ENG 3165 LECTURE 8

THERMODYNAMICS COMPONENT

Gas Power Cycles

Introduction

- In this lecture, the **Otto Cycle,** the **Diesel Cycle** and the **Dual Cycle** are discussed.
- An overview of the **Brayton cycle** is also given.
- **They are all based on the Carnot Engine** as the **ideal thermodynamic cycle**.

BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

- The cycles encountered in actual devices are difficult to analyze because of the presence of complicating effects, such as friction, and the absence of sufficient time for establishment of the equilibrium conditions during the cycle.
- When the actual cycle is stripped of all the internal irreversibilities and complexities, we end up with a cycle that resembles the actual cycle closely but is made up totally of internally reversible processes. Such a cycle is called an ideal cycle
- Recall that heat engines that operate on a totally reversible cycle, such as the Carnot cycle, have the highest thermal efficiency of all heat engines operating between the same temperature levels. That is, nobody can develop a cycle more efficient than the Carnot cycle

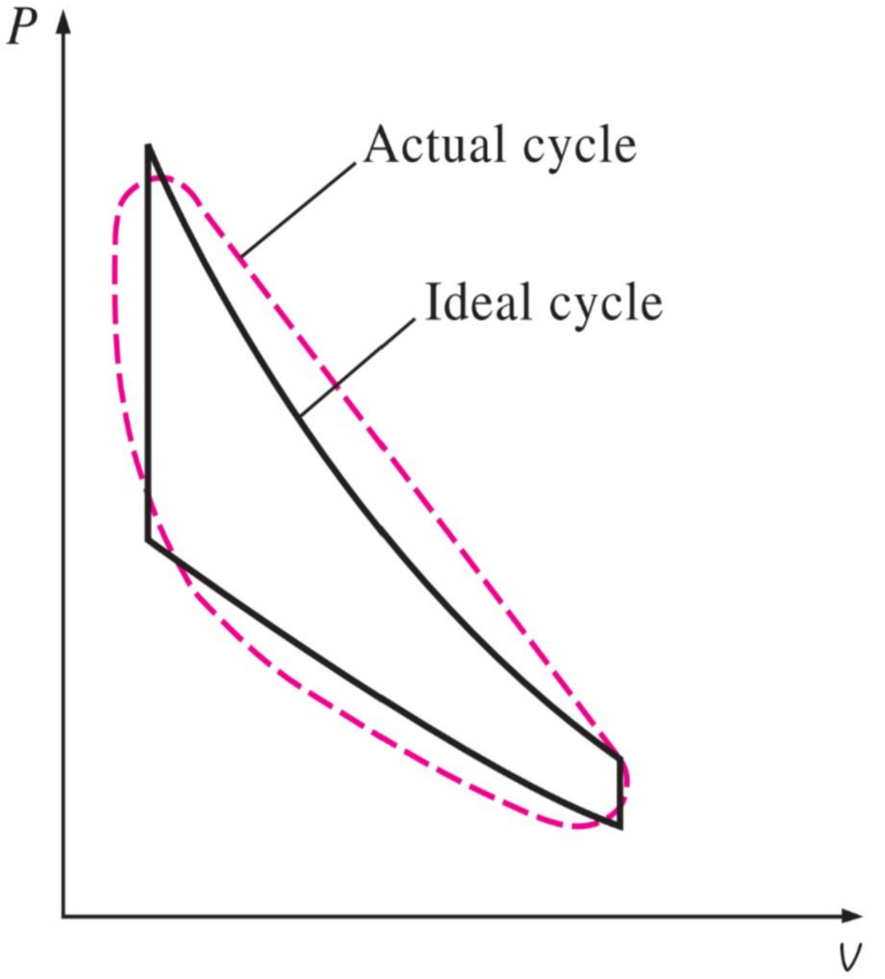
BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

- Most cycles encountered in practice differ significantly from the Carnot cycle, which makes it unsuitable as a realistic model. Each ideal cycle is related to a specific work-producing device and is an idealized version of the actual cycle.
- The ideal cycles are internally reversible, but, unlike the Carnot cycle, they are not necessarily externally reversible

THE IDEALIZATIONS AND SIMPLIFICATIONS COMMONLY EMPLOYED IN THE ANALYSIS OF POWER CYCLES CAN BE SUMMARIZED AS FOLLOWS:

- The cycle does not involve any friction. Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.
- All expansion and compression processes take place in a quasie-quilibrium manner.
- The pipes connecting the various components of a system are well insulated, and heat transfer through them is negligible.
- Neglecting the changes in kinetic and potential energies of the working fluid is another commonly utilized simplification in the analysis of power cycles. This is a reasonable assumption since in devices that involve shaft work, such as turbines, compressors, and pumps, the kinetic and potential energy terms are usually very small relative to the other terms in the energy equation.

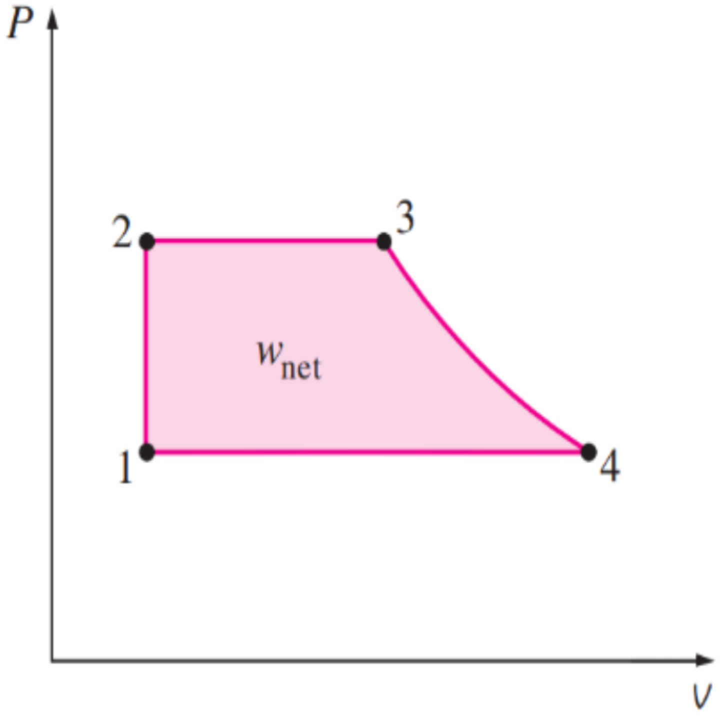
THE IDEALIZATIONS AND SIMPLIFICATIONS COMMONLY EMPLOYED IN THE ANALYSIS OF POWER CYCLES CAN BE SUMMARIZED AS FOLLOWS:

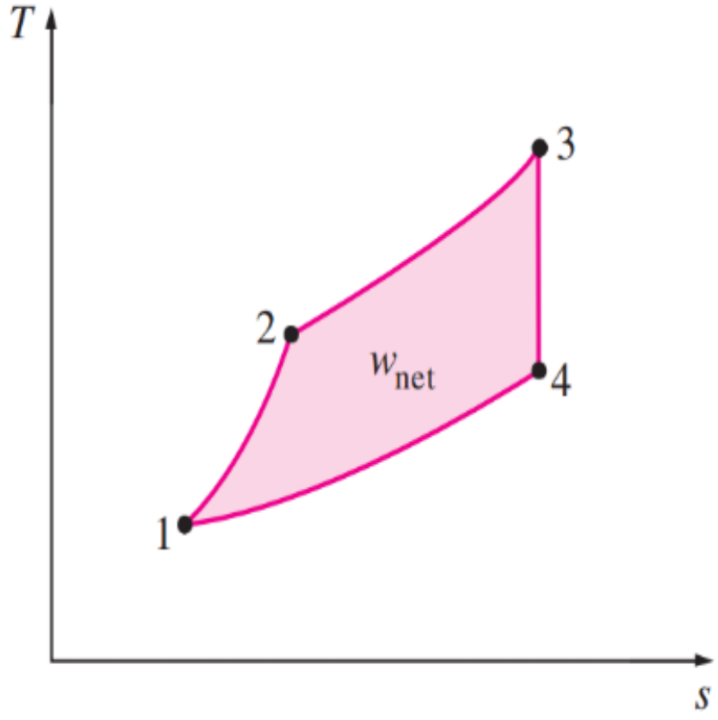


PROPERTY DIAGRAMS

- On both the P-v and T-s diagrams, the area enclosed by the process curves of a cycle represents the net work produced during the cycle, which is also equivalent to the net heat transfer for that cycle
- On a T-s diagram, a heat-addition process proceeds in the direction of increasing entropy, a heat-rejection process proceeds in the direction of decreasing entropy, and an isentropic (internally reversible, adiabatic) process proceeds at constant entropy

