APPLIED THERMODYNAMICS

Internal Combustion Engines (Module III)

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List of Topics

1. Internal Combustion Engine – Components, Nomenclature and Classifications
2. Basic Engine Cycle and Engine Kinematic Analysis
3. Engine Operating Characteristics
4. Thermodynamic Analysis of Air Standard Cycles ✓
5. Valve Timing Diagram and Fuel – Air Cycle
6. Thermochemistry and Fuel Characteristics
7. Combustion Phenomena in Engines
8. Heat Transfer Analysis in Engines
9. Exergy Analysis and Engine Emission/Pollution

Lecture 3

Thermodynamic Analysis of Air Standard Cycles

- Otto Cycle (Ideal cycle for SI engines)
- Diesel Cycle (Ideal cycle for CI engines)
- Dual Cycle (Combination of Otto and Diesel cycle)

Air Standard Cycles

The cyclic process experienced in the IC engines (both four-stroke and two-stroke) is very complex.

- The intake charge (air for CI engine or premixed fuel-air mixture for SI engine) is ingested and mixed with residual exhaust of previous cycle.
- After compression and combustion, the intake charge turns into exhaust product largely with CO_2, H_2O, N_2 and other residue.
- Thus, the real process involves charges with changing composition and these open cycles are difficult to analyze.

A manageable option is to consider ideal "air standard cycles" with certain assumptions. Based on this approach the following air standard cycles are considered for thermodynamic analysis for SI and CI engines:

- Otto cycle (SI engines)
- Diesel cycle (CI engine)
- Dual cycle

Air Standard Cycles

Assumptions

- The gas mixture in the cylinder is taken as 'air' and its constant property value is considered for entire cycle analysis. In real cycle, the charge contains mostly air with around 7% of fuel vapour.
- The real open cycle is changed to closed cycle by assuming that the combustion products exhausted to atmosphere are fed back with fresh charge in the intake system.
- Since combustion is not possible with air alone, its equivalent thermodynamic heat addition process is considered.
- The exhaust process carries lot of enthalpy out of the system, so that the closing process can be considered as heat rejection process.
- The actual engine processes are approximated as near optimal ideal thermodynamic reversible process.

Air Standard Cycles

Assumptions

- Almost constant pressure intake & exhaust strokes.
- Compression and expansion strokes are treated as isentropic.
- Combustion process is idealized as constant-volume (for SI cycle) and constant pressure (for CI cycle)
- Exhaust blow down is treated as constant volume process
- Air is treated as ideal gas and the following ideal gas relations can be used

Notations:

p: Gas pressure in cylinder; V: Volume of the gas in cylinder; v: Specific volume of the gas;
R: Gas constant for air; T: Temperature; $\dot{m}$: Mass flow rate of the gas; ρ: Density;
h: Specific enthalpy; u: Specific internal energy; w: Specific work; $\dot{q}$: Heat transfer rate per unit mass;
Q_{hv}: Heating value of fuel; $\dot{W}$: Power; r_c: Compression ratio; AF: Air-fuel ratio; c_p, c_v: Specific heats

$$pv = RT;\ pV = mRT;\ p = \rho RT;\ h = c_p T;\ u = c_v T;\ k = \frac{c_p}{c_v}$$

Isentropic process: $pv^k = \text{constant};\ Tv^{k-1} = \text{constant};\ Tp^{\frac{1-k}{k}} = \text{constant};\ w = \frac{p_1 v_1 - p_2 v_2}{k-1} = \frac{R(T_1 - T_2)}{k-1}$

Air: $c_p = 1.005\ \text{kJ/kg.K};\ c_v = 0.718\ \text{kJ/kg.K};\ c_p - c_v = 0.287\ \text{kJ/kg.K};\ k = 1.4$

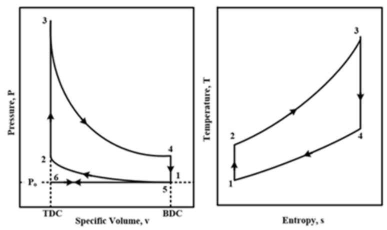
Combustion products: $c_p = 1.108\ \text{kJ/kg.K};\ c_v = 0.821\ \text{kJ/kg.K};\ c_p - c_v = 0.287\ \text{kJ/kg.K};\ k = 1.35$

