Q. A Royal Enfield Thunderbird 500cc bike engine has the following specifications

- ✓ Engine type: Single cylinder, 04stroke, air cooled.
- ✓ Engine displacement: 499cc
- ✓ Compression ratio: 8.5:1
- ✓ Maximum power: 27.2BHP@5250RPM
- ✓ Maximum torque: 41.3Nm@4000 RPM
- ✓ Air to fuel ratio: 14.7:1
- ✓ Calorific value of gasoline: 45200kJ/kg

Consider an air-standard Otto cycle with variable property and determine

a) Temperature/Pressurw at all process end points in the cycle
b) Air standard efficiency
c) Fuel consumption rate (kg/s)
d) Air consumption rate (kg/s)
e) Actual efficiency at maximum power
f) Actual and ideal power
g) Narrate the possible reason for the differences between actual efficiency for maximum power condition and air standard efficiency. Consider a polytropic index of 1.2 to determine the pre and post-combustion temperature and heat losses. Compare it with air standard data.
h) Repeat the step (g) for actual efficiencies determined at maximum torque condition. Why efficiencies vary at maximum power to maximum torque condition?

Q. Consider a Hyundai Creta Diesel variant with the following specifications.

- ✓ Engine type: 1.6-litre, 04Cylinder, 126.2bhp, 16V, U2, CRDI, VGT Engine
- ✓ Displacement: 1582cc
- ✓ Maximum Power: 126 BHP@4000 RPM
- ✓ Maximum Torque: 265 Nm@1900 RPM
- ✓ Compression ratio: 17:1
- ✓ Diesel calorific value: 42800kJ/kg
- ✓ Fuel cut-off ratio:2.6
- ✓ Fuel consumption rate:12kg/hr

Consider an air-standard Diesel cycle with variable property and determine

a) Temperature/Pressure at all process end points in the cycle with a comparative emphasis on variable property.
b) Determine air standard efficiency and ascertain the effect of cut-off ratio.
c) Compute air and fuel consumption under various conditions.
d) Suggest a suitable polytropic index and cut-off ratio that roughly yield the same performance of the actual engine.

Q.1:- Royal Enfield Thunderbird 500 specifications

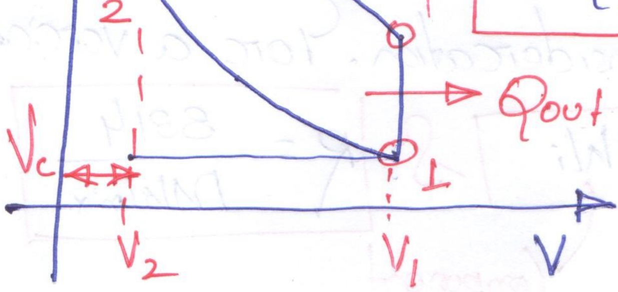
- Engine Type - Single cylinder, four stroke, air cooled
- Engine displacement - 499cc
- Compression ratio - 8.5 : 1
- Maximum power - 27.2 BHP @ 5250 RPM
- Maximum torque - 41.3 Nm @ 4000 RPM
- Air to fuel ratio - 14.7 : 1
- Gasoline calorific value - 45,200 kJ/kg

Developing the solution:-

Ambient pressure (Delhi) = P_1 = 96.8 KPa

Ambient temperature = 310K

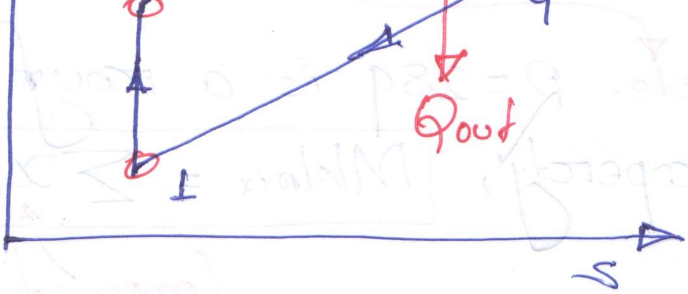
Air-standard Otto-cycle



$$\frac{V_1}{V_2} = 8.5$$

$$\frac{V_s + V_c}{V_c} = 8.5$$

$$\boxed{V_c = 66.53 \text{ cc}}$$



$$P_2 = P_1 (rc)^{\gamma} = 96.8 \times (8.5)^{1.4} \Rightarrow \boxed{P_2 = 1936.69 \text{ KPa}}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 310 \times \left(\frac{1936.69}{96.8}\right)^{\frac{1.4-1}{1.4}} \Rightarrow \boxed{T_2 = 729.67 K}$$

