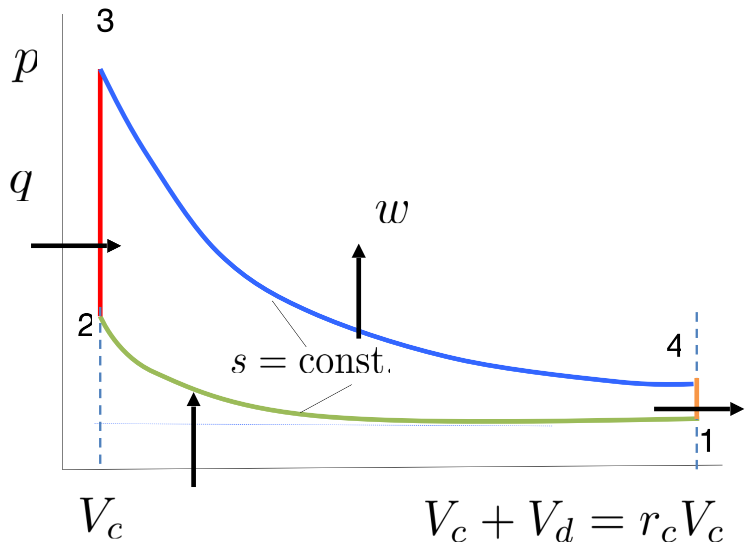
SI engines

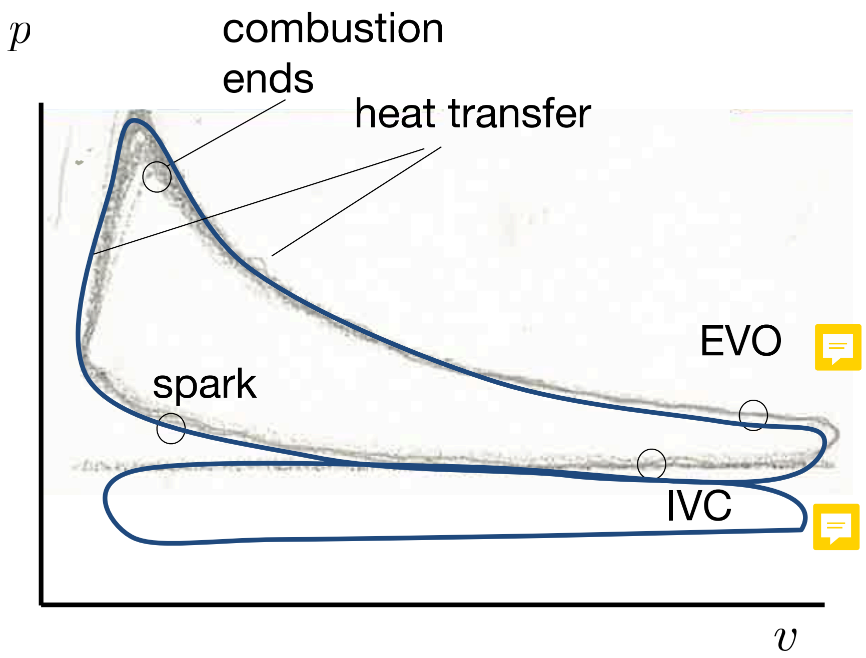
4A13

Heywood: Ch 5. Ideal Models of Engine Cycles, Ch 6. Gas Exchange Processes, 9. Combustion in Ignition Engines
Stone: Ch. 4. Spark Ignition Engines, Ch. 5, Direct Injection Spark Ignition Engines, Ch. 8. Induction and Exhaust Processes

Ideal cycles



Real cycles - SI engines



- **Real gases:** fresh, residual and burned, variable properties with temperature
- **Gas exchange**: mixing and pumping
- **Heat transfer**: end of compression and combustion, exhaust enthalpy
- **Combustion phasing**: around but not *at* TDC (spark or injection)
- **Pumping**: pressure and enthalpy losses at intake and exhaust
- **Incomplete combustion**: 1-2%
- **Leakage**: rings
- …

Air-fuel ratio in IC engines

$$AFR = \frac{\dot{m}_a}{\dot{m}_f}$$

SI: 12-18
CI: 18-70

$$\lambda = \frac{AFR}{AFR_s} = \frac{1}{\phi}$$

>1 fuel-lean
<1 fuel-rich

AFR largely controls:

- Mixture properties: (air+fuel, products)
- Rate of combustion
- Peak temperature
- Emissions

SI: λ~1.0 except cold start /accel/deceleration
Catalytic converter requirement for NOx and HC removal

CI: λ~1.2-1.5
Excess air required for complete fuel burnout/ lower soot

Inlet conditions conditions and performance

Volumetric efficiency

$$\eta_v = n_R \frac{\dot{m}_a}{\rho_{a,i} V_d N}$$

Power

$$P = \eta_f \dot{m}_f q_f = \eta_f \frac{\dot{m}_a}{AFR} q_f = \eta_f \eta_v \frac{\rho_{a,i} V_d N q_f}{AFR n_R}$$

imep

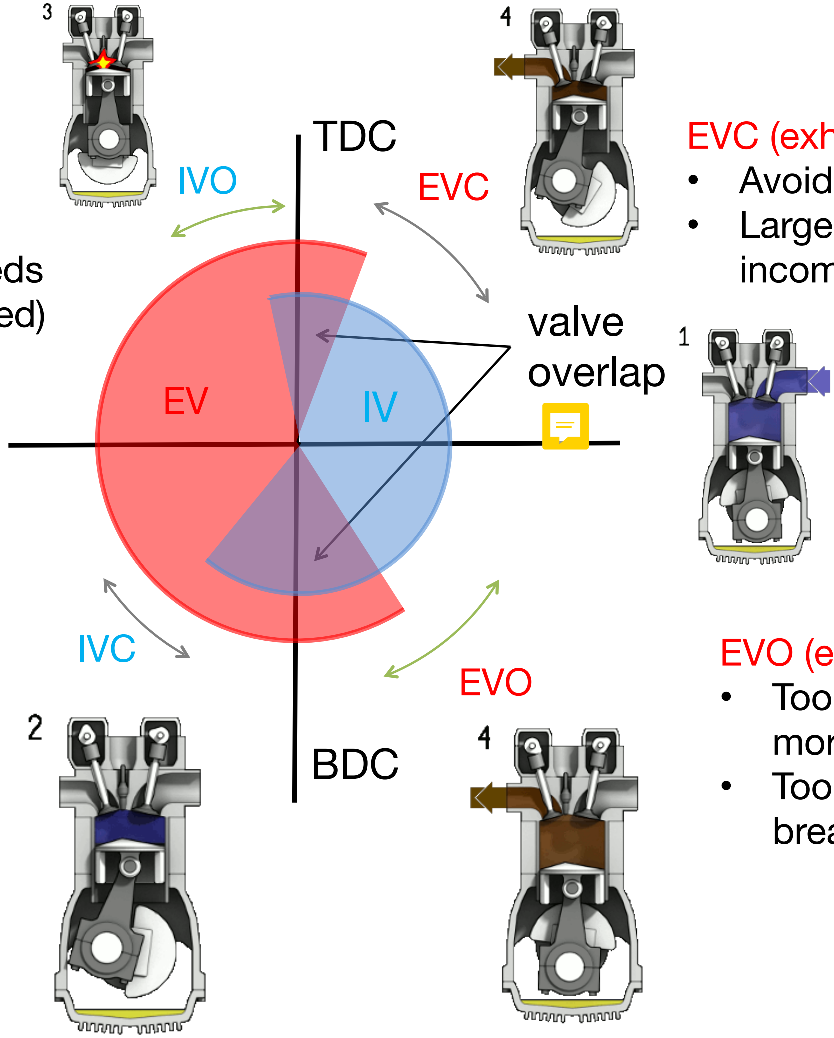
$$\text{imep} = \frac{P n_R}{N V_d} = \eta_f \eta_v \frac{\rho_{a,i} q_f}{AFR}$$

For a fixed AFR and inlet conditions, the imep varies only with the fuel efficiency and volumetric efficiency.

Gas transfer processes: valve timing

IVO (intake valve open):

- Before TDC (large opening area)
- Earlier for for higher power/speeds
- Earlier for high residuals
- Ideally longer IVO-IVC for higher speeds (variable valve timing (VVT) can be used)



EVC (exhaust valve close):

- Avoid compression after EVC
- Larger valve overlap for heating incoming injected fuel

EVO (exhaust valve open):

- Too early: less gross work (but more turbo)
- Too late: poor overall scavenging/ breathing

IVC (intake valve close):

- Optimum for volumetric efficiency (with IVO) vs. speed

Valve timing notes

Intake

- The intake valve opens just before TC after the exhaust gases have been expelled. Fresh mixture is induced into the cylinder by the pressure drop as the piston pulls air across the intake valves as they open.
- A long dwell period (EVC-IVC) is useful to let sufficient air for high power needs, whereas at idle only a short opening period is needed: this motivates the use of variable valve timing (VVT), with e.g. two stage valves.
- The process of air induction also plays a role in determining the motion of the gases, swirl (rotation around the cylinder axis) and tumble (rotation around the radial axis), and a differential design feature in determining engine performance.
- The early success of the Honda Civic valve (CVCC - Compound vortex controlled combustion) was due to the use of a variable control for the intake: partial blocking of the incoming flow during low load operation led to good performance at low speeds/low loads (as high speed flows increased burning velocity), and change to a larger opening for higher loads.
- As the speed increases, there is a limit to the speed of the flows induced into the system, as the valve eventually chokes: the design of engines and intakes is carefully tuned acoustically to optimise the rate at which air can flow into the cylinder.

