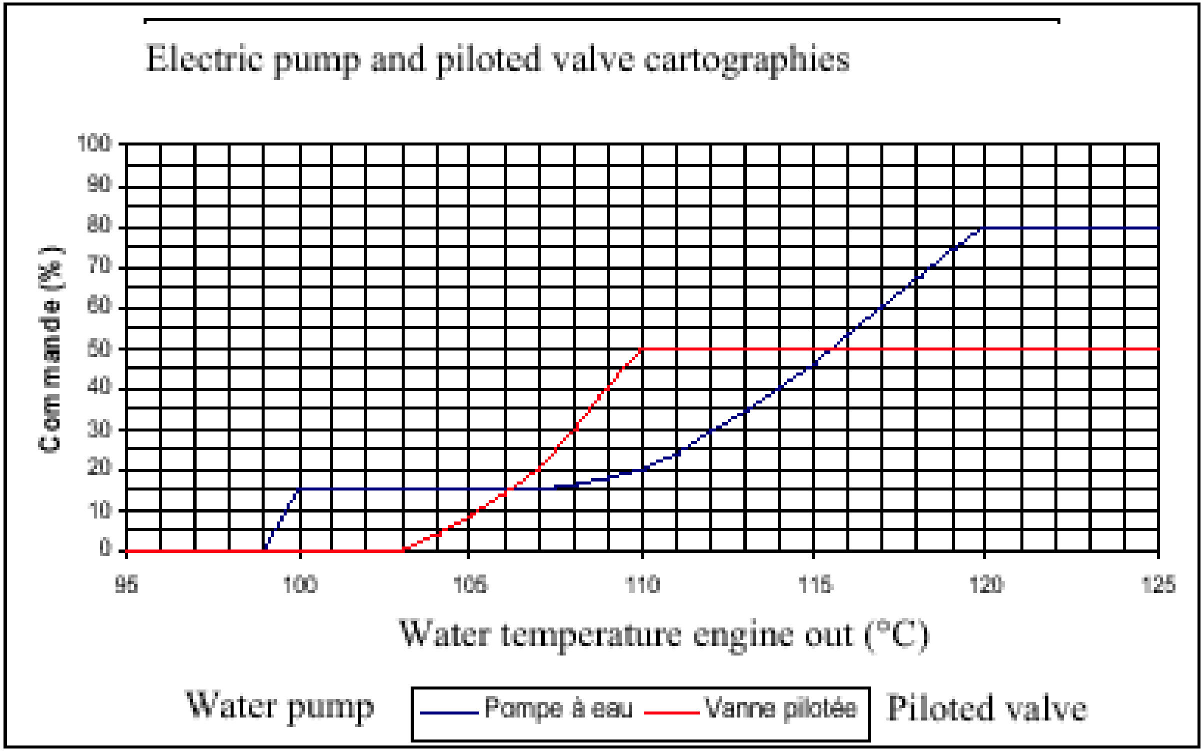
STRATEGY TO ACCELERATE THE ENGINE'S WARM-UP

est of three different configurations on a controlled ignition engine:

