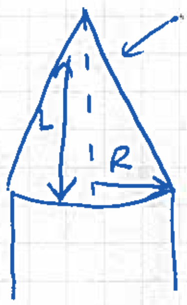
① (a) NO : produced through Zel'dovich mechanism (operational at high T) described by the reactions

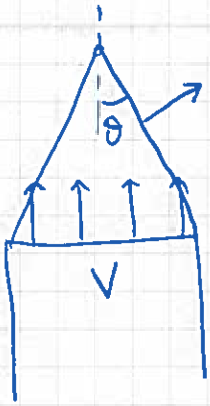
$$N_2 + O \rightarrow NO + N$$
$$N + O_2 \rightarrow NO + O$$
$$O_2 = 2O$$

It is also produced throught the "prompt" mechanism in the reaction zone through HCN reactions. To reduce NO, we need to burn lean (hence low T) or use other ways to reduce flame temperature, for example Exhaust Gas Recirculation.

CO: if T becomes too low for combustion to complete, CO may be emitted from regions of high fluid mechanical strain or close to walls. This is avoided by ensuring lean combustion, flames away from walls, not too low residence times, and good mixing in non-premixed systems.

(b)





$V \sin\theta = S_L$

(this is the key flame stabilisation idea)

Hence $L = \frac{R}{\tan\theta}$, with $\theta = \arcsin\left(\frac{S_L}{V_0(1 + A\cos\omega t)}\right)$

Maximum L when V is a maximum

Minimum L when V is a minimum

Hence: for V max, $V = 0.9$ m/s $\Rightarrow L = 20.2$ mm

V min, $V = 0.7$ m/s $\Rightarrow L = 14.4$ mm

(c) If in the above problem V becomes zero, the flame will propagate into the pipe with a planar shape. If R is too small, wall effects cannot be neglected anymore and we have heat losses (hence chemical reaction rate drops) and even radical losses (recombination of radicals at the low T of the wall). This means quenching at the wall. If this quenching region becomes large relative to R, the whole flame will fail to propagate into the pipe and then the pipe diameter is called "quenching distance". It is of the order of laminar flame thickness.

(2) (a) In the context of 1-step chemistry, constant c_p & P, $\dot{w}_{fu} = -A' \rho^2 Y_{fu} Y_{ox} \exp\left(-\frac{E}{R^0 T}\right)$. Writing $E/R^0 = T_{act}$, A' a constant absorbing all molecular masses & pre-exponential factors.

From 1st Law, temperature rise <u>must</u> be accompanied by reactant consumption, hence

$$Q\left(Y_{fu,0} - Y_{fu}\right) = c_p (T - T_0)$$

$$\Rightarrow Y_{fu} = Y_{fu,0} - c_p (T-T_0)/Q \qquad Q\text{: Lower calorific value}$$

For oxidiser, $Y_{ox} = Y_{ox,0} - S \cdot c_p (T - T_0)/Q$

Hence fuel mass fraction and temp rise are connected. Assuming negligible reactant consumption until autoignition implies very small T rise.

The starting point of the derivation is 1st Law,

i.e. $\rho c_p \frac{dT}{dt} = \underbrace{Q A' \rho^2 Y_{fu,0} Y_{ox,0}}_{B' \,\approx\, \text{constant throughout induction time}} \exp\left(-\frac{T_{act}}{T}\right)$

$$\Rightarrow \frac{dT}{dt} = B \exp\left(-\frac{T_{act}}{T}\right)$$

Writing $T = T_0 + \Delta T$, $\exp\left(-\frac{T_{act}}{T}\right) = \exp\left[-\frac{T_{act}}{T}\left(1 - \frac{\Delta T}{T_0} + \ldots\right)\right]$

$$\approx \exp\left(-\frac{T_{act}}{T}\right) \exp\left[\frac{(T-T_0) T_{act}}{T_0^2}\right]$$

$$\Rightarrow \frac{dT}{dt} = B \exp\left(-\frac{T_{act}}{T}\right) \exp\left[\frac{T_{act}(T-T_0)}{T_0^2}\right]$$

subject to $T = T_0$ at $t = 0$. The solution is

$$t = \frac{1}{B} \frac{T_0^2}{T_{act}} \exp\left(\frac{T_{act}}{T_0}\right)\left[1 - \exp\left(-\frac{T_{act}(T-T_0)}{T_0^2}\right)\right]$$

which gives $t = \tau_{ign}$ when $T \to \infty$.