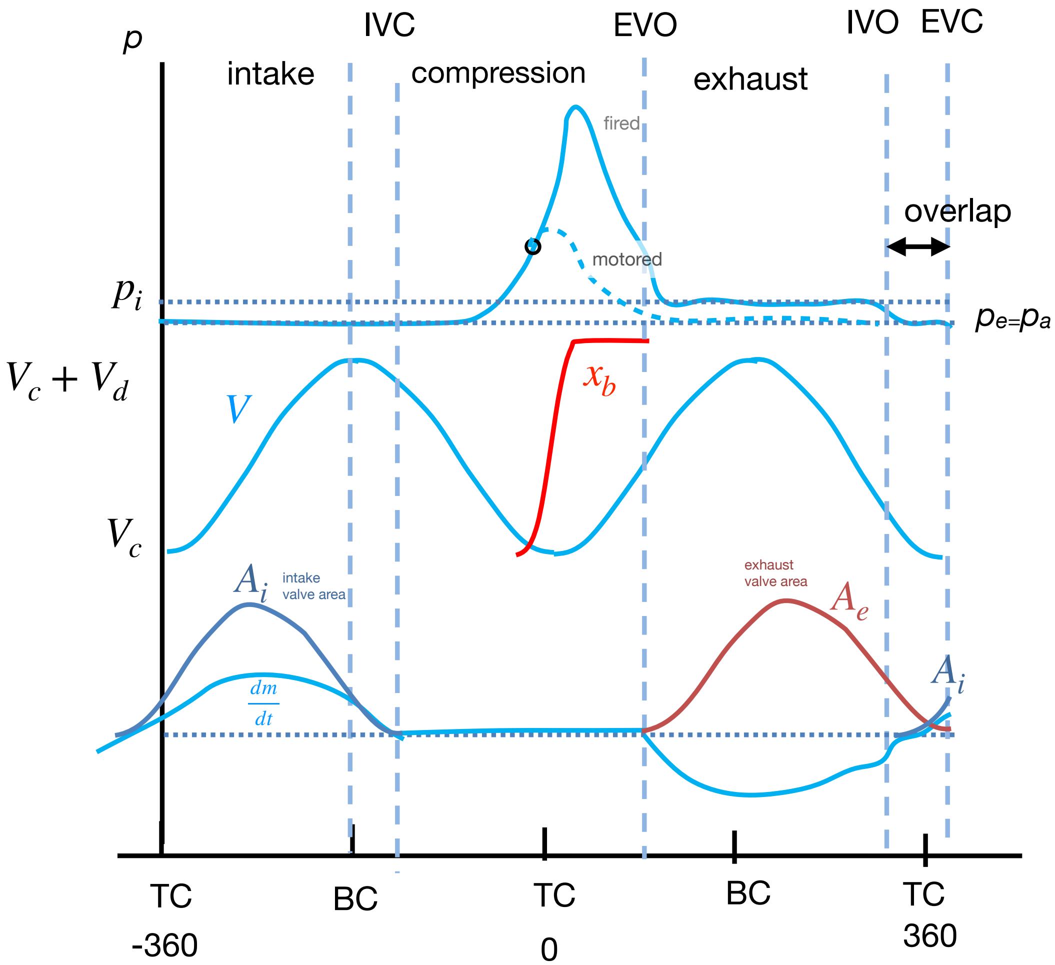
Exhaust

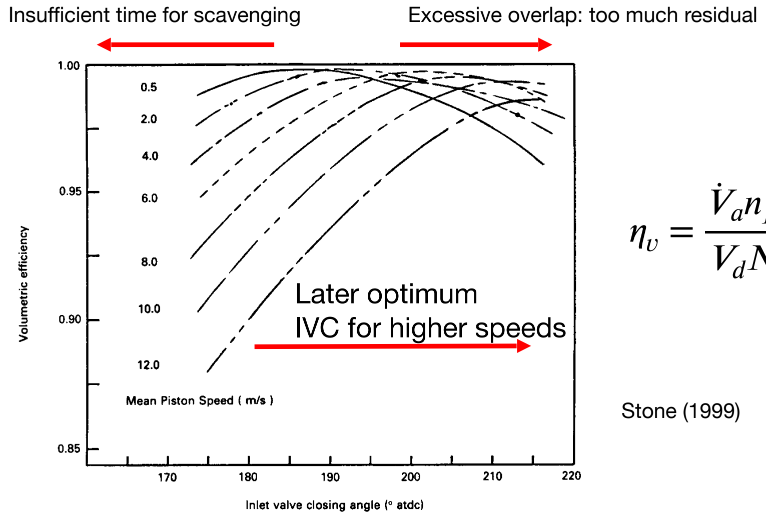
- The exhaust valve opens after a sufficient amount of work has been extracted from the expanding burned gases, typically near bottom center of the power stroke. The burned gases are displaced by the pressure difference across the valve, as well as the upward motion of the piston.
- The exhaust valve closes near top center, leading to an overlap with the open intake valve. In port fuel injected (PFI) engines, the overlap between exhaust and intake is usually designed to push a sufficient amount of heated exhaust gases back into the intake, so as to vaporise the incoming fuel, which is injected on the back of the intake valve.
- The valve overlap period can also be designed to extract more or less work from the power cycle, extending the power stroke with an early or late valve opening, and thus controlling load in lieu of throttling.
- The fraction of hot residual burned gases trapped in the cylinder is also a function of the valve overlap. Residual gases have higher specific heat than fresh mixtures, and can be used to control final burned gas temperatures (internal exhaust gas recirculation).

Two-stroke

- In two-stroke engines, the valve overlap is determined by the position of the ports, the speed of the engine and the throttle. Some motorcycle engines use the hot residual gas to drive autognition in a CI mode (ATAC cycle). The processes of gas exchange are much more important in (nearly unvalued) two-stroke engines in determining performance and emissions.

Gas exchange and volumetric efficiency



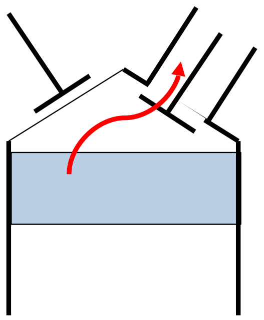


$$\eta_v = \frac{\dot{V}_a n_R}{V_d N} = \frac{\dot{m}_a n_R}{\rho_{a,i} V_d N}$$

Stone (1999)

Variable valve timing (VVT):
- allows optimisation of gas exchange for a compromise between efficiency, residual (emissions) and power
- large overlap can allow great dilution of charge, which minimises the need for throttling, thus increasing efficiency

Residual mass fraction of burned gases in cylinder



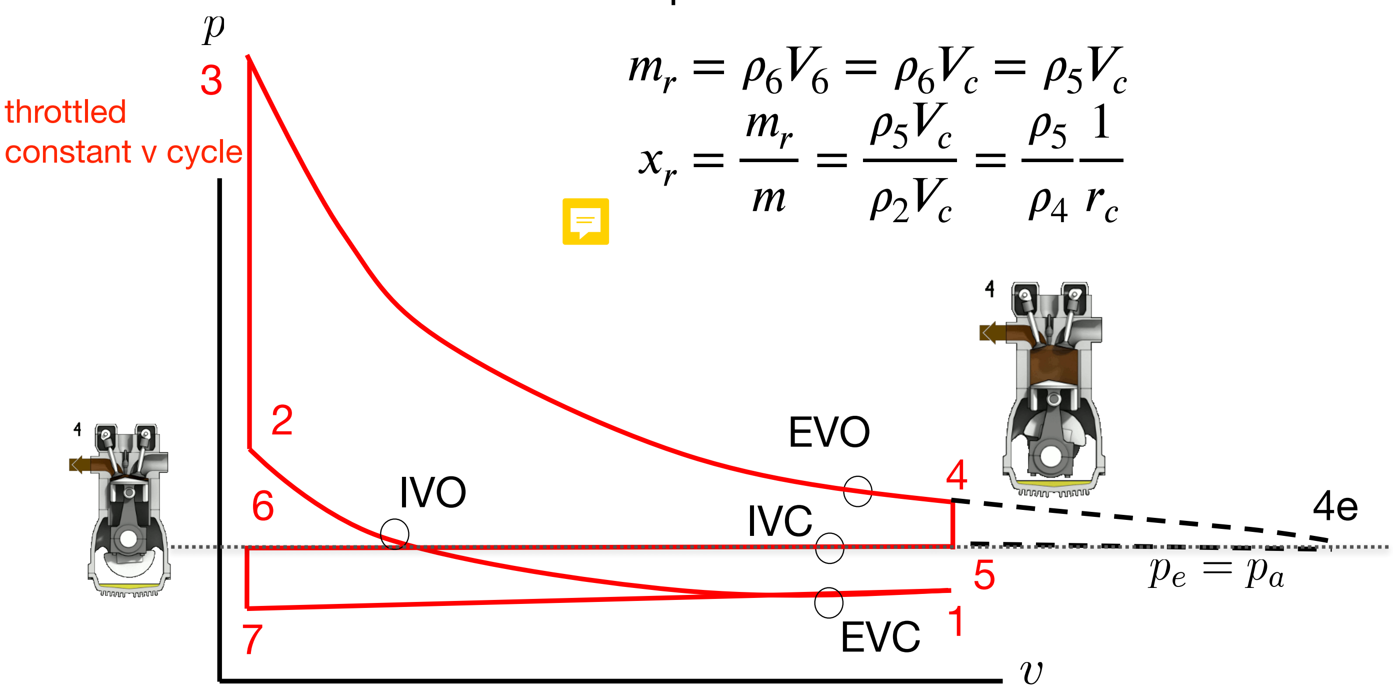
- What is the composition and temperature of the charge mass in the cylinder before compression starts?
- Air and fuel from the manifold intake, plus the residual burned gas from the previous cycle
- Let us consider a simple model for the state and total mass of the residual gas, assuming that the state is the same as the full expanded exhaust gas at point $4e$

