University of
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Advanced propulsion systems
MMME4066

Lubrication system and Friction

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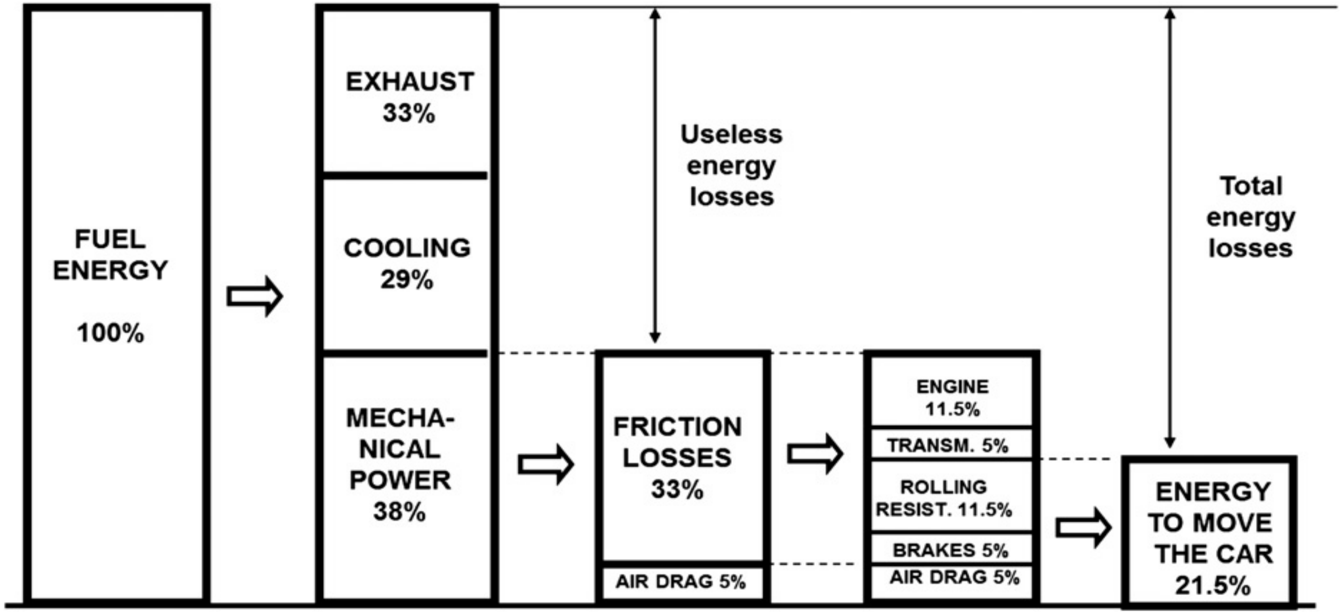
- Friction and lubrication
- Engine lubrication system
- Viscosity characteristics
- Stribeck diagram
- Journal bearings

Rubbing Friction in Engines



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Friction is the force resisting the relative motion of two surfaces and giving rise to the dissipation of work into heat

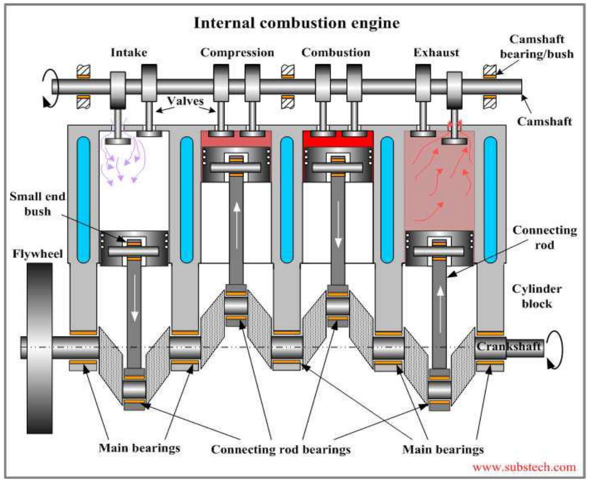
Lubricated friction occurs at the interface between solid surfaces separated by lubricating fluid. Most rubbing friction in engines is of this type

Dry friction occurs at the interface between solid surfaces. *Static friction (stiction)* occurs between non-moving surfaces, and *kinetic friction* between moving surfaces. Dry friction occurs in rubbing seals on the crankshaft. Dry friction makes a minor contribution to total engine friction

Sites of friction

Net Indicated Work = Rubbing Friction+ Ancillary Work+ Brake OutputWork

$$imep_n - bmep = fmep + amep = fmep^{\#}$$



Sites of Rubbing friction

Valve train, FEAD chains/belts (~20%)

Piston assembly: liner interface with rings and skirt; gudgeon pin, big end bearing. (~50%)

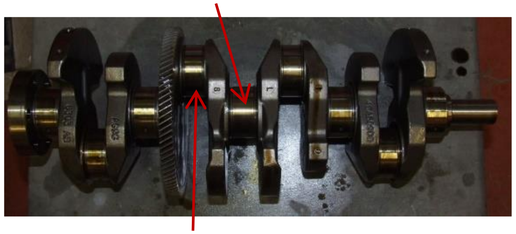
Crankshaft assembly: big end bearings, main bearings, seals (~30%)

Ancillaries

Coolant and oil pumps, alternator,

Anything else driven of the **Front End Accessory Drive (FEAD) system**

Journal bearings



Big end bearings

Impact of friction on engine behaviour

FMEP is work dissipated by friction per unit displacement of the engine per cycle

Power dissipated by friction $\dot{W}_f = fmep \times (nV_s) \times \frac{N}{2}$ $N\ in\ \frac{rev}{s}$

$$fmep = imep_n - amep - bmep$$

Typically, for a fully warm engine, fmep is around 0.5bar. This might account for typically 15% of vehicle fuel consumption in town

Under sub-zero, cold start conditions, fmep can be 5-10bar. This can cause long start up times or failure to start.

Friction can give raise to wear of components.

Methods of determining friction in firing engines

Measurement of friction obtained by subtracting brake power from the indicated power determined by in-cylinder pressure measurements

$$fmep^{\#} = fmep + amep = imep_n - bmep$$

Two common measurement methods are:

Willians line: fuel consumption is measured and plotted against load. The line extrapolated to find the negative load (*fmep*$^{\#}$) at which fuel consumption is zero. Measurements at constant speed. Plot slightly curve tricky to extrapolate.

Morse test: in a multi-cylinder engine running at constant speed, each cylinder is in turn disabled and the drop in *bmep* recorded

$$fmep^{\#} = \sum \Delta bmep - \overline{bmep}$$

Sum of drop in *bmep* recorded $= \sum \Delta bmep$

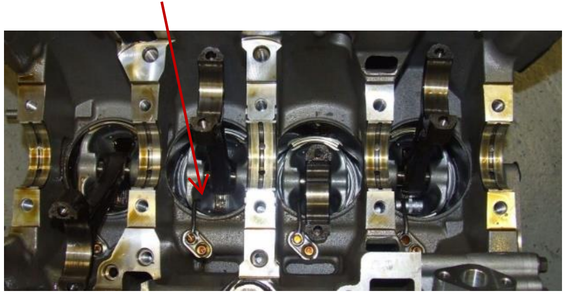
$$\overline{bmep} = engine\ bmep$$

Oil lubrication system

The principal function of the oil lubrication system is to reduce friction and wear at rubbing surfaces.