#2

For constant volume heat addition part 2-3

$$(M_{air} + M_{fuel}) \times C_v (T_3 - T_2) = M_{fuel} \times \text{Calorific value}$$

$$\Rightarrow (1 + A/F) \times C_v (T_3 - T_2) = \text{Calorific value}$$

$$\Rightarrow (1 + 14.7) \times 0.717 (T_3 - 729.67) = 45,200$$

$$\Rightarrow T_3 = 729.67 + \frac{45,200}{15.7 \times 0.717} \Rightarrow \boxed{T_3 = 4744.98K}$$

*Note:-

For variable property calculation, be careful here :- $\boxed{C_{v,mix} = f(\text{composition and temperature})}$

Now ideal air-standard efficiency

$$\eta_{otto} = 1 - \frac{1}{r_c^{\gamma-1}} = 1 - \frac{1}{(8.5)^{0.4}} \Rightarrow \boxed{\eta_{otto} = 57.5\%}$$

Invoking ideal gas equation at intake

$$(M_{air} + M_{fuel}) = \frac{P_1 V_1}{R T_1} = \frac{96.8 \times (\overbrace{499 + 66.53}^{V_s + V_c}) \times 10^{-6} \times 10^3}{289 \times 310}$$

$$\Rightarrow 15.7 \times M_{fuel} = 6.11043 \times 10^{-5} \text{ kg} \Rightarrow \boxed{M_{fuel} = 3.891 \times 10^{-5} \text{ kg}}$$

*Note- $R = 289$ is a rough consideration. For a variable property, $\boxed{MW_{mix} = \sum x_i MW_i}$ & $\boxed{R = \frac{8314}{MW_{mix}}}$

Component mole fraction

Component mol. wt.

Now for maximum power condition, RPM = 5250

$$\therefore \dot{M}_{fuel} = M_{fuel} \times \frac{N}{2 \times 60} = (3.89e{-5}) \times \frac{5250}{120}$$

#3

$$\therefore \boxed{\dot{m}_{fuel} = 1.702e-3\ kg/s}$$ Fuel consumption rate at maximum power

Now heat input rate, $\dot{Q}_{in} = \dot{m}_{fuel} \times$ Calorific value

$$\Rightarrow \dot{Q}_{in} = 1.702e-3 \times 45{,}200 \Rightarrow \boxed{\dot{Q}_{in} = 76.93\ kJ/s}$$

$$\therefore \eta_{actual}\Big|_{max,p} = \frac{\text{Maximum power}}{\text{Rate of Heat input at maximum power}} = \frac{27.2/0.746 = 36.46kW}{76.93}$$

$$\Rightarrow \boxed{\eta_{act}\Big|_{max,p} = 47.39\%}$$ Compare it with $\boxed{\eta_{Otto} = 57.5\%}$

So ideal maximum power should have been

$$P_{max} = 76.93 \times 0.575 \Rightarrow \boxed{P_{max}\Big|_{ideal} = 44.23kW}$$

The differences in the ideal & actual power may be attributed to the losses as the process is never isentropic.

A first law perspective

$$\delta Q = \overset{0\ (\text{isentropic})}{\cancel{T ds}} = \delta U + P\delta V, \text{ but}$$

Realistic

$$\boxed{Q_{loss} = mC_v(T_2 - T_1) + \frac{mR(T_2 - T_1)}{1-\gamma}}$$

$\gamma < 1.4$

Let's assess the losses with $\gamma < 1.4$

Say $n = 1.2$

#4

$m_{fuel} = 3.891e{-5}$ kg $\quad (A/F) = 14.7$

$\therefore m = m_{fuel} + m_{air} = 15.7 \times m_{fuel} \Rightarrow \boxed{m = 6.1e{-4}\ \text{kg}}$