$$m_r = \rho_6 V_6 = \rho_6 V_c = \rho_5 V_c$$

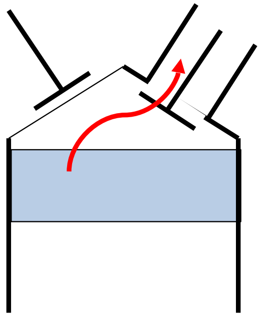
$$x_r = \frac{m_r}{m} = \frac{\rho_5 V_c}{\rho_2 V_c} = \frac{\rho_5}{\rho_4} \frac{1}{r_c}$$

Assuming same density 5=6

Since $\quad m = \rho_2 V_c = \rho_4 V_c r_c$



Residual mass fraction of burned gases in cylinder

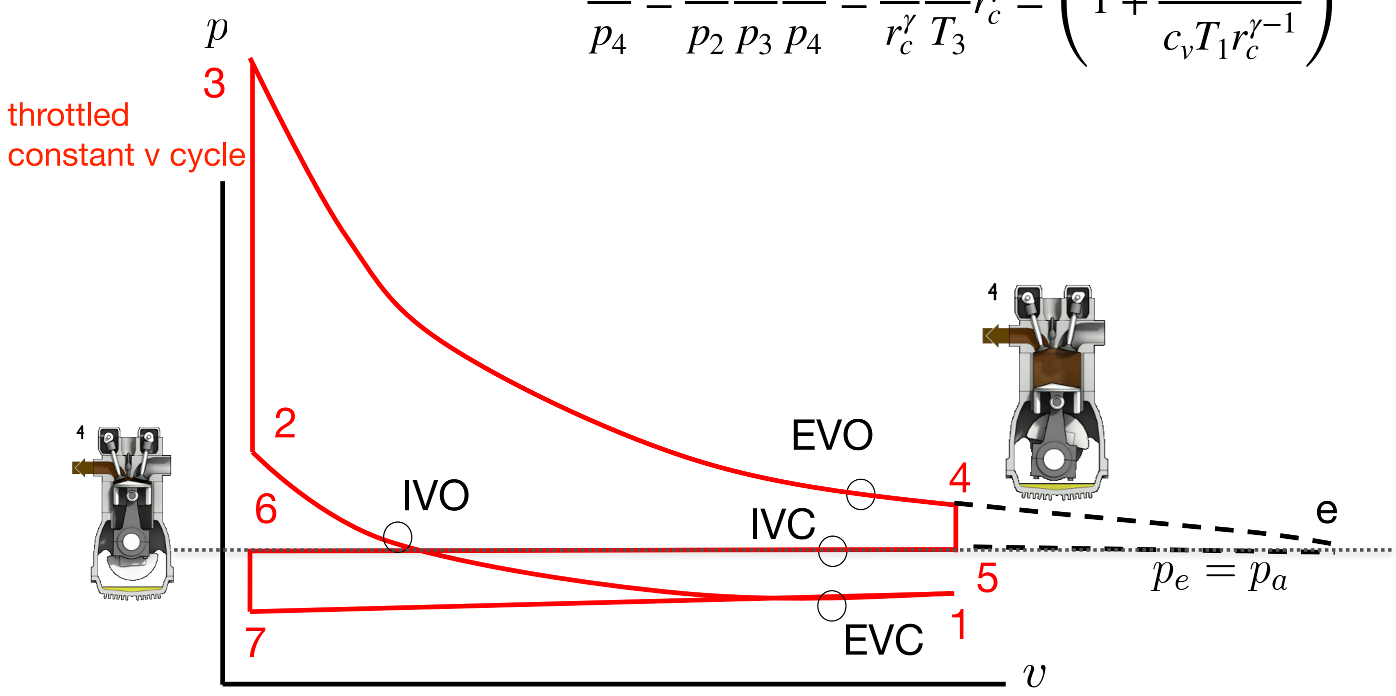


$$\frac{1}{r_c}\frac{\rho_5}{\rho_4} = \frac{1}{r_c}\left(\frac{p_e}{p_4}\right)^{1/\gamma} = \frac{1}{r_c}\left(\frac{p_e}{p_1}\right)^{1/\gamma}\left(\frac{p_1}{p_4}\right)^{1/\gamma}$$

Isentropic expansion 4-4e (mass expands into the exhaust and intake), and state 5 = 4e

$$\frac{p_1}{p_4} = \frac{p_1}{p_2}\frac{p_2}{p_3}\frac{p_3}{p_4} = \frac{1}{r_c^\gamma}\frac{T_2}{T_3}r_c^\gamma = \left(1 + \frac{Q^*}{c_v T_1 r_c^{\gamma-1}}\right)^{-1}$$

$$\boxed{x_r = \frac{1}{r_c}\frac{\left(p_e/p_i\right)^{1/\gamma}}{\left[1 + Q^*/(c_v T_1 r_c^{\gamma-1})\right]^{1/\gamma}}}$$



- **Residual mass fraction**: contains CO_2, H_2O, which have higher c_v : impact on mean and final temperature and therefore emissions.
- Combined with **EGR** (exhaust gas recirculation) for NOx emission control.

Example: residual gas calculation

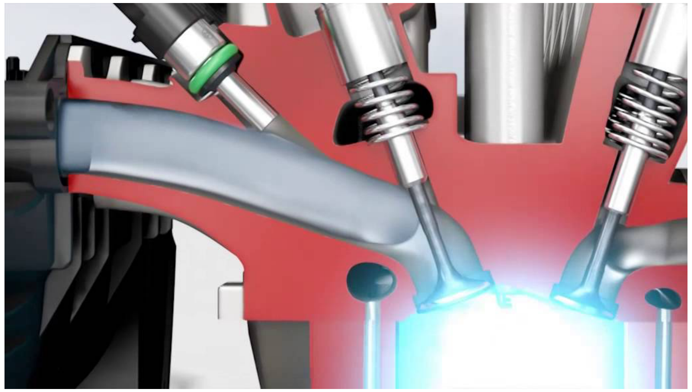
(a) Estimate the mass fraction of residual gases remaining in the cylinder for a cycle with the following conditions:
p_e = 1 bar
p_i = 0.5 bar,
T_1= 300 K,
q_f=45 MJ/kg,
Y_f= 0.05,
r_c= 10,
γ= 1.35,
c_v= 1.2 kJ/kg K.

(b) The fraction of mass of fuel in the fresh charge and the temperature at state 1 in the fresh charge are affected by dilution by residual gases. Assume at first that both charge and the gases have the same
γ and c_v. Determine how to get a better estimate, if the ambient temperature is 290 K, and the mass fraction of fuel in the fresh charge is 0.05.

Solution: ICE-1.ipynb

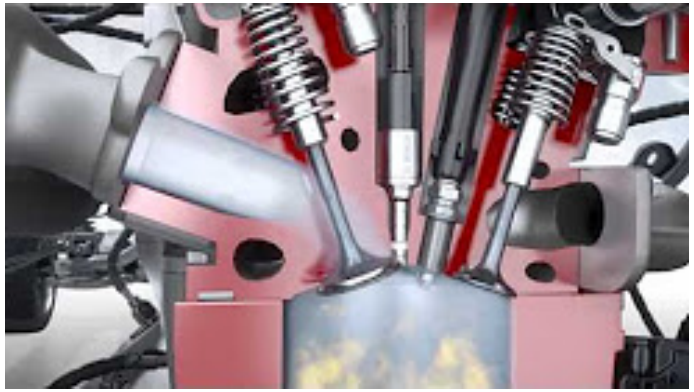
SI: Intake and fuel injection systems

Port fuel injection (PFI)



Most common system for 4-stroke
Spray onto heated valve
Good quality premixing
Optimised injection timing for low emissions, low cycle-to-cycle variability

