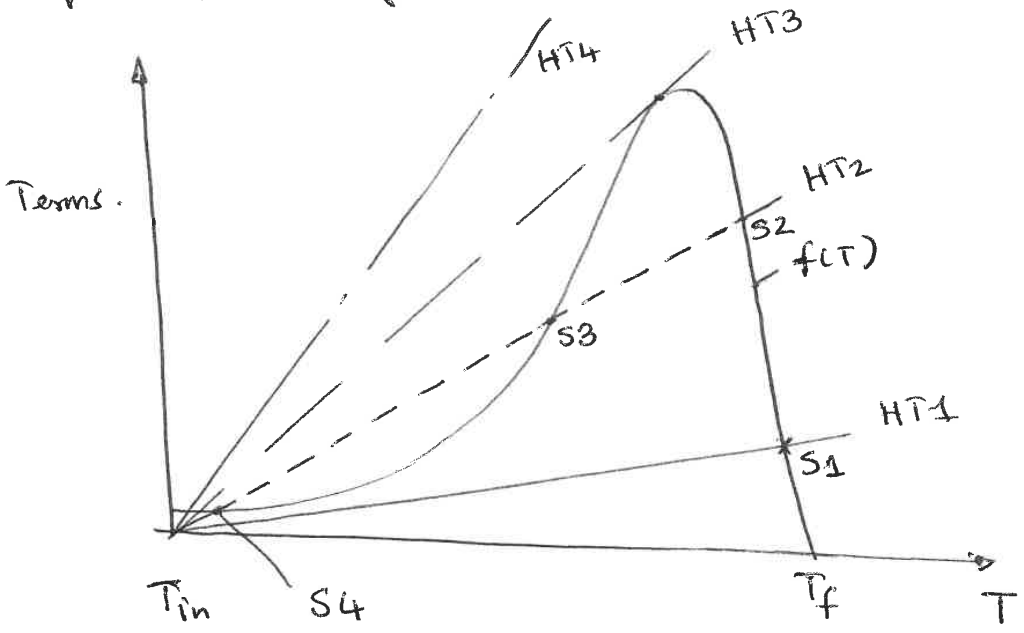
Q1. (a) For well-stirred reactor, the energy balance is

$$\frac{\dot{m}}{\rho V}(T - T_{in}) = -\dot{\omega}_f = C\, f(T) \quad \text{— (A)}$$

$\dot{m}$ – mass flow rate

ρ – density of reactant mixture

V – Volume of the reactor



$f(T)$ – heat generation by chemical reaction

H.T → L.H.S. of equation (A)

with slope $(\dot{m}/\rho V) = 1/t_{res}$.

This means that the residence time, t_{res}, is smaller for larger slope

For HT 1 → larger residence time

Stable flame @ S1

HT2 — intermediate residence time

If T is large, Stable flame @ S2

If T is low no flame @ S4

S3 - is unstable & will move to S4.

HT3 — Critical residence time

Any small increase in $\dot{m}$ or decrease in t_{res} will extinguish the flame leading to flame blow-off. This depends on reactant temperature, equivalence ratio, pressure and it is because of competing effects of heat release rate and heat loss. This can be written using a chemical time scale

$$t_{chem} > B\, t_{res}.$$

HT4 — residence time is too ~~sh~~ short for chemical reactions to occur.

[5 marks]

(b) $\dfrac{\tau_{chem}}{\tau_{res}} = C$ @ blow-off.

$$\tau_{chem} = \frac{\lambda}{\rho_0 c_p S_L^2} = \frac{\lambda_{ref}\left(T_0/T_{ref}\right)^{1/2}}{\rho_0 c_p S_{L,ref}^2 \left(T_0/T_{ref}\right)^{4}}$$

$$= \frac{\lambda_{ref}}{c_p S_{L,ref}^2 \rho_0}\left(\frac{T_{ref}}{T_0}\right)^{7/2}$$

$$\tau_{res} = \frac{L}{v} = \frac{L \rho_0 A}{\dot{m}}$$ A — C.S. area of the combustor

$$\frac{\tau_{chem}}{\tau_{res}} = \frac{\lambda_{ref}}{c_p S_{L,ref}^2 A}\frac{\dot{m}}{L\rho_0^2}\left(\frac{T_{ref}}{T_0}\right)^{7/2} \qquad \rho_0 = \frac{P}{RT_0}$$

$$= C_1 \frac{\dot{m}}{L T_0^{3/2}}$$ C_1 is constant.