For $n = 1.2$

$$T_2 = \left(\frac{P_1 \times r_c^{n}}{P_1}\right)^{\frac{n-1}{n}} \times T_1 = \left(\frac{96.8 \times 8.5^{1.2}}{96.8}\right)^{\frac{0.2}{1.2}} \times 310$$

$$\Rightarrow \boxed{T_2 = 478.91\text{K}}$$

Note:- T_2 was 729.67k at same compression ratio when $n = 1.4$ initially.

$$Q_{loss} = \frac{mR(T_2 - T_1)}{1-\gamma} + m c_v (T_2 - T_1)$$

$$= \frac{6.1e^{-4} \times 289\,(478.9 - 310)}{-0.2} + 6.1e{-4} \times 717\,(478.9 - 310)$$

$$= -148.87 + 73.871 \Rightarrow \boxed{Q_{loss} = 74.9\text{ J}}$$

Now let's find out what is the $\dot{m}_f$ rate per cycle?

$\dot{m}_f = 1.702e{-3}$ kg/s - For max power at 5250 rpm.

$$\therefore \text{No. of cycles per second} = \frac{5250}{60 \times 2}$$

For four stroke

$$\Rightarrow \boxed{N_s = 43.75 \text{ cycles/s}}$$

$$\therefore \dot{m}_{f,cycle} = \frac{\dot{m}_f}{N_s} = \frac{1.702e{-3}}{43.75} = 3.890e{-5} \text{ kg/cycle}$$

Now $\dot{m}_{f,cycle} \times \text{Calorific value} \times x_{loss} = Q_{loss}$

$\therefore \; x_{loss} = \dfrac{74.9}{3.890e-5 \times 45200 \times 1000} \Rightarrow \boxed{x_{loss} = 4.26\%}$

$\therefore$ Loss during combustion will be

$$m \times C_V (T_3 - T_2) = (1 - x_{loss}) \, m_f \times CV$$

$$\Rightarrow T_3 = T_2 + \frac{45,200 \times 0.957}{15.7 \times 0.717}$$

$$\Rightarrow T_3 = 478.91 + 3842.657$$

$$\Rightarrow \boxed{T_3 = 4321.567}$$

Note - It was 4744.98K for $n = 1.4$

n	T_2 (K)	Q_{loss} (J)	x_{loss}	T_3 (K)
1.2	478.91	74.9	4.26%	4321.567
1.4	729.67	0	0	4744.98K

Repeat the above procedure for actual efficiencies determined at maximum torque condition.

2. A Hyundai Creta Diesel variant has the following specifications.

- Engine type: 1.6L, CRDI, VGT Engine
- Displacement: 1582cc
- Maximum Power: 126 BHP @ 4000 RPM
- Maximum Torque: 265 Nm @ 1900 RPM
- Compression ratio: 17 : 1
- Diesel calorific value: 42800 kJ/kg
- Fuel cut-off ratio: 2.6
- Fuel consumption rate: 12 kg/hr

Air standard diesel cycle thermal efficiency

$$\eta_{diesel} = 1 - \frac{1}{rc^{\gamma-1}}\left[\frac{1}{\gamma}\cdot\frac{(rc_c^{\gamma}-1)}{(rc_c-1)}\right] \quad (rc_c = \text{cut off ratio})$$

$$\Rightarrow \eta_{diesel} = 1 - \frac{1}{(17)^{0.4}}\left[\frac{1}{1.4}\cdot\frac{(2.6^{1.4}-1)}{(2.6-1)}\right]$$

$$\Rightarrow \eta_{diesel} = 1 - 0.3219\,(1.254612)$$

$$\Rightarrow \boxed{\eta_{diesel} = 59.6\,\%}$$

$\dot{m}_f = 12\,kg/hr = 0.0033\,kg/s \quad \therefore \dot{Q}_f = \dot{m}_f \times \text{Cal. value}$

$\Rightarrow \dot{Q}_f = 0.0033 \times 42{,}800 \Rightarrow \boxed{\dot{Q}_f = 141.24\,kW}$ ← For four cylinders

$\boxed{\dot{Q}_{f,pc} = \frac{141.24}{4} = 35.31\,kW}$ ← Heat input per cylinder