PROPERTY DIAGRAMS

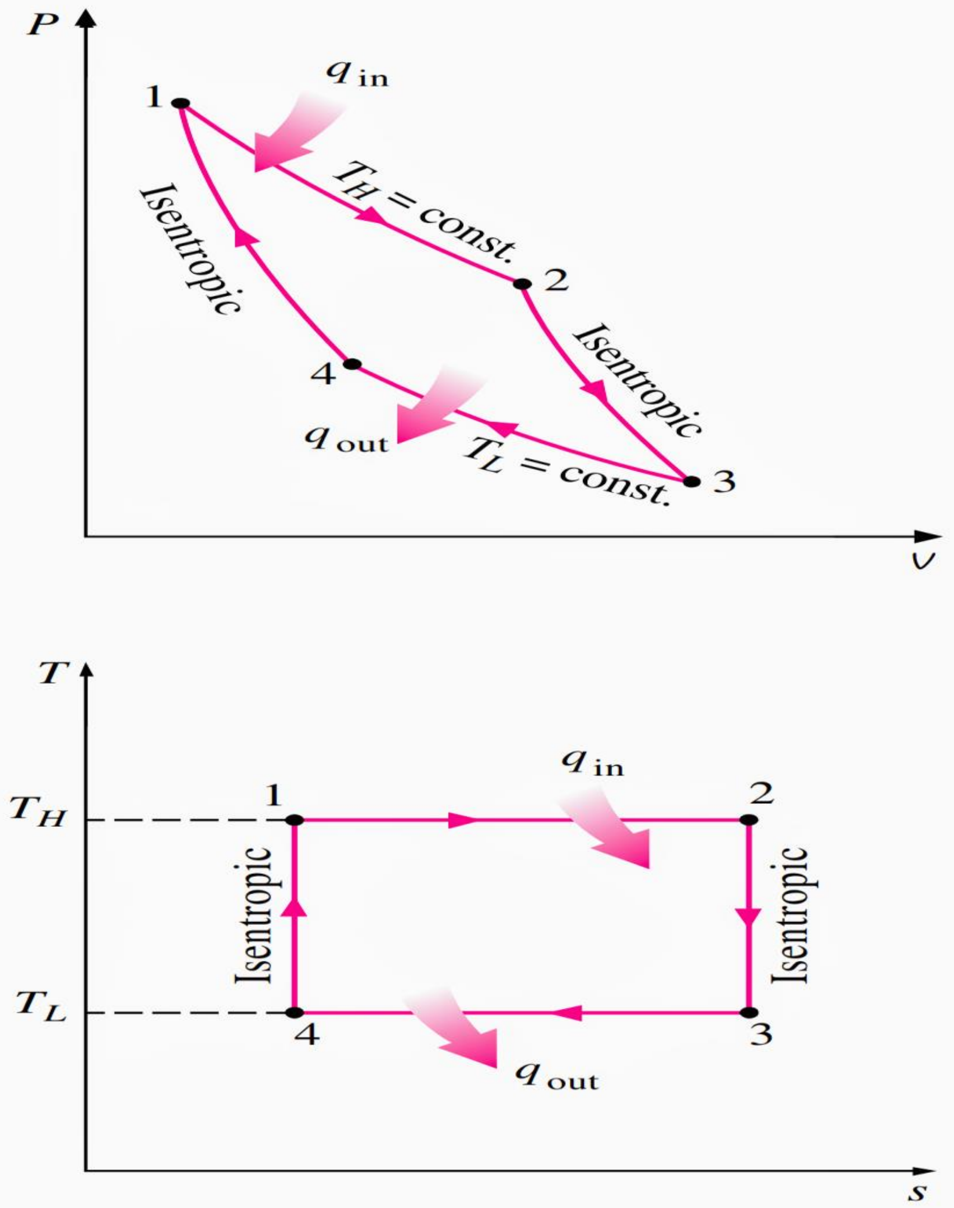




THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING

- The Carnot cycle is composed of four totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression
- The real value of the Carnot cycle comes from its being a standard against which the actual or the ideal cycles can be compared.
- *Thermal efficiency increases with an increase in the average temperature at which heat is supplied to the system or with a decrease in the average temperature at which heat is rejected from the system.*
- The source and sink temperatures that can be used in practice are not without limits, however. The highest temperature in the cycle is limited by the maximum temperature that the components of the heat engine, such as the piston or the turbine blades, can withstand. The lowest temperature is limited by the temperature of the cooling medium utilized in the cycle such as a lake, a river, or the atmospheric air.

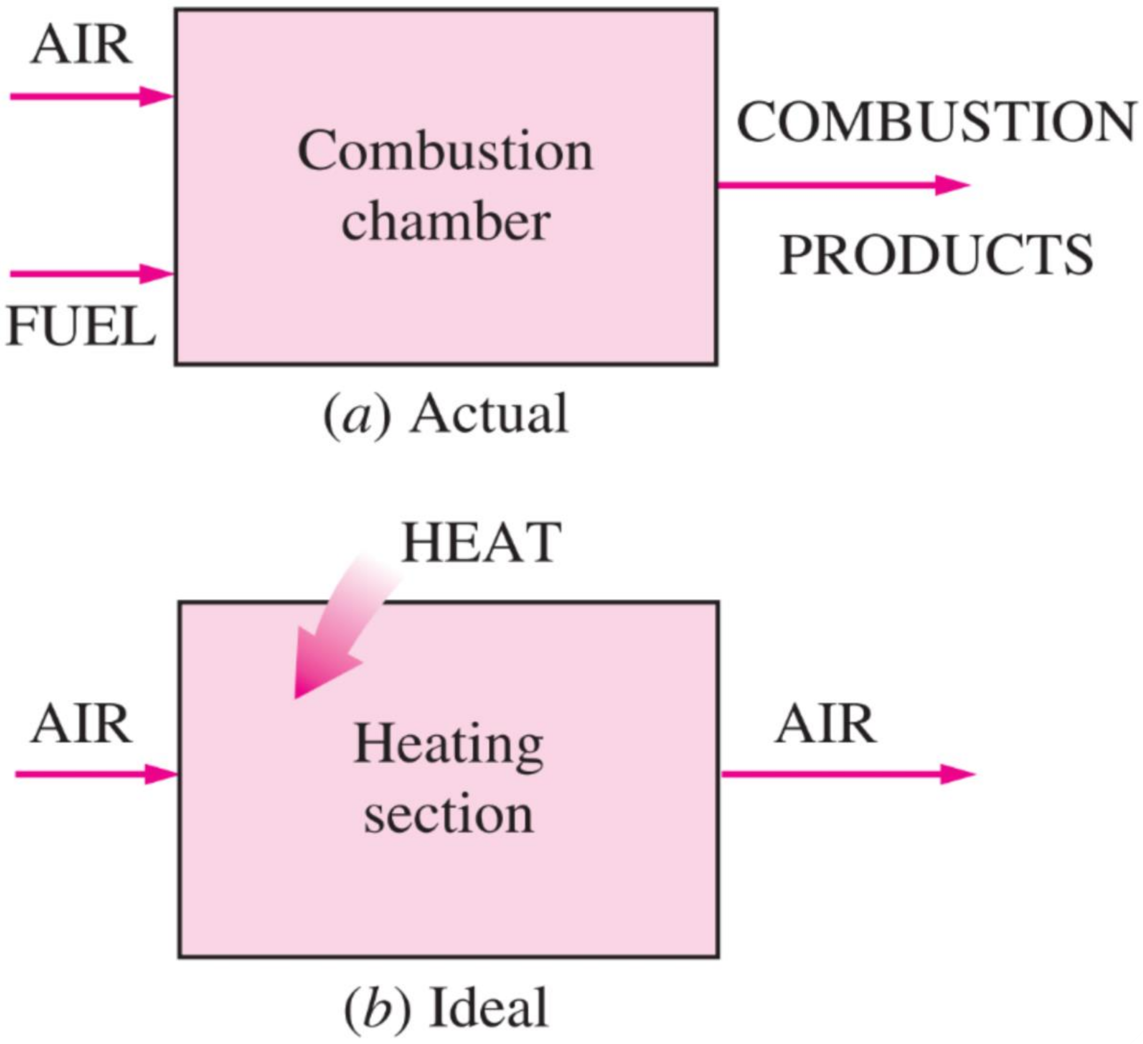
THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING



AIR-STANDARD ASSUMPTIONS

- In gas power cycles, the working fluid remains a gas throughout the entire cycle. Spark-ignition engines, diesel engines, and conventional gas turbines are familiar examples of devices that operate on gas cycles.
- In all these engines, energy is provided by burning a fuel within the system boundaries. That is, they are internal combustion engines.
- The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the ***air-standard assumptions***:
    - The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
    - All the processes that make up the cycle are internally reversible.
    - The combustion process is replaced by a heat-addition process from an external source
    - The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state. Another assumption that is often utilized to simplify the analysis even more is that air has constant specific heats

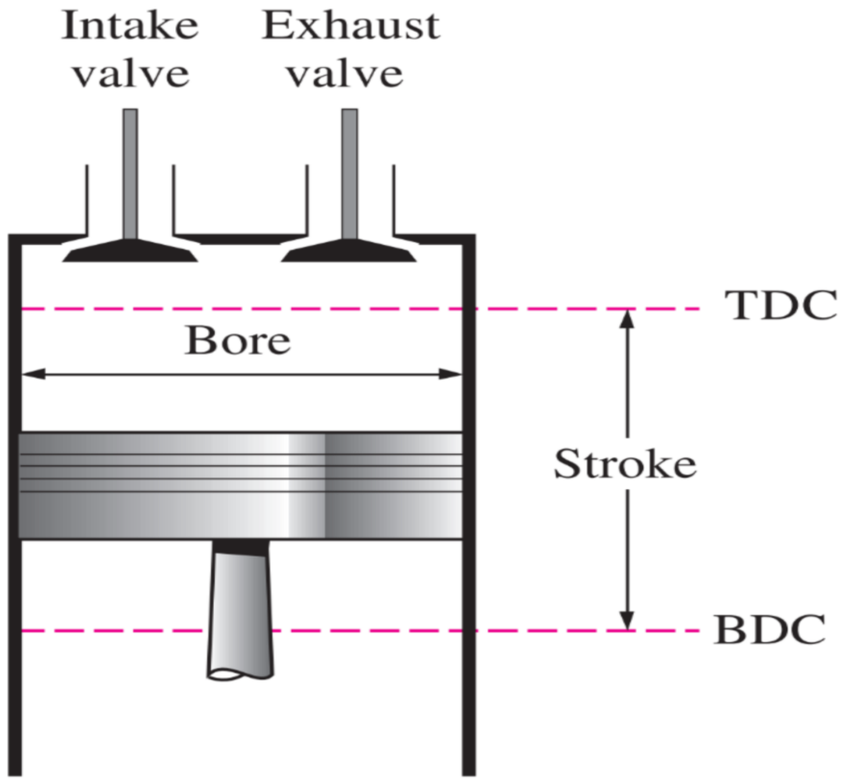
AIR-STANDARD ASSUMPTIONS

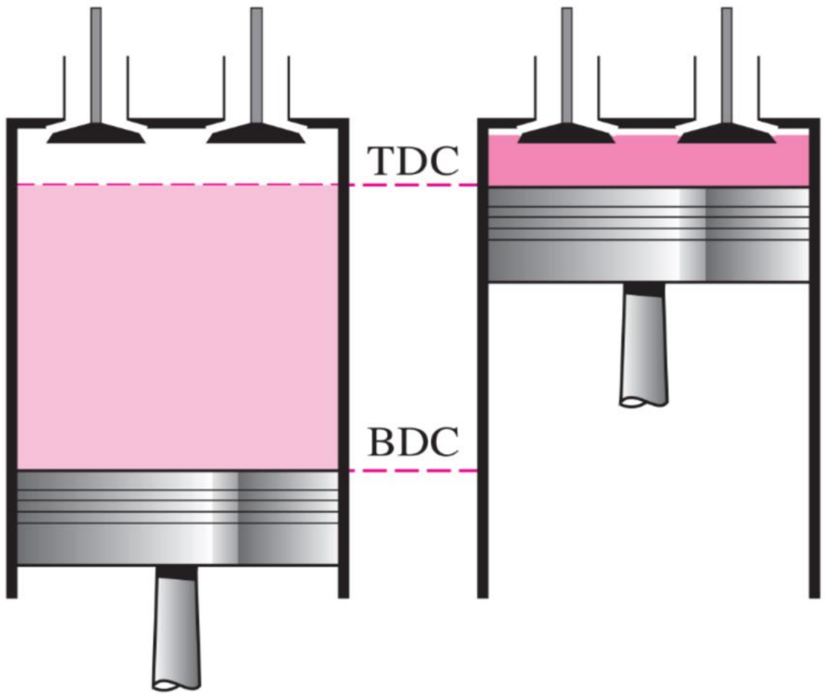


AN OVERVIEW OF RECIPROCATING ENGINES

- The piston reciprocates in the cylinder between two fixed positions called the **top dead center (TDC)**—the position of the piston when it forms the smallest volume in the cylinder—and the **bottom dead center (BDC)**—the position of the piston when it forms the largest volume in the cylinder.
- The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called **the stroke** of the engine.
- The diameter of the piston is called **the bore**.
- The air or air–fuel mixture is drawn into the cylinder through the **intake valve**, and the combustion products are expelled from the cylinder through the **exhaust valve.**
- The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume.**
- The volume displaced by the piston as it moves between TDC and BDC is called the **displacement volume.**