Cooling: Additional jets of oil may be directed onto underside of the piston to cool this and limit its temperature under high load conditions

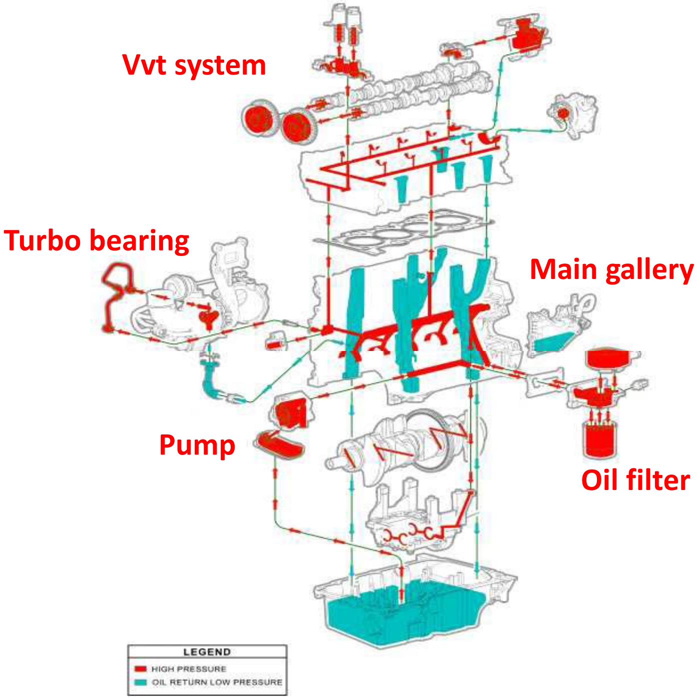
Piston cooling jet



The oil is also used as an actuating fluid for variable valve timing and lift system

This is the typical and current lubrication system of a 4-cylinder spark ignition engine

It is a wet sump system: oil is drawn from and returned to a sump reservoir



Red: pressurised part of the system
Blue: low pressure, flow due to gravity

Engine failure caused by lubrication problems

Oil between engine’s moving parts: reduces friction & carries away heat.

Oil is the primary means by which the rod and main bearings are cooled, as well as the pistons. Any reduction in oil flow may cause these parts to run hot and seize.

Engine failures might result from low oil pressure. (worn oil pump and/or excessive clearances in the main and rod bearings as a result of high mileage wear or neglect (not changing the oil and filter often enough)

Oil starvation is almost always fatal to any engine (failed oil pump, a plugged oil pickup screen inside the oil pan, or plugged oil passage).

Oil starvation: very shiny surfaces and evidence of wiping

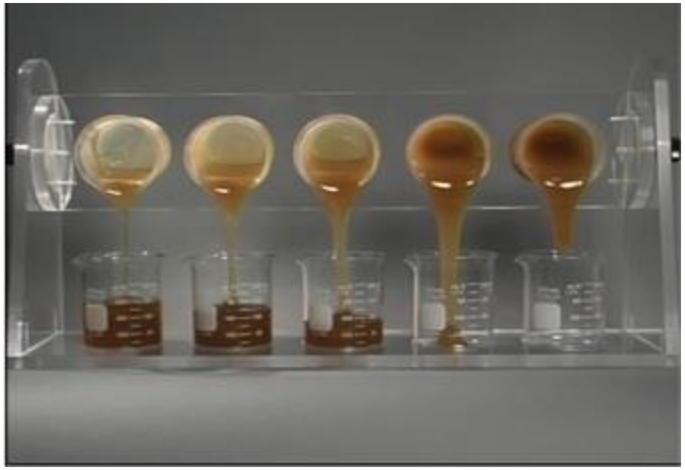
Oil Viscosity

Oil viscosity can be expressed as **kinematic** viscosity or **dynamic** viscosity.

These are related by density:

dynamic viscosity (μ) [Pa.s, N.s/m2] = kinematic (v) * density (ρ)

Values vary with temperature, shear and pressure. In engines, the oil performs under high shear conditions and over a temperature range from ambient to ~140ºC (on the valve deck)



The variation of kinematic viscosity with temperature is determined from flow through a capillary under low shear conditions.

Other tests - cold crank simulation (CCS) at low temperatures and high temperature high shear (HTHS) tests – give viscosity under high-shear conditions.

Typically the Vogel or Walther equations are used to describe the variation with temperature under low-shear