Otto Cycle

It is the air-standard model of most four-stroke SI engines and involves following thermodynamic processes:

Intake stroke (Process 6-1)

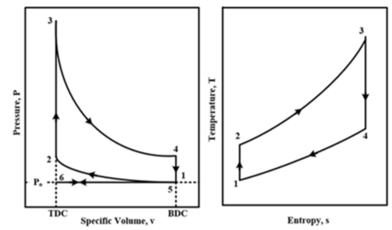
- Starts with piston at TDC: constant pressure process. But, the inlet process is slightly lower than atmospheric pressure)
- Temperature of air is increases as the air passes through hot intake manifold (i.e. air at state '1' is hotter than '6' around 25-35° higher)



Otto Cycle

Compression stroke (Process 1-2)

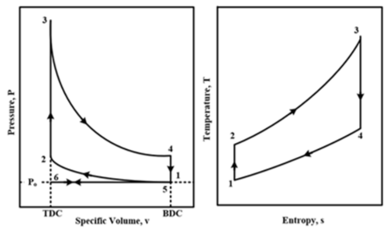
- Piston moves from BDC to TDC: Isentropic process (good approximation for real engine except at the beginning & end of stroke)
- At the beginning of compression stroke, the intake valve is not fully closed and the end of compression stroke is affected by sparkplug firing
- Temperature and pressure increases substantially



Otto Cycle

Heat addition (Process 2-3)

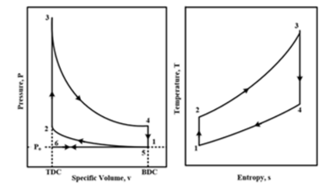
- This heat input process at constant volume replaces combustion process at TDC. In real engines, combustion starts slightly bTDC, reaches maximum speed at TDC and terminates little aTDC.
- The energy added to air rises to peak cycle pressure and temperature at '3'.



Otto Cycle

Power/expansion stroke (Process 3-4)

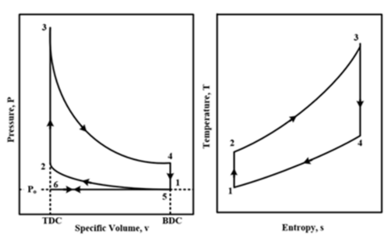
- Very high pressure and enthalpy values within the system at TDC generate high pressure on the piston face that forces back the piston & produces power output for the engine.
- In real engines, the beginning of power stroke is affected by the last part of combustion process while end of power stroke is affected by opening of exhaust valve bBDC.
- Both temperature and pressure decrease as volume increases from TDC to BDC.



Otto Cycle

Heat rejection (Process 4-5)

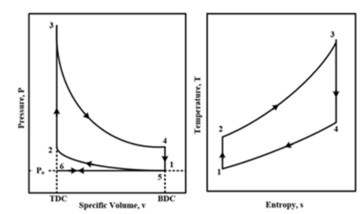
- The replacement of exhaust blowdown open system process of a real cycle is replaced with equivalent pressure reduction constant volume process of a closed system. The enthalpy loss is catered as heat rejection.
- The pressure & temperature in the cylinder at the end of exhaust blow down has been reduced to atmosphere.



Otto Cycle

Exhaust stroke (Process 5-6)

- The piston travels from BDC to TDC where the pressure is slightly higher than surroundings for real engines.
- At the end of exhaust stroke, the engine experiences two revolutions of crankshaft (four-stroke engine). Piston is back to TDC to begin a new cycle with closing of exhaust valve and opening of intake valve.
- Processes 5-6 & 6-1 cancel each other and do not contribute thermodynamically.



Otto Cycle

Thermodynamic cycle analysis

- **At maximum throttle opening condition, only CR is required to calculate the indicated thermal efficiency, commonly known as Otto cycle efficiency.**
- **Thermal efficiency increases with increase in CR.**

Process 6−1: $p_1 = p_6 = p_0$; $w_{6\text{-}1} = p_0(v_1 - v_6)$; Process 5−6: $p_5 = p_6 = p_0$; $w_{5\text{-}6} = p_0(v_6 - v_5) = p_0(v_6 - v_1)$