$$\Rightarrow \tau_{ign} = (A \, \rho_0 \, Y_{fu,0} \, Y_{ox,0})^{-1} \frac{T_0^2}{T_{act}} \exp\left(\frac{T_{act}}{T}\right)$$

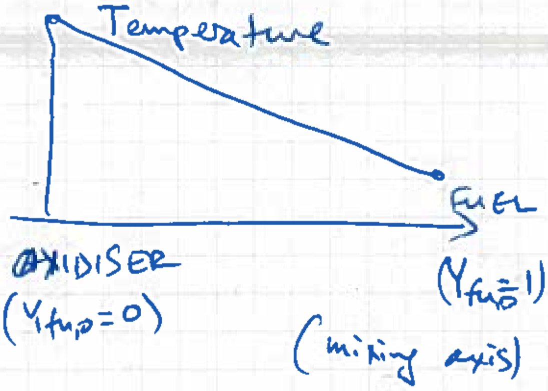
A absorbing all constants, but we leave ρ_0 & $Y_{fu,0}$ & $Y_{ox,0}$ visible.

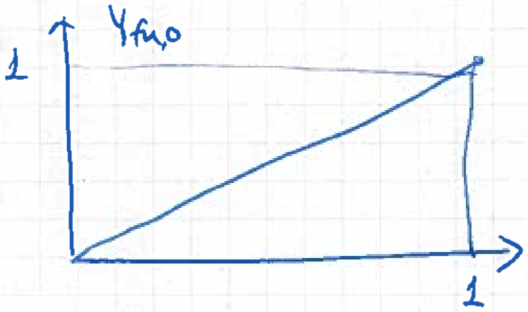
(b) If there are small heat losses, $\frac{dT}{dt}$ will rise less quickly $\Rightarrow$ τ_{ign} will be increased. If heat losses are excessive, $\frac{dT}{dt}$ rises slowly but the system does not reach high T: it stabilises at at T just a little higher than T_0, so that the heat release balances heat loss.

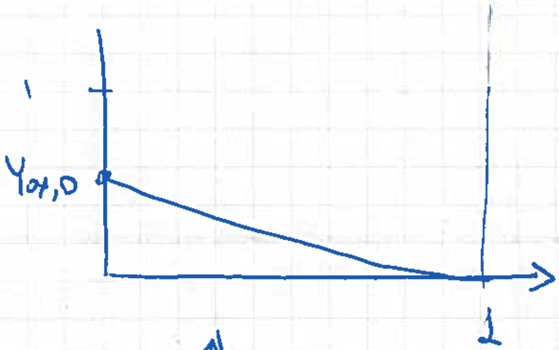
(c) if $\tau_{ign} = (A \rho_0 Y_{fu,o} Y_{ox,o})^{-1} \dfrac{T_0^{\,2}}{T_{act}} \exp\left(\dfrac{T_{act}}{T_0}\right)$,

for the inhomogeneous problem in this part

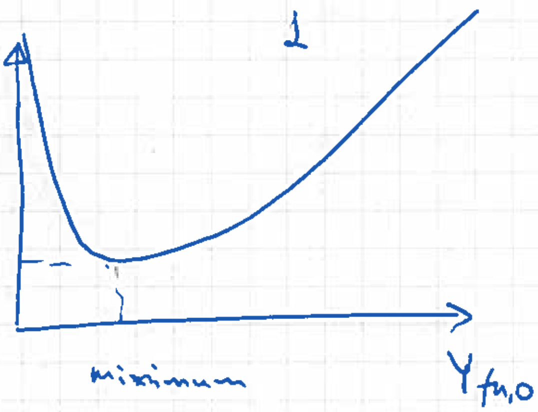
τ_{ign} will depend on Y_{fu}. Schematically,