onfiguration 1 – Mechanical water pump + Wax thermostat 90°C

onfiguration 2 – Mechanical water pump + Piloted thermostatic valve 110°C

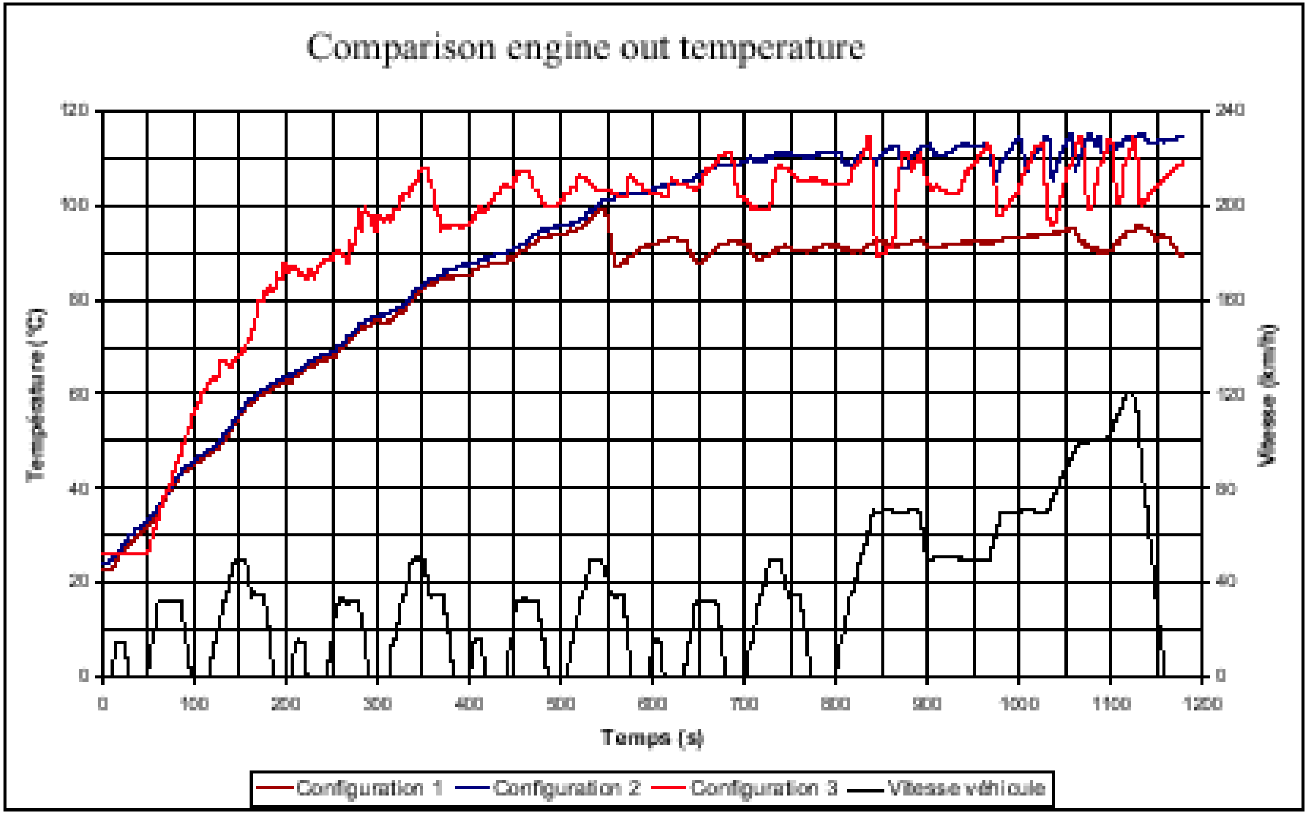
onfiguration 3 – Electric water pump + Piloted thermostatic pump 110°C (in this configuration, ne pump is used only when the water temperature at engine outlet reaches 100°C)



STRATEGY TO ACCELERATE THE ENGINE'S WARM-UP: ENGINE BEHAVIOR

The use of a piloted valve enables preventing the momentum effects encountered with wax thermostats.

With the electric pump, engine warm-up is much faster, but after the valve's opening, temperature oscillations appear caused by the arrival and mixing with cold liquid from the ho loop.



STRATEGY TO ACCELERATE THE ENGINE'S WARM-UP: ENGINE CONSUMPTI

Configuration (n° essais)	Consommation en l/100 km				
	UDC1	UDC2-3-4	UDC	EUDC	EURO 2000
1 - (V4a10)	10.41	8.14	8.71	5.41	6.63
2 - (V4a37)	10.60	8.29	8.87	5.44	6.71
3 - (V4a86)	10.47	7.93	8.57	5.42	6.59
Ecart 2/1	1.8%	1.8%	1.8%	0.6%	1.2%
Ecart 3/1	0.6%	-2.6%	-1.6%	0.2%	-0.6%
Ecart 3/2	-1.2%	-4.3%	-3.4%	-0.4%	-1.8%

When comparing the consumption obtained in configuration 3 with configuration 1, it appears that:

•Consumption in the UDC1 cycle is virtually identical, the water temperature in the engine being insufficiently high to impact the engine's thermal losses,

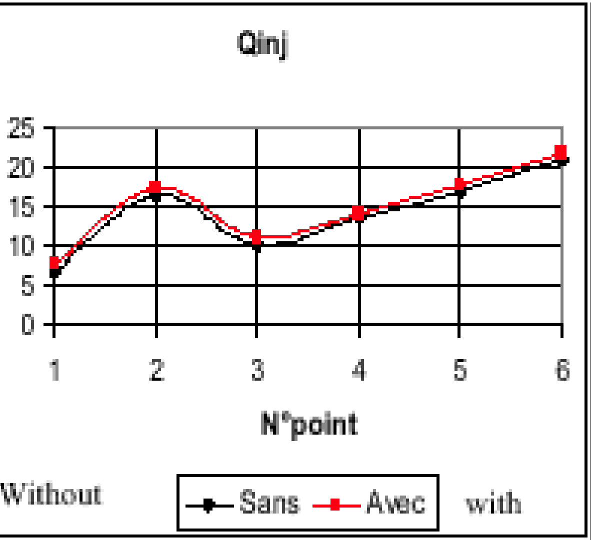
•However, during the UDC2, 3, and 4 cycles, in which the temperature delta increases, consumption gains are much higher (2.6 %),