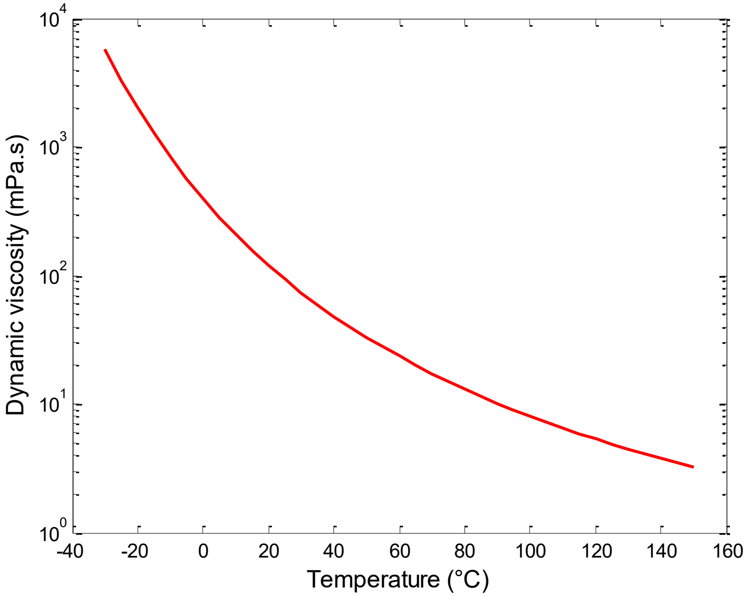
Dynamic viscosity & Vogel equation

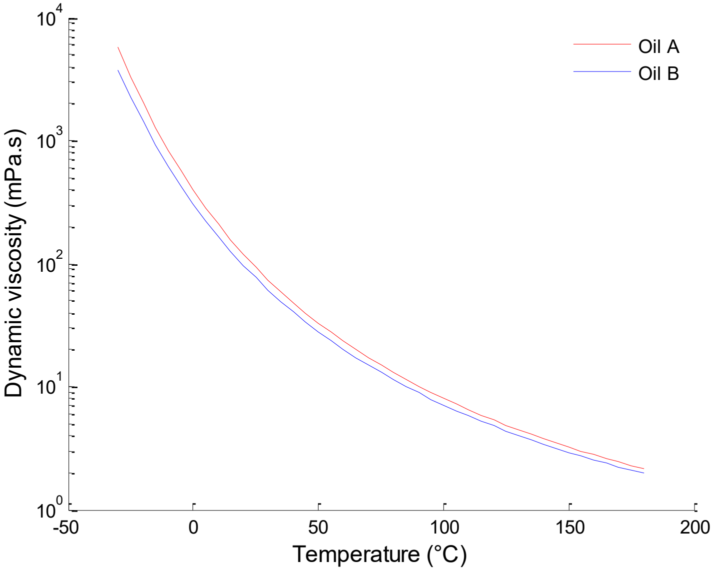
Vogel equation

$$\mu = k \exp\left(\frac{\theta_1}{\theta_2 + T}\right)$$

variation with temperature under low-shear

For example, oil 10 W / 30: *k* =0.078 mPa.s, θ_1 = 1180°C and θ_2 = 133°C





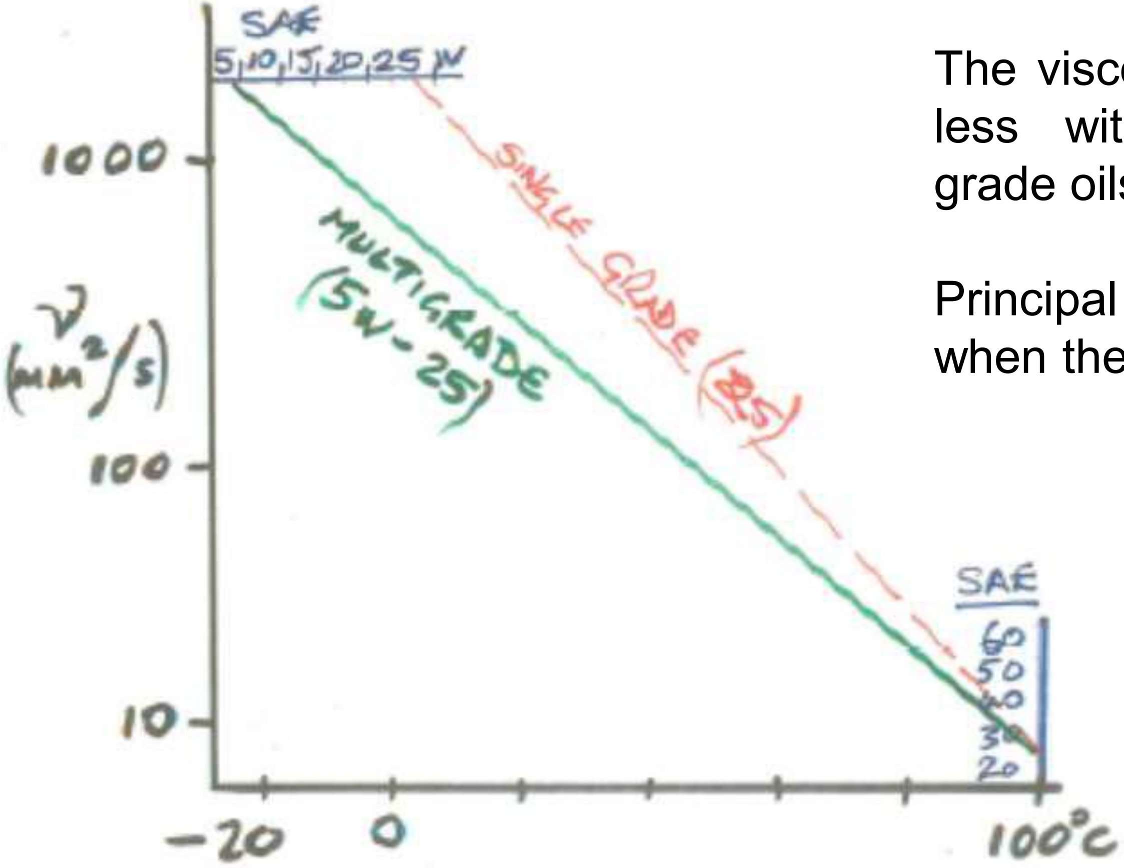
5 % different Vogel coefficients

SAE Viscosity Grades

The SAE grades defines how kinematic viscosity varies with temperature under low shear conditions

A multigrade oil is defined by two numbers giving the winter grade and the summer grade

Example 5w-25: Winter grade ‘5w’ defines viscosity at a low temperature (between -10 and -35ºC). Summer grade ’25’ defines viscosity at 100ºC



The viscosity of multigrade oils varies less with temperature than single grade oils

Principal benefit: less engine friction when the engine is cold.

Group activity - Viscosity

A 10W/30 oil with *k* =0.078 mPa.s, θ_1 = 1180°C and θ_2 = 133°C is at 40ºC and its temperature is increased to 85ºC. Calculate the dynamic viscosity at the two temperatures

Using Vogel equation at 40

$$\mu = k\exp\left(\frac{\theta_1}{\theta_2 + T}\right) = 0.078\exp\left(\frac{1180}{133 + 40}\right) = 71.5mPa.s$$

Using Vogel equation at 85

$$\mu = k\exp\left(\frac{\theta_1}{\theta_2 + T}\right) = 0.078\exp\left(\frac{1180}{133 + 85}\right) = 17.5mPa.s$$

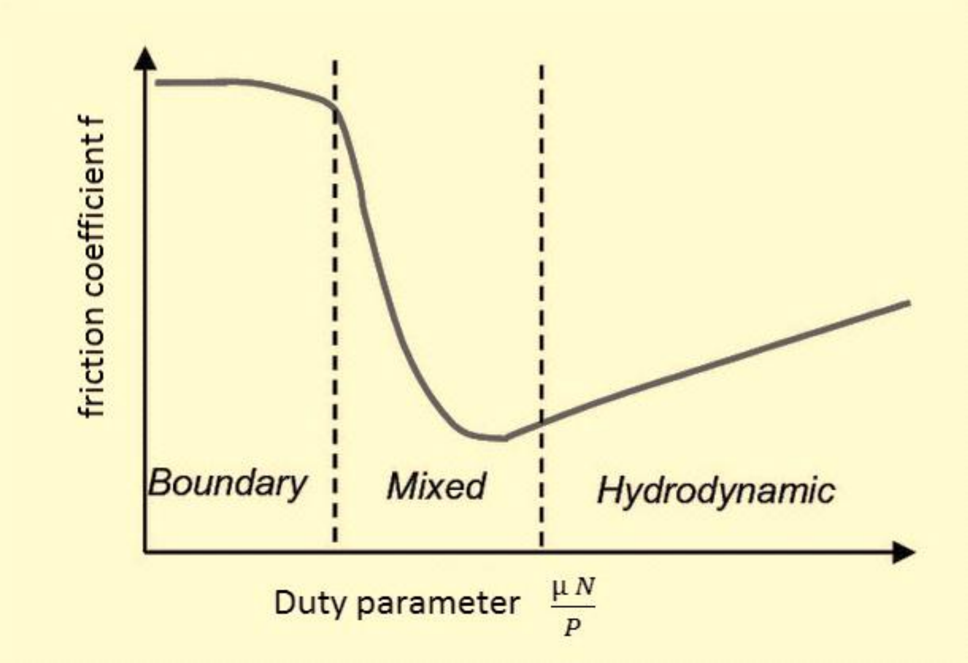
Dynamic viscosity decreased by a factor of 4

Stribeck diagram

The different regimes of lubricated friction can be illustrated in a diagram

- The Stribeck diagram is a plot of friction relating it to viscosity, speed and load
- Vertical axis friction coefficient (f)
- Horizontal axis dimensionless parameter combining viscosity (μ), speed (N) and load (P) – duty parameter (μ N / P)
- As you move to the right on the horizontal axis, the effects of speed increases friction
- Low speed and low viscosity with high load will produce boundary lubrication
- Boundary lubrication is characterized by little fluid in the interface and large surface contact, resulting in very high friction.

Stribeck diagram



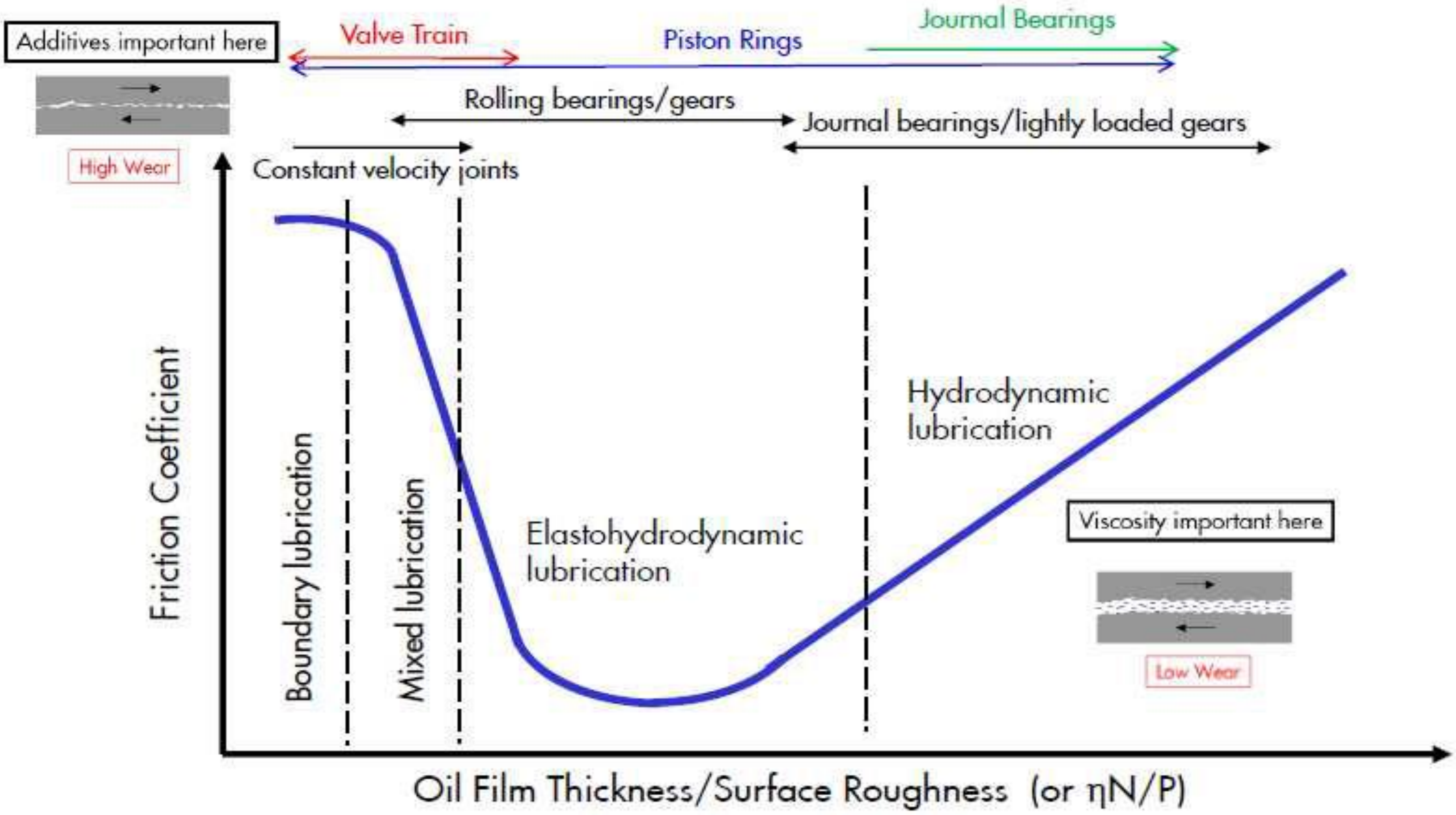
Different regimes of lubricated friction