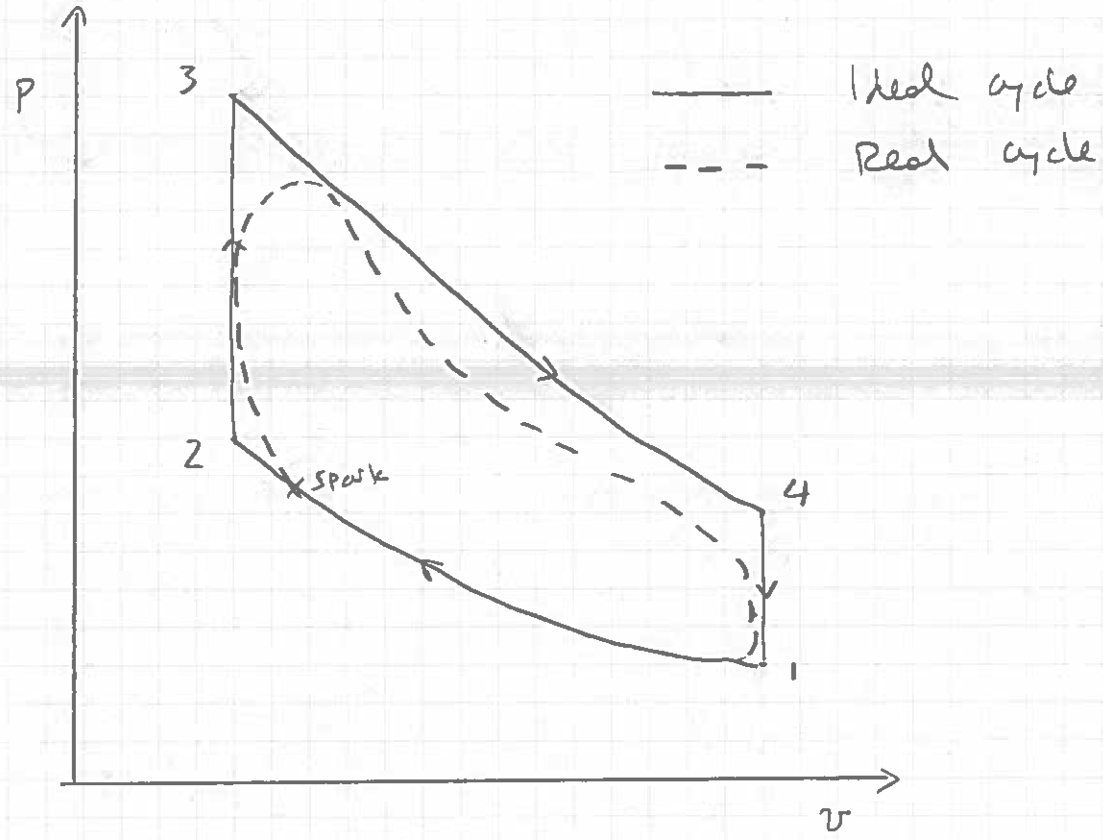


$\Rightarrow$ τ_{ign}



at some intermediate value, leaning towards high T side

Q3 (a)