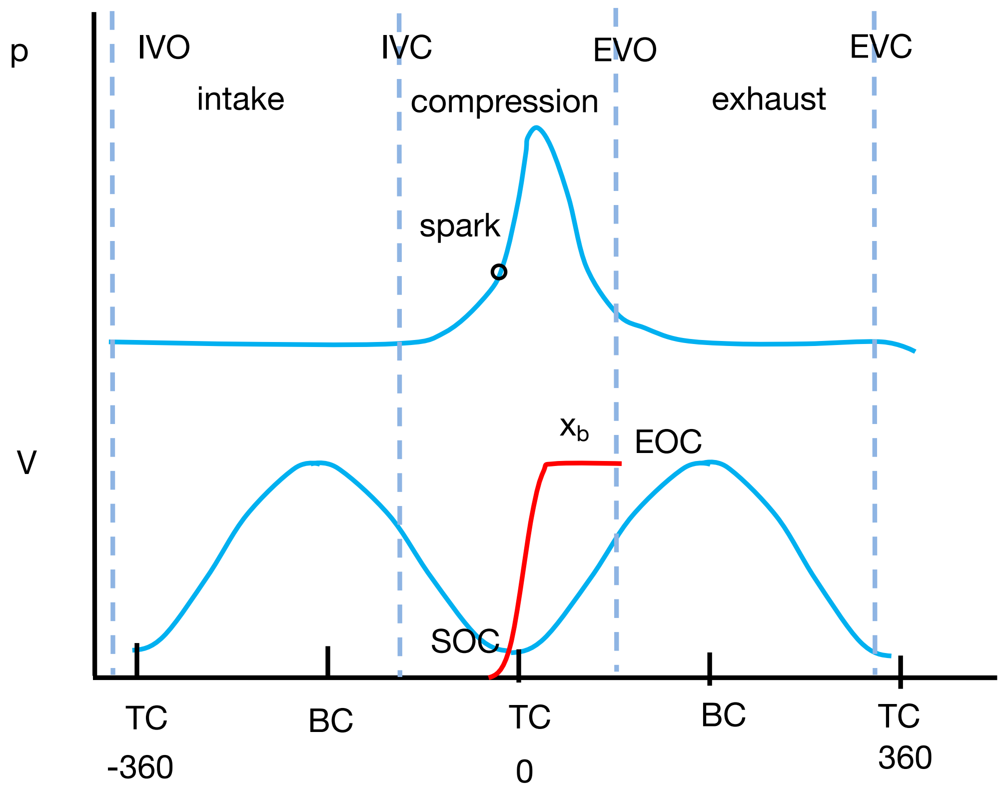
Direct injection (Gasoline DI)



Used for high power, high performance engines
Early injection: premixing, cooling leads to higher density and power
Higher PM, HC (sometimes NOx) emissions than PFI (unvaporized pockets of fuel)
Often used with PFI/GDI (more $$)

Images: Bosch Mobility Solutions

Combustion in SI engines: overview



Mixture properties

	Reactant	Products	Notes
Species	air + fuel + residual	$CO_2 + H_2O + N_2$	residual composition = products
Typical composition	0.80 + 0.05 + 0.15	0.14 + 0.14 + 0.72	1-2% CO, hydrocarbons
λ (-)	1.0	1.0	Occasional lean/rich excursions during transients
γ (-)	1.35	1.30	Smaller molecules have higher γ
c_v (kJ/kg K)	0.8-1.5	1.0 - 2.0	Temperature dependence leads to variation

Product composition not far from chemical equilibrium (except CO, NO)
Dissociation, high c_v lowers adiabatic T
Lower γ for products: efficiency higher for leaner mixtures
(but lean operation a problem for NOx removal)

Compression limits: knock

Autoignition:

- Hydrocarbon chain reaction during compressive heating: leads to high pressure oscillations (bad for noise, efficiency)
- Ignition time characteristic of the fuels: controlled by complex chain
- Octane number (ON): rating correlated with resistance to autoignition (heptane = 0; iso-octane = 100)
- Limits compression ratio to ~12-13 in SI engines for ON~90
- Advancing the spark moves the peak of combustion closer to TDC (desirable) but can lead to knock
- “Superknock” can also appear as pre-ignition in supercharged engines (typically due to lube oil)

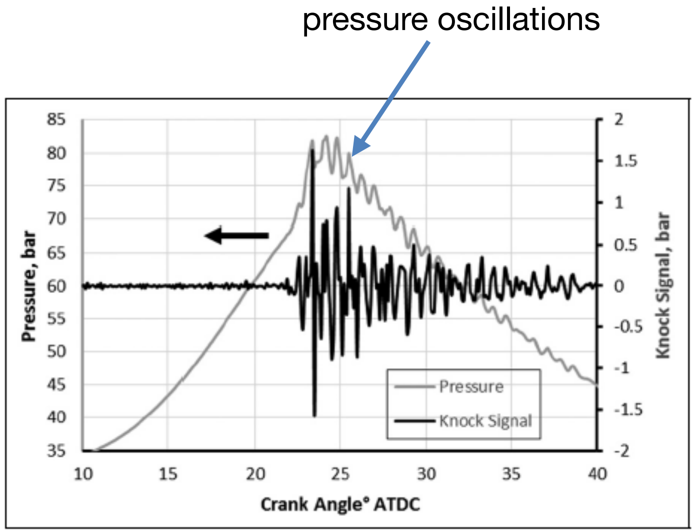


Figure 1. An example of a pressure signal and a knock signal for a single knocking cycle.

G.T. Kalghatgi , International J of Engine Research 2018, Vol. 19(1) 7–20
DOI: 10.1177/1468087417736430

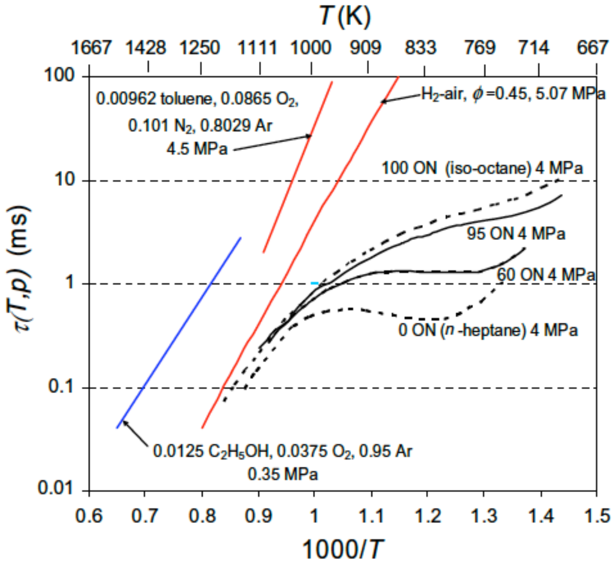


Fig. 2. Ignition delay times of various fuels with air at $\phi = 1.0$, unless otherwise stated. See Table 1 for source references.

D. Bradley, G.T. Kalghatgi / Combustion and Flame 156 (2009) 2307–2318

Combustion, flame propagation and phasing

- Spark propagates from the plug, transitioning from a laminar to a turbulent flame
- Propagation speed proportional to laminar flame speed of mixture, and function of local turbulent conditions

OH laser sheet visualisation

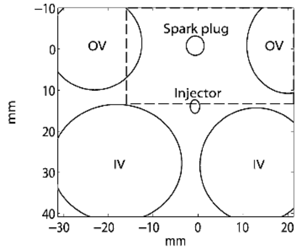


Fig. 3 Field of view within the cylinder. Intake valves (IV) are in the lower part and exhaust gas valves (OV) in the upper. The injector is mounted centrally and the spark plug is located between the exhaust valves. The *dashed rectangle* shows the enlarged portion for Figs. 2, 4–6

0% EGR

slice across a flame

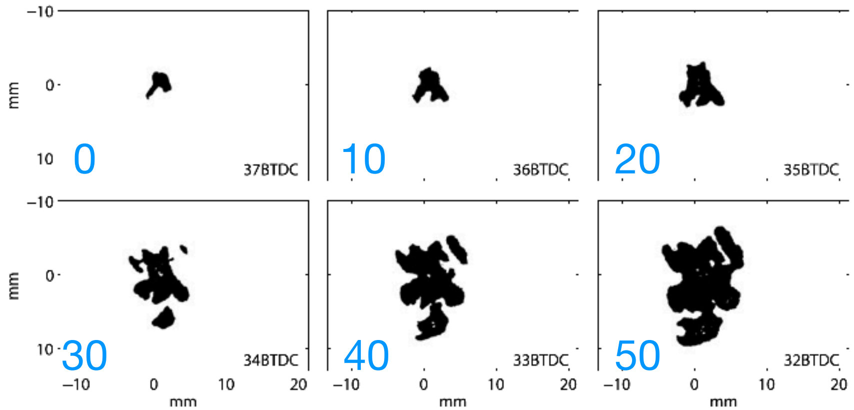


Fig. 4 Flame kernel development during an individual engine cycle (1000 rpm). The kernel growth is derived from transient OH radical distributions recorded with a repetition rate of 6 kHz. Times are denoted in ° CA before top dead center (BTDC). In this case no exhaust gas was recirculated (0% EGR)