Process 1−2: $T_2 = T_1(v_1/v_2)^{k-1} = T_1(r_c)^{k-1}$; $p_2 = p_1(v_1/v_2)^k = p_1(r_c)^k$; $q_{1\text{-}2} = 0$; $w_{1\text{-}2} = c_v(T_1 - T_2)$

Process 2−3: $v_3 = v_2 = v_{TDC}$; $w_{2\text{-}3} = 0$; $q_{2\text{-}3} = q_{in} = c_v(T_3 - T_2)$; $T_3 = T_{max}$ & $p_3 = p_{max}$

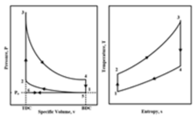
$Q_{2\text{-}3} = Q_{in} = m_f Q_{hv}\eta_c$; $Q_{2\text{-}3} = m_m c_v(T_3 - T_2) = (m_f + m_a)c_v(T_3 - T_2) \Rightarrow Q_{hv}\eta_c = (AF+1)c_v(T_3 - T_2)$

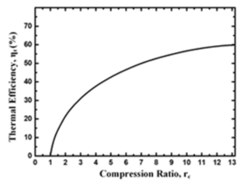
Process 3−4: $T_4 = T_3(v_3/v_4)^{k-1} = T_3(1/r_c)^{k-1}$; $p_4 = p_3(v_3/v_4)^k = p_3(1/r_c)^k$; $q_{3\text{-}4} = 0$; $w_{3\text{-}4} = c_v(T_3 - T_4)$

Process 4−5: $v_5 = v_4 = v_1 = v_{BDC}$; $w_{4\text{-}5} = 0$; $q_{4\text{-}5} = q_{out} = c_v(T_5 - T_4) = c_v(T_1 - T_4)$; $Q_{4\text{-}5} = Q_{out} = m_m c_v(T_1 - T_4)$

$$v_1 = v_4;\ v_2 = v_3;\ \frac{T_2}{T_1} = (v_1/v_2)^{k-1} = (v_4/v_3)^{k-1} = \frac{T_3}{T_4} \quad \Rightarrow \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

$$\eta_t = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1}{T_2};\ \eta_t = 1 - \frac{1}{r_c^{k-1}}$$

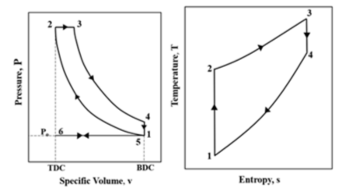




Diesel Cycle

It is the air-standard model of most four-stroke CI engines and involves following thermodynamic processes:

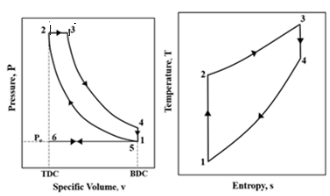
- **Intake stroke (Process 6-1): Intake valve open and exhaust valve closed**
- **Compression stroke (Process 1-2): All valves closed**
- **Heat addition (Process 2-3): All valves closed**
- **Power/expansion stroke (Process 3-4): All valves closed**
- **Heat rejection (Process 4-5): Exhaust valve open and intake valve closed**
- **Exhaust stroke (Process 5-6): Exhaust valve open and intake valve closed**



Diesel Cycle

Note:

- **Fuel is injected into the combustion chamber very late in the compression stroke and there is an ignition delay between fuel injection and combustion.**
- **Fuel "cutoff ratio" is defined as the change in volume occurring during combustion.**
- **Due to finite time requirement, combustion lasts into expansion stroke.**
- **Pressure is kept with peak level at TDC so that combustion process (2-3) can be approximated as "constant-pressure" process.**



Diesel Cycle

Thermodynamic cycle analysis

Process 6−1: $w_{6\text{-}1} = p_0(v_1 - v_6)$; Process 5−6: $w_{5\text{-}6} = p_0(v_6 - v_5) = p_0(v_6 - v_1)$

Process 1−2: $T_2 = T_1(v_1/v_2)^{k-1} = T_1(r_c)^{k-1}$; $p_2 = p_1(v_1/v_2)^k = p_1(r_c)^k$; $q_{1\text{-}2} = 0$; $w_{1\text{-}2} = c_v(T_1 - T_2)$; $V_2 = V_{TDC}$