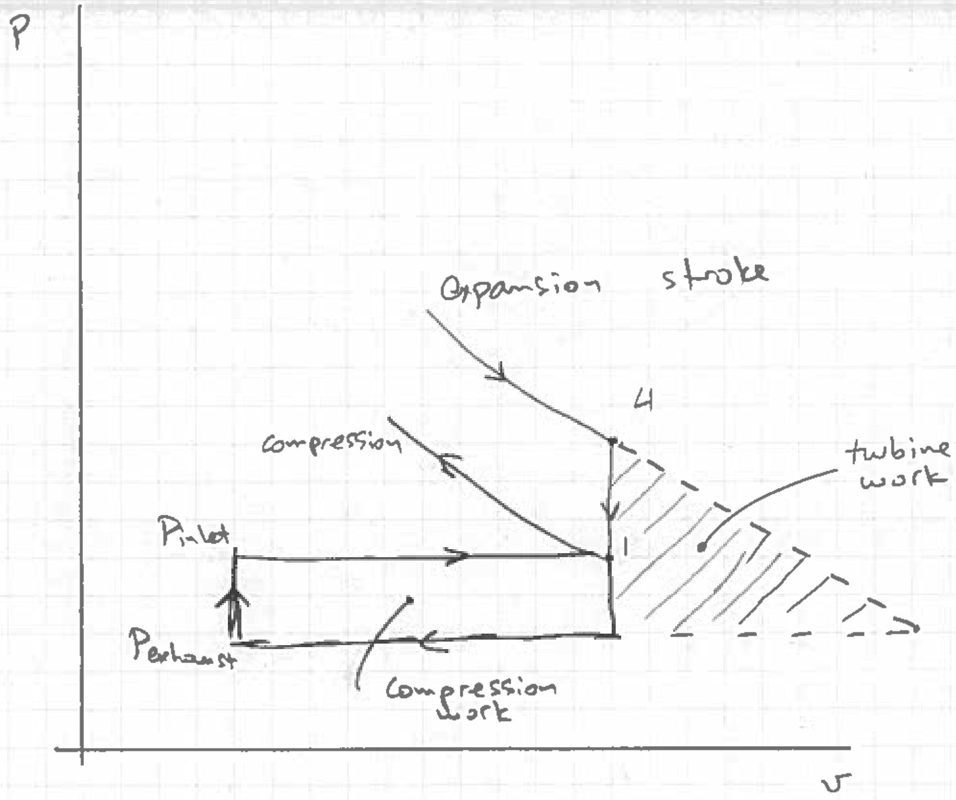


The differences are due to (i) heat losses (hence compression & expansion are not exactly isentropic, with the difference much larger for the expansion stroke because the gases in the cylinder are hot); (ii) finite burn rate (flame needs time to travel across cylinder); (iii) some mass loss (blow-by); (iv) pressure losses associated with gas exchange. In practice, cycle efficiency η is smaller than η_{Otto}, with $\eta/\eta_{Otto} \sim 0.8$-$0.9$ across many engines.

(b) With turbocharging, enthalpy from the exhaust gases is used in a turbine to drive a compressor that increases the density of the air into the cylinder. This increases the thermodynamic efficiency of the engine (waste heat is used) and a smaller engine can be constructed for the same power.

3 (b) cont

Both spark-ignition & compression-ignition engines can be used with turbocharging. For SI, too much boost can cause knock, while for CI too high pressure may have mechanical problems.



(c) If same amount of fuel is injected from many small nozzles as opposed to a single larger hole, the diesel spray will have smaller droplet sizes. This means quicker evaporation & mixing with air before autoignition. This, in turn, means less proportion of the fuel close to stoichiometric, therefore smaller NO_x & smoke. The claim is correct.

4 (a) Volumetric efficiency is defined as the ratio of the rate of volumetric flow into the cylinder per unit cycle, to the displaced volume per unit time during intake:

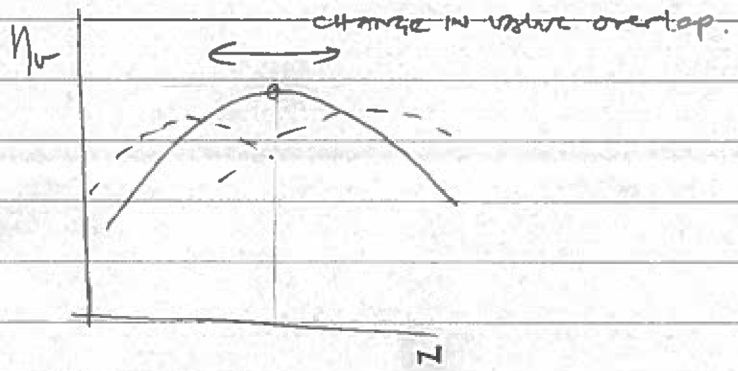
$$\eta_v = \frac{\dot{m}_a/\rho_{a,i}}{V_d N/n_R}$$

The total power produced is proportional to the volumetric efficiency times thermal efficiency:

$$P = \eta_v \eta_f Q_{LHV} \dot{m}_f$$

So that the total power increases with increasing η_v.