Mueller et al., Appl Phys B (2010) 100: 447–452

20% EGR: slower rate of growth

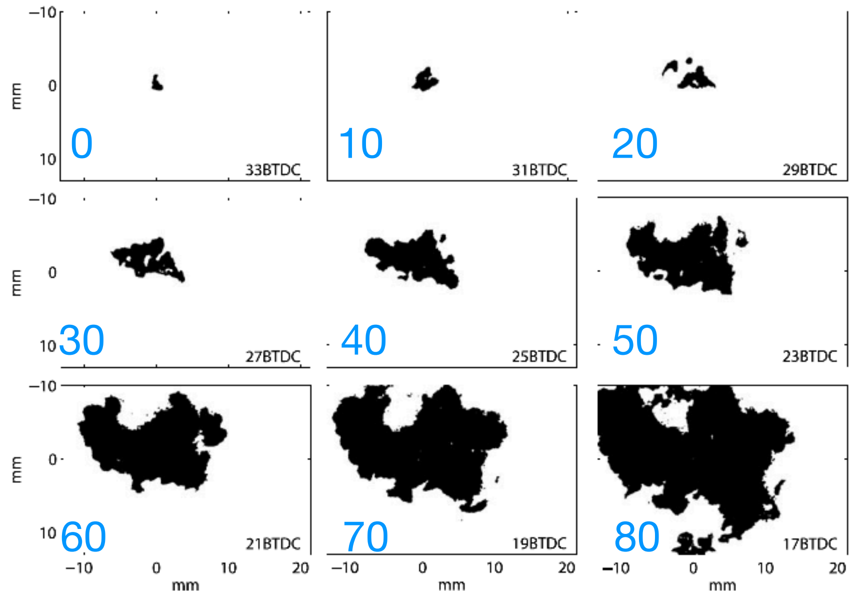
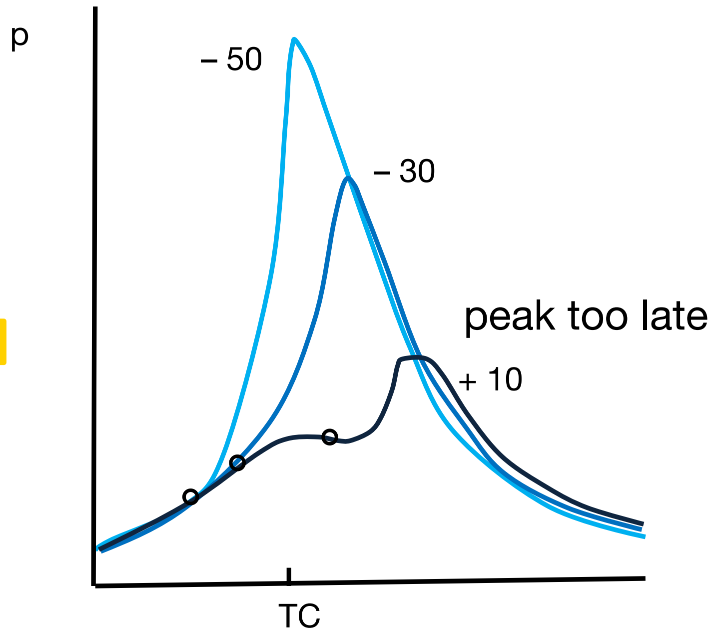
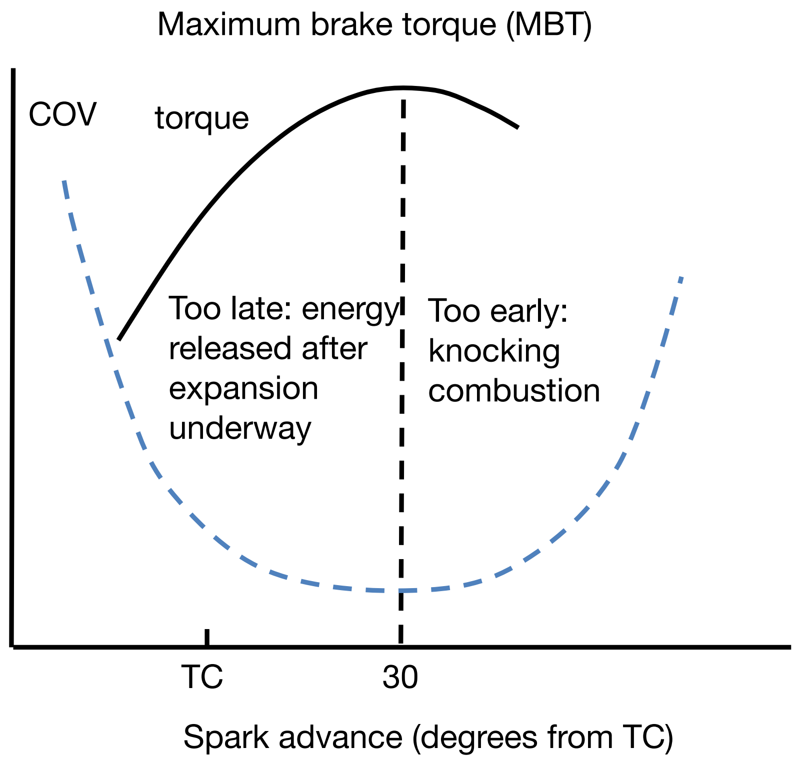


Fig. 6 Individual engine cycle with 20% EGR. Obviously flame growth is retarded significantly compared to the cases with 0 and 10% EGR

Combustion duration, phasing and cycle stability



early spark: pressure increases and can lead to knock. sometimes used to produce high exhaust temperatures and warm up catalyst.
late spark: not enough work extracted. Sometimes used at high power to avoid overheating components.



COV = coefficient of variability (rms of imep)

Quiz 1

https://www.vle.cam.ac.uk/mod/quiz/view.php?id=11754182

Notes on SI flame propagation

- Ignition and propagation: Combustion in SI engines is initiated when the spark initiates a flame kernel. The flame propagates through the premixed charge, engulfing the unburned mixture over a few milliseconds, or a few tens of crank angle degrees.

- Spark timing: usually designed to provide the maximum efficiency, i.e. the maximum torque.
  - If the spark goes off too early, combustion takes place while the piston is compressing the mixture, and the compression work increases against the expanding flame.
  - If the spark goes off too late, not enough work is transferred from the gas to the piston. An optimum (Maximum Brake Torque) is usually found by adjusting the spark 10-30 degrees before top centre, to allow enough time for a flame kernel to develop and full combustion to be completed shortly (10-20 degrees) after TC.

- The combustion duration in SI engines is relatively independent of speed: the increased turbulence compensates for the shorter available time. In general, combustion is completed within 40-60 degrees regardless of speed.

- Combustion efficiency (i.e. the total fraction of fuel conversion) is within 98-99 percent at most conditions. Only 1-2 % of the fuel remains unburned. However, this is not necessarily reflected in the cycle, as significant amount of the available energy (~ 30%) is released exhaust enthalpy or lost via heat transfer.

Notes on SI flame propagation

- Effect of speed and load: A goal of sound engine design is to enable robust combustion both at low loads (when the engine is idling) as well as at high loads (high imep).
  - At low speeds and loads (i.e. at low intake pressures), when the mass flow rates and velocities in the engine are low, and the temperatures at the point of spark are also low, the challenge is to keep the mixture burning fast enough.
  - At high speeds and loads, the velocities are high, and so is the burning rate – the limiting factor becomes the fact that the high pressures and temperatures in the unburned portion of the mixture can lead to the onset of autoignition, or knock.
  - The flow motion across the inlet valves is often designed to provide for enhanced swirl (the bulk motion along the cylindrical axis) or tumble (across the cylindrical access) at low speeds and loads to help the flame propagate. Many modern engine developments, such as two-position valve cams, or fully variable inlet valve timing, can help tune the flow in and out of the cylinders to maximize power and fuel economy, while minimising emissions.

- Cycle to cycle variation is a measure of the stability of combustion. Too early spark timing can lead to knock and unsteady combustion; too late means that the flame reaches maximum rate of heat release after expansion is already under way, and the flame can be stretched and extinguish. Mixtures that are dilute with high residual gas or exhaust gas recirculation show higher cycle to cycle variations, which are created by misfires, or poor flame propagation.

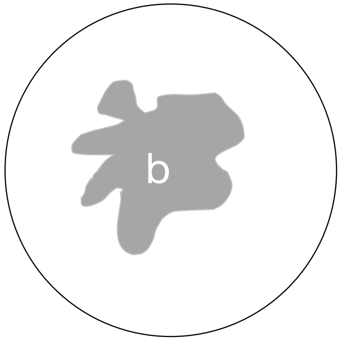
Heat release rate analysis: energy balance models