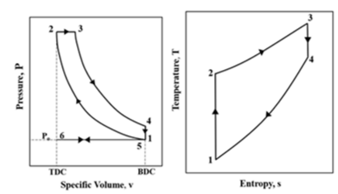
Process 2−3: $q_{2\text{-}3} = q_{in} = c_p(T_3 - T_2) = h_3 - h_2$; $w_{2\text{-}3} = q_{2\text{-}3} - (u_3 - u_2) = p_2(v_3 - v_2)$; $T_3 = T_{max}$; $\beta = \frac{V_3}{V_2} = \frac{v_3}{v_2} = \frac{T_3}{T_2}$

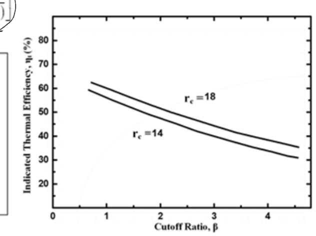
$Q_{2\text{-}3} = Q_{in} = m_f Q_{hv}\eta_c$; $Q_{2\text{-}3} = m_m c_p(T_3 - T_2) = (m_f + m_a)c_p(T_3 - T_2) \Rightarrow Q_{hv}\eta_c = (AF+1)c_p(T_3 - T_2)$

Process 3−4: $T_4 = T_3(v_3/v_4)^{k-1}$; $p_4 = p_3(v_3/v_4)^k$; $q_{3\text{-}4} = 0$; $w_{3\text{-}4} = c_v(T_3 - T_4)$

Process 4−5: $v_5 = v_4 = v_1 = v_{BDC}$; $w_{4\text{-}5} = 0$; $q_{4\text{-}5} = q_{out} = c_v(T_5 - T_4) = c_v(T_1 - T_4)$; $Q_{4\text{-}5} = Q_{out} = m_m c_v(T_1 - T_4)$

$$\eta_t = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)};\quad \eta_t = 1 - \frac{1}{r_c^{k-1}}\left[\frac{\beta^k - 1}{k(\beta - 1)}\right]$$

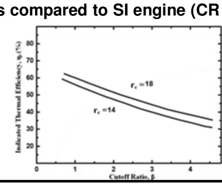




Diesel Cycle

Thermodynamic cycle analysis

- Only CR & fuel cut-off ratio are required to calculate the indicated thermal efficiency, commonly known as diesel cycle efficiency.
- Thermal efficiency increases with increase in CR and decreases with increase in fuel cutoff ratio.
- For a given CR, the indicated thermal efficiency of Otto cycle is higher than the diesel cycle.
- Constant-volume combustion at TDC is more efficient than constant-pressure combustion.
- CI engines operate with higher CR (12 to 24) as compared to SI engine (CR 8 to 11) and thus have higher thermal efficiency.



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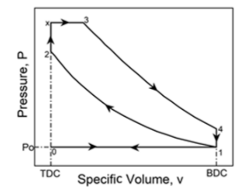
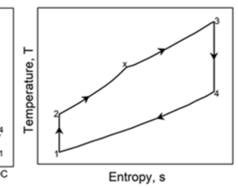
Dual Cycle

- Modern high speed CI engines accomplish dual mode of combustion process first by constant-volume process followed by constant-pressure.
- Fuel injection starts earlier in the cycle (20° bTDC) so that first fuel ignites late in compression stroke as that of Otto cycle followed by diesel cycle.
- The peak pressure still remains high into expansion stroke due to finite time required to inject the fuel.
- The last part of fuel injected at TDC so that combustion of this fuel keeps pressure high into the expansion stroke.
- This concept of combustion process analysis is known as dual cycle or limited pressure cycle. Many a times it is also called as modified Otto cycle with limited upper pressure.





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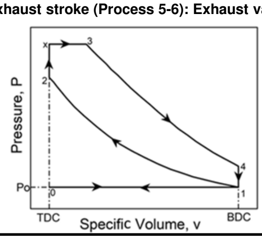
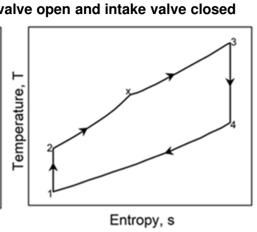
Dual Cycle

It is the air-standard model of most four-stroke high-speed CI engines and involves following thermodynamic processes:

- Intake stroke (Process 6-1): Intake valve open and exhaust valve closed
- Compression stroke (Process 1-2): All valves closed
- Heat addition (Process 2-x): All valves closed, Combustion Part I ✓
- Heat addition (Process x-3): All valves closed, Combustion Part II ✓
- Power/expansion stroke (Process 3-4): All valves closed
- Heat rejection (Process 4-5): Exhaust valve open and intake valve closed
- Exhaust stroke (Process 5-6): Exhaust valve open and intake valve closed





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Dual Cycle

Thermodynamic cycle analysis

Process 6−1: $w_{6\text{-}1} = p_0(v_1 - v_6)$; Process 5−6: $w_{5-6} = p_0(v_6 - v_5) = p_0(v_6 - v_1)$

Process 1−2: $T_2 = T_1(v_1/v_2)^{k-1} = T_1(r_c)^{k-1}$; $p_2 = p_1(v_1/v_2)^k = p_1(r_c)^k$; $q_{1-2} = 0$; $w_{1-2} = c_v(T_1 - T_2)$; $V_2 = V_{TDC}$

Process 2−x: $V_x = V_2 = V_{TDC}$; $w_{2-x} = 0$; $q_{2-x} = c_v(T_x - T_2) = u_x - u_2$; $Q_{2-x} = m_m c_v(T_x - T_2) = (m_f + m_a)c_v(T_x - T_2)$;

$$p_x = p_{\max} = p_2\left(\frac{T_x}{T_2}\right);\quad \alpha = \frac{p_x}{p_2} = \frac{p_3}{p_2} = \frac{T_x}{T_2} = \left(\frac{1}{r_c^k}\right)\left(\frac{p_3}{p_1}\right)$$ ✓

Process x−3: $p_x = p_3 = p_{\max}$; $T_3 = p_{\max}$; $q_{x-3} = c_p(T_3 - T_x) = h_3 - h_x$; $Q_{x-3} = m_m c_p(T_3 - T_x) = (m_f + m_a)c_p(T_3 - T_x)$

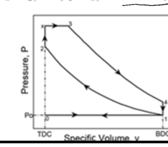
$$w_{x-3} = q_{x-3} - (u_3 - u_x) = p_x(v_3 - v_x) = p_3(v_3 - v_x);\quad \beta = \frac{V_3}{V_2} = \frac{v_3}{v_2} = \frac{v_3}{v_x} = \frac{T_3}{T_x}$$

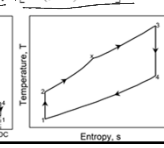
$Q_{in} = Q_{2-x} + Q_{x-3} = m_f Q_{hv}\eta_c$; $q_{in} = q_{2-x} + q_{x-3} = (u_x - u_2) + (h_3 - h_2)$

Proess 3−4: $T_4 = T_3(v_3/v_4)^{k-1}$; $p_4 = p_3(v_3/v_4)^k$; $q_{3-4} = 0$; $w_{3-4} = c_v(T_3 - T_4)$

Process 4−5: $v_5 = v_4 = v_1 = v_{BDC}$; $w_{4-5} = 0$; $q_{4-5} = q_{out} = c_v(T_5 - T_4) = c_v(T_1 - T_4)$; $Q_{4-5} = Q_{out} = m_m c_v(T_1 - T_4)$

$$\eta_t = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{(T_4 - T_1)}{(T_x - T_2) + k(T_3 - T_x)};\quad \eta_t = \left(1 - \frac{1}{r_c^{k-1}}\right)\left[\frac{\alpha\beta^k - 1}{k\alpha(\beta - 1) + \alpha - 1}\right]$$





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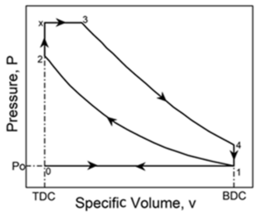
Dual Cycle

Thermodynamic cycle analysis

- Fuel "cutoff ratio" is defined as the change in volume occurring during combustion. "Pressure ratio" is defined as rise in pressure during combustion.
- The air standard thermal efficiency for CI engine obtained through Diesel cycle is slightly higher than Otto cycle.
- The real engine cycle has less indicated thermal efficiency with respect to its air standard efficiency of corresponding cycle. It is mainly because of changing composition, heat losses, valve overlap and finite time required for cycle process.