AN OVERVIEW OF RECIPROCATING ENGINES



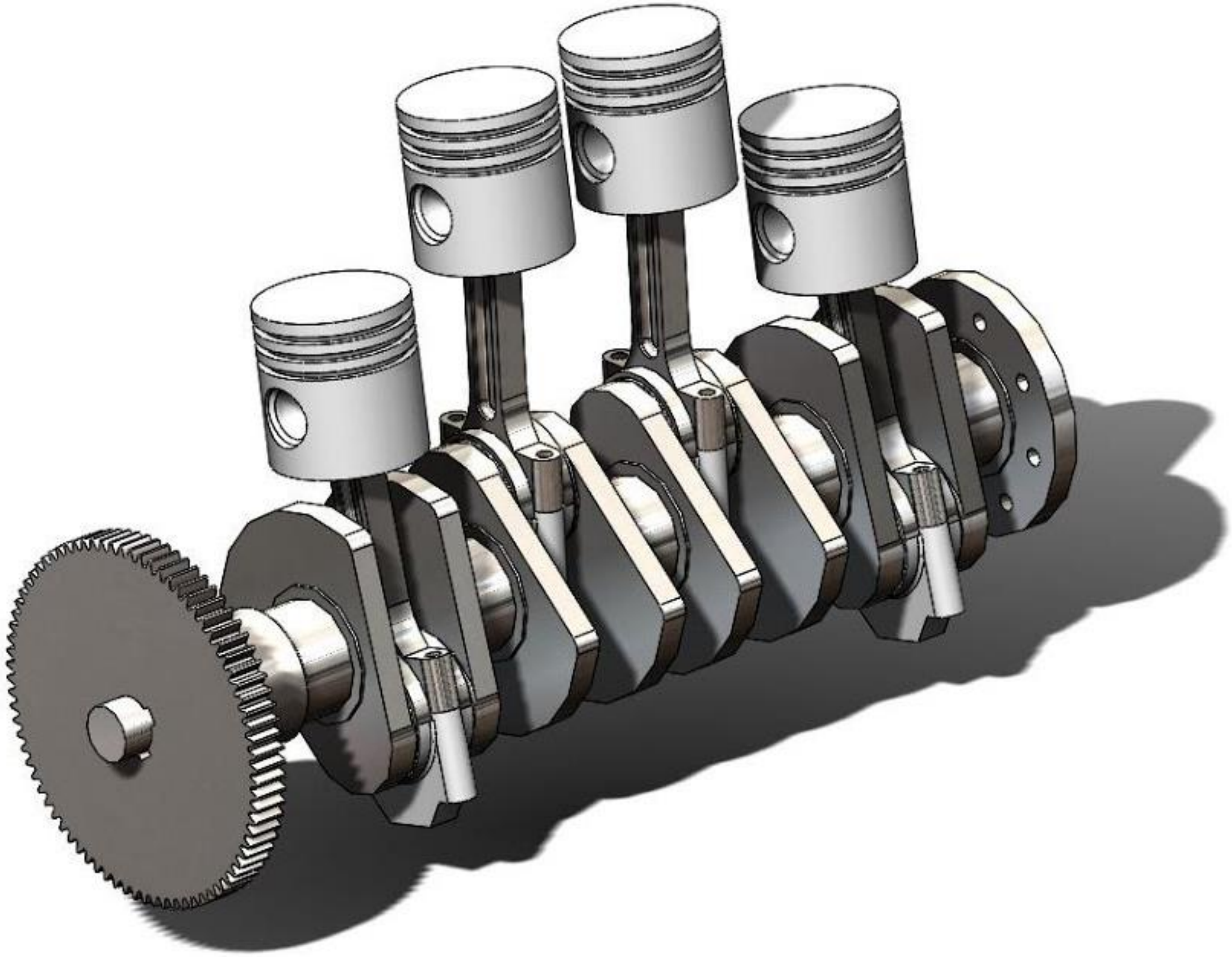


(*a*) Displacement volume

(*b*) Clearance volume

AN OVERVIEW OF RECIPROCATING ENGINES

Piston – Connecting Rod – Crankshaft Assembly



AN OVERVIEW OF RECIPROCATING ENGINES

- The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the *compression ratio r* of the engine:

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$$

- Notice that the compression ratio is a volume ratio and should not be confused with the pressure ratio.
- Another term frequently used in conjunction with reciprocating engines is the mean effective pressure *(MEP).* It is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle. That is,

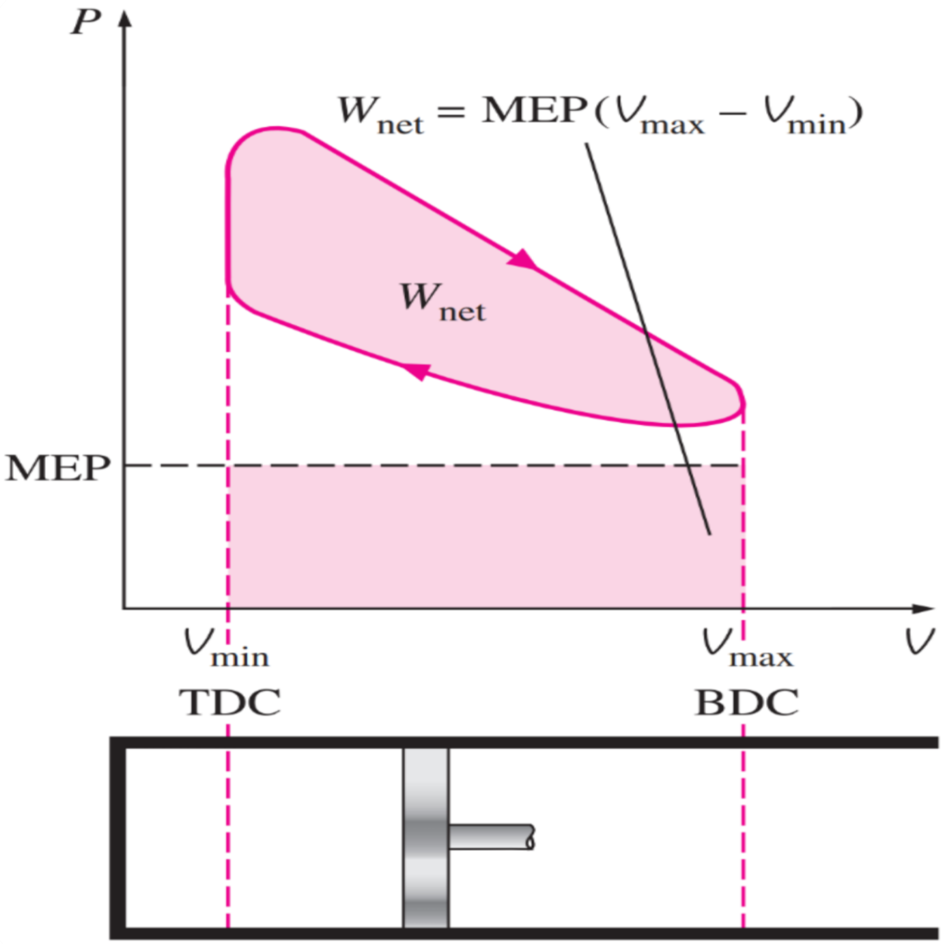
$$W_{net} = MEP \times Piston\ Area \times Stroke$$
$$= MEP \times Displacement\ Volume$$

or

$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{mib}} \qquad \text{(kPa)}$$

- The mean effective pressure can be used as a parameter to compare the performances of reciprocating engines of equal size. The engine with a larger value of MEP delivers more net work per cycle and thus performs better

AN OVERVIEW OF RECIPROCATING ENGINES



AN OVERVIEW OF RECIPROCATING ENGINES

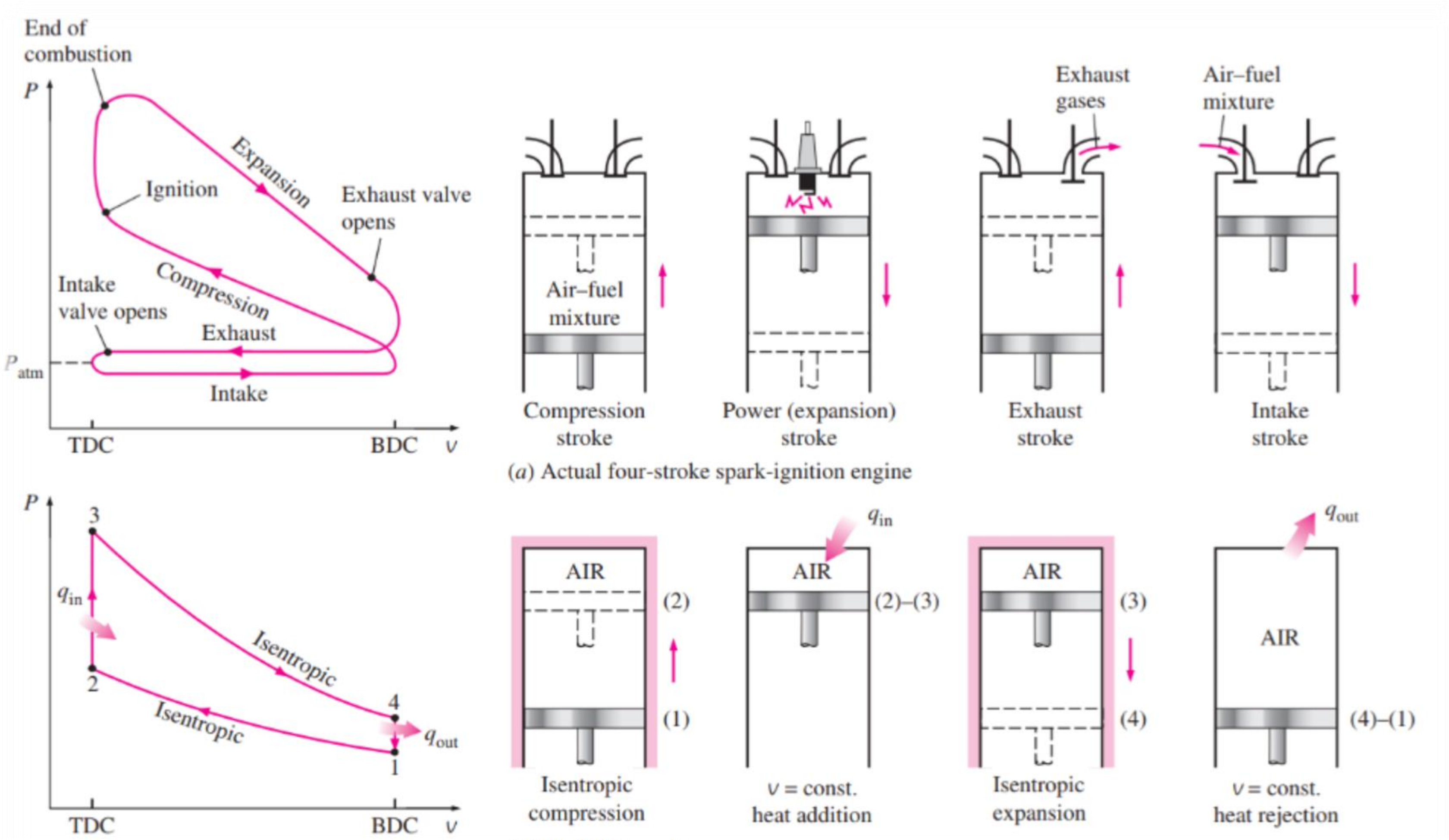
- Reciprocating engines are classified as ***spark-ignition (SI)*** *engines* or ***compression-ignition (CI) engines***, depending on how the combustion process in the cylinder is initiated.
- In SI engines, the combustion of the air–fuel mixture is initiated by a **spark plug**.
- In CI engines, the air–fuel mixture is self-ignited as a result of **compressing** the mixture above its self-ignition temperature