Boundary lubrication, where the load-bearing surfaces comes into contact with each other, occurring at low speeds, high loads or changes in direction

Hydrodynamic lubrication is where the oil separates two load-bearing surfaces and can increase with increasing film thickness

Mixed lubrication where peaks of two surfaces come into contact through lubricating film, combination of boundary and hydrodynamic lubrication

Stribeck curve



- As the speed and viscosity increase, or the load decreases, the surfaces will begin to separate, and a fluid film begins to form
- The film is still very thin, but acts to support more of the load
- Mixed lubrication is the result, and is easily seen on the Stribeck curve as a sharp drop in friction coefficient
- The drop in friction is a result of decreasing surface contact and more fluid lubrication
- The surfaces will continue to separate as the speed or viscosity increase until there is a full fluid film and no surface contact

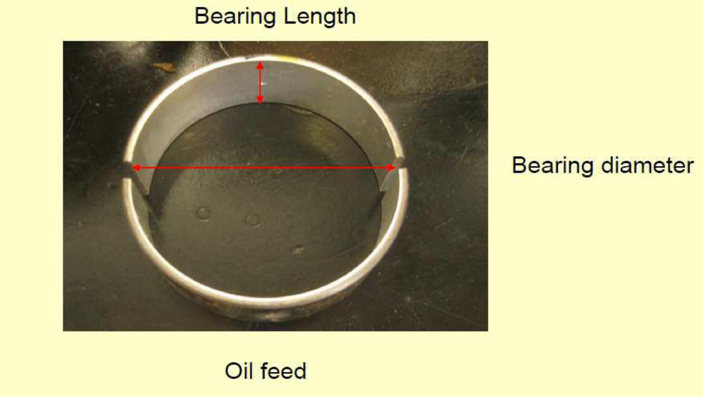
Plain (or journal) Bearings

Journal bearings are used for the crankshaft main bearings, the piston big end bearings, and the camshaft journal bearings. (May also be used to support a balancer shaft, if used)

These operate in hydrodynamic regime under most conditions, but may move into the boundary regime on start up and shut down

The inner part (the journal) rotates within a pair of shells; the journal and shells are separated by an oil film.

Crankshaft and big end journals rotate at engine speed; camshaft bearings rotate at half engine speed

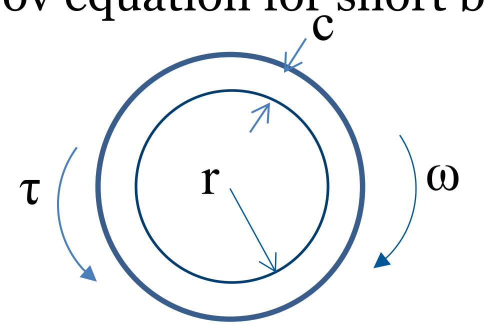


Friction torque on journal bearings

Engine journal bearings generally conform to short bearing approximations, **Petroff equation** gives a simple estimate of friction torque τ

Petroff's equation describes friction torque in flooded journal bearings; two main assumptions: short bearing; full fluid film

Petrov equation for short bearings;



Journal rotation ω (rad/s)
Journal radius r (m)
Radial clearance c (m)
Oil dynamic viscosity μ

short-bearing approximation is based on the assumption that the ratio of the journal's length to its diameter is less than or equal to 0.5

Friction dissipation in journal bearings

Let’s consider a liquid film between two surfaces in relative motion.
For small clearance, c, assume linear radial velocity gradient. The shear stress is

$$\tau = \mu\left(\frac{du}{dy}\right) = \mu\left(\frac{r\omega}{c}\right)$$

Friction force

$$F = \tau \times 2\pi r L = \frac{2\pi r^2 L\omega\mu}{c} = \frac{4\pi^2 r^2 L\mu N}{c}$$

where L is the bearing length and $\omega = 2\pi N$

Friction torque $T = F \times r = \dfrac{4\pi^2 r^3 L\mu N}{c}$

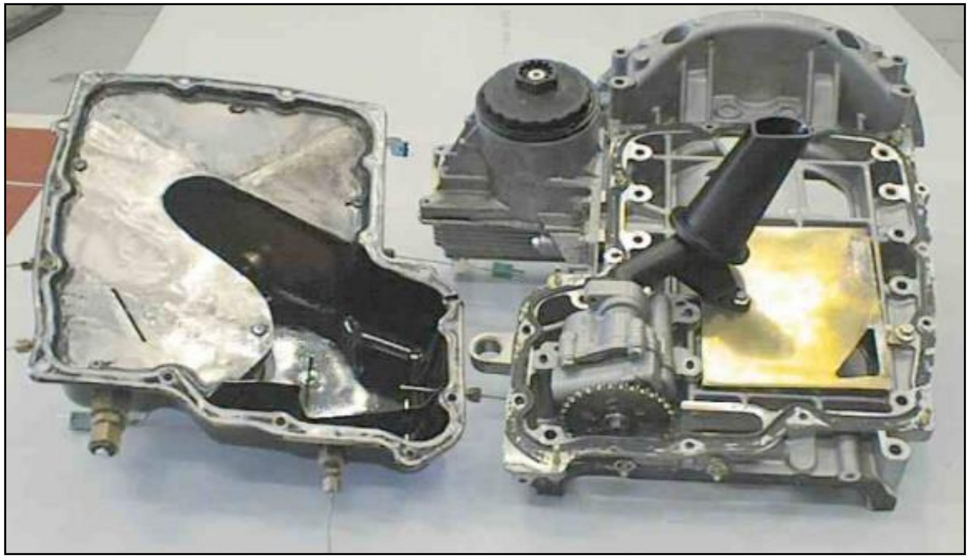
Friction dissipation $= T \times \omega = T \times 2\pi N$

Tribological means of reducing friction

Low friction coatings on mechanical components

The friction of components with new advanced surface coatings has been decreased by more than 90% for dry contacts and by 10–50% for boundary-lubricated contacts

New techniques, e.g. physical vapor deposition (PVD), chemical vapor deposition (CVD), and thermal spraying (TS), make it possible to industrially deposit coatings on car components.



Novel sump design to accelerate engine warm up

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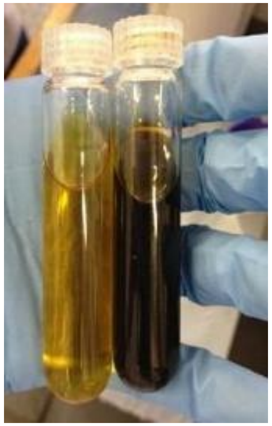
Tribological means of reducing friction

Surface topography and texturing

The topography of a surface in sliding contact has a remarkable influence on friction in both dry and lubricated sliding.
The most promising results originate from investigations with modifying the contacting surface in a very controlled way by partial laser surface texturing.
Microdimples can, when correctly designed, produce a wedge flow effect, and hydrodynamic pressure in the contact may reduce friction by about 25%



Lubricants

Research @ Nottingham has shown fmep $\propto \mu^{0.4}$.

If the lubricant viscosity could be reduced without detriment to the other functions of the lubricant, large energy savings could be achieved.

Additives such as the friction modifiers can be used

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Effect of soot on the Viscosity of Lubricant Oil

Soot particles are produced by direct injection diesel engines in fuel-rich regions during the combustion process & a small proportion is transferred to the lube oil.

3-4%wt of soot in oil can lead to the deterioration of lubricant performance, causing severe engine wear and penalising fuel economy and increasing CO_2 emissions.