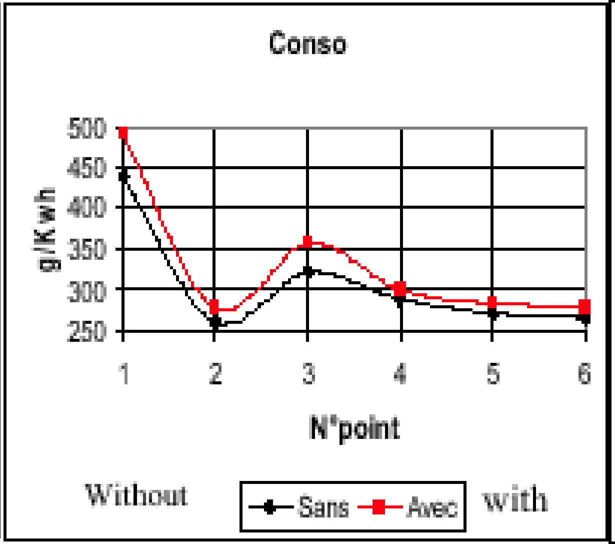
•In the EUDC cycle, consumption is identical (0.2 % gap meaningless). This results from the fact that the average regulation temperature is about 105°C, i.e. about 10°C more than the reference.

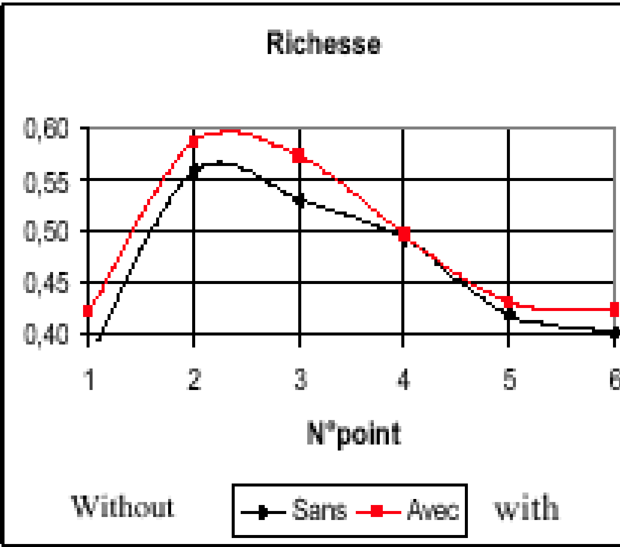
INFLUENCE OF ENGINE TEMPERATURE AT A STABILIZED POINT

With the same mechanical pump (Configurations 1 and 2), the increase in the engine's water regulation temperature from 90 to 110°C results in a consumption gain of 2.7% stabilized at 50 km/h. The tests performed with the electric water pump (configuration 3) and regulation temperatures of 90 and 110°C respectively confirm and adjust the consumption gains obtained when increasing the engine water temperature: 3.4% at 32 km/h; 2.9% at 50 km/h, and 4.9% at 70 km/h.
These values, ranging from 3 to 5%, comply with the magnitudes usually accepted by motorists for an increase in engine water temperature of 20°C.
These tests show that shifting from the mechanical water pump driven directly by the engine shaft to the electric water pump, the driving power of which is conditioned by the yield of the alternator and the pump's electric motor, results in a consumption increase of 0.8% at 50 km/h, in spite of a reduced water flow.