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES (GASOLINE ENGINES)

- The Otto cycle is the ideal cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876.
- In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle. These engines are called four-stroke internal combustion engines

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

Actual and ideal cycles in spark-ignition engines and their P-v diagrams.



(a) Actual four-stroke spark-ignition engine

(b) Ideal Otto cycle

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

How Gasoline Engine Works

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

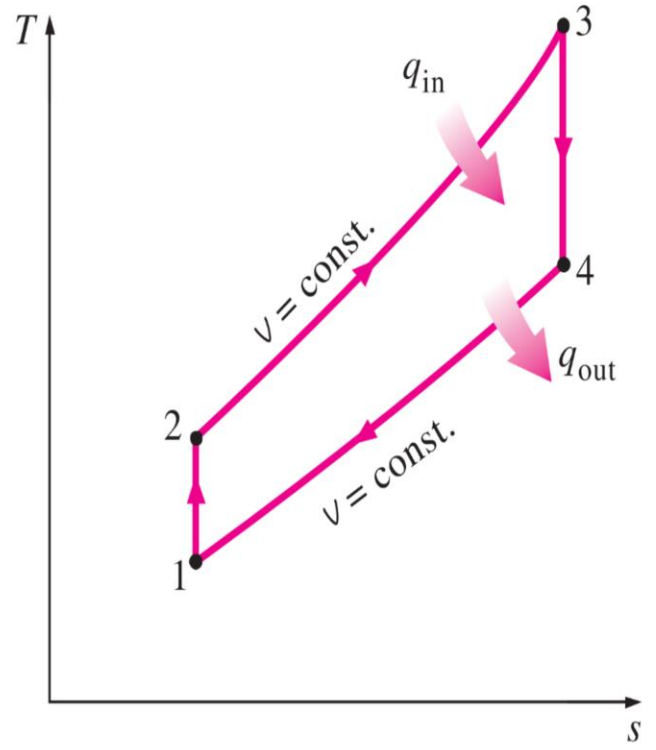
- The thermodynamic analysis of the actual four-stroke or two-stroke cycle can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal Otto cycle.
- It consists of four internally reversible processes:

1-2 Isentropic compression

2-3 Constant-volume heat addition

3-4 Isentropic expansion

4-1 Constant-volume heat rejection



OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

Related formula based on basic thermodynamics:

Process	*Description*	*Related formula*
1-2	Isentropic compression	$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n = \left(\frac{T_1}{T_2}\right)^{\frac{n}{n-1}}$
2-3	Constant volume heat addition	$Q_{in} = mC_v\left(T_3 - T_2\right)$
3-4	Isentropic expansion	$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n = \left(\frac{T_1}{T_2}\right)^{\frac{n}{n-1}}$
4-1	Constant volume heat rejection	$Q_{out} = mC_v\left(T_4 - T_1\right)$

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis, as

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \quad \text{(kJ/kg)}$$

- No work is involved during the two heat transfer processes since both take place at constant volume. Therefore, heat transfer to and from the working fluid can be expressed as

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

Then the thermal efficiency of the ideal Otto cycle under the cold air standard assumptions becomes:

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and v2 = v3 and v4 = v1. Thus,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{V_3}{V_4}\right)^{k-1} = \frac{T_4}{T_3}$$

Substituting these equations into the thermal efficiency relation and simplifying give

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

Where

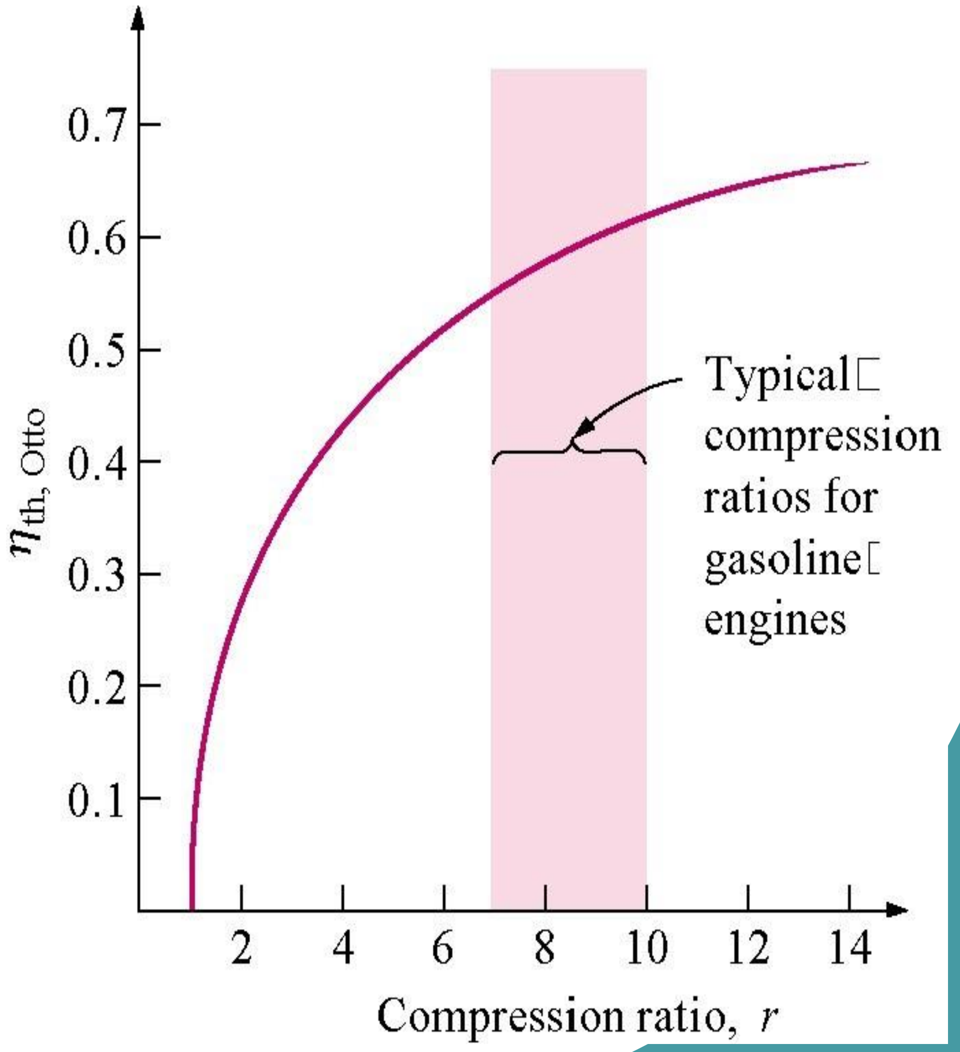
$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$$

is the compression ratio and k is the specific heat ratio cp/cv.

