APPLIED THERMODYNAMICS

Internal Combustion Engines (Module III)

Prof. Niranjan Sahoo
Department of Mechanical Engineering
Indian Institute of Technology Guwahati

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

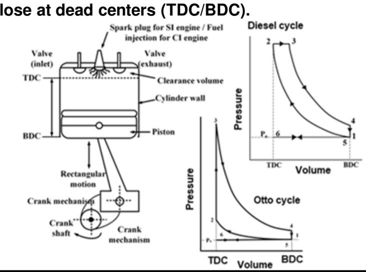
Lecture 5

Valve Timing Diagram and Fuel – Air Cycle

- Valve Timing Diagram (SI Engine & CI Engine)
- Fuel – Air Cycle
  - Significance
  - Composition of cylinder Gas
  - Variation of specific heat
  - Effects of dissociation
  - Characteristics curves

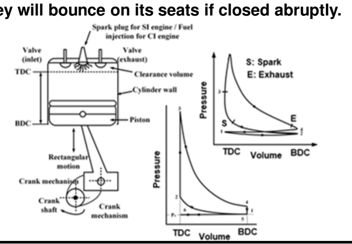
Valve Timing Diagram

- One of the prime deviation of fuel-air cycle with respect to air-standard cycle is the time of opening and closing the valves for SI and CI engines. The fuel-air mixture as fresh charges are inducted in SI engines with combustion initiated by spark. For CI engines, the combustion is initiated by auto-ignition and fuel injection controls the ignition.
- The theoretical p-v diagram for Otto/Diesel cycles have sharp edges i.e. the valves open/close at dead centers (TDC/BDC).



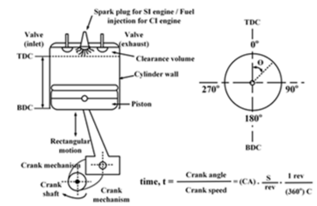
Valve Timing Diagram

- In actual scenario, every corner of *p-v* diagram is rounded which means there is finite time required for opening/closing of valves. Moreover, they need to be opened/closed, before/after the dead centers (TDC/BDC) because of mechanical and dynamic factors of gas flows.
- Typically, a poppet valve/mushroom valve is used in engines that allows a fixed quantity of charge into the engines.
- The valves are opened/closed by cam-mechanism and spreads over certain crank angle. They will bounce on its seats if closed abruptly.



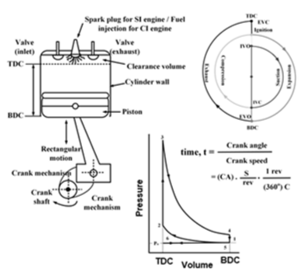
Valve Timing Diagram

- The *valve-timing* is the precise timing of opening and closing of valves for fresh charge induction and combustion products coming out of engine.
- The diagram represents the angular crank positions in terms of crank angle with respect to TDC and BDC of the piston.
- The crank angle displacements can be translated into time values if the rotational speed of the engine is known and can be assumed to be constant.
- The time (s/°CA) of opening or closing can be defined as ratio of crank angle (CA) to the crank speed. One revolution refers to 360°CA.



Valve Timing Diagram

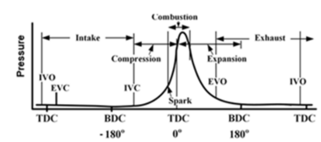
- The time required to open and close the valve means that each valve will be fully open for a crank angle movement much less than that indicated in the valve-timing diagram.
- Due to configuration of crank-connecting rod geometry, the considerable amount of crank movement only produces only small movement of piston at the dead center positions (TDC/BDC).



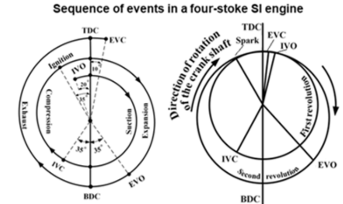
Valve Timing Diagram – SI Engines

The points on *valve-timing* diagram for a four-stroke petrol engine are indicated as follows:

- IVO: Inlet valve opens with actual position 10°bTDC and 15°aTDC.
- IVC: Inlet valve closes with actual position as 20-40°aBDC to take the advantage of momentum of rapidly moving gas, inducting into the engine. This phenomena is known as "ram effect" and it increases the volumetric efficiency of the engine.