DIFFICULTIES FOR A WARM UP STRATEGY: ACCESSORY INFLUENCE

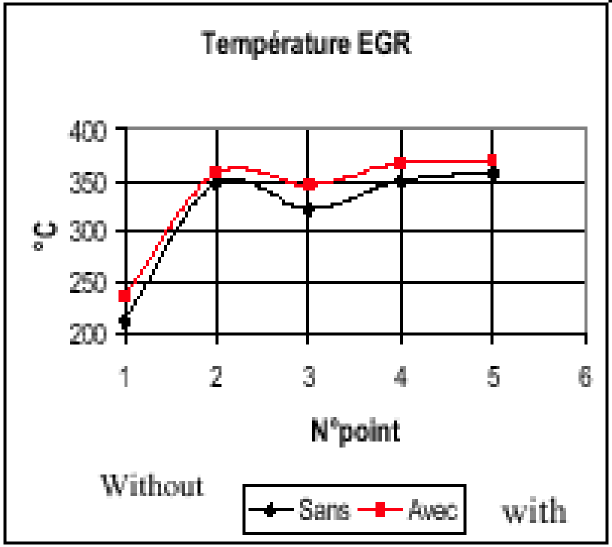


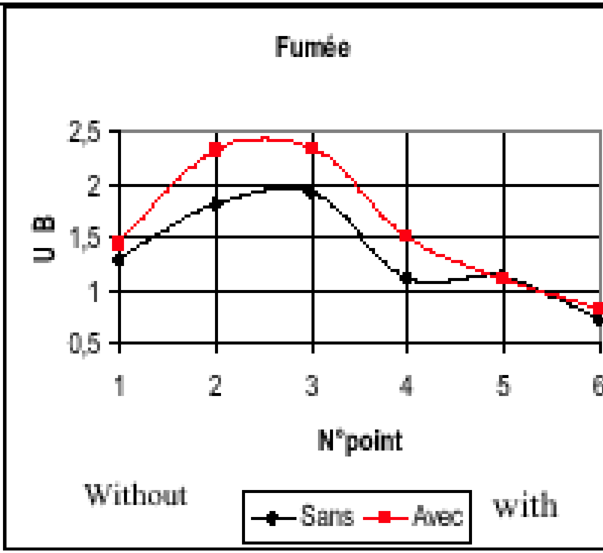




1: 1500 rev/min Pme 1 bar
2: 1500 rev/min Pme 4 bar
3: 2000 rev/min Pme 2 bar
4 :2500 rev/min Pme 3 bar
5 :3000 rev/min Pme 4 bar
6 :3500 rev/min Pme 5 bar

DDI Engine





COOLING CIRCUITS
CONSTITUTIVE ELEMENTS

Generally, regardless of the engine considered, the cooling circuit includes:

a pump,
a radiator,
a fan,
an aerotherm,
a calorstat,
an expansion tank (with circulation),
Not on all engines
an oil cooler: Diesel engine and auto gearbox,
a turbocharging air cooler: the RAS,
an EGR cooler.

COOLING CIRCUITS: FUNCTIONS ENSURED

ump : It ensures the flow of liquid in the various circuit configurations.

adiator : It ensures the heat exchange between the cooling liquid and the outside air.

an : It restores a sufficient level of heat exchange when the vehicle's speed is insufficient.

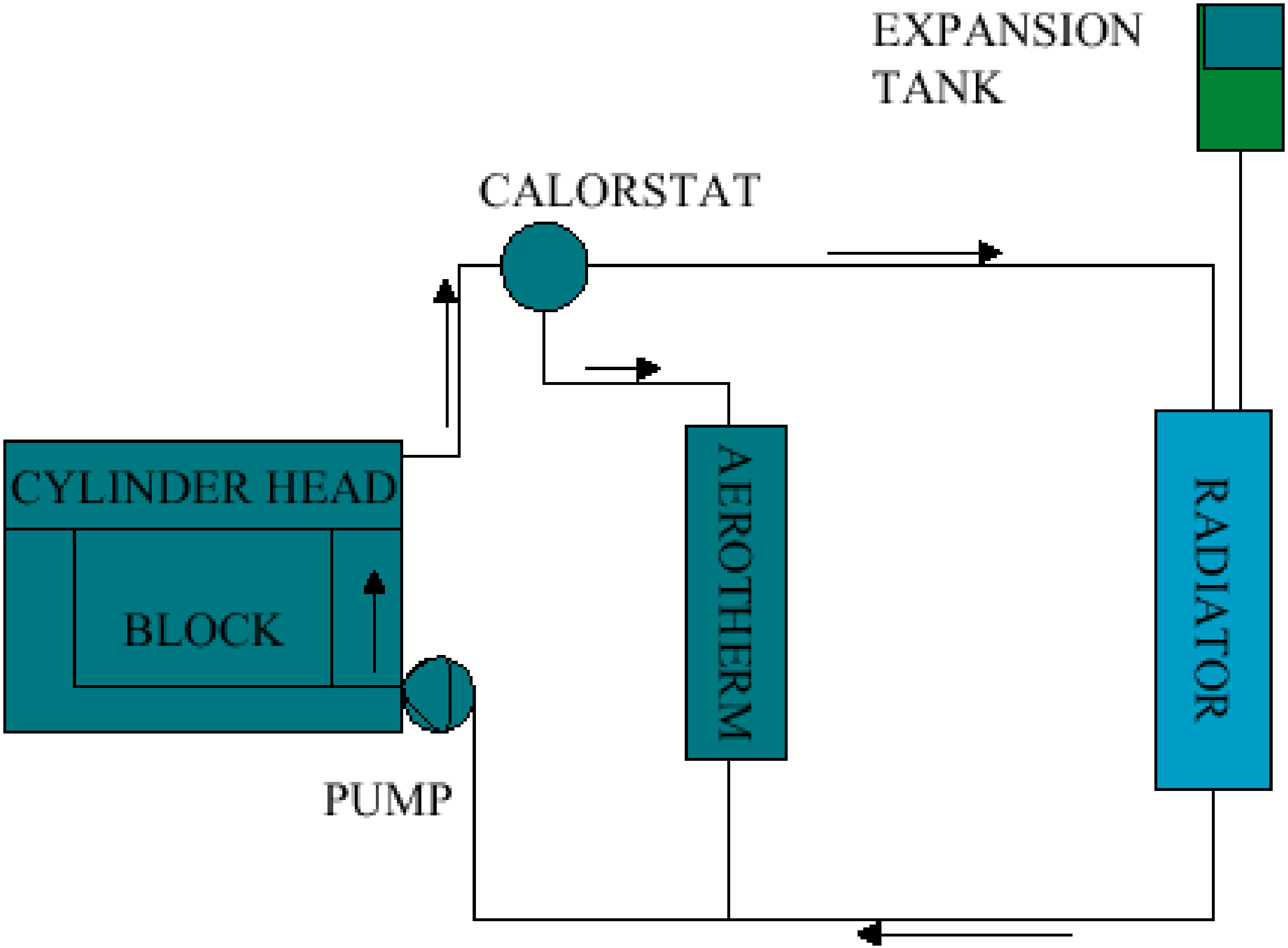
erotherm : Today, it participates in cooling the liquid, and enables recovering calories for the cab.

