

1) a) laminar burning velocity:

The speed at which a planar laminar unstrained premixed flame propagates into a stagnant reactant mixture.

Flammability limits: & their origins:

The limits in equivalence ratio,  $\phi$ , (also fuel to air ratio) between which a mixture can sustain a self-propagating flame. These limits can also be expressed as values of fuel content usually given in volume percentage.

The lean limit is the smallest possible fuel to air ratio. The existence of lean flammability limit is due to the low flame temperature, as the mixture becomes leaner, which leads to the dominance of chain-terminating reaction over chain propagating reactions. This means that there is not enough of radicals to sustain flame propagation.

For the rich side, temperature is low and not enough oxygen, leading to low radical concentration and thus no flame propagation.

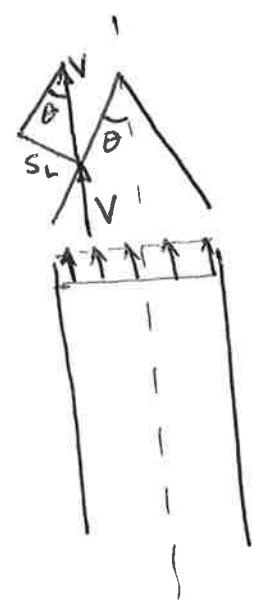
Thus the flame propagation occurs for

$$\phi_{lean} \leq \phi \leq \phi_{rich}$$

Quenching distance: is the diameter of the smallest tube in which a premixed flame of a given equivalence ratio can propagate freely.

(b)

(i)



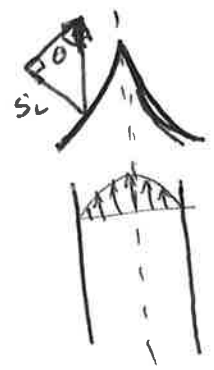
$$V \sin \theta = S_L$$

$$\therefore \sin \theta = \left( \frac{S_L}{V} \right)$$

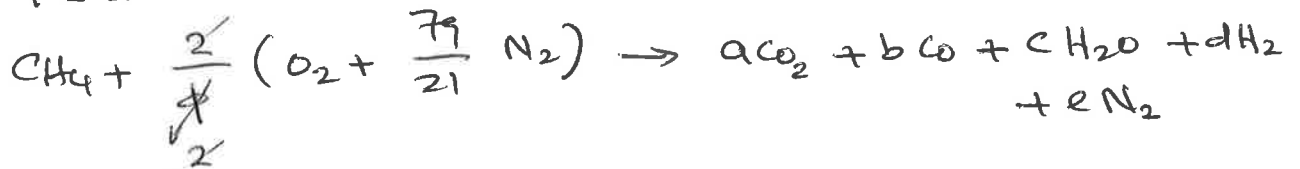
if  $V$  is uniform  $\theta(r) = \theta$   
 $\Rightarrow$  the flame is conical.

$$(ii) \sin \theta = \frac{S_L}{V(r)} = \frac{S_L}{V_0 \left(1 - \frac{r^2}{R^2}\right)}$$

$\Rightarrow \theta \neq \text{const} \Rightarrow \theta = \theta(r)$   
the flame is not conical.



$$(c) \phi = 2$$



$$\text{N atom conservation: } 2 \frac{79}{21} = 2e \Rightarrow \boxed{e = \frac{79}{21}}$$

$$\text{C atom: } 1 = a + b \quad \text{--- (1)}$$

$$\text{H: } 4 = 2(c + d) \Rightarrow c + d = 2 \quad \text{--- (2)}$$

$$\text{O: } 2 = 2a + b + c \quad \text{--- (3)}$$

4 unknowns and three equations.

4<sup>th</sup> equation comes from  $K_p$  relation of

water-gas shift reaction



because of rich combustion.

$$\Rightarrow K_p = \frac{(P_{\text{H}_2}/P) (P_{\text{CO}_2}/P)}{(P_{\text{CO}}/P) (P_{\text{H}_2\text{O}}/P)} = \frac{X_{\text{H}_2} X_{\text{CO}_2}}{X_{\text{CO}} X_{\text{H}_2\text{O}}}$$

$$= \frac{da}{bc} = \exp(-1.091) \quad \text{from the data book.}$$

$$\text{from (1): } a = 1 - b$$

$$\text{(2): } d = 2 - c = 2 - (2 - 2(1 - b) - b) \quad \text{using (3)}$$

$$\Rightarrow d = 2 - b \Rightarrow \underline{\underline{c = b}}$$

$$\Rightarrow \frac{(2 - b)(1 - b)}{b^2} = \exp(-1.091)$$

$\Rightarrow b = 0.81298$  or  $3.70441$

$\uparrow$   
This isn't possible  
because  $d < 0$  &  $a < 0$ .