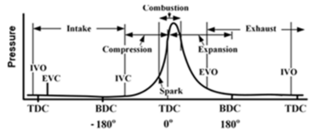
Sequence of events in a four-stoke SI engine



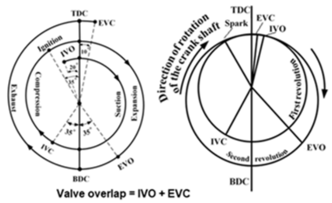
Valve overlap = IVO + EVC

Valve Timing Diagram – SI Engines

- S: Spark occurs 20-40°bTDC, when the ignition is fully advanced. At TDC, the ignition is fully retarded. This allows time delay between spark and commencement of combustion.
- EVO: Exhaust valve opens with its position 50°bBDC or else the pressure will rise enormously. Hence, work required to expel the gas during exhaust blow down will increase significantly.
- EVC: Exhaust valve closes within 10°aTDC to increase volumetric efficiency.



Sequence of events in a four-stroke SI engine

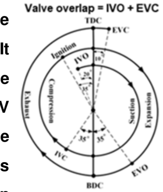


Valve overlap = IVO + EVC

Valve Timing Diagram – SI Engines

Valve overlap: The EV opens bBDC so as to expel the combustion products efficiently. By virtue of excess pressure (above atmospheric), some exhaust gas leaves the cylinder. It makes exhaust gas to flow freely from the cylinder by the time piston commences the exhaust stroke. Again by closing EV after TDC, the kinetic energy of the combustible gases can be utilized to assist in maximizing exhaust of gases. In this process, IV begins to open before EV closes. This is known as "valve overlap" i.e. time during which both the valve remains open (typically on average 30-40°CA).

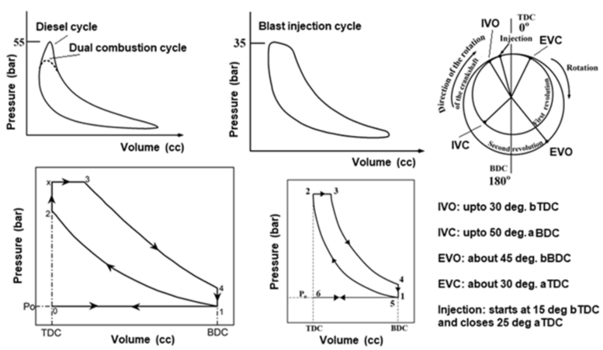
Valve overlap = IVO + EVC



Valve	Engines	Open	Close	Duration	Valve overlap (IVO + EVC)
Inlet	Conventional	5°bTDC (IVO)	45°aBDC (IVC)	230°CA	15°CA (Conventional engines) 65°CA (High performance engines)
	High performance	30°bTDC (IVO)	75°aBDC (IVC)	285°CA	
Exhaust	Conventional	45°bBDC(EVO)	10°aTDC (EVC)	235°CA	
	High performance	70°bBDC (EVO)	35°aTDC (EVC)	285°CA	

Valve Timing Diagram – CI Engines

- The ideal cycle for CI engines is the diesel cycle or a dual cycle.
- Fuels to CI engines can be injected by means of air blast to maintain constant pressure at early part of return stroke.



IVO: upto 30 deg. bTDC

IVC: upto 50 deg. aBDC

EVO: about 45 deg. bBDC

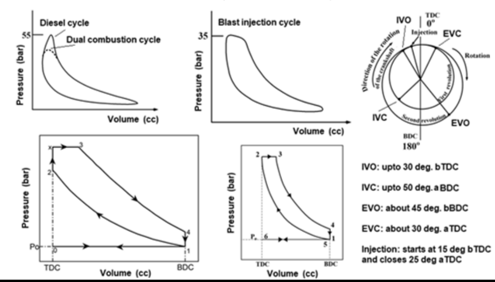
EVC: about 30 deg. aTDC

Injection: starts at 15 deg bTDC and closes 25 deg aTDC

Valve Timing Diagram – CI Engines

- Advanced CI engines use dual combustion cycle in which the heat input is predominantly at constant volume followed by constant pressure.
- A typical valve-timing diagram for a four-stroke CI engine has average values of valve position as follows:

IVO: Up to 30°bTDC; IVC: Up to 50°aBDC; EVO: About 45°bBDC;
EVC: About 30°aTDC; Fuel Injection: during 15°bTDC and 25°aTDC