xpansion tank : It enables degassing the circuit, provides for the liquid's expansion during heating, and ensures the circuit's pressurizing by the cap's setting. The circuit's pressure depends on the saturating vapor pressure $P = (T/100)^4$ P= 1 bar for 100°C, 1.4 bar for 108.8°C and 1.6 bar for 112.5°C.

COOLING CIRCUITS: FUNCTIONS ENSURED

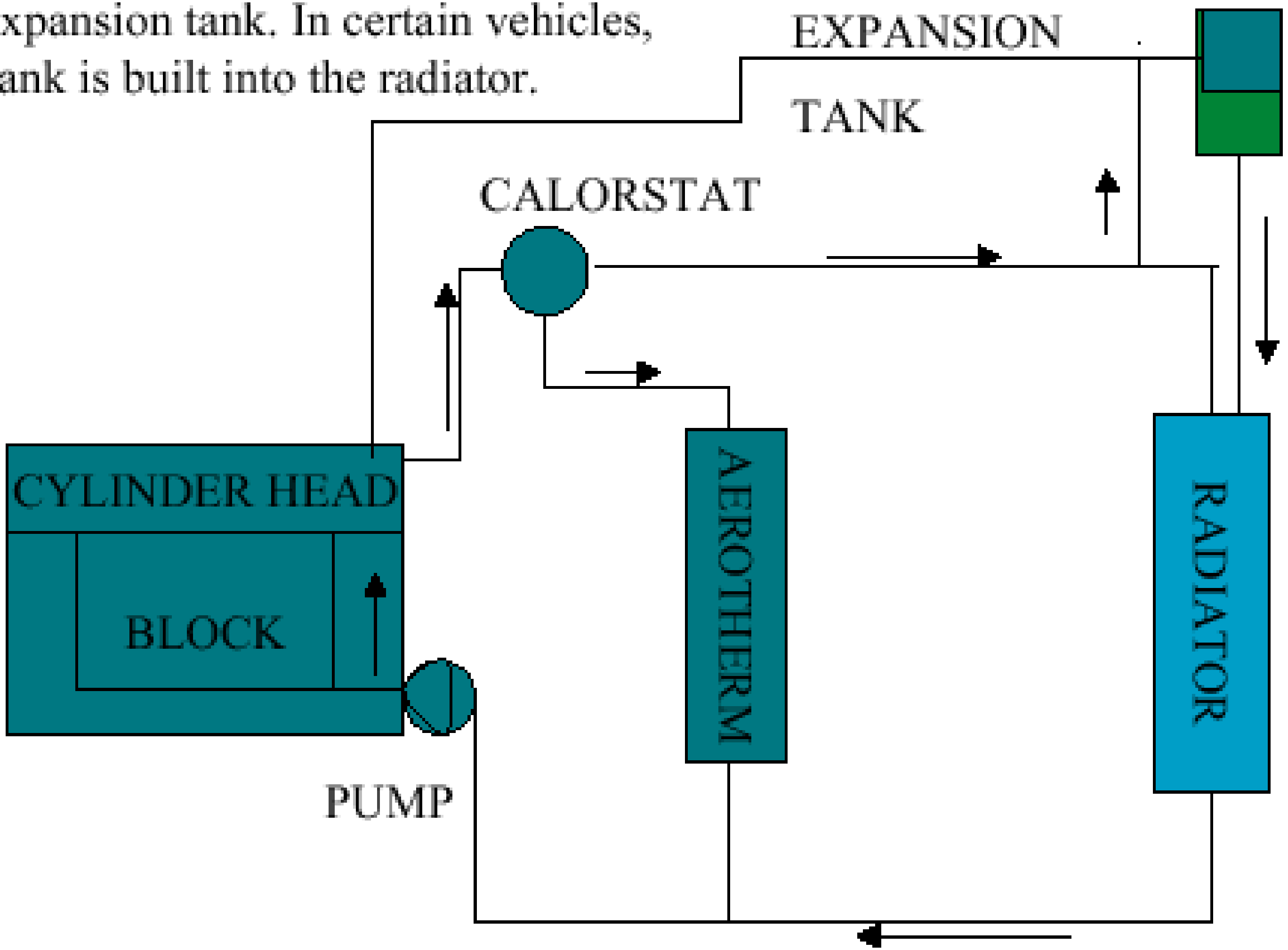
Water/oil exchanger :	It conditions the oil thermally, heats it during the start up phase, and then cools it.
Calorstat	: It enables to shift from the cold loop circuit to the hot loop circuit. It ensures the engine's fast warm up. Reduction of polluting emissions and consumption during cold start - cab heating -
RAS	**:** It enables reducing the gas temperature for compressed air, in order to increase the engine's air filling and lower exhaust temperatures.
EGR cooler	**:** It enables reducing the temperature of the gases recycled at the intake, maintaining the engine's fresh air filling, and supporting the reduction in nitrogen oxides.