This equation shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Otto cycle depends on the compression ratio of the engine and the specific heat ratio of the working fluid. The thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

- A plot of thermal efficiency versus the compression ratio is given in the figure for k = 1.4, which is the specific heat ratio value of air at room temperature.
- For a given compression ratio, the thermal efficiency of an actual spark-ignition engine is less than that of an ideal Otto cycle because of the irreversibilities, such as friction, and other factors such as incomplete combustion.



OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

- Therefore, the increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios.
- Also, when high compression ratios are used, the temperature of the air–fuel mixture rises above the auto-ignition temperature of the fuel during the combustion process, causing an early and rapid burn of the fuel
- This premature ignition of the fuel, called auto-ignition, produces an audible noise, which is called engine knock. Auto-ignition in spark-ignition engines cannot be tolerated because it hurts performance and can cause engine damage. The requirement that auto-ignition not be allowed places an upper limit on the compression ratios that can be used in spark ignition internal combustion engines.

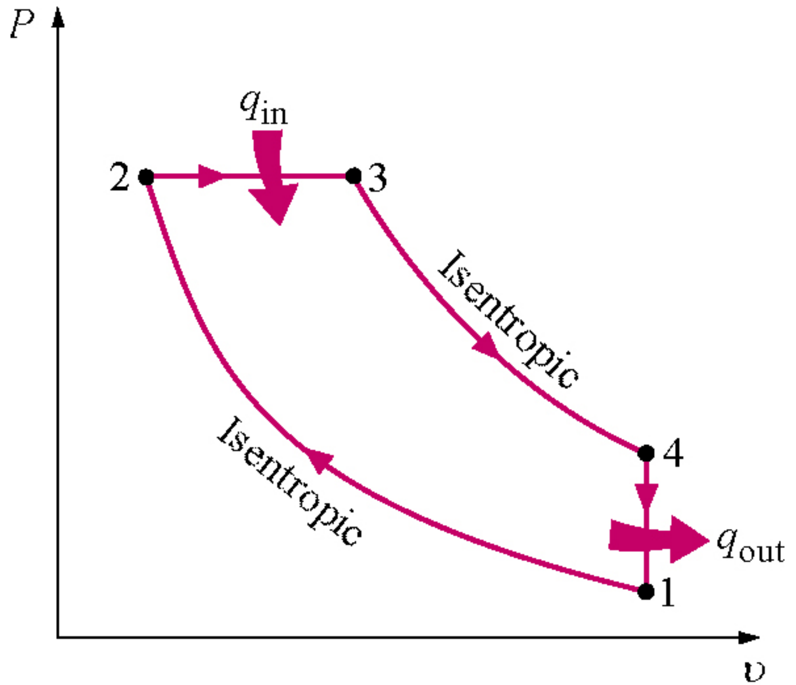
DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- The Diesel cycle is the ideal cycle for CI reciprocating engines. The CI engine, first proposed by Rudolph Diesel in the 1890s, is very similar to the SI engine discussed in the last section, differing mainly in the method of initiating combustion
- In CI engines (also known as diesel engines), the air is compressed to a temperature that is above the auto-ignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines

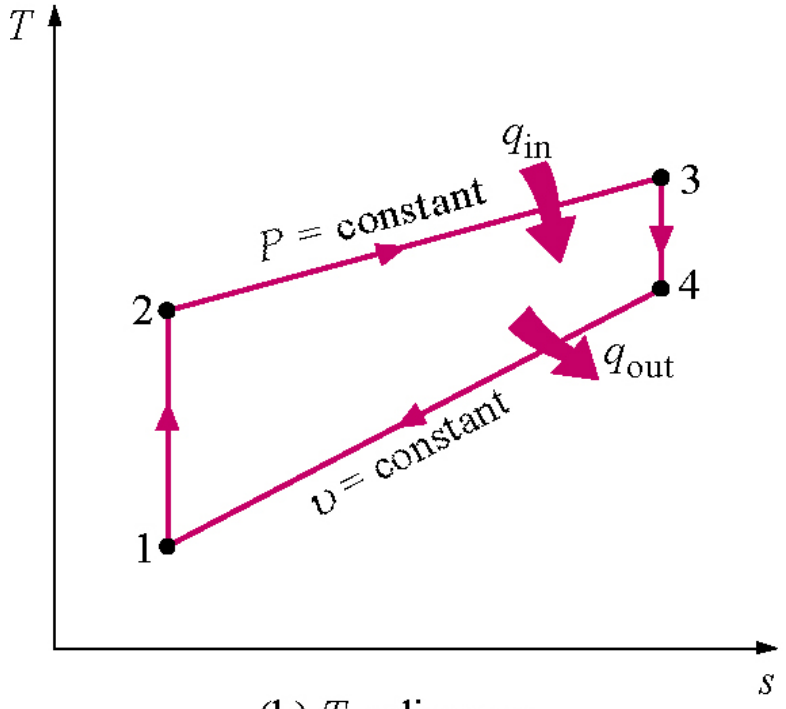
DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of auto-ignition or engine knock.
- In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of auto-ignition.
- Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24.
- Furthermore, and fuels that are less refined (thus less expensive) can be used in diesel engines

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)



(a) P-υ diagram



(b) T-s diagram

$$\frac{v_1}{v_2} = Compression\ ratio, r_v \quad and \quad \frac{v_3}{v_2} = Cut-off\ \ ratio, r_c$$

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DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval.
- Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process.

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- This is the only process where the Otto and the Diesel cycles differ. The remaining three processes are the same for both ideal cycles.
- That is,

1-2 is isentropic compression,

2-3 is constant-pressure heat-addition

3-4 is isentropic expansion, and

4-1 is constant-volume heat rejection..

Related formula based on basic thermodynamics:

Process	*Description*	*Related formula*
1-2	Isentropic compression	$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n = \left(\frac{T_1}{T_2}\right)^{\frac{n}{n-1}}$
2-3	Constant pressure heat addition	$Q_{in} = mC_P(T_3 - T_2)$
3-4	Isentropic expansion	$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n = \left(\frac{T_1}{T_2}\right)^{\frac{n}{n-1}}$
4-1	Constant volume heat rejection	$Q_{out} = mC_v(T_4 - T_1)$

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- Noting that the Diesel cycle is executed in a piston–cylinder device, which forms a closed system, the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as

$$q_{in} - w_{b,out} = P_2(V_3 - V_2) + (u_3 - u_2) = h_3 - h_2 = c_p(T_3 - T_2)$$

- And

$$-q_{out} = u_4 - u_1 = h_3 - h_2 = c_v(T_4 - T_1)$$

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- Then the thermal efficiency becomes:

$$\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$

We now define a new quantity, the cutoff ratio , as the ratio of the cylinder volumes after and before the combustion process

Where

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2}$$

$$\frac{P_4V_4}{T_4}=\frac{P_1V_1}{T_1} \quad \text{where } V_4=V_1$$

$$\frac{T_4}{T_1}=\frac{P_4}{P_1}$$

- Recall processes 1-2 and 3-4 are isentropic, so

$$\boxed{P_1V_1^k=P_2V_2^k \quad \text{and} \quad P_4V_4^k=P_3V_3^k}$$

- Since $V_4 = V_1$ and $P_3 = P_2$, we divide the second equation by the first equation and obtain

$$\frac{P_4}{T_4}=\left(\frac{V_3}{V_2}\right)^k=r_c^k$$

Therefore,

$$\boxed{\eta_{th,Diesel}=1-\frac{1}{r^{k-1}}\frac{r_c^k-1}{k\left(r_c-1\right)}}$$

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- The efficiency of a Diesel cycle differs from the efficiency of an Otto cycle by the quantity in the brackets. This quantity is always greater than 1. Therefore:

$$\eta_{th,Otto} > \eta_{th,Diesel}$$

- Also, as the cutoff ratio decreases, the efficiency of the Diesel cycle increases. For the limiting case of rc = 1, the quantity in the brackets becomes unity, and the efficiencies of the Otto and Diesel cycles become identical.