IVO: upto 30 deg. bTDC

IVC: upto 50 deg. aBDC

EVO: about 45 deg. bBDC

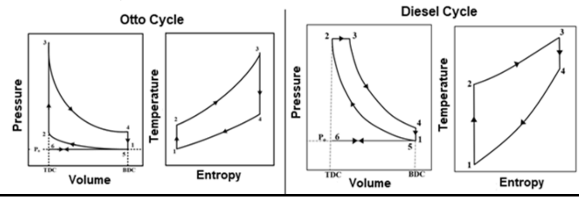
EVC: about 30 deg. aTDC

Injection: starts at 15 deg bTDC and closes 25 deg aTDC

Fuel Air Cycles

Assumptions (Air standard cycle)

- There is no chemical change in either fuel or air prior to the combustion and the charge is always in chemical equilibrium subsequent to combustion.
- There is no heat exchange between the gases and the cylinder walls in any process (i.e. adiabatic).
- The compression and expansion processes are frictionless.
- With respect to constant-volume fuel-air cycle, the fuel is completely vaporized and perfectly mixed with air. The burning takes place instantaneously at TDC (at constant volume).



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Fuel Air Cycles

Significance

- The air-standard cycle analysis rely on simplified assumptions for which the estimated engine performance is always on higher side as compared to actual performance.
- In actual engines, the working fluid is a mixture of air, fuel vapour and residual gases from previous cycle.
- The specific heats of working fluid are not constant but increases with temperature.
- The products of combustion are subjected to certain dissociation at high temperature.
- The actual pressure and temperature within the engine cylinder are different.

A realistic approach is to consider the analysis based on the actual properties of working medium (i.e. fuel and air) is called as "fuel-air cycle".

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Fuel Air Cycles

Significance

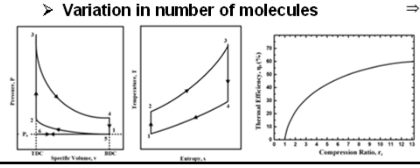
- An air-standard cycle can bring out the effect of compression ratio on indicated thermal efficiency but it will not justify the effect of air-fuel ratio on thermal efficiency.
- A fuel-air cycle can bring out the effect of fuel-air ratio on thermal efficiency and can study the effect of pressure and temperature. It considers the following aspects:
  - Actual composition of cylinder gases
  - Variation of specific heat with temperature
  - Effect of dissociation
  - Variation in number of molecules

$$q_{2-3} = q_{in} = c_v\left(T_3 - T_2\right);\ \eta_t = 1 - \frac{1}{r_c^{k-1}}$$

$$Q_{2-3} = Q_{in} = m_f Q_{in} \eta_c;$$

$$Q_{2-3} = m_m c_v\left(T_3 - T_2\right) = \left(m_f + m_a\right) c_v\left(T_3 - T_2\right)$$

$$\Rightarrow Q_{in}\eta_c = \left(AF + 1\right) c_v\left(T_3 - T_2\right)$$



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Fuel Air Cycles

Composition of cylinder gases

- The cylinder gases contains fuel, air, water vapour and residual gases.
- The fuel-air ratio varies during operation of engine that changes relative to amount of CO_2 and water vapours.
- The change in air-fuel ratio affects the composition of gases before and after combustion particularly for percentage carbon dioxide, carbon monoxide, water vapours in exhaust gases.
- In four-stroke engines, when the fresh charges enters the cylinder, it comes in contact with trapped burnt gases within the clearance space of previous cycle.
- The fuel-air cycle takes into account of the trapped gases and analyzes the effect of cylinder composition on performance of engine through suitable numerical simulation.

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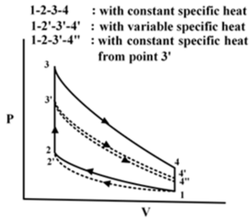
Fuel Air Cycles

Variation of specific heat in the cylinder

- All the gases show increase in specific heat with temperature particularly in the range 300-2000K.
- The behavior is linear up to 1500K beyond which the specific heat increases more rapidly with temperature.
- The net effect is difference in both specific heat remains constant but specific heat ratio increases with temperature.