BASIC CIRCUIT



IMPROVING DEGASSING

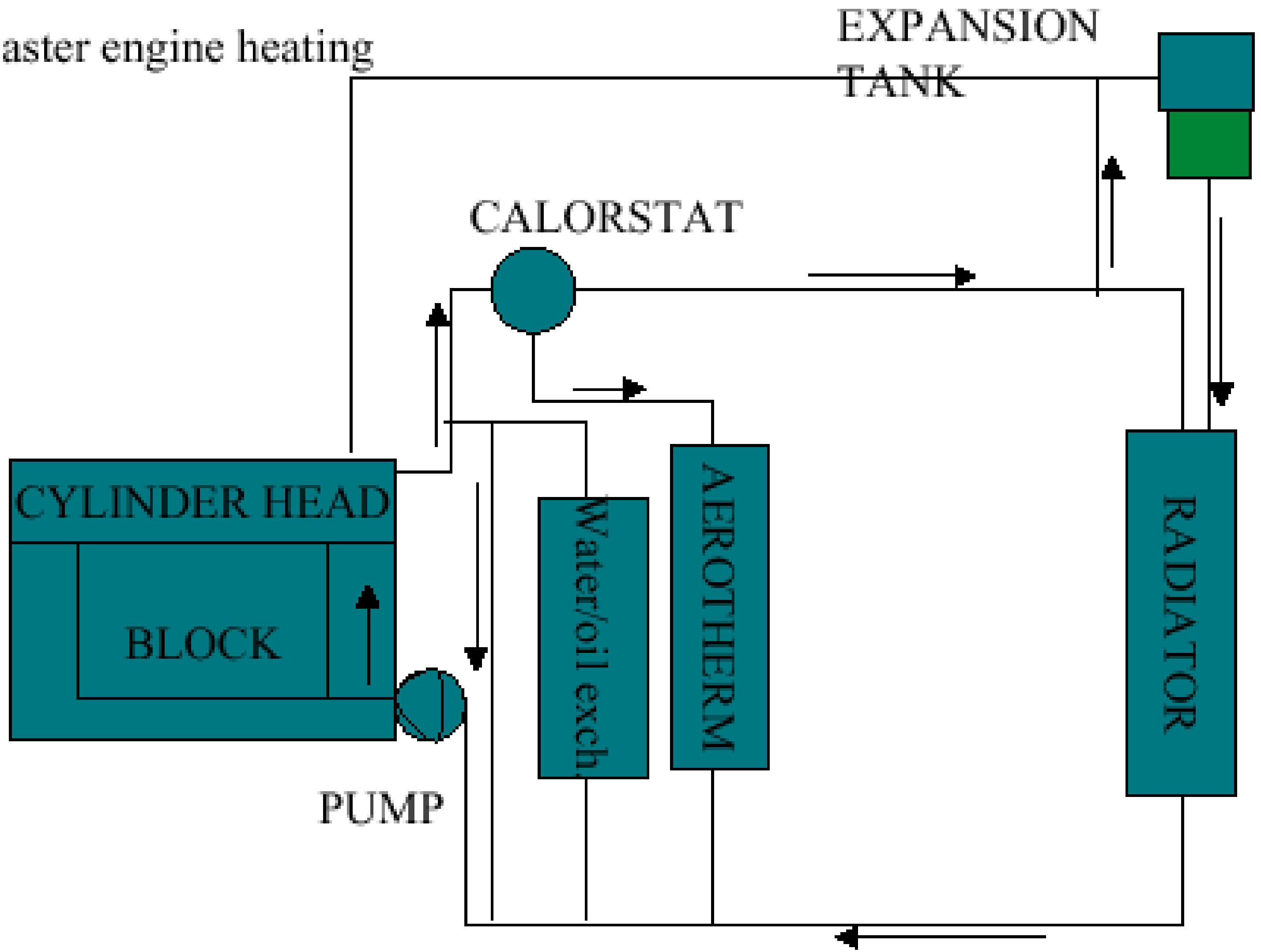
Use of a flow expansion tank. In certain vehicles, the expansion tank is built into the radiator.