blow-off occurs @ both conditions

$$\Rightarrow \frac{\dot{m}_1}{L_1 T_{0,1}^{3/2}} = \frac{\dot{m}_2}{L_2 T_{0,2}^{3/2}} \qquad \begin{pmatrix} 1 - 300K \\ 2 - 600K \end{pmatrix}$$

$$\Rightarrow \frac{\dot{m}_2}{\dot{m}_1} = \frac{L_2 T_{0,2}^{3/2}}{L_1 T_{0,1}^{3/2}} = 2\left(\frac{600}{300}\right)^{3/2} = 4\sqrt{2} = 5.66$$

$\Rightarrow$ The mass flow rate, $\dot{m}_2$, is increased by

466 %

[6]

(C) There are three mechanisms.

1. Fuel NO:

Nitrogen in the fuel, such as coal, contributes to this NO, formed through reactions with hydrocarbon radicals such as CH, CH_3, forming CHN.

One can avoid this NO by removing N from coal or using coal containing very low N with oxygen rather than air. (Oxy-coal combustion).

2. Prompt NO:

Air bound N reacting with hydrocarbon radicals as above. We cannot do anything to limit its generation, unless air is replaced with oxygen.

3. Thermal NO:

This is common and major contributor to NO formation in combustion. Described by Zeldovich mechanism

$$N_2 + O \rightarrow NO + N$$

$$N + O_2 \rightarrow NO + O$$

The overall reaction is $N_2 + O_2 \rightarrow 2NO$.

The O-atom is provided by the dissociation of O_2 at high temperature ($> 1600\,K$). This dissociation reaction is typically slower compared to combustion reactions. Thus, the residence time at high temperature

plays a vital role in the total amount of NO formed. Reducing this residence time reduces the NO emitted finally from the combustion.

Mitigation methods:

Spark ignition engines:

1) Catalytic reduction, - NO to N_2, and O_2

2) Lean burning — reduced peak/flame temperature

3) EGR → reduced temperature and reduced N_2 & O_2 levels

Diesel engines:

1) EGR, (2) Selective catalytic reduction using ammonia.

Gas turbines:

1) Lean burning, (2) Good mixing to avoid hot spots & diffusion combustion.

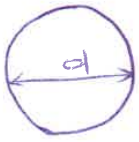
Industrial burners, Incinerators & Coal Power Station:

1) EGR, 2) NOx reburn

3) Selective catalytic reduction using ammonia [7]

(d) No, the strategies wouldn't change, since the NO formation is expected to be through thermal, N_2O and NNH pathways. Prompt NO would be zero since there wouldn't be any hydrocarbon radicals. [2]

Q2)



air @ 1 bar

T_a = 700K

$\Rightarrow Y_{f,\infty} = 0$

(a) $-\frac{dm_\ell}{dt} = A\dot{m}''$

$m_\ell = \frac{4}{3}\pi\left(\frac{d}{2}\right)^3 \rho_\ell = \frac{\pi}{6} d^3 \rho_\ell$

$A = \pi d^2$

$$\frac{\pi}{6} d^2 \rho_\ell \frac{dd}{dt} = -\pi d^2 \frac{\rho_\ell \beta}{4d}$$

$$\Rightarrow 2d\frac{dd}{dt} = -\beta \Rightarrow \boxed{d^2 = d_0^2 - \beta t}$$

[4]

(b) $d = 0$ when the droplet is fully evaporated.

$$\Rightarrow t_v = \frac{d_0^2}{\beta} = \frac{\rho_\ell d_0^2}{8\rho D \ln\left(\frac{1-Y_{f\infty}}{1-Y_{fo}}\right)}$$

$\because Y_{f,\infty} = 0 \Rightarrow$

$$\boxed{t_v = \frac{\rho_\ell d_0^2}{8\rho D \ln\left(\frac{1}{1-Y_{fo}}\right)}}$$

[3]