$$dU = dQ \; - dW$$

$$dW = p\, dV$$

$$dU = m\, c_v\, dT$$

$$dT = \frac{d\,(pV)}{mR}$$

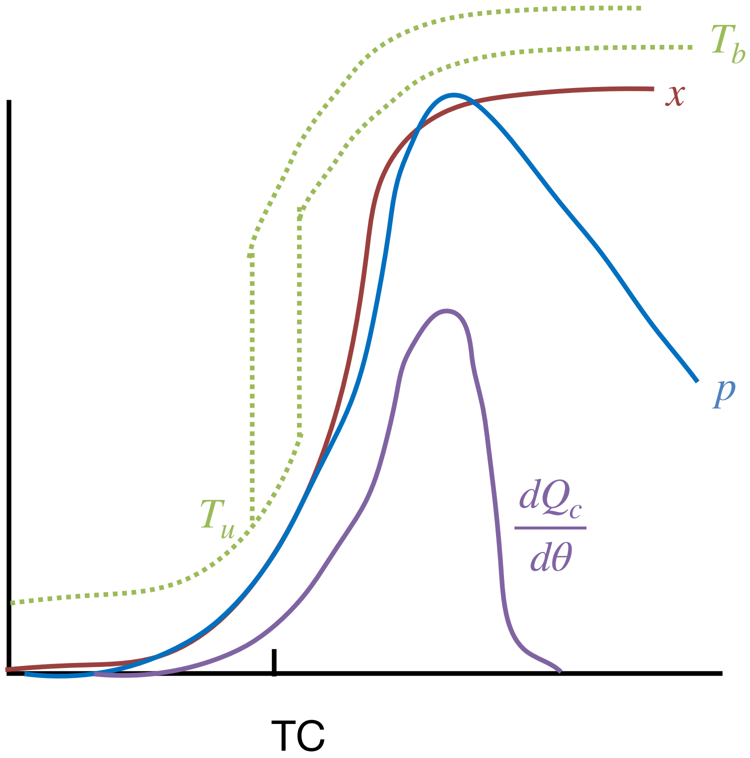


$$dQ = dQ_c - dQ_l = p\, dV + \frac{c_v V}{R} dp + \frac{c_v\, p}{R} dV$$

$$dQ_c = \frac{\gamma}{\gamma - 1} p\, dV + \frac{1}{\gamma - 1} V\, dp + dQ_l$$

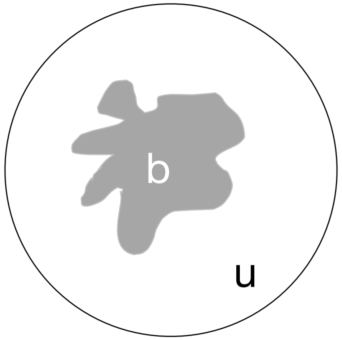
$$\boxed{\frac{dQ_c}{d\theta} = m_f Q_f \frac{dx_b}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + \frac{dQ_l}{d\theta}}$$

heat release rate | measured | measured | estimated



Heat release models (0D)

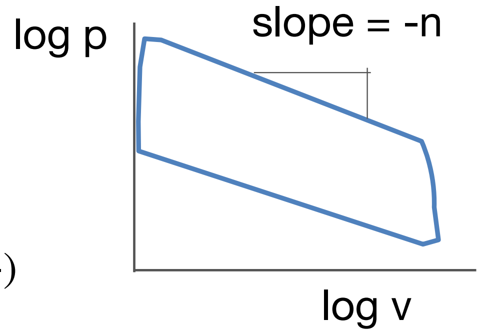
Heat release analysis: polytropic model of pressure and fraction burned



Burned gas fraction $$x = \frac{m_b}{m} = \frac{V_b}{m v_b}$$

Connect the initial and final pressures for $x = 0$ and $x = 1$, assuming isentropic or fitted polytropic compression n (which avoids assumptions about heat loss, by taking an average):

$$v_u = v_{u,0}\left(\frac{p_0}{p}\right)^{1/n} \qquad v_b = v_{b,f}\left(\frac{p_f}{p}\right)^{1/n}$$



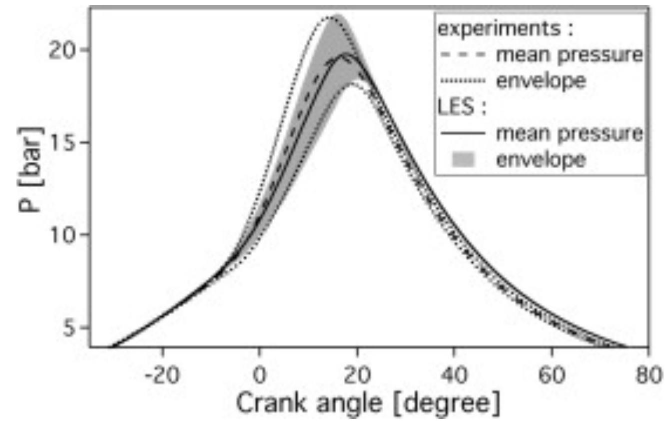
Using $V_u + V_b = V$

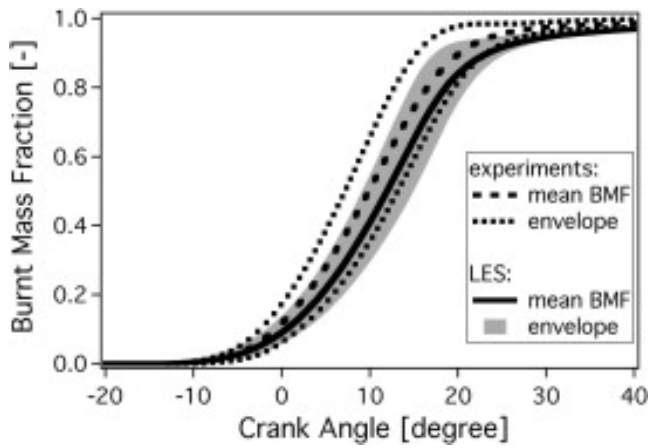
$$x = \frac{p^{1/n}v - p_0^{1/n}v_{u,0}}{p_f^{1/n}v_f - p_0^{1/n}v_{u,0}} \qquad \frac{dx}{d\theta} = \frac{1}{p_f^{1/n}v_f - p_0^{1/n}v_{u,0}}\left(vp^{\frac{1}{n}-1}\frac{dp}{d\theta} - p^{1/n}\frac{dv}{d\theta}\right)$$

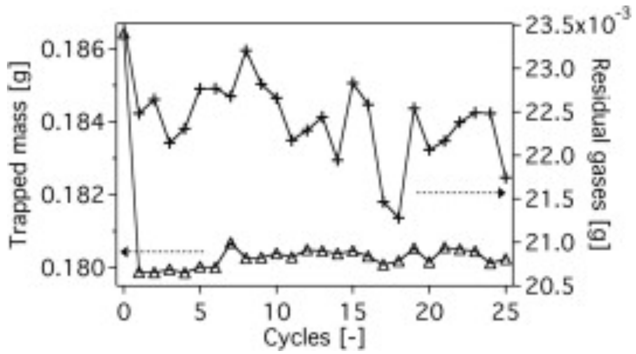
For a known v(θ), p(θ), the rate of combustion can be calculated, given initial (0) and final (f) burn conditions

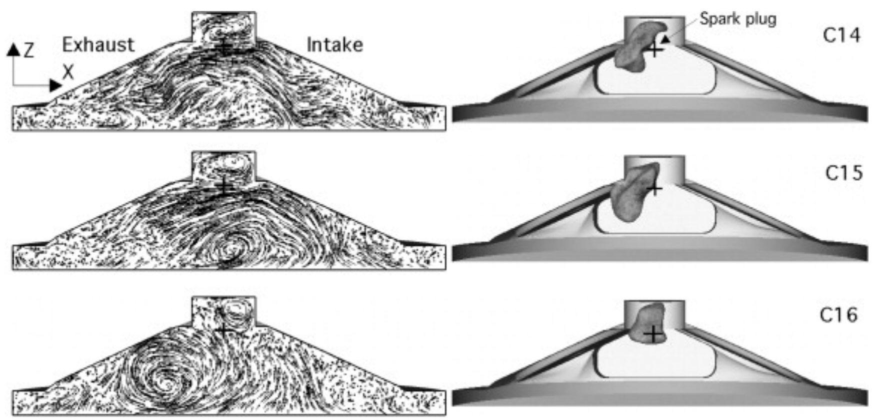
More sophisticated models use corrections for heat loss, variable properties, etc.

Cycle to cycle variation: experiments and simulations



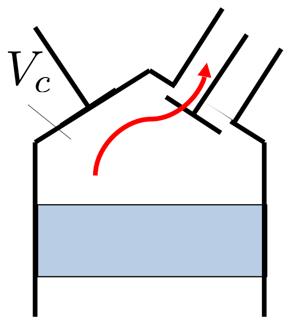






Enaux et al., LES study of cycle-to-cycle variations in a spark ignition engine
[Proceedings of the Combustion Institute](#)
[Volume 33, Issue 2](#), 2011, 3115–3122
http://dx.doi.org/10.1016/j.proci.2010.07.038

Effect of speed and throttling on overall mep



Heat losses: $Q_l \propto \Delta T \, \Delta p \, N^n \quad n \approx 0.3 - 0.8$

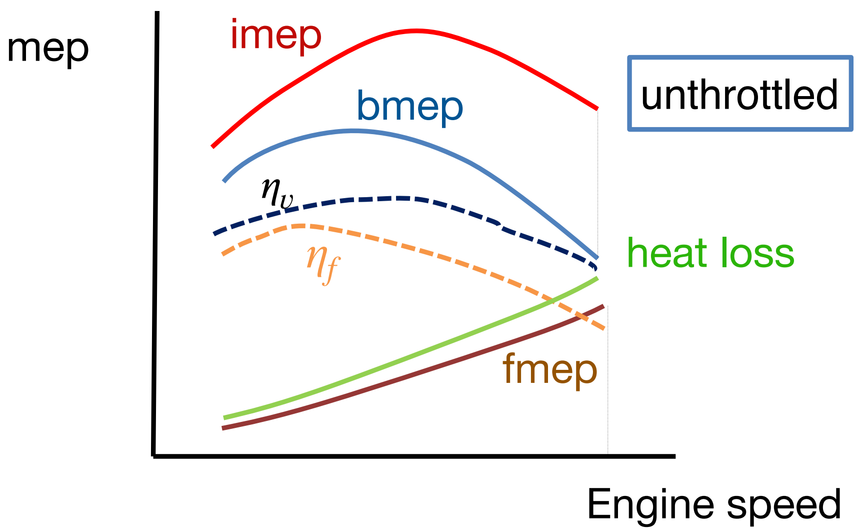
heat losses increase with higher Re, exhaust enthalpy loss