$$c_p = a_1 + k_1 T;\ c_v = b_1 + k_1 T;\ R = c_p - c_v = a_1 - b_1$$

$$c_p = a_1 + k_1 T + k_2 T^2;\ c_v = b_1 + k_1 T + k_2 T^2$$

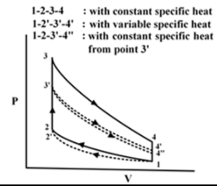
$$\text{At } 300\,\text{K},\ c_p = 1.005\ \text{kJ/kg.K};\ c_v = 0.717\ \text{kJ/kg.K}$$

$$\text{At } 2000\,\text{K},\ c_p = 1.345\ \text{kJ/kg.K};\ c_v = 1.057\ \text{kJ/kg.K}$$

Fuel Air Cycles

Variation of specific heat in the cylinder

- When the variation in specific heat is taken into account for compression stroke, the final temperature and pressure would be lower than if constant specific heat are considered (process 1-2' instead of 1-2)
- Similarly, for expansion stoke, the final temperature after expansion will be higher for variable specific heat case (process 3-4' instead of 3-4'')
- For given initial conditions, the constant volume combustion will lower the peak temperature and pressure of the cycle (process 2'-3' instead of 2-3)
- There is always loss of power due to change in specific heats.



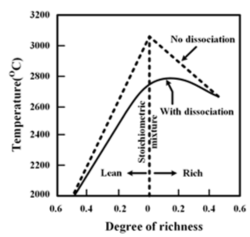
$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma-1};\ T_{2'} = T_1\left(\frac{v_1}{v_{2'}}\right)^{k-1};\ r_c = \frac{v_1}{v_2} = \frac{v_1}{v_{2'}}$$

$$T_{4''} = T_{3'}\left(\frac{v_3}{v_4}\right)^{k-1};\ T_{4'} = T_{3'}\left(\frac{v_1}{v_{2'}}\right)^{\gamma-1};\ \gamma = 1.4\ \&\ c_p - c_v = k$$

Fuel Air Cycles

Effect of dissociation

- The dissociation process is the disintegration of combustion products at high temperature and considered as reverse process of combustion in which heat is absorbed.
- The maximum extent of dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperature are expected to be high but decreases with lean and rich mixture.

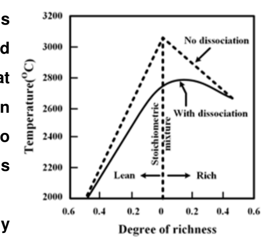


Fuel Air Cycles

Effect of dissociation

- The heat transfer to the cooling medium causes reduction in maximum temperature and pressure. As the temperature falls during expansion stroke, the separated constituents recombine – the heat absorbed during dissociation is again released, but it is too late in the stroke to recover the lost power. A portion of this heat is carried away by exhaust gases.
- In IC engines, the dissociation mainly occurs for carbon dioxide (above 1000°C) and little effect for water (above 1300°C)

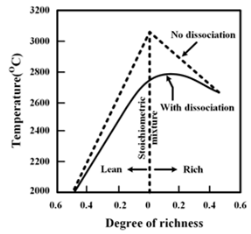
$CO_2 \leftrightarrow 2CO + O_2$ $H_2O \leftrightarrow 2H_2 + O_2$



Fuel Air Cycles

Effect of dissociation

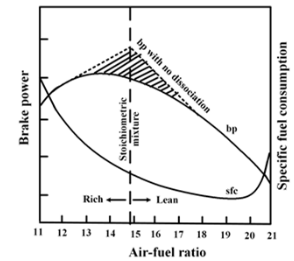
- The first effect of dissociation is the reduction in temperature of exhaust gas mixtures with respect to air-fuel ratio.
- The dissociation could reduce the maximum temperature by about 300°C even at chemically correct air-fuel ratio.
- When the air-fuel ratio decreases (fuel rich), the maximum temperature rises and maximum dissociation occurs at chemically correct mixture strength. The dissociation effect tends to decline due to incomplete combustion for rich mixture.



Fuel Air Cycles

Effect of dissociation

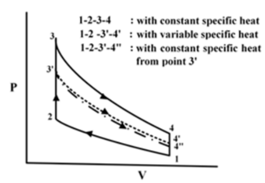
- The second effect of dissociation is the reduction brake power for a four-stroke SI engine.
- If there is no dissociation, the brake power output is maximum when the mixture is stoichiometric. With lean mixture, the dissociation is insignificant.
- The dissociation effect are not pronounced in CI engines due to presence of heterogeneous mixture and excess air availability for combustion.