The volumetric efficiency is a function of speed (only weakly of load), and is typically between 80-90% for SI engines @ full load.



(b) RESIDUAL GAS:

REMAINING PRODUCT GAS INSIDE CYLINDER AFTER EACH CYCLE,

usually expressed as a volumetric or mass percentage of the total mass in the cylinder.

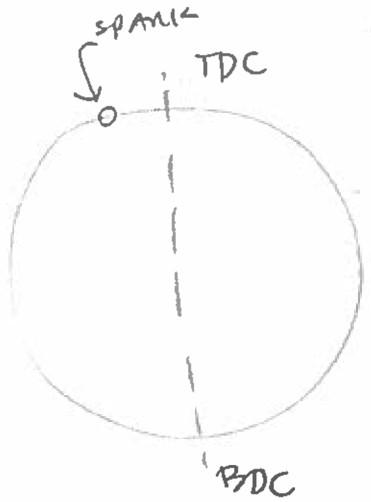
Larger amounts of residual gas reduce the total power, but can be helpful in lowering NO emissions.

EGR / EXHAUST GAS RECIRCULATION refers to a fraction of the total exhaust stream that is purposely recirculated into the intake manifold in order to reduce the peak burned gas temperatures.

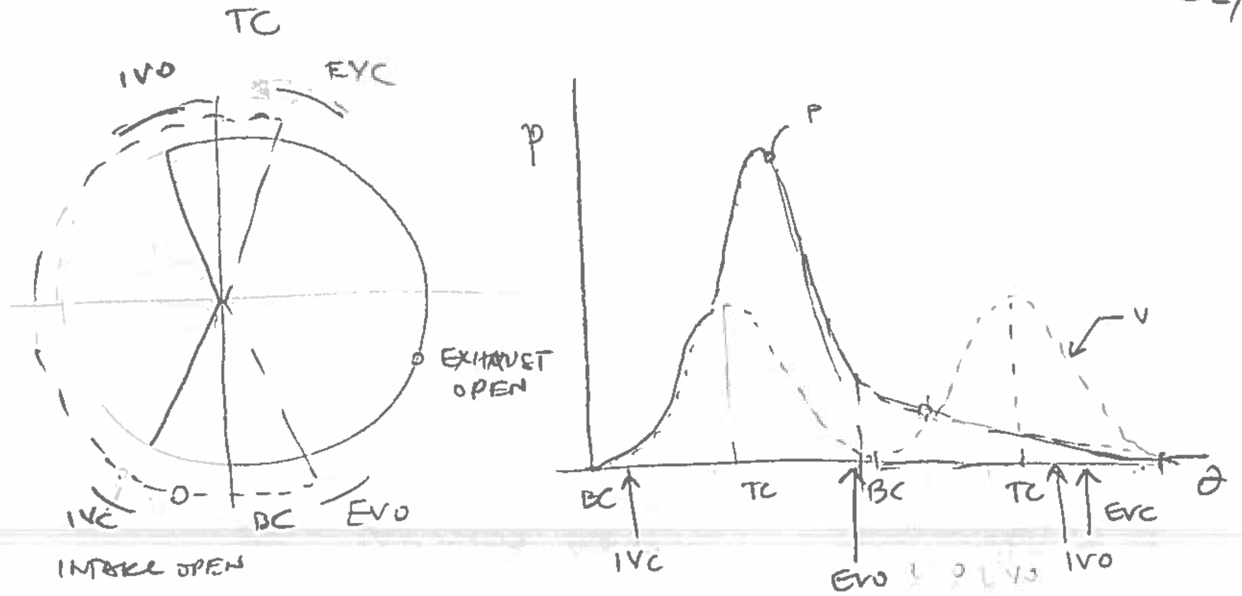
Residual gas % are ~5-10% depending on compression ratio and operating condition.

EGR % can be up to 30% before combustion irregularities appear owing to mixture dilution.