The possible & meaningful root is

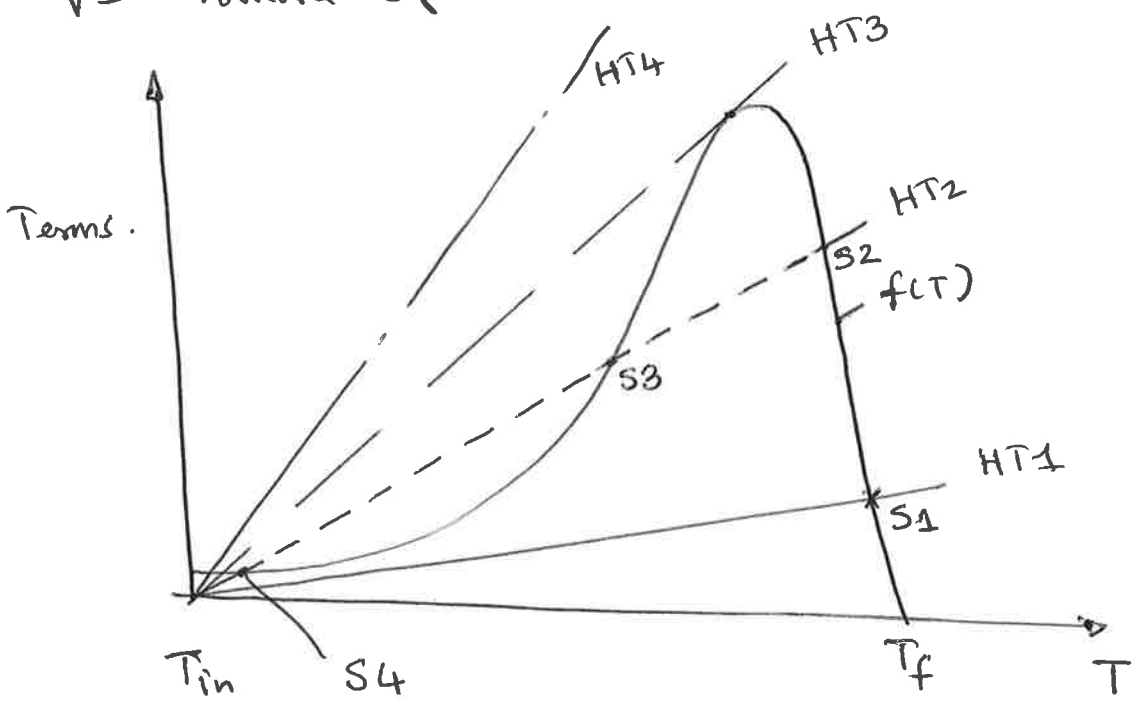
$b = 0.81298$	}	=>
$a = 0.18702$		
$c = 0.81298$		
$d = 1.18702$		
$e = 3.76190$		
$\text{Total} = 6.76190$		

$X_{CO_2} = 0.0277$
$X_{CO} = 0.1202$
$X_{H_2O} = 0.1202$
$X_{H_2} = 0.1755$
$X_{N_2} = 0.5567$
<u><u>Total = 1.00</u></u>

2. (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
- $\rho$  - density of reactant mixture
- $V$  - Volume of the reactor



$f(T)$  - heat generation by chemical reaction.

H.T  $\rightarrow$  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 HT1  $\rightarrow$  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  $\tau$  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

$$t_{chem} > c t_{res}$$

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

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

$$\tau_{chem} = \frac{1}{\rho_0 c_p S_L^2} = \frac{A_{ref} (T_0/T_{ref})^{1/2}}{\rho_0 c_p S_{ref}^2 (T_0/T_{ref})^{3/2}}$$

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

$$\tau_{res} = \frac{L}{U} = \frac{L \rho_0 A}{\dot{m}} \quad \text{A C.S. Area of the combustor}$$

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

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

blow-off occurs @ both conditions

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

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

$\Rightarrow$  The mass flow rate,  $\dot{m}_2$ , is increased by 466%

(C) There are three mechanisms.