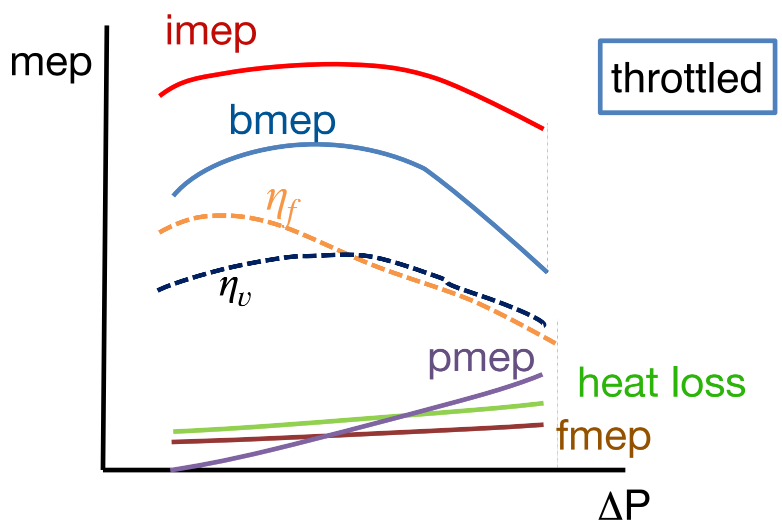
Friction: $P_f \propto N^2$

Friction power increases as viscous losses

Pumping: $P_p \propto \Delta p \, N$

Pumping losses increase with the mass flow rate x pressure rise





Indicated and brake torque-speed curves

IMEP

$$\text{imep} = \frac{P\ n_R}{NV_d} = \eta_f \eta_v \frac{\rho_{a,i} V_d q_f}{AFR}$$

Indicated torque

$$T = \frac{\text{imep} V_d}{2\pi n_R} = \eta_f \eta_v \frac{\rho_{a,i} V_d q_f}{2\pi AFR n_R}$$

Brake torque

$$T_b = T_i \eta_m$$

Power

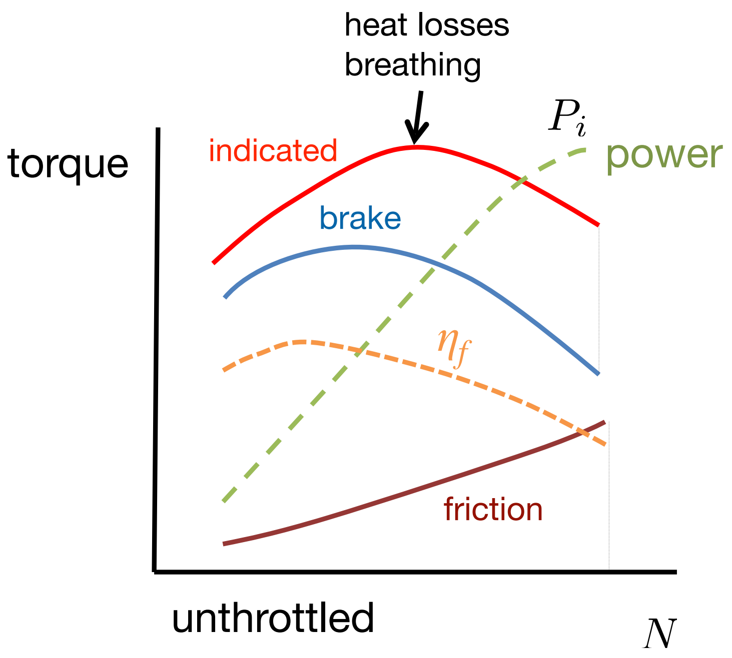
$$P = \frac{\eta_f \dot{m}_a q_f}{AFR} = \eta_f \eta_v \frac{\rho_{a,i} V_d q_f}{AFR\ n_R} N$$

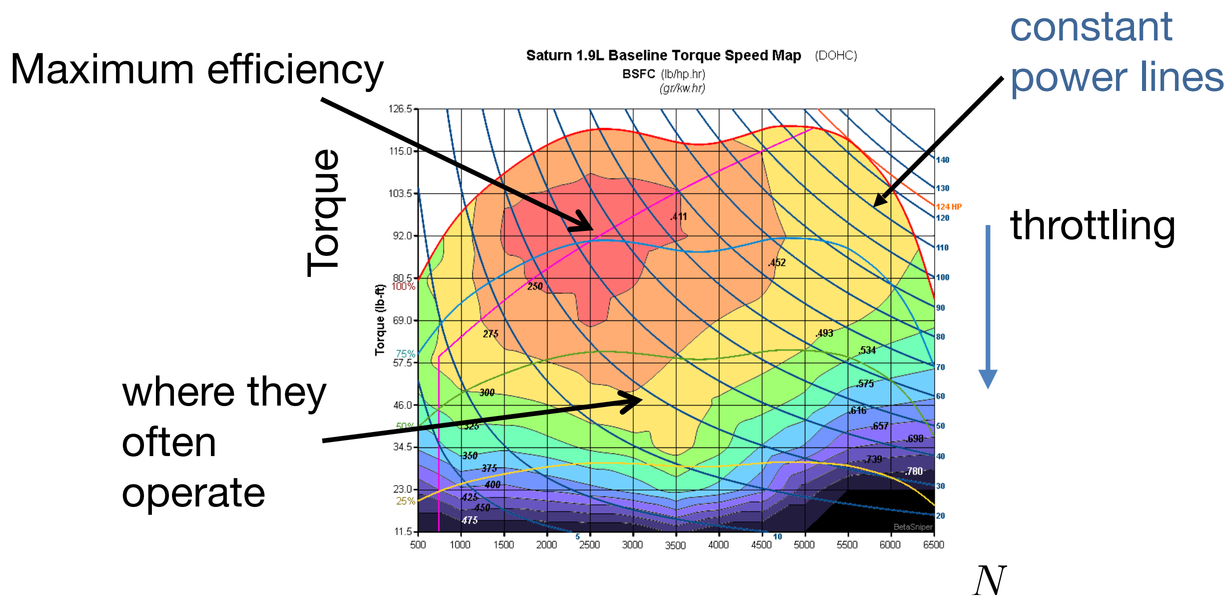
Indicated torque/imep peak at a speed where the product of **volumetric and indicated efficiency peak** at full throttle.

Brake torque is net of frictional and pumping losses, so peaks at slightly lower speeds