(c) $\dot{q}'' = \frac{2\lambda(T_a - T_s)}{d}$

energy flux at the surface: heat conduction = evapo heat flux

$A\dot{q}'' = A\dot{m}'' h_{fg} \Rightarrow \dot{q}'' = \dot{m}'' h_{fg}$

$$2\lambda\frac{(T_a - T_s)}{d} = \frac{\rho_\ell \beta}{4d} h_{fg}$$

$$\Rightarrow \beta = \frac{g\lambda(T_a - T_s)}{\rho_l h_{fg}} = \frac{8\rho D}{\rho_l} \ln\left(\frac{1}{1-Y_{fo}}\right)$$

$$\Rightarrow \ln\left(\frac{1}{1-Y_{fo}}\right) = \frac{\lambda(T_a - T_s)}{8\rho D h_{fg}}$$

$$Le = \frac{\alpha}{D} = \frac{\lambda}{\rho c_p D} = 1 \Rightarrow \frac{\lambda}{\rho D} = c_p$$

$$\therefore \ln\left(\frac{1}{1-Y_{fo}}\right) = c_p \frac{(T_a - T_s)}{h_{fg}}$$

$$\Rightarrow \boxed{Y_{f,o} = 1 - \exp\left[-c_p \frac{(T_a - T_s)}{h_{fg}}\right]} \text{ as required.}$$

(a) $\rho_l = 750\ kg/m^3$ $c_p = 1.075\ KJ/kg\text{-}K$

$D = 1.78 \times 10^{-5}\ m^2/s$ $h_{fg} = 256\ KJ/kg$ $T_s = 490\ K$

$d_o = 20 \times 10^{-6}\ m$

$$Y_{f,o} = 1 - \exp\left[-\frac{1.075}{256} \times (700 - 490)\right] = 0.58598$$

$$\ln\left(\frac{1}{1-Y_{fo}}\right) = 0.8818$$

$$\therefore t_v = \frac{750 \times (20 \times 10^{-6})^2}{8 \times 1.78 \times 10^{-5} \times \rho \times 0.8818} = \frac{2.389 \times 10^{-3}}{\rho}\ s$$

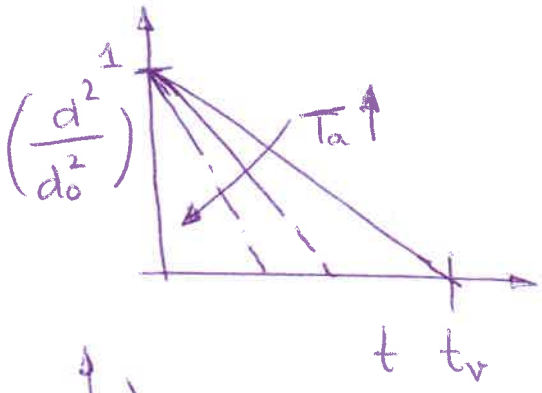
$\rho \approx$ density of air @ 1 bar & 700 K

$$\rho = \frac{1 \times 10^5 \times 28.8}{8.314 \times 10^3 \times 700} = 0.495\ kg/m^3$$

$\therefore t_v = 4.83 \times 10^{-3}$ Sec. $\approx$ 5 ms

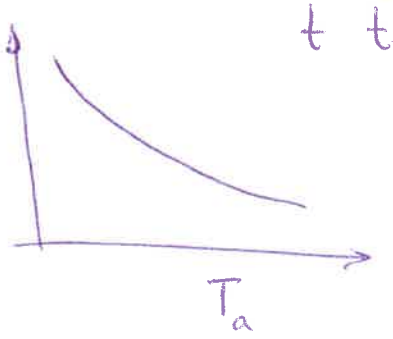
If $T_a \uparrow$ $Y_{f,0} \uparrow$ $\Rightarrow$ $t_v \downarrow$

$d_0 \downarrow$ $t_v \downarrow$



With Pressure:

As $p \uparrow$ $\rho \uparrow$ $\Rightarrow$ $t_v \downarrow$ if everything else remain Const.

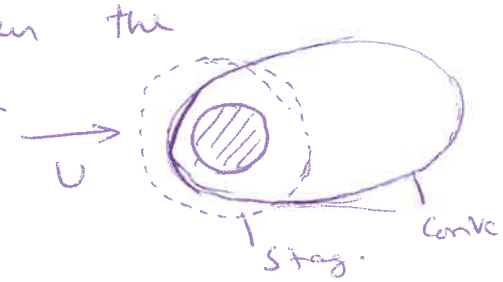


But T_s depends on p, also h_{fg} decreases with p increasing.

(Saturation temperature or boiling point)

Thus, the effect of ~~[illegible]~~ changing p on t_v is highly non-linear, but t_v generally decreases as $p \uparrow$

(e) If there is convective flow then the heat flux at the droplet surface is through convective mechanism



& thus $q'' = h(T_a - T_s)$

h - comes from Nu Correlations & $t_v \downarrow$

Y_f = const. iso-surface. in convective flow.

The rest of the analysis remains the same.