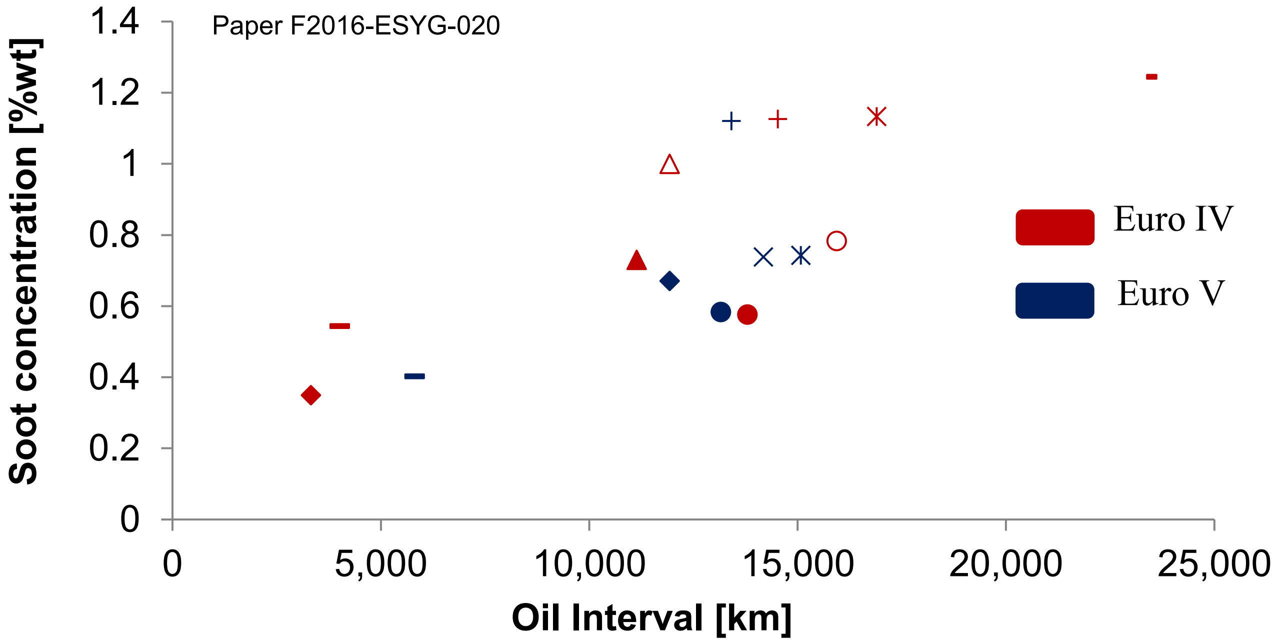
'sealed for life' engines requires a better understanding of soot transfer mechanisms & to reduce rates of soot transfer to oil.

Soot concentration near the cylinder walls more likely to be available for transfer to the oil film at the liner.

Exhaust Gas Recirculation (EGR) can influence both the total amount and distribution of soot in the cylinder.

Soot content in used oil from Euro IV & V vehicles

Soot-in-oil increased fairly linearly with mileage: average value of 1.0wt% after 15,000 km



Similar soot deposition rates for both Euro IV & V vehicles

Average deposition of 3.18 mg/km.

Typical urban and extra-urban driving cycle vehicle speeds (NEDC): 18.35km/h and 62.6km/h.

Rate of deposition: varied from **58mg/h** (urban cycle) to **200mg/h** (extra-urban cycle).

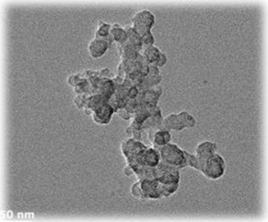
Soot-in-Oil Contamination

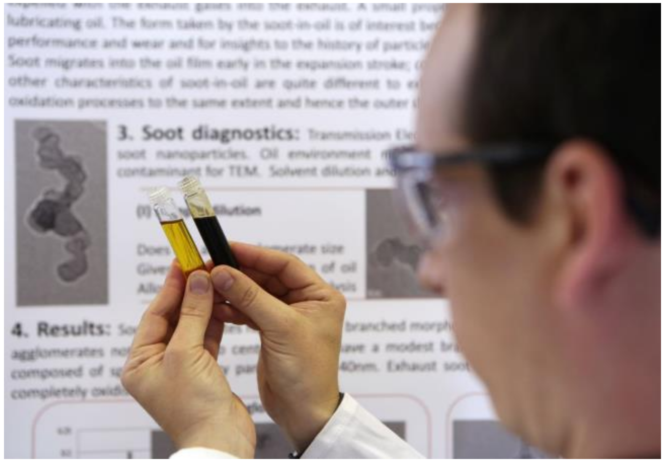
Soot forms in the combustion chamber. A small proportion is transferred to the lubricating oil.

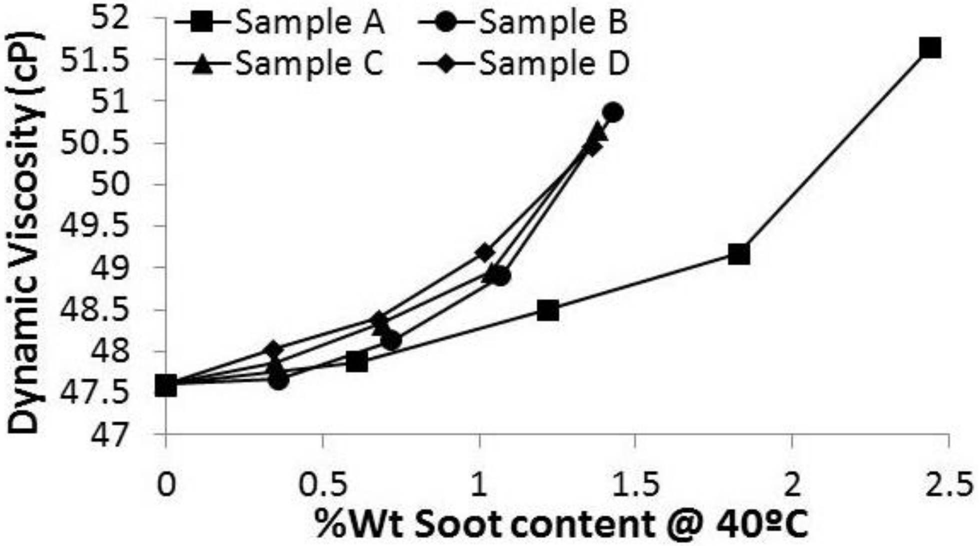
Oil thickening increases viscosity

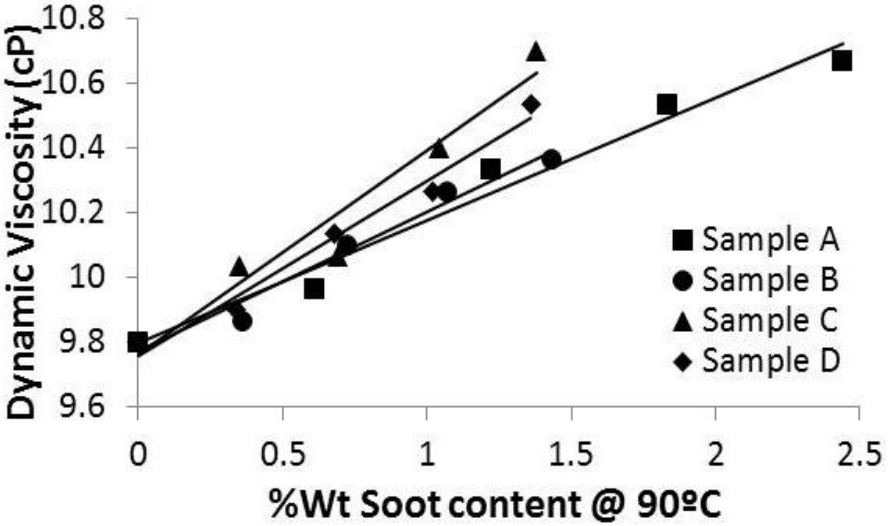
Soot build-up in oil reduces performance raising fuel consumption and CO_2 emissions

Soot thickening studied extensively, yet it is still not well understood









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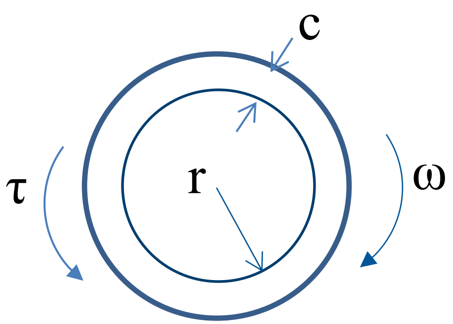
Petroff equation for short bearings



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Journal radius r (m)
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Friction force

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Friction torque $T = F \times r = \dfrac{4\pi^2 r^3 L\mu N}{c}$

Friction dissipation $= T \times \omega = T \times 2\pi N$

Group Activity

The crankshaft rig at Nottingham University has 5 main journal bearings, a diameter of 70 mm and a length 20 mm. At 20 and 90° C the shaft rotates at 1000 rpm and has a 35 μm clearance of oil. The viscosities at 20 and 90° C are 120 and 10 mPa.s respectively. What are the torques at these temperatures?