Fuel Air Cycles

Effect of number of moles

- The number of molecules present in the cylinder after combustion depends on fuel-air ratio, type and extent of reaction in the cylinder.
- The pressure depends on number of moles present in the cylinder. It has direct effect on the amount of work, the cylinder gas can impact on the piston leading to loss of power/efficiency.
- Because of lower maximum temperature due to dissociation, the maximum pressure is reduced to 3' and the expansion process is indicated as 3'-4''. If there is re-association, the expansion process is indicated as 3'-4'.



Fuel Air Cycle

Characteristics curves:

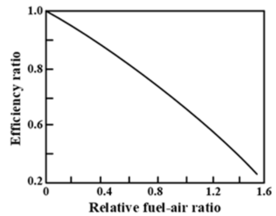
- The magnitude of difference between air-standard cycle and fuel-air cycle arises due to three major factors:
  - ➢ Character of the cycle
  - ➢ Relative air fuel ratio (Ratio of actual F/A to stoichiometric F/A)
  - ➢ Chemical composition of the fuel
- There are two important concerns due to the above factors:
  - ➢ Effect of relative fuel-air ratio on efficiency ratio (i.e. the ratio between fuel-air cycle efficiency to the air-standard cycle efficiency)
  - ➢ Effect of specific fuel consumption with respect to mean effective pressure

Fuel Air Cycle

Characteristics curves

1. Effect of relative fuel-air ratio on efficiency ratio at constant speed and throttle setting

- The efficiency ratio increases as the mixture becomes lean and tend to approach towards air-standard efficiency.
- At low F/A ratio, the mixture behaves like perfect gas with constant specific heat. Since both pressure and temperature increases, the chemical reactions tend to complete.

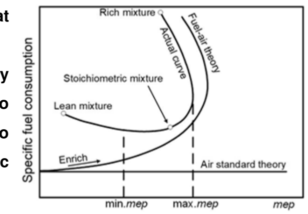


Fuel Air Cycle

Characteristics curves

2. Effect of specific fuel consumption with respect to mean effective pressure at constant speed and throttle setting

- For fuel-air cycle, the thermal efficiency deteriorates as the mixture supplied to the engine is enriched. It is mainly due to increase in losses due to variable specific heats and dissociation.
- At the same time, beyond certain leaning of mixture, the combustion becomes erratic with loss in efficiency.
- The maximum efficiency is in the lean zone closure to stoichiometric ratio.



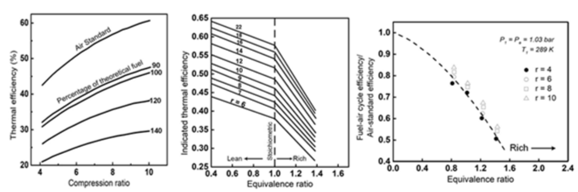
Fuel Air Cycle

Characteristics curves

3. The effect of common engine operating variables on pressure and temperature within the cylinder can be understood by fuel-air cycle analysis.

(a) Compression ratio (CR):

- The fuel-air cycle efficiency increases with CR in same manner as that of air standard efficiency.
- The maximum pressure and temperature increases with CR.
- The efficiency ratio is independent of CR for a given relative fuel-air ratio for constant volume cycle.

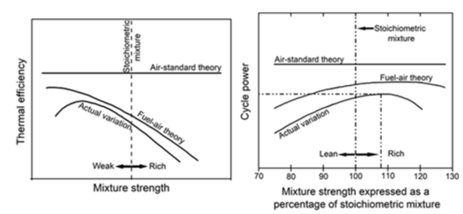


Engine Operating Variables

Characteristics curves

3 (b) Fuel-air ratio:

- When the mixture is made lean (less fuel), the temperature rise due to combustion, is lowered as result of reduced energy input per unit mass. It lowers the losses due to dissociation and variation in specific heat. The cycle efficiency approaches to air standard efficiency.
- At rich mixture, there is a drop in cycle efficiency and maximum power due to higher specific heat, dissociation and insufficient air.