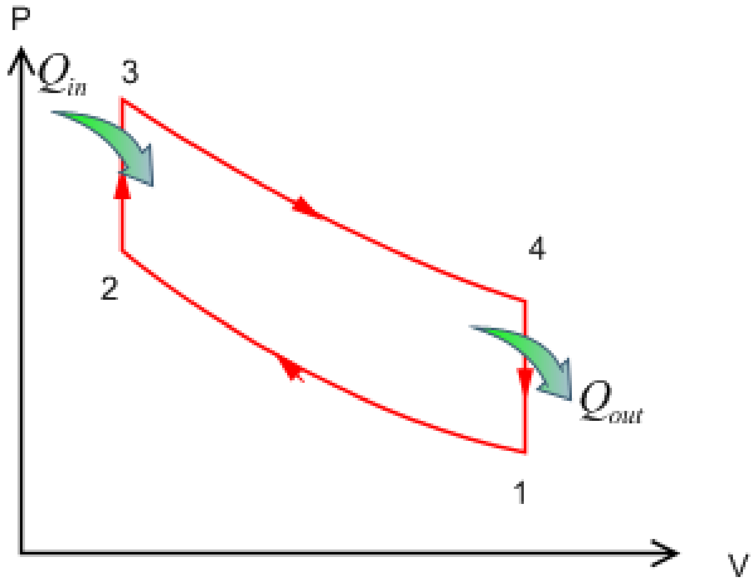
DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES (DIESEL ENGINES)

- Remember, though, that diesel engines operate at much higher compression ratios and thus are usually more efficient than the spark-ignition (gasoline) engines. The diesel engines also burn the fuel more completely since they usually operate at lower revolutions per minute and the air–fuel mass ratio is much higher than spark-ignition engines.
- Thermal efficiencies of large diesel engines range from about 35 to 40 percent. The higher efficiency and lower fuel costs of diesel engines make them attractive in applications requiring relatively large amounts of power, such as in locomotive engines, emergency power generation units, large ships, and heavy trucks.

Example 1

An Otto cycle having a compression ratio of 9:1 uses air as the working fluid. Initially P_1 = 95 kPa, T_1 = 17°C, and V_1 = 3.8 liters. During the heat addition process, 7.5 kJ of heat are added. Determine all T's, P's, η_{th}, the back work ratio and the mean effective pressure.

Solution:



Data given:

$$T_1 = 290K$$

$$V_1 / V_2 = 9$$

$$Q_{23} = 7.5kJ$$

$$P_1 = 95kPa$$

$$V_1 = 3.8 Litres$$

Example 1

$$\boxed{\text{Process } 1-2\,(\textit{isentropic compression})}$$

$$\frac{T_2}{T_1}=\left(\frac{V_1}{V_2}\right)^{k-1} \Rightarrow T_2 = 290(9)^{0.4} = 698.4K$$

$$\frac{P_2}{P_1}=\left(\frac{V_1}{V_2}\right)^{k-1} \Rightarrow P_2 = 95(9)^{1.4} = 2059kPa$$

$$\boxed{\text{Process } 2-3\,(\textit{Const.volume heat addition})}$$

$$1^{st}\ law: Q_{net} - \cancel{W_{net}}^{\,0} = \Delta U$$

$$Q_{23} = mC_v(T_3 - T_2)$$

$$IGL: \quad P_1 v_1 = RT_1 \Rightarrow v_1 = \frac{0.2871(290)}{95} = 0.875\,\tfrac{m^3}{kg}$$

$$q_{23} = \frac{Q_{23}}{m} = Q_{23}\frac{v_1}{V_1} = 1727\,\tfrac{kJ}{kg}$$

Example 1

$Back\ to\ IGL:$

$q_{23} = C_v(T_3 - T_2)$

$= 0.718(T_3 - 698.4)$

$T_3 = 3103.7K$

$But\ V_3 = V_2$

$\frac{P_3}{T_3} = \frac{P_2}{T_2}$

$P_3 = 9.15MPa$

$\boxed{\text{Process } 3-4\ (isentropic\ expansion)}$

$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} \Rightarrow T_4 = T_3(1/9)^{0.4} = 1288.8K$

$\frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^{k} \Rightarrow P_4 = P_3(1/9)^{1.4} = 422kPa$

Example 1

$$\boxed{Process\ 4-1\left(Const.volume\ heat\ rejection\right)}$$

$$Q_{41} = mC_v\left(T_4 - T_1\right)$$

$$q_{41} = C_v\left(T_4 - T_1\right)$$

$$= 0.718\left(1288.8 - 290\right)$$

$$= 717.1\tfrac{kJ}{kg}$$

Then:

$$W_{net} = q_{in} - q_{out}$$

$$= q_{23} - q_{41}$$

$$= 1009.6\ \tfrac{kJ}{kg}$$

$$\eta_{th,Otto} = \frac{W_{net}}{q_{in}} = \underline{\underline{0.585\left(58.5\%\right)}}$$

Example 1

What else?

$$MEP = \frac{W_{net}}{V_{\max} - V_{\min}} = \frac{w_{net}}{v_{\max} - v_{\min}}$$

$$= \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1\left(1 - v_2/v_1\right)}$$

$$= \frac{w_{net}}{v_1\left(1 - \frac{1}{r}\right)} = \frac{1009.6}{0.875\left(1 - 1/9\right)} = \underline{\underline{1298\,kPa}}$$

$$r_{bw} = \frac{w_{compr}}{w_{expans}} = \frac{\Delta u_{12}}{-\Delta u_{34}} = \frac{\cancel{C_v}\left(T_2 - T_1\right)}{\cancel{C_v}\left(T_3 - T_4\right)}$$

$$= \underline{\underline{0.225\left(22.5\%\right)}}$$

Example 2

An air-standard Diesel cycle has a compression ratio of 18 and a cut-off ratio of 2.5. The state at the beginning of compression is fixed by P = 0.9 bar ant T = 300K. Calculate:

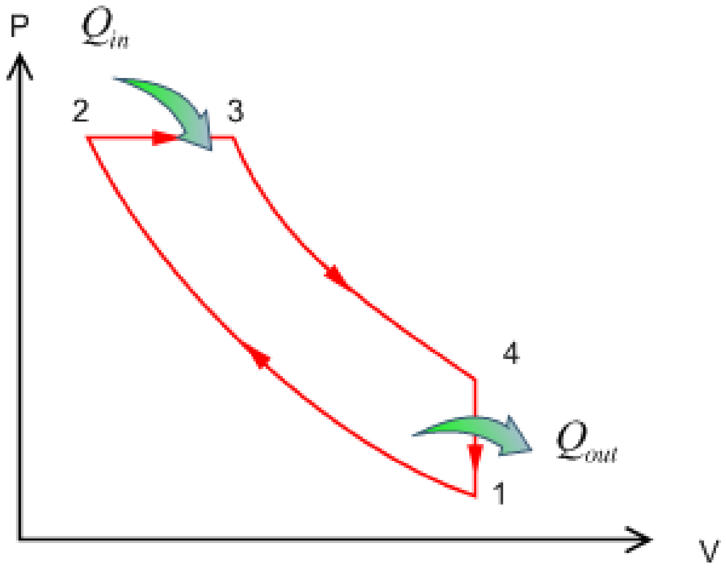
i. the thermal efficiency of the cycle,
ii. the maximum pressure, $P_{max,}$ and
iii. The mean effective pressure.

Solution:

Data given:

$$\frac{V_1}{V_2} = 18$$

$$\frac{V_3}{V_2} = 2.5$$



Example 2

$$\boxed{\text{Process } 1-2 \; (\textit{isentropic compression})}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} \Rightarrow T_2 = 300(18)^{0.4} = 953.3K$$

$$\boxed{\text{Process } 2-3 \; (\textit{Const. pressure heat addition})}$$

$$P_2 = P_3 \Rightarrow \frac{V_2}{T_2} = \frac{V_3}{T_3} \Rightarrow T_3 = T_2\left(\frac{V_3}{V_2}\right) = 2383.3K$$

$$\boxed{\text{Process } 3-4 \; (\textit{isentropic expansion})}$$

$$\frac{V_4}{V_3} = \frac{V_1}{V_2} \cdot \frac{V_2}{V_3} = 18(1/2.5) = 7.2$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} \Rightarrow T_4 = 2383.3(1/7.2)^{0.4} = 1082K$$