Bearing radius 35×10^{-3} m with a clearance of 35×10^{-6} m

Bearing length 20×10^{-3} m and a speed of 16.67 rev per second

Using Petroff equation the friction torque is:

$$T = F \times r = \frac{4\pi^2 r^3 L \mu N}{c}$$

Group Activity

At 20ºC the torque for one bearing is:

$$T = F \times r = \frac{4\pi^2 r^3 L\mu N}{c} = \frac{4\pi^2 (35 \times 10^{-3})^3 \times 20 \times 10^{-3} \times 120 \times 10^{-3} \times 16.67}{35 \times 10^{-6}} = 1.9Nm$$

Total torque for 5 bearings: 1.9 x 5 = 9.6 Nm

At 90ºC the torque for one bearing is:

$$T = F \times r = \frac{4\pi^2 r^3 L\mu N}{c} = \frac{4\pi^2 (35 \times 10^{-3})^3 \times 20 \times 10^{-3} \times 10 \times 10^{-3} \times 16.67}{35 \times 10^{-6}} = 0.16Nm$$

Total torque for 5 bearings: 0.16 x 5 = 0.8 Nm

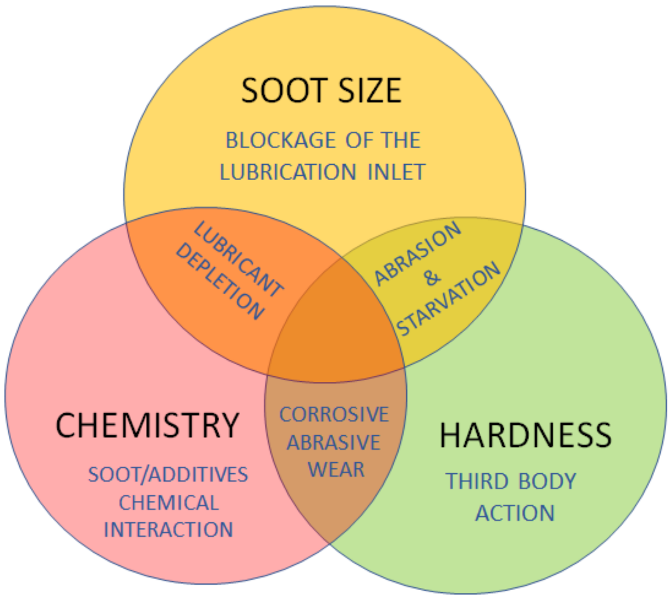
Summary

Friction in Engines

The oil lubrication system

Viscosity & its temperature dependence

Wear seems to be the result of coupled mechanisms