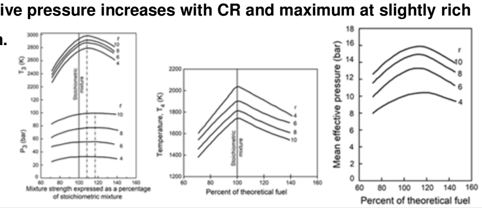


Engine Operating Variables

Characteristics curves

3 (b) Fuel-air ratio:

- **At a given CR, the maximum temperature after combustion reaches maximum when the mixture is slightly rich. The rich mixture causes more fuel to combine with oxygen. The maximum pressure curve follow similar trend but significantly start to decrease until mixture is significantly rich.**
- **The exhaust temperature is low at high CR because the increased expansion causes the gas to do more work on the piston.**
- **The mean effective pressure increases with CR and maximum at slightly rich mixture strength.**



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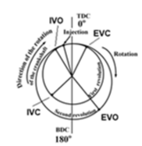
Numerical Problems

Q1. A CI engine operates with a variable valve-timing control mechanism. When the engine is idling at 400 rpm, the intake valve opens at 11°bTDC and closes at 17°aBDC while the exhaust valve opens at 20°bBDC and closes at 7°aTDC. At operational speed of 3500 rpm, the intake valve opens at 32°bTDC and closes at 57°aBDC while the exhaust valve opens at 52°bBDC and closes at 21°aTDC. Calculate, (i) valve overlap in crankcase degree for operational and idle speed; (ii) valve overlap in real time for operational and idle speed.



Soln Idle 400 rpm. (N)

IVO – 11° bTDC IVC : 17° aBDC

EVO – 20° bBDC EVC : 7° aTDC

VO = ∠IVO + ∠EVC = 18°

Operational speed 3500 rpm (N) → VO = 32° + 21° = 53° | °CA

(ii) $t = \frac{\text{Crank angle}}{\text{Crank speed}}$

400 rpm.

$t = \frac{\left(\frac{18}{360}\right) \text{rev}}{\left(\frac{400}{60}\right) \text{rev/s}} = 7.5 \text{ ms}$

At 3500 rpm

$t = \frac{(53/360) \text{ rev}}{(3500/60) \text{ rev/s}} = 2.5 \text{ ms}$

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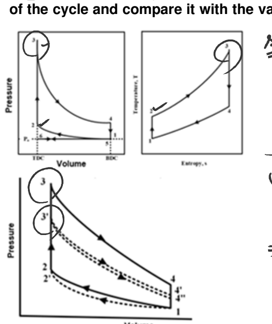
Numerical Problems

Q2. A petrol engine has a compression ratio of 6 and uses air-fuel ratio of 15. The pressure and temperature at the beginning of suction stroke is 1 bar and 57°C. The specific heat (kJ/kg-K) at constant volume for the working fluid varies with temperature, Cv = 0.678 + 0.00013 T, (T is in Kelvin). Calculate the maximum pressure and temperature of the cycle and compare it with the values obtained through fixed value of Cv.



Data.

$Q_{cv} = 42$ MJ/

$k = 1.35$

$c_v = 0.821$ kJ/kg

Soln $p_1 = 1$ bar $T_1 = 57°C = 330$ K

$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^k = 11.23$ bar

$T_2 = T_1 \left(\frac{p_2 V_2}{p_1 V_1}\right) = 330 \times 11.23 \times \frac{1}{6} = 618$ K

$(c_v)_m = 0.678 + 0.00013 \left(\frac{T_3 + T_2}{2}\right)$

$Q_{23} = (c_v)_m \, m_m \, (T_3 - T_2)$

$\Rightarrow 2800 = 0.72(T_3 - 618) + 0.00007(T_3^2 - 618^2)$

$T_3 = 3400$ K

$Q_{23} = \frac{42}{15} = 2.8$ MJ

$m_m = 1 + \frac{1}{15} = \frac{16}{15}$

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Numerical Problems

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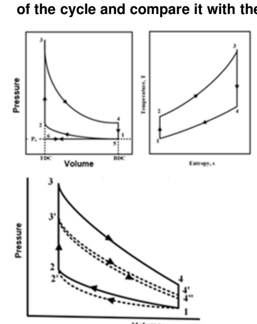

$c_v = f(T)$

$p_3 = p_2 \left(\frac{T_3}{T_2}\right)$ $T_3 = 3400$ K

$p_3 = 11.23 \left(\frac{3400}{618}\right) = 61.8$ bar

Const. sp. heat $c_v = 0.821$ kJ/kg.K

$Q_{23} = c_v (m_m)(T_3 - T_2)$

$\Rightarrow 2800 = 0.821 \times \frac{16}{15} (T_3 - 618)$

$\Rightarrow T_3 = 3815$ K

$p_3 = 11.23 \left(\frac{3815}{618}\right) = 69.3$ bar

c_v = Const.

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THANK YOU

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