Q3(a)

Fossil kerosene: CO_2, CO, NO_x, soot

SAF: same combustion behaviour in terms of CO_2, CO & NO_x. But CO_2 may be neutral in a lifecycle sense because the C may have been captured from the air or from biomass.

Combustor performance virtually the same.

Soot could be different due to the different aromatic content, which means PM emission & contrail formation could be improved. [6]

(b) $NH_3 \rightarrow \frac{1}{2}N_2 + \frac{3}{2}H_2$ We need heat to achieve this cracking, but in an adiabatic system this energy is chemical energy in the H_2. Therefore, the adiabatic flame T is independent from degree of cracking. The H_2 has higher LCV, but the N_2 acts like an "absorber" to keep T_{ad} the same.

Flame speed probably increases with x due to the presence of H_2. Therefore, qualitatively,

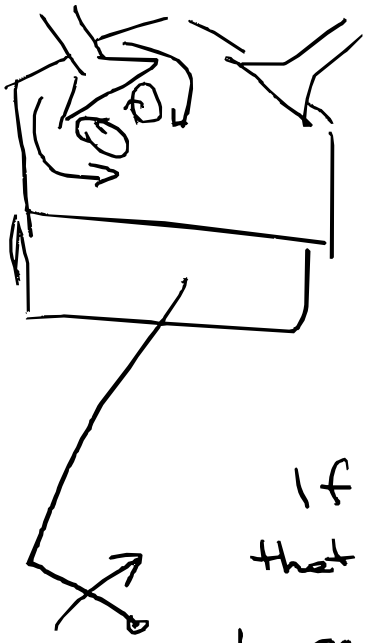
[6]

f(c) Compression ignition: likely to be like diesel where pilot fuel (eg a hydrocarbon) is injected first to autoignite & then the NH_3 is injected. In such a system, NH_3 may escape due to incomplete mixing. If the NH_3 + air are already premixed at the moment of ignition of the pilot fuel, unburnt NH_3 may escape as in spark-ignition engines that rely on flame propagation, i.e. due to fuel in the piston crevices or due to heat loss & quenching at walls.

In gas turbines, typically we have a staged system & NH_3 may escape due to incomplete mixing with air at the second stage.

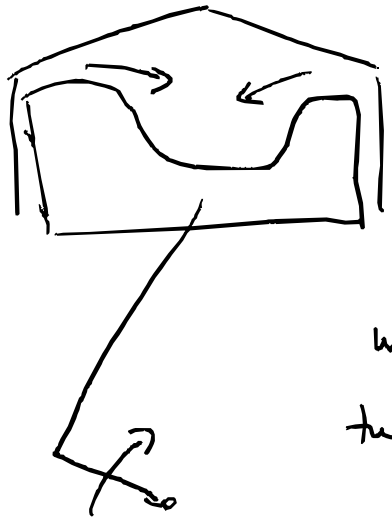
[8]

Q4 (a)



Tumble: the jet-like flows at the inlet valve produce vortex under the valve; as the piston moves these vortices are stretched & the jets break ⇒ turbulence

If the jet orientation is such that the incoming flow has a tangential component, swirl is produced (i.e. a vortex with axis parallel to the axis of the cylinder).



Squish: shape of piston bowl allows inward flow which causes jet-like flow from large radii towards the centre. As the piston moves, this breaks up & causes turbulence.

[6]

(b) $P = \dot{m}_f \, LHV \cdot \eta_{comb} \eta_c$

η_{comb} ← combustion efficiency

η_c ← cycle efficiency $= \eta_{ther} \cdot \eta$

η_{comb}: Combustion efficiency < 1 because we may have some unburnt fuel at crevices or due to too late combustion at power stroke.

η_{theor}: theoretical cycle efficiency

η = fraction of the theoretical efficiency < 1 due to finite rate of combustion, heat losses to walls, pumping losses

η_{theor} can increase by having higher compression ratio.

Pumping losses may decrease by careful design of valves & ports

Heat release rate could approach theoretical (otto or diesel) by fast combustion.

Heat losses to walls are not easy to reduce. [8]

(c) Marine diesel engines tend to use heavy hydrocarbons $\Rightarrow$ more sooting tendency from chemistry. Also, they contain aromatics $\Rightarrow$ even higher soot. The low vapour pressure implies poor atomisation $\Rightarrow$ slower mixing.

Automotive diesel engines burn lighter diesel $\Rightarrow$ smaller droplets & easier to have more premixed combustion phase $\Rightarrow$ lower soot. [6]