SOOT SIZE

BLOCKAGE OF THE LUBRICATION INLET

LUBRICANT DEPLETION

ABRASION & STARVATION

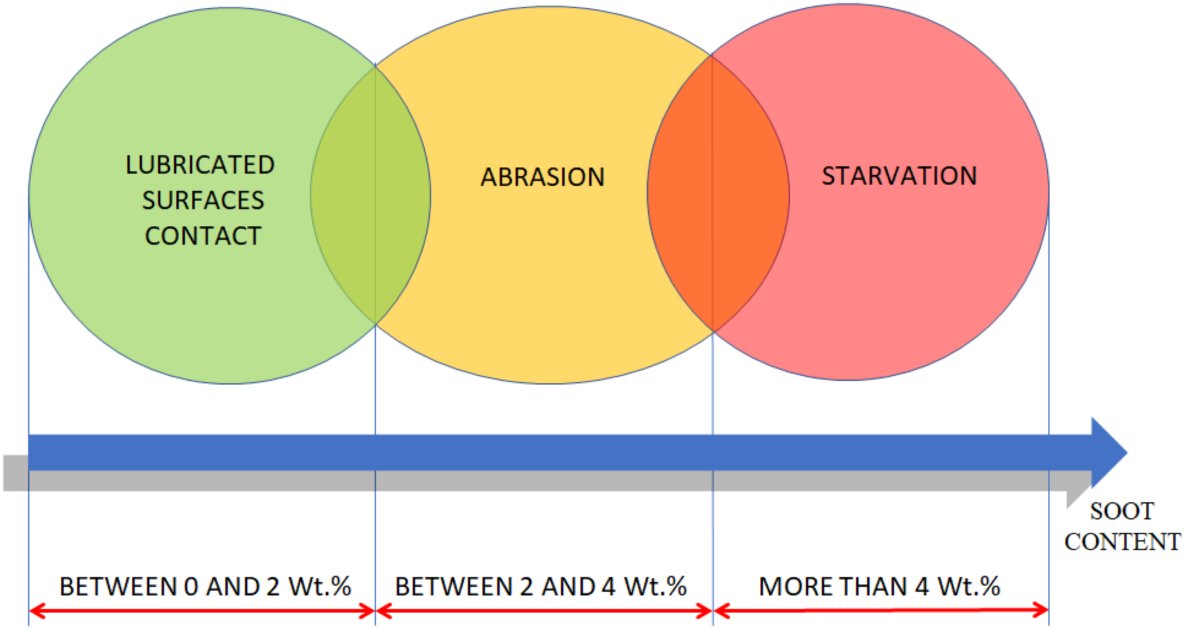
CHEMISTRY

SOOT/ADDITIVES CHEMICAL INTERACTION

CORROSIVE ABRASIVE WEAR

HARDNESS

THIRD BODY ACTION



LUBRICATED SURFACES CONTACT

ABRASION

STARVATION

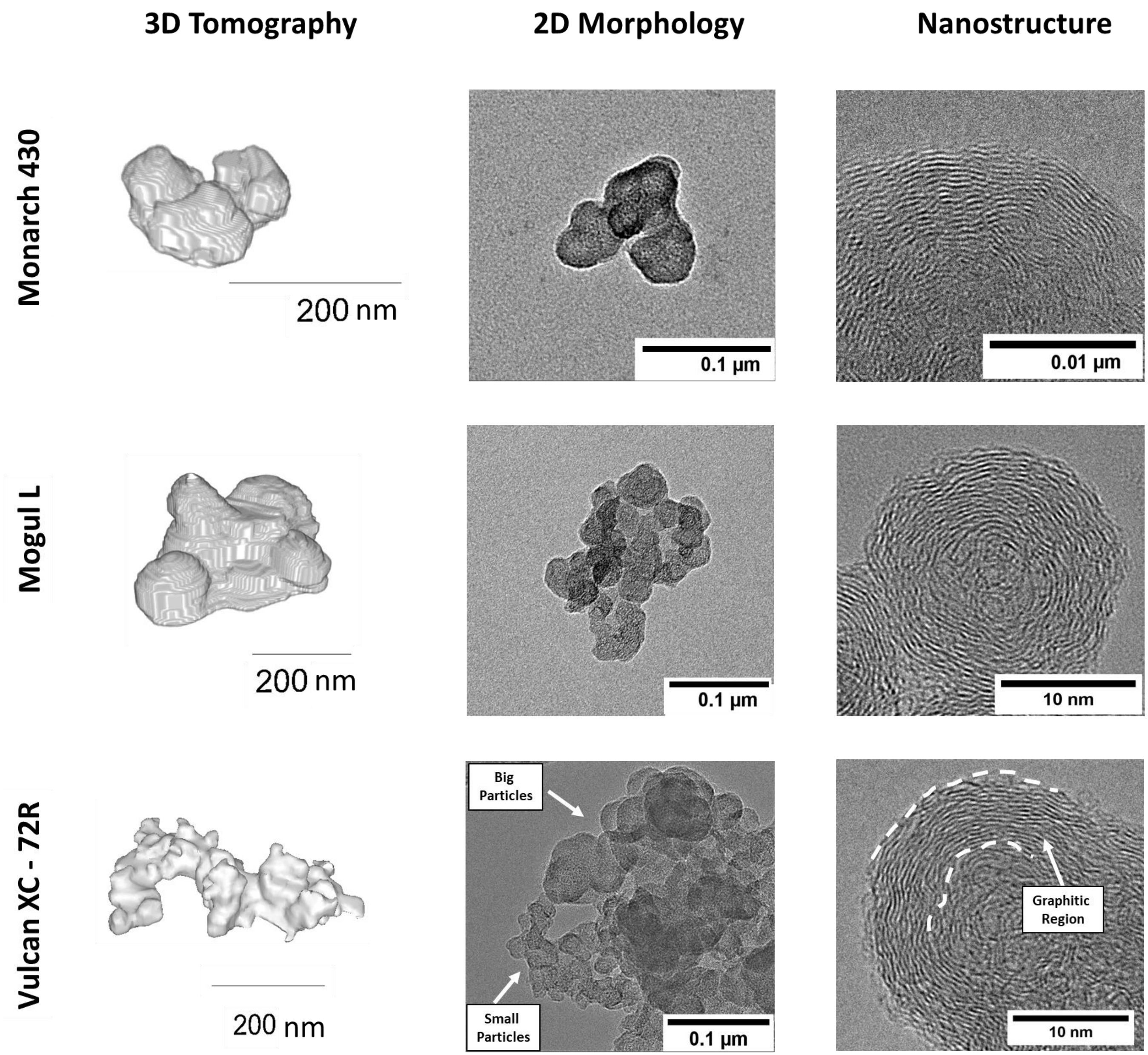
SOOT CONTENT

BETWEEN 0 AND 2 Wt.%

BETWEEN 2 AND 4 Wt.%

MORE THAN 4 Wt.%

Carbon Blacks: Chain-like carbon structures with high graphitic character


3D Tomography

2D Morphology

Nanostructure

- carbon aggregates in the 100-500 nm range
- chain-like structures
- Vulcan: mix of small and big graphitic PP*
- Mogul & Monarch: smaller PP with lower degree of graphitization

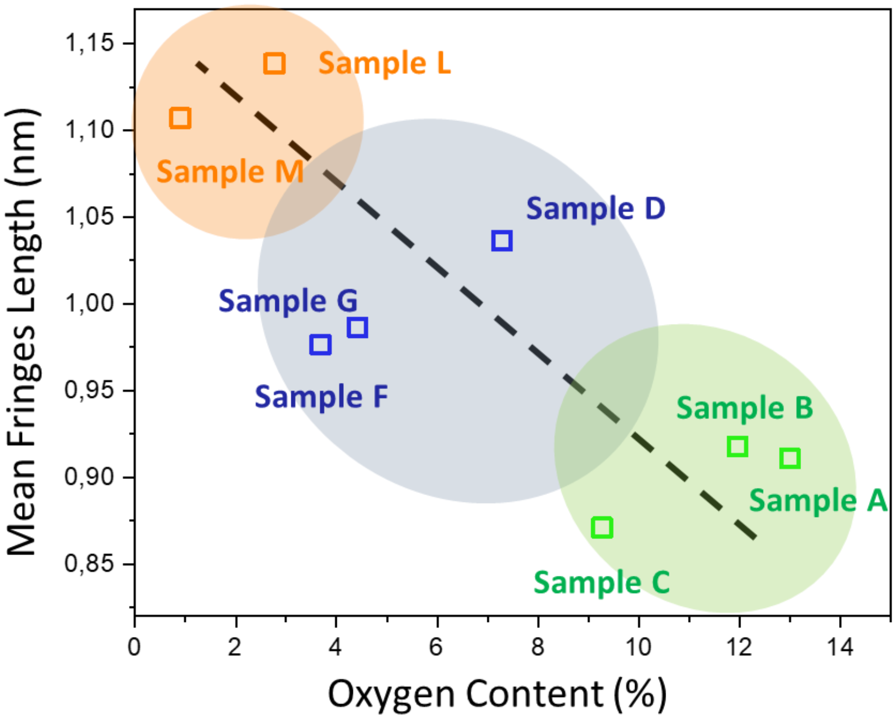
Correlation between surface oxygen content and fringes parameters

○ Carbon blacks show the higher graphitization

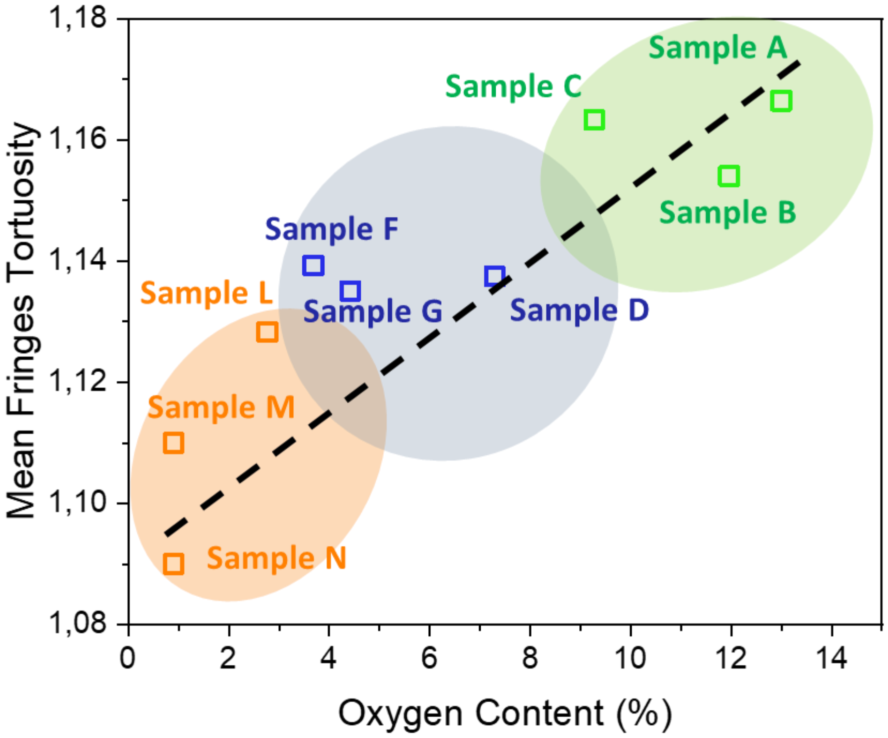
○ Class II samples show higher oxygen content