IMPROVING START UP

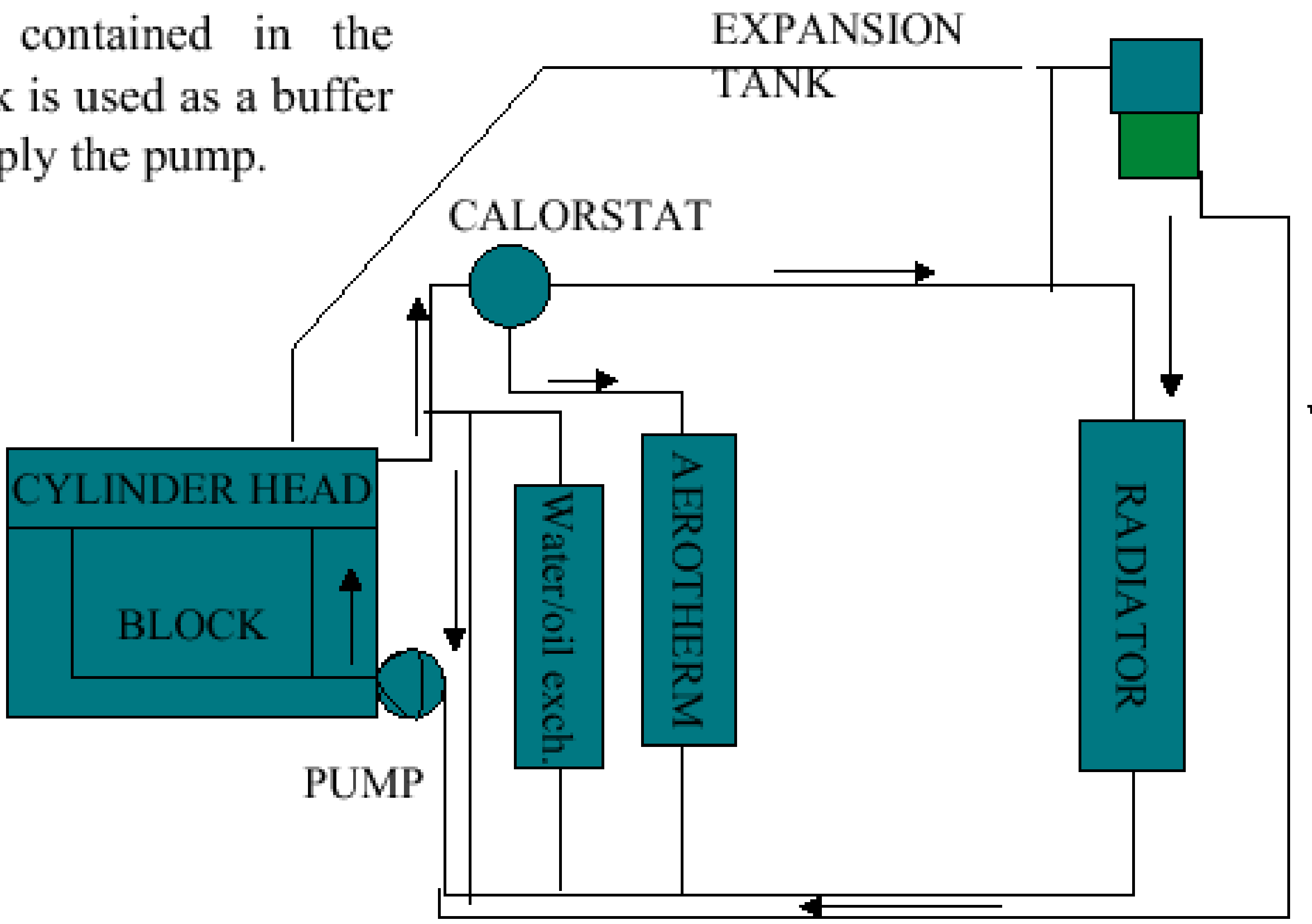
Oil heating: reduction in the PMF

Use of a bypass: Faster engine heating



PREVENTING WATER PUMP CAVITATION

The volume contained in the expansion tank is used as a buffer volume to supply the pump.



Abbreviations/Acronyms

- ATDC
- After Top Dead Centre
- BDC
- Bottom Dead Centre
- BMEP
- Brake Mean Effective Pressure
- BSEC
- Brake Specific Energy Consumption
- BSFC
- Brake Specific Fuel Consumption
- BTDC
- Before Top Dead Centre
- CA
- Crank Angle
- CARS
- Coherent Anti-Stokes Raman Spectroscopy
- CC
- Close-coupled
- CCD
- Charge-coupled Device

Abbreviations/Acronyms

- CFD
- Computational Fluid Dynamics
- CI
- Compression Ignition
- CO
- Carbon Monoxide
- COV
- Coefficient of Variation
- CVCC
- Constant Volume Combustion Chamber or Compound Vortex Controlled Combustion
- DI
- Direct Injection
- DISC
- Direct Injection Stratified-charge
- DISI
- Direct Injection Spark-ignited
- DMI
- Direct Mixture Injection
- DNS
- Direct Numerical Simulation

Abbreviations/Acronyms

- DOHC
- Dual Overhead Cam
- ECU
- Electronic Control Unit
- EFI
- Electronic Fuel Injection
- EGR
- Exhaust Gas Recirculation
- EOI
- End of Injection
- EOS
- Equation of State
- EVC
- Exhaust Valve Closing
- EVO
- Exhaust Valve Opening
- FID
- Flame Ionization Detector
- FV
- Finite Volume
- FVM
- Finite Volume Method
- GDI
- Gasoline Direct Injection

Abbreviations/Acronyms

- HC
- Hydrocarbon