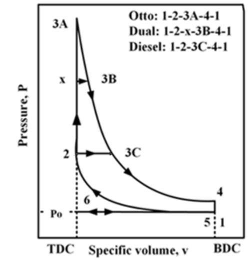


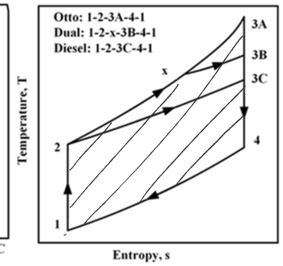
$$(\eta_t)_{actual} \approx 0.85(\eta_t)_{diesel}$$
$$(\eta_t)_{actual} \approx 0.85(\eta_t)_{dual}$$

Comparison of Air Standard Cycles

- Otto, Diesel and Dual cycles are compared with same inlet conditions and same compression ratio in p-v and T-s diagram.
- The area under T-s diagram is equal to heat transfer. For each cycle, the heat rejection is same but heat input is different.

$$\eta_t = 1 - \frac{q_{out}}{q_{in}};\ (\eta_t)_{Otto} > (\eta_t)_{Dual} > (\eta_t)_{Diesel}$$

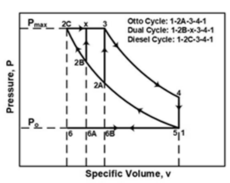


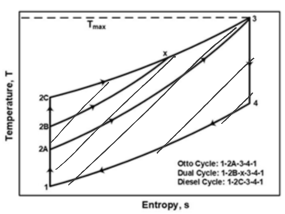


Comparison of Air Standard Cycles

- The SI engines and CI engines are normally operated with different CRs. The realistic way of Otto, Diesel and Dual cycle comparison is to consider same inlet conditions and same peak pressure.
- The area under T-s diagram is equal to heat transfer. For each cycle, the heat rejection is same but heat input is different.
- Thus, thermodynamically, the most efficient engine would have combustion as close as possible to constant volume but it would be CI engine operating at higher CRs.

$$(\eta_t)_{Diesel} > (\eta_t)_{Dual} > (\eta_t)_{Otto}$$

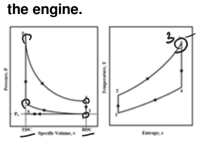




Numerical Problems

Q1. A four-cylinder, 2.5 litre SI engine with compression ratio of 8.6, operates on Otto cycle. At the start of compression stroke, the fresh charge is at condition of 100 kPa and 60°C. The engine uses isooctane as fuel with air-fuel ratio 15 and heating value of 44 MJ/kg with combustion efficiency of 97%. Carry out complete thermodynamic analysis of the engine.

$C_p = 1.108$ kJ/kg.K $\quad C_v = 0.821$ kJ/kg.K $\quad k = 1.35$



$V_d = \frac{2.5}{4} = 0.625\ L = 0.000625\ m^3.$

$r_c = 8.6 = \frac{V_d + V_c}{V_c} \Rightarrow V_c = 0.0000822\ m^3.$

$V_1 = V_d + V_c = 0.000707\ m^3.$, $p_1 = 100$ kPa $\quad T = 60°C.$

$m_m = \frac{pV}{RT} = 0.00074$ kg. $(m_f + m_a).$

$m_a = \frac{15}{16} m_m = 0.00069$ kg. $\quad m_f = \frac{1}{16} m_m = 0.00005$ kg.

State 2

$p_2 = p_1 (r_c)^k = 1826$ kPa

$T_2 = T_1 (r_c)^{k-1} = 707$ K.

State 3

$Q_{in} = m_f Q_{cv} \eta_c = m_m (c_v)(T_3 - T_2)$

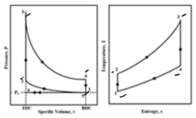
$Q_{cv} = 44$ MJ/kg. $\Rightarrow T_3 = 3915$ K.

$\eta_c = 97\%$ $\quad V_3 = V_2$

$p_3 = p_2\left(\frac{T_3}{T_2}\right) = 10111$ kPa

Numerical Problems

Q1. A four-cylinder, 2.5 litre SI engine with compression ratio of 8.6, operates on Otto cycle. At the start of compression stroke, the fresh charge is at condition of 100 kPa and 60°C. The engine uses isooctane as fuel with air-fuel ratio 15 and heating value of 44 MJ/kg with combustion efficiency of 97%. Carry out complete thermodynamic analysis of the engine.