○ More interaction with the engine oil is expected for the class II samples

● Class I
● Class II
● CB

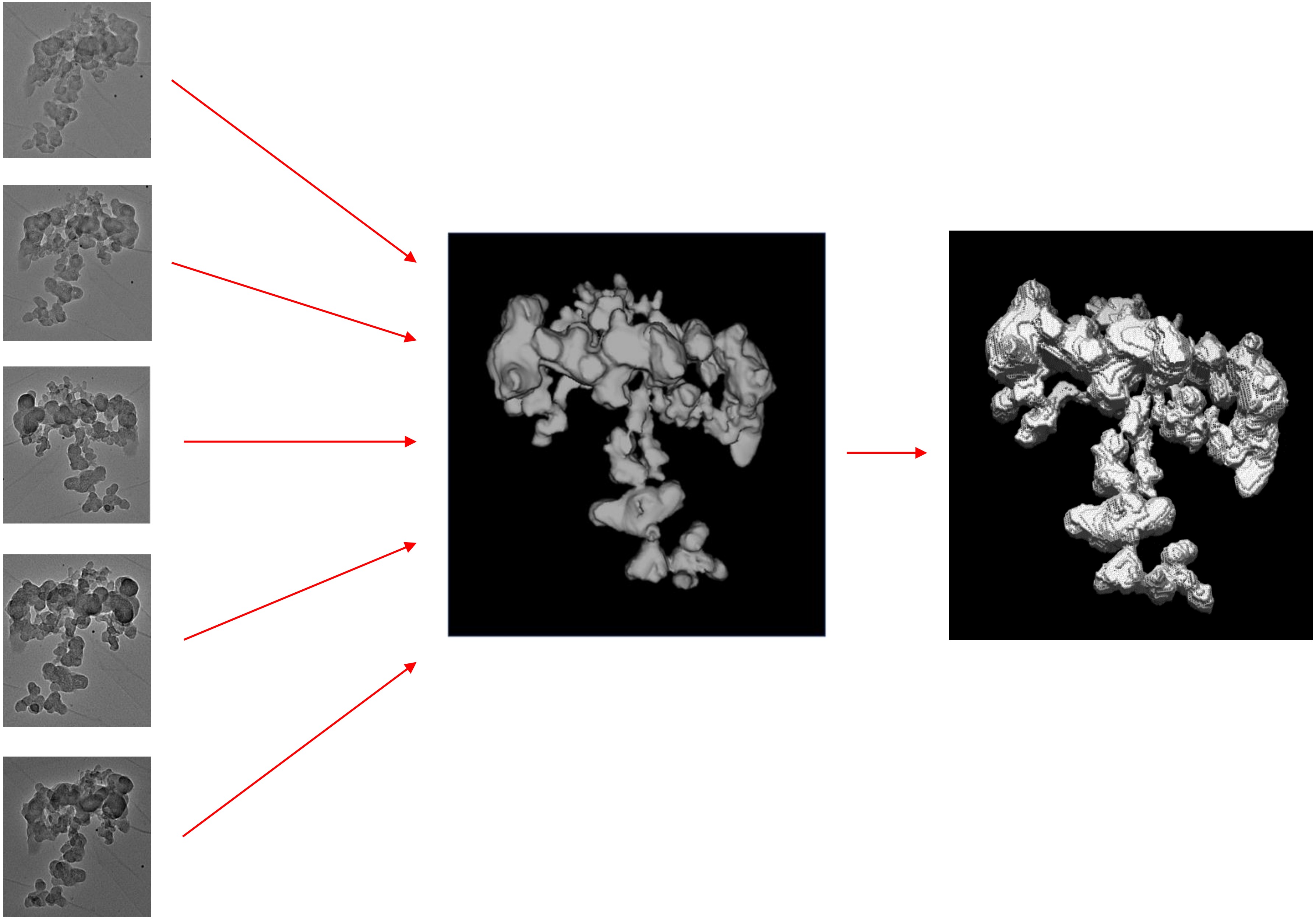


(a)

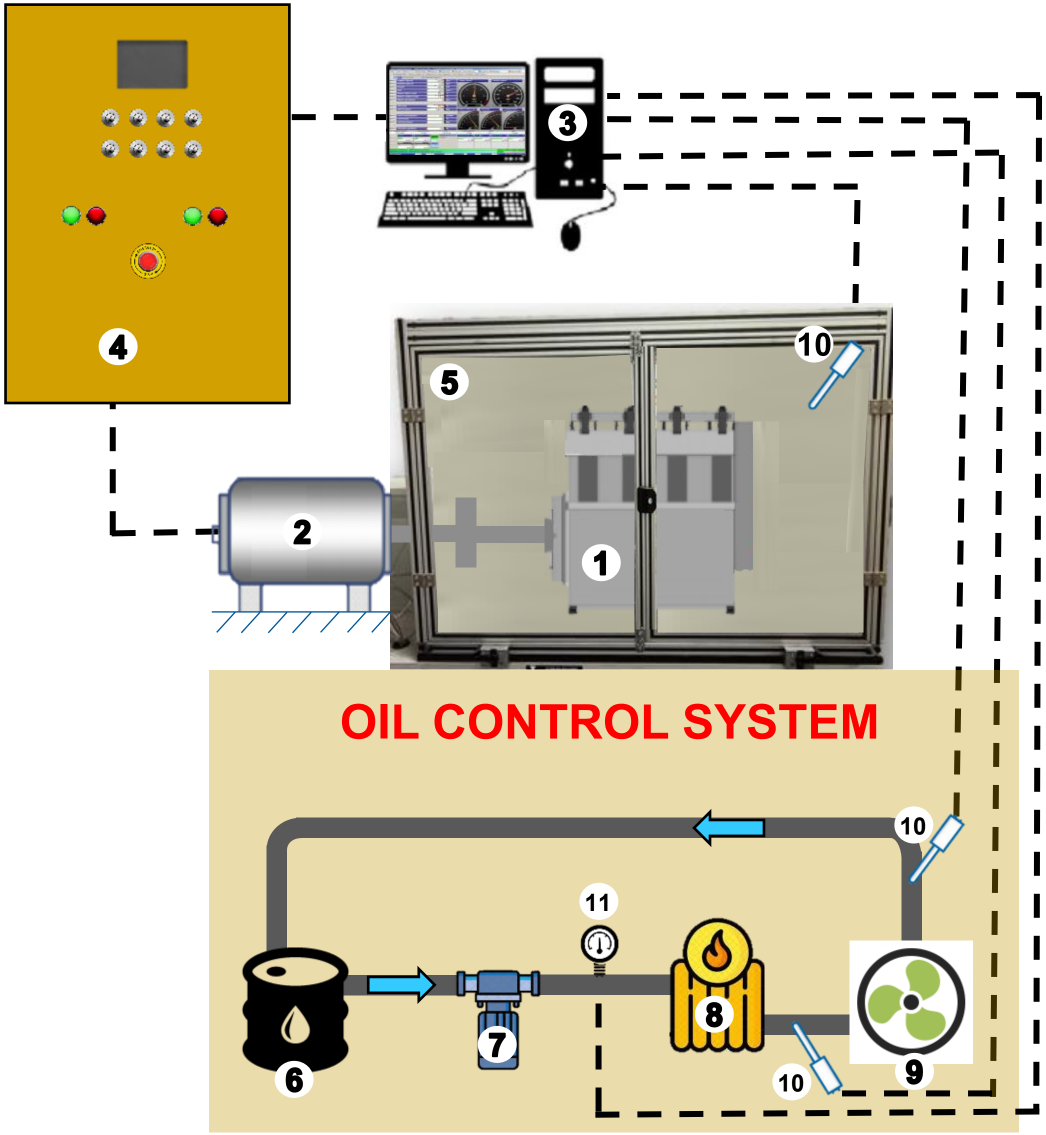


(b)

3D Reconstruction of a soot nanoparticle



Wear Test rig – Engine Test Facility

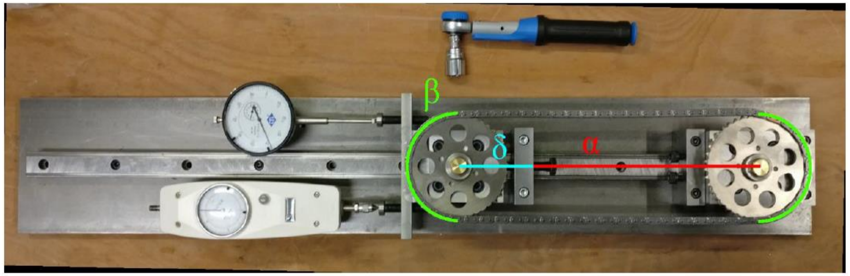


Components:

1. Internal Combustion Engine
2. Dynamometer
3. PC
4. Dyno Control System
5. Test cell
6. Oil Tank
7. Oil Pump
8. Heater
9. Fan
10. Thermocouples
11. Pressure Trasducer

Testing Facilities:

- ✓ CADET 14 software
- ✓ WLTC Test cycle
- ✓ Oil temperature control system
- ✓ Up to 100 h of testing
- ✓ 10L oil system capacity



Content

1. Introduction to coolant systems
2. Heat transfer
3. Conduction through the walls
4. Worked examples

Why does an engine need to be cooled?

Principally to limit the operating temperature of engine parts to values consistent with durability and integrity requirements. Typically this requires temperatures to be <200°C .

Compare this to temperatures of gases in the cylinder:

Towards the end of the compression stroke, the charge temperature is typically 600K in an SI engine and 675K in a diesel

After charge combustion, early in the expansion stroke the the products of combustion have a temperature typically 2000- 2500K

During the exhaust stroke, the charge (exhaust) gas temperatures vary typically between 500K (idling) to ~1300K (full load)