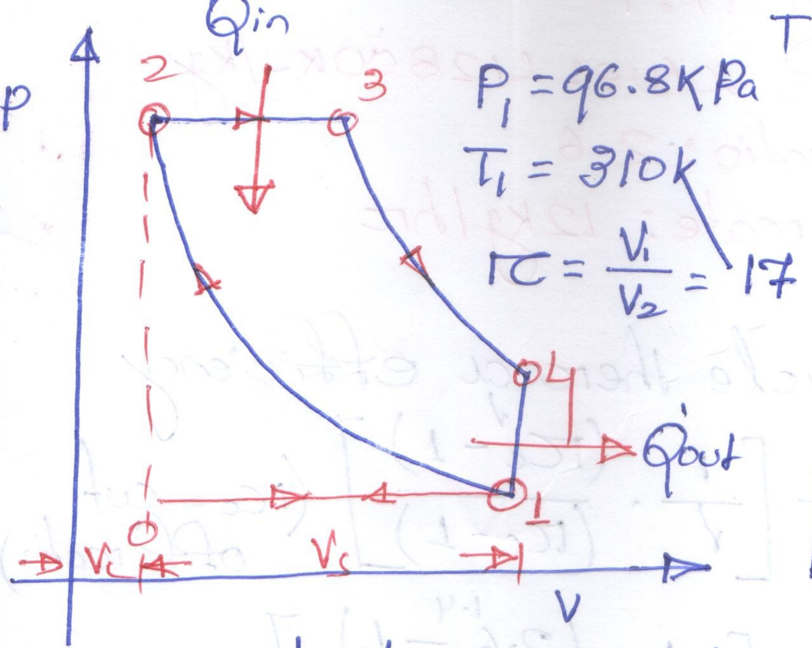
On the basis of ideal diesel cycle, the power to be realized is

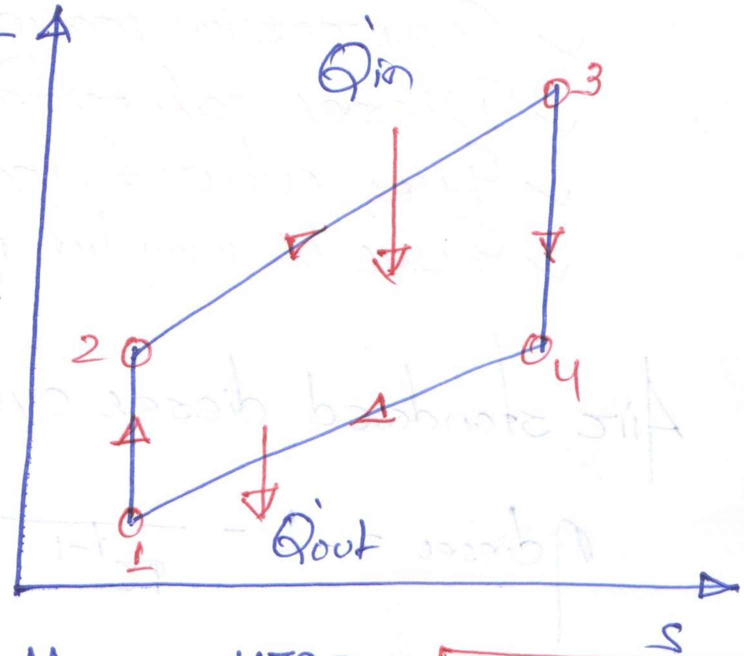
$$\boxed{P_{ideal,\,pc} = 35.31 \times 0.596 = 21.04\,kW}$$

However, the actual (rated) engine power is

$$\boxed{P_{actual,\,pc} = \frac{126}{4 \times 0.746} = 42.225\,kW}$$

Deviation from ideal cycle





$$r_c = \frac{V_s + V_c}{V_c} = 17 \Rightarrow V_c = \frac{V_s}{r_c - 1} = \frac{1582}{16} \Rightarrow \boxed{V_c = 98.87\,cc}$$

→ Clearance volume

Possible scenarios of post compression temp.

$$\boxed{T_2 = T_1 (r_c)^{\gamma - 1}}$$

	n	T_2 (k)
1.	1.4	962.81
2.	1.3	725.26
3.	1.2	546.32

Possible scenarios of post compression pressure

$$\boxed{P_2 = P_1 (r_c)^{\gamma}}$$

	n	P_2 (KPa)
1.	1.4	5110.976
2.	1.3	3849.984
3.	1.2	2900.107

#8

Determination of rate of heat addition per cylinder per cycle:—

Now $\boxed{\dot{Q}_{f,pc} = 35.31\,kW}$ ← $\frac{kJ}{s}$ Amount of heat added per cycle (kJ/cycle) is a function of engine RPM.