- HSDI
- High Speed Direct Injection
- IDI
- Indirect Injection
- IMEP
- Indicated Mean Effective Pressure
- IPC
- Inlet Port Closing
- IPO
- Inlet Port Opening
- IPTV
- Incidents per Thousand Vehicles
- ISFC
- Indicated Specific Fuel Consumption
- IVC
- Inlet Valve Closing
- IVO
- Inlet Valve Opening
- kPa
- kiloPascal
- LDA
- Laser Doppler Anemometry

Abbreviations/Acronyms

- LDV
- Laser Doppler Velocimetry
- LES
- Large Eddy Simulation
- LEV
- Low Emission Vehicle
- LHS
- Left Hand Side
- LIF
- Laser-induced Fluorescence
- MAP
- Manifold Absolute Pressure
- MBT
- Maximum Brake Torque
- MON
- Motor Octane Number
- MPa
- MegaPascal
- MPI
- Multipoint Injection
- NO
- Nitrous Oxide
- NO_x
- Nitric Oxides

Abbreviations/Acronyms

- OEM
- Original Equipment Manufacturer
- PCP
- Peak Cylinder Pressure
- PDA
- Phase Doppler Anemometry
- PFI
- Port Fuel Injection
- PIV
- Particle Imaging Velocimetry
- PLIF
- Planar Laser-imaging Fluorescence
- RIF
- Representative Interactive Flamelet
- RMS
- Root Mean Square
- ROHR
- Rate of Heat Release
- RON
- Research Octane Number
- SCV
- Swirl Control Valve
- SI
- Spark Ignition

Abbreviations/Acronyms

- SIDI
- Spark-Ignited Direct-injection
- SMD
- Sauter Mean Diameter
- SOHC
- Single Overhead Cam
- SOI
- Start of Injection
- TDC
- Top Dead Centre
- TWC
- Three-way Catalyst
- UHC
- Unburnt Hydrocarbon
- ULEV
- Ultra-low Emission Vehicle
- VVT
- Variable Valve Timing
- WOT
- Wide Open Throttle