(c)



(c)



IVO — INTAKE VALVE OPEN

THIS TYPICALLY IS SET TO JUST BEFORE TOP CENTER — TYPICALLY THE VALVES OPEN TO LET FRESH MIXTURE IN. THE PRESSURE IN THE CYLINDER IS SLIGHTLY HIGHER THAN THE MANIFOLD, AND SOME OF THE BURNED GASES ENTER THE INTAKE, HELPING VAPORIZE FUEL DURING SI ENGINE INJECTION. THE DOWNWARD MOTION OF THE PISTON ENTRAINS THE AIR/FUEL MIXTURE FROM THE MANIFOLD ACROSS THE OPEN VALVE.

THE MOTION OF THE CHARGE DURING THIS PERIOD IS IMPORTANT, AS IT SETS UP THE LEVEL OF TURBULENCE, SWIRL + TUMBLE.

THE EXTENT OVERLAP (IVO TO EVC) HAS AN INFLUENCE ON HOW WELL THE FRESH CHARGE REPLACES THE BURNED GAS. TURBOCHARGED SYSTEM CAN HAVE EFFICIENCIES HIGHER THAN 1 DEPENDING ON OPERATING CONDITION.

IVC — INTAKE VALVE CLOSES JUST BEFORE COMPRESSION STARTS. LONGER IVO-IVC REQUIRED FOR HIGHER MASS FLOW RATES.

EVC — TYPICALLY BEFORE IVC, CONTROLS THE EQUILIBRIUM OF PRESSURES DURING CHARGE INTAKE.

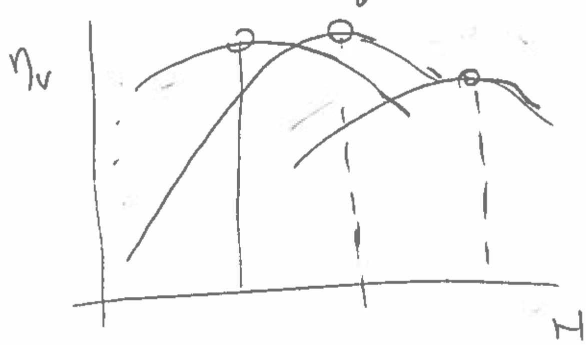
10

(c) (CONT.)

EVO : CONTROLS GROSS WORK – LATER IS BETTER FOR HIGHER EXPANSION WORK, BUT CAN PRODUCE POOR SCAVENGING.
FOR TURBOCHARGING – EARLIER EVO LEADS TO MORE ENERGY FOR TURBOCHARGER.

THE VOLUMETRIC EFFICIENCY IS AFFECTED BY TIMING AND SPEED.

↙ OPTIMUM FOR A GIVEN VALVE TIMING



OPTIMUM TIMING MOVES WITH SPEED – WIDER $\Delta\theta_{IVC-IVO}$ NEEDED FOR HIGHER MASS FLOW RATES.

(d) COMPRESSION-IGNITION → THE SAME PRINCIPLES APPLY, BUT (i) THERE IS NO ROLE FOR EXHAUST GASES IN EVAPORATING THE INCOMING FUEL, (ii) RESIDUAL GASES ARE LESS IMPORTANT (COOLER + LESS RICH IN PRODUCT GASES); (iii) TYPICALLY DESIGNED FOR LOWER SPEEDS + TURBOCHARGED, SO GAS EXCHANGE + PRESSURES MORE DEFINED BY TURBO CHARACTERISTICS.

Q1 Pollution from natural gas combustion and laminar flame speed

Very popular question, answered very well. However, many students listed all possible pollutants, rather than focusing on the ones from natural gas. The basic stabilization idea for a laminar flame was well handled. Quenching was well understood.

Q2 Autoignition

This was the least popular question. The theoretical part in the first part of the question was done well, and the intuition concerning heat losses was very good. The third part of the question was very difficult, but many students had the correct physical intuition about this problem.

Q3 Real vs. ideal cycle, turbocharging, emissions

A qualitative question, with quite satisfactory performance. The details on the reasons why an ideal cycle was not the same as the real cycle were not answered completely.

Q4 Gas exchange processes

This proved to be the most difficult question, with few students doing very well at it. The details on how valve overlap affects gas exchange and residual gases were not well understood.