State-4

$T_4 = T_3 \left(\frac{1}{r_c}\right)^{k-1} = 1844\ K.$

$p_4 = p_3 \left(\frac{1}{r_c}\right)^{k} = 554\ kPa.$

$V_4 = \frac{m R T_4}{p_4} = 0.000707\ m^3.$

$W_{34} = \frac{mR(T_3 - T_4)}{k-1} = 1.257\ kJ.$

$W_{12} = \frac{mR(T_2 - T_1)}{k-1} = 0.227\ kJ$

W_{net}

$Q_{in} = m_f Q_{cv} \eta_c = 2.134\ kJ.$

$\eta_t = \frac{W_{net}}{Q_{in}} = \frac{1.03}{2.134} = 48\%$

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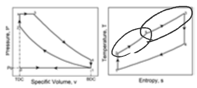
Numerical Problems

Q2. A four-cylinder, 4 litre truck engine operates on dual cycle with air-fuel ratio of 18. The compression ratio is 16 and the cylinder bore diameter is 100 mm. At the start of compression stroke, the fresh charge is at condition of 100 kPa and 60°C. It can be assumed that half of the heat input from combustion is added at constant volume and the other half at constant pressure. Calculate, the pressure and temperature at each state of the cycle, indicated thermal efficiency and exhaust temperature.

$c_p = 1.108\ kJ/kg\cdot K$

$c_v = 0.821\ kJ/kg\cdot K$

$k = 1.35$



Soln

$V_d = \frac{4}{4} = 1\ L. = 0.001\ m^3.$

$r_c = 16 \Rightarrow V_c = 0.0000667\ m^3.$

$V_1 = V_d + V_c.$ $p_1 = 100\ kPa$ $T_1 = 60^\circ C.$

$m_m = \frac{pV}{RT} = 0.00112\ kg.,$ $AF = 18.$

$m_f = 0.0000578\ kg.$

State 2

1-2

$T_2 = 879\ K$

$p_2 = 4222\ K.$

$Q_{in} = m_f Q_{cv} \eta_c = 2.46\ kJ.$

$\longrightarrow \frac{Q_{2x}}{1.23}$ & $\frac{Q_{x-3}}{1.23.}$

$Q_{x3} = m_m c_p (T_3 - T_2). \Rightarrow T_3 = 3208\ K.$

$V_3 = \frac{m_m R T_3}{p_3} = 0.000097\ m^3.$

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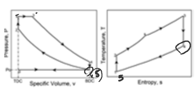
Numerical Problems

Q2. A four-cylinder, 4 litre truck engine operates on dual cycle with air-fuel ratio of 18. The compression ratio is 16 and the cylinder bore diameter is 100 mm. At the start of compression stroke, the fresh charge is at condition of 100 kPa and 60°C. It can be assumed that half of the heat input from combustion is added at constant volume and the other half at constant pressure. Calculate, the pressure and temperature at each state of the cycle, indicated thermal efficiency and exhaust temperature.

$Q_{cv} = 42.5\ MJ/kg$

$\eta_c = 97\%$



$Q_{2x} = m_m c_v (T_x - T_2) \Rightarrow T_x = 2217\ K.$

$p_x = \frac{m R T_x}{V_x} = 10650\ kPa.$

State-4

$V_4 = V_1 = 0.0010667\ m^3.$

$T_4 = T_3 \left(\frac{V_3}{V_1}\right)^{k-1} = 1386\ kPa$

$p_4 = p_3 \left(\frac{V_3}{V_4}\right)^{k} = 418\ kPa.$

State-5

$T_5 = T_4 \left(\frac{p_5}{p_4}\right)^{\frac{k-1}{k}}$

$\Rightarrow T_5 = 957\ K$

$W_{net} = W_{x3} + W_{34} - W_{12}$

$W_x = p(V_3 - V_x)$

$W_{34} = \frac{mR(T_4 - T_3)}{1-k}$

$W_{12} = \frac{mR(T_1 - T_2)}{1-k}$

$W_{net} = 1.495\ kJ.$

$Q_{in} = m_m Q_{cv} \eta_c = 2.46\ kJ.$

$\eta_t = 60.7\%.$

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THANK YOU

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