Power increases approximately linearly with speed

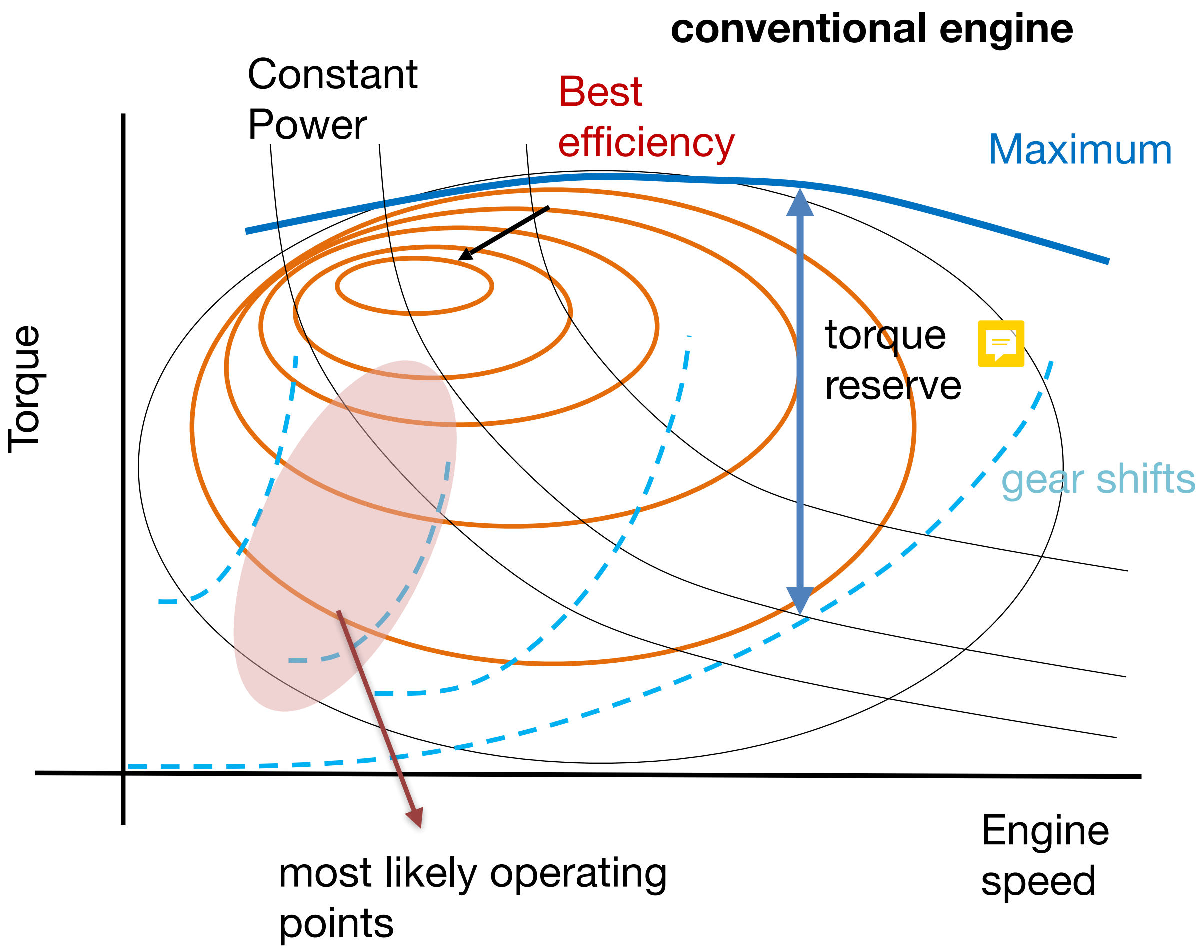
Performance maps





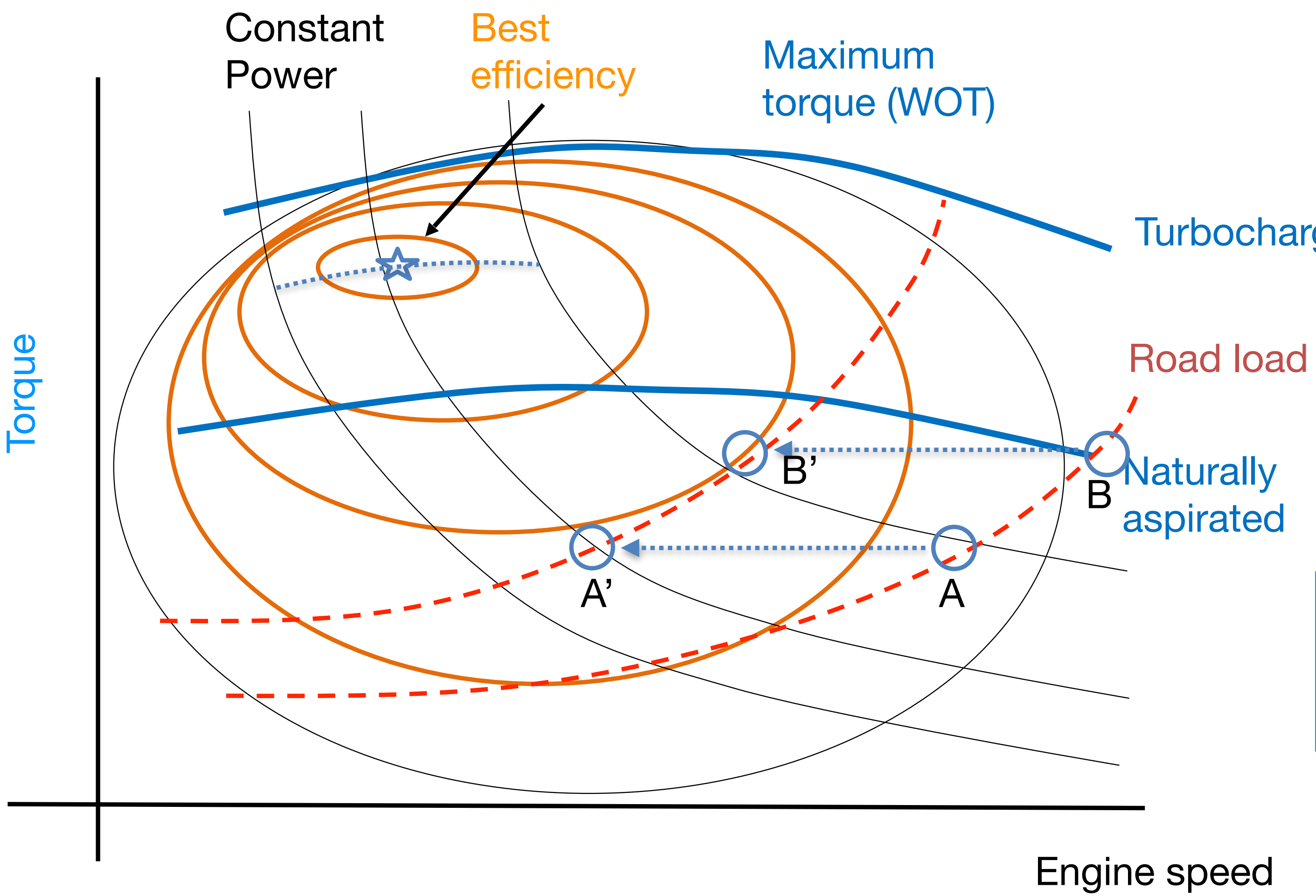
Dynamic operation of IC engines leads to sub-optimal part load operation
Optimisation can come from downsizing/supercharging (right-torqueing) and active control of exhaust gas recirculation.
Even more potential for multiple injection strategy/stratified compression ignition
Emission control severely limits performance (limits to T, AFR)

Engine rightsizing



- The point of best efficiency/lowest fuel consumption is not necessary where the engine spends most of its operating time: at low loads in traffic or cruising down the highway, one does not need much power or torque.
- A typical road load is lower than the maximum the engine can provide. A torque reserve exists to allow e.g. overtaking at high speeds for a short period of time. The engine design characteristics are such that it is impossible to have the best fuel efficiency at all operating points.
- Engines are designed for the maximum conceivable torque (e.g. overtaking quickly) rather than best efficiency. This creates a problem. Part of that problem can be overcome by downsizing the base engine, then using a turbocharger when necessary.
- There are different ways of using a smaller displacement engine with a small baseline torque, but producing methods that can reach higher torque during certain operating conditions (hybrid, continuous variable transmission, turbocharging) when additional torque is necessary, without compromising efficiency.
- All of these solutions have a direct cost and a direct weight penalty, which must be weighted against its advantages.

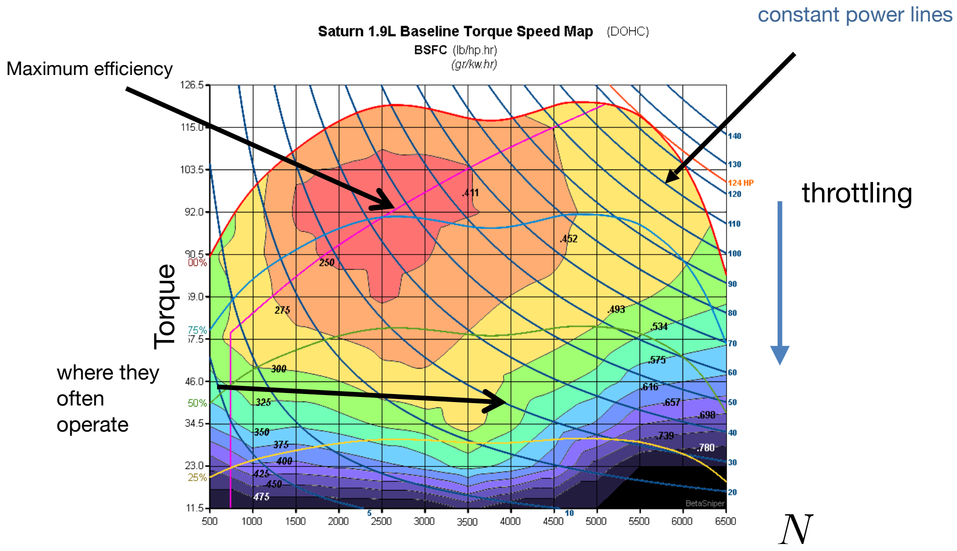
Downsizing and turbocharging

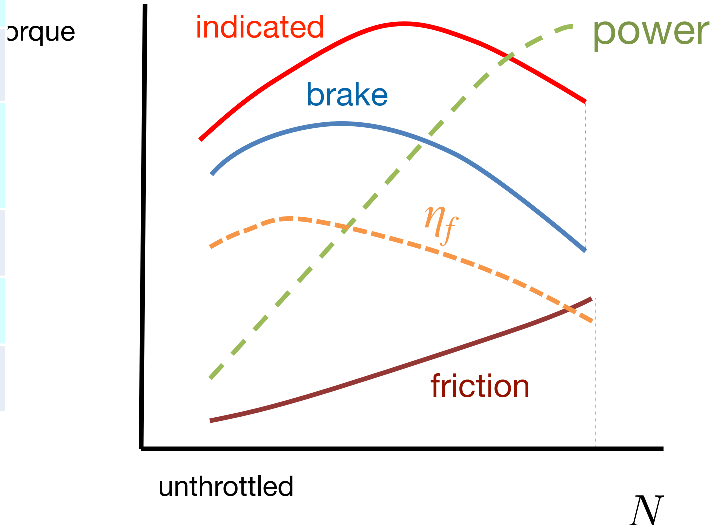


- The efficiency curves are moved by adding turbocharging; the operating point can be shifted to a higher overall efficiency
- The penalty is the extra cost and weight of the turbocharger

Summary: SI engine operation

	SI
Compression ratio	6-11 (limited by onset of knock and NOx production)
Equivalence ratio φ	1.0 (to allow operation of 3-way catalytic converter)
Throttling	Yes (associated pumping losses)
Efficiency	Max. 35%
Bore (mm)	50-450 (70-100 for light duty)
Maximum speed (rpm)	4500-7500 (limited by heat transfer and stresses)
Maximum bmep (bar)	7-12
Weight/power ratio (kg/kW)	2-6
Power densities (kW/litre)	20-60





Quiz 2

https://www.vle.cam.ac.uk/mod/quiz/view.php?id=11781912