∴ Let's consider the maximum power condition ⇒ 4000 RPM.

$$\boxed{\therefore \quad N_{ps} = \frac{4000}{60 \times 2} = 33.33}$$ ← No. of cycles per second for max power

$$\therefore \quad \dot{Q}_{f,pc,pc} = \frac{\dot{Q}_{f,pc}}{N_{ps}} = \frac{35.31}{33.33} \Rightarrow \boxed{\dot{Q}_{f,pc,pc} = 1.059\,kW}$$

A scenario of rate of heat added per cylinder per cycle

RPM	$\dot{Q}_{f,pc,pc}$ (kW)
4000 –	1.059
3000 –	1.412
2000 –	2.118

Note – Under the same load, reduced RPM generates higher workdone per cycle.

Determination of post combustion temperature:-

At intake – $P_1 V_1 = m R T_1$ ← Mass of air as no fuel for diesel suction stroke.

$$\Rightarrow m_a = \frac{P_1 V_1}{R T_1}$$

$V_1 \times N_{ps}$ gives the suction rate in m^3/s

$$\Rightarrow m_a = \frac{P_1 \times (V_1 \times N_{ps})}{R T_1} \ (kg/s)$$

Different scenarios of air-breathing rate-

Air breathing rate is a function of engine RPM and volumetric efficiency at constant intake conditions.

N	η_v (%)	$\dot{m}_a$ (kg/s)	m_a (kg/cycle)
4000	100	0.0605	0.001816
4000	80	0.0484	0.001453
3000	100	0.0454	0.001816
3000	80	0.0363	0.001543
2000	100	0.0302	0.001816
2000	80	0.0242	0.001543

Note they are same

Now the fuel flow rate at maximum power condition-

$$\boxed{\dot{m}_{f,pc,pc} = \frac{\dot{Q}_{f,pc,pc}}{\text{Cal. value}} = \frac{1.059}{42,800} = 2.474e-05}$$

$\Rightarrow$ per cycle per cylinder fuel mass flow rate at 4000 RPM

$\therefore$ Invoking the energy balance during constant pressure heat addition-

Note- C_p is not constant

$$\boxed{C_{p\,mix} = \sum x_i C_{pi}}$$

$$(m_{a,pc,pc} + \dot{m}_{f,pc,pc}) \times C_p \times (T_3 - T_2) = \dot{m}_{fuel,\,pc,pc} \times \text{Calorific value}$$

$$\Rightarrow (2.474e-5 + 0.001816) \times 1005 \times (T_3 - 962.81)$$

$$= 2.474e-5 \times 42,800 \times 1000$$

$$\Rightarrow T_3 = 962.81 + \frac{2.474e-5 \times 42,800 \times 1000}{(2.474e-5 + 0.001816) \times 1005}$$

$$\Rightarrow \boxed{T_3 = 1535.191\,K}$$

Note $C_p = 1005\ J/kgK$ is a crude assumption. The actual value is always higher & variable in nature

#10

Different scenarios of post combustion temperature

n	RPM	T_2 (K)	T_3 (K)
1.4	4000	962.81	1535.191
1.3	4000	725.26	1297.641
1.2	4000	546.32	1118.701
1.4	2000	962.81	~~1675.89~~ 2092.39
1.3	2000	725.26	1854.84
1.2	2000	546.32	1675.89

Based upon the above analysis, try to figure out

1. Effect of cut-off ratio on engine performance
2. A suitable polytropic index & cut-off ratio that will roughly yield the same performance of actual diesel engine.
3. Carry out a similar analysis using maximum torque condition.
4. Figure out the combustion duration in (ms) (based upon cut-off ratio) at various RPMs & put a comment.