1. Fuel NO:

Nitrogen in the fuel, such as coal, contribute to this NO, formed through reactions with hydrocarbon radicals such as CH, CH3, 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



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

The O-atom is provided by the dissociation of O<sub>2</sub> at high temperature (> 1800K). 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<sub>2</sub>, and O<sub>2</sub>
- 2) Lean burning - reduced peak/flame temperature
- 3) EGR → reduced temperature and reduced N<sub>2</sub> & O<sub>2</sub> levels

Diesel engines:

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

Gas turbines:

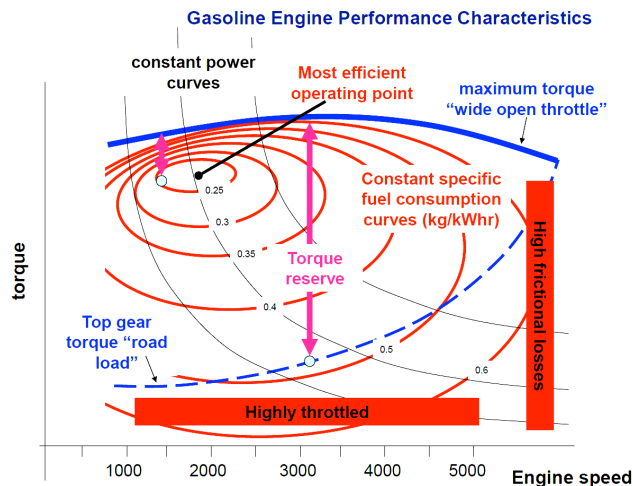
- 1) Lean burning, (2) Good mixing to avoid hot spots & diffusion combustion.

Industrial burners, Incinerators &

Coal Power Station:

- 1) EGR, 2) NOx reburn
- 3) Selective catalytic reduction using ammonia

3. (a)



Maximum torque characteristic. At low engine speeds scavenging is less effective as during valve overlap there is relatively little momentum in the exhaust gases. At high engine speed, pressure losses through the pipework and valves leads to a reduced charge pressure, and hence less oxygen. Thus in the mid engine speed range, the optimum bmep/torque is developed. Variable valve timing helps to mitigate these issues to some extent.

- (b) The maximum torque is limited by breathing, as above, and by knocking combustion. If the compression ratio is too high, then optimum spark timing cannot be used, because damaging auto-ignition takes place in the end gas regions. This leads to excessive heat transfer and engine damage
- (c) The top gear road load characteristic is dominantly parabolic, as the vehicle drag increases as the square of the speed. Due to tyre rolling resistance, engine auxiliaries (water pump, oil pump, etc) it does not pass through zero.
- (d) The dominant effects on sfc are throttling losses and engine friction torque. The latter increases significantly with engine speed so the friction power increases very rapidly with engine speed. Throttling losses increase as one drops below the maximum torque curve.
- (e)
  - (i) Downsizing +turbochargers
  - (ii) Start-stop – especially beneficial for city driving
  - (iii) Mechanical CVT – better matching of engine to power requirement – essentially down-speeding
  - (iv) Electro/mechanical Hybrid (Prius etc)
  - (v) Light weighting
- (f) All of the above can in principle apply to Diesel engines. On a like-for-like basis however, the diesel engine is significantly more expensive that a gasoline engine, so (iii) and (iv) are not popular.

4 (a) Isentropic compression,  $pV^\gamma = \text{const}$ , so we have  $pV^\gamma = p_1V_1^\gamma = p_2V_2^\gamma = k$

The work done is given by  $W_{12} = \int_1^2 p dV = \int_1^2 \frac{k}{V^\gamma} dV = \frac{k}{1-\gamma} [V_2^{(-\gamma+1)} - V_1^{(-\gamma+1)}]$

So

$$W_{12} = \frac{1}{1-\gamma} (p_2V_2^\gamma V_2^{1-\gamma} - p_1V_1^\gamma V_1^{1-\gamma}) = \frac{p_1V_1}{\gamma-1} \left(1 - \frac{p_2V_2}{p_1V_1}\right) = \frac{p_1V_1}{\gamma-1} \left(1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right)$$

$$(b) W_{12} = \frac{p_i V_m}{\gamma-1} \left(1 - \left(\frac{V_m}{V_c}\right)^{\gamma-1}\right), \quad W_{34} = \frac{p_3 V_c}{\gamma-1} \left(1 - \left(\frac{V_c}{V_m}\right)^{\gamma-1}\right)$$

(c) For the constant volume combustion,  $\frac{p_2}{T_2} = \frac{p_3}{T_2 + \Delta T_c}$ , and  $p_2 = p_i \left(\frac{V_m}{V_c}\right)^\gamma$ ,  $T_2 = T_i \left(\frac{V_m}{V_c}\right)^{\gamma-1}$  therefore

$$p_2 = 0.5(10)^{1.4} = 12.56 \text{ bar}, \quad T_2 = 288 \times 10^{0.4} = 723.4 \text{ K}$$

$$\therefore p_3 = \frac{(723.4 + 1300) \times 12.56}{723.4} = 35.13 \text{ bar}$$

$$W_{12} = \frac{0.5V_m}{0.4} (1 - (10)^{0.4}) = -1.89V_m \quad W_{34} = \frac{p_3 \cdot V_c}{0.4} (1 - (1/10)^{0.4}) = 5.286V_m$$

$$\text{gross imep is } = \frac{(5.26 - 1.89)V_m}{(V_m - V_c)} = 3.78 \text{ bar}$$

(d) The pumping work is equal to  $W_{\text{pump}} = (p_i - p_e)(V_m - V_c) = p_i V_m \left(1 - \frac{p_e}{p_i}\right) \left(1 - \frac{V_c}{V_m}\right)$

$$V_8 = V_m \left(\frac{p_i}{p_e}\right)^{1/\gamma} \quad W_{86} = p_e (V_6 - V_8) = p_e V_m \left(\frac{V_c}{V_m} - (p_i/p_e)^{1/\gamma}\right) \quad W_{67} = 0$$

$$W_{71} = p_i (V_1 - V_7) = p_i V_m \left(1 - \frac{V_c}{V_m}\right)$$

Using the data given

$$W_{18} = \frac{0.5}{0.4} V_m [1 - (1.05/0.5)^{0.4/1.4}] = -0.295V_m \quad W_{86} = 1.05V_m \left(\frac{1}{10} - (0.5/1.05)^{1/1.4}\right) = -0.513V_m$$

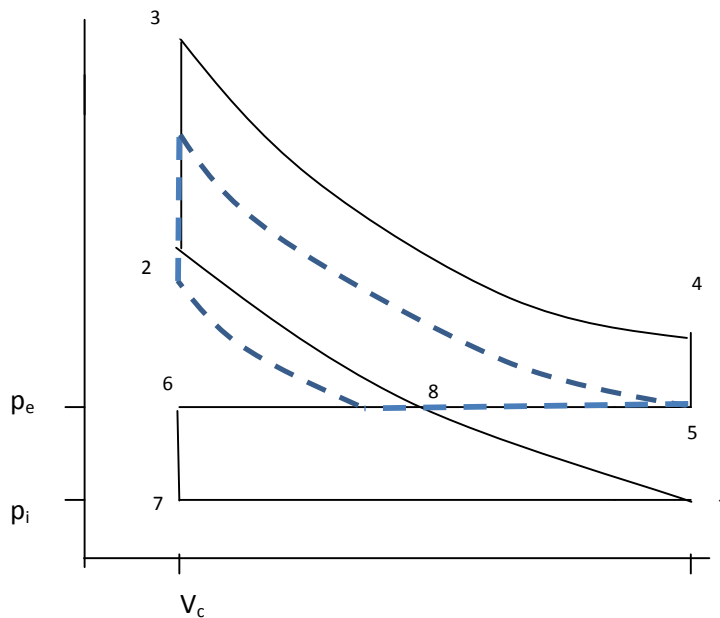
$$W_{71} = 0.5V_m \left(1 - \frac{1}{10}\right) = 0.45V_m$$

So the net pumping work is  $W_{\text{pump}} = V_m (-0.295 - 0.513 + 0.45) = -0.358V_m$

Therefore the  $p_{mep}$  is  $= \frac{-0.358V_m}{V_m - V_c} = -0.398 \text{ bar}$

Net  $imep$  is  $= 3.78 - 0.398 = 3.382 \text{ bar}$

(e) Un-throttled cycle producing the same work:-



The key thing to note here is that the area of the new, unthrottled cycle has to be the same as the net area of the throttled one – so the inlet valve closure needs to happen after “8” in order to get less mixture in the cylinder.

## **Examiners' comments**

### **Q1 Premixed flame and equilibrium composition**

A straightforward question and good answers were found for parts (a) and (c). Many students showed trouble in rearranging the algebraic relation to get the required quadratic equation in part (c) and this could perhaps be because of examination pressure. Part (b) was not answered well, in general, as it required some extended thinking conceptually, although a similar question was there in the example paper of this module.

### **Q2 Flame blow-off and NO<sub>x</sub> emission**

No major issues in answering this question

### **Q3 Engine map characteristics for gasoline and diesel engines**

This was the most popular and no major issues in answering this question.

### **Q4 Theoretical analysis of 4-stroke engine cycle**

This was a popular question and there were some good answers. Most of the students were able to apply the isentropic relations to deduce the work done during the compression and expansion strokes, but had algebraic error while doing the calculations for part (c). The answers to parts (c) and (d) are mostly incomplete and the common mistake was ignoring the path 1-8 while calculating the pumping work.