Example 2

$$Q_{in} = Q_{23} = mC_P\left(T_3 - T_2\right) \Rightarrow \; q_{in} = C_p\left(T_3 - T_2\right) = 1437.15\,\tfrac{kJ}{kg}$$

$$Q_{out} = Q_{41} = mC_P\left(T_4 - T_1\right) \Rightarrow \; q_{out} = C_p\left(T_4 - T_1\right) = 561.48\,\tfrac{kJ}{kg}$$

$$w_{net} = q_{in} - q_{out} = 875.67\,\tfrac{kJ}{kg}$$

What we need?

$$(i)\ \eta_{th,diesel} = \frac{w_{net}}{q_{in}} = \underline{\underline{0.6093\,(60.93\%)}}$$

$$(ii)\ P_{max} = P_2 = P_3$$

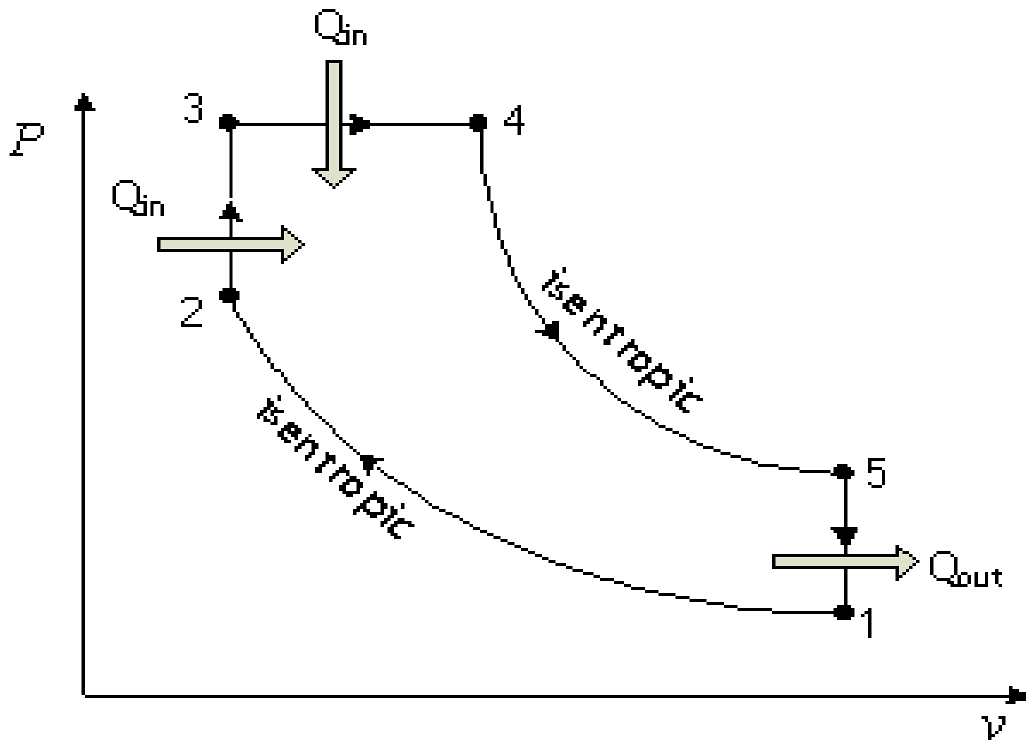
$$\left(\frac{P_2}{P_1}\right) = \left(\frac{T_2}{T_1}\right)^{\frac{k-1}{k}} \Rightarrow P_2 = \underline{\underline{5148\,kPa\,(P_{max})}}$$

$$(iii)\ MEP = \frac{w_{net}}{V_1\left(1 - 1/r\right)} = \frac{875.67}{0.9566\left(1 - 1/18\right)} = \underline{\underline{969.1\,kPa}}$$

IDEAL DUAL CYCLE

- Approximating the combustion process in internal combustion engines as a constant-volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic.
- A better approach would be to model the combustion process in both gasoline and diesel engines as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure.
- The ideal cycle based on this concept is called the dual cycle,
- The relative amounts of heat transferred during each process can be adjusted to approximate the actual cycle more closely.
- Note that both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle

IDEAL DUAL CYCLE



Dual cycle gives a **better approximation to a real engine**. The heat addition process is partly done at a constant volume and partly at constant pressure. From the P-v diagram, it looks like the heat addition process is a combination of both Otto and Diesel cycles.

IDEAL DUAL CYCLE

Process	*Description*
1-2	Isentropic compression
2-3	Constant volume heat addition
3-4	Constant pressure heat addition
4-5	Isentropic expansion
5-1	Constant volume heat rejection

- The same procedure as to Otto and Diesel cycles can be applied to Dual cycle. Upon substitutions, the thermal efficiency of Dual cycle becomes

$$\eta_{th} = 1 - \frac{r_p r_c^{\,k} - 1}{\left[\left(r_p - 1\right) + kc_p\left(r_c - 1\right)\right] r_v^{\,k-1}}$$

Example 3

At the beginning of the compression process of an air-standard dual cycle with a compression ratio of 18, the temperature is 300 K and the pressure is 1 bar. The pressure ratio for the constant volume part of the heating process is 1.5 to 1. The volume ratio for the constant pressure part of the heating process is 1.2 to 1. Determine (a) the thermal efficiency and (b) the mean effective pressure.

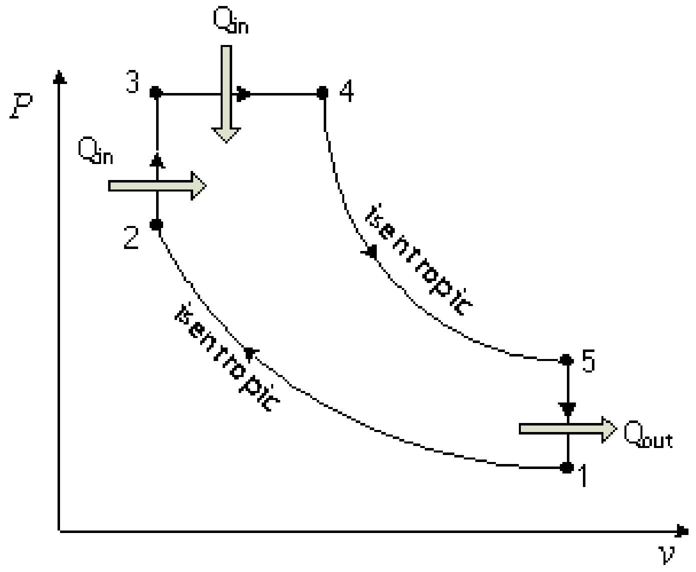
Solution:

Data given:

$$\frac{V_1}{V_2} = 18 \qquad \frac{P_1}{P_2} = 1.5$$

$$T_1 = 300K \qquad \frac{V_4}{V_3} = 1.2$$

$$P_1 = 1bar$$



$$\boxed{\text{Process } 1-2\ (\textit{isentropic compression})}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} \Rightarrow T_2 = 300(18)^{0.4} = 953.3K$$

$$\boxed{\text{Process } 2-3\ (\textit{Const. pressure heat addition})}$$

$$P_2 = P_3 \Rightarrow \frac{V_2}{T_2} = \frac{V_3}{T_3} \Rightarrow T_3 = T_2\left(\frac{V_3}{V_2}\right) = 2383.3K$$

$$\boxed{\text{Process } 3-4\ (\textit{isentropic expansion})}$$

$$\frac{V_4}{V_3} = \frac{V_1}{V_2} \cdot \frac{V_2}{V_3} = 18(1/2.5) = 7.2$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} \Rightarrow T_4 = 2383.3(1/7.2)^{0.4} = 1082K$$

$$\boxed{Process\ 4-5\left(isentropic\ expansion\right)}$$

$$\frac{T_5}{T_4}=\left(\frac{V_4}{V_5}\right)^{k-1}\Rightarrow T_5=T_4\left(\frac{V_4}{V_5}\right)^{k-1}=T_4\left[\left(\frac{V_4}{V_3}\right)\left(\frac{V_3}{V_5}\div\right)\right]^{k-1}$$

$$=1715.94\left[(1.2)\left(\tfrac{1}{18}\right)\right]^{0.4}$$

$$=584.85K$$

Information needed?

$$Q_{out}=Q_{51}=mC_v\left(T_5-T_1\right)=204.52\ \tfrac{kJ}{kg}$$

$$Q_{in}=Q_{23}+Q_{34}=mC_v\left(T_3-T_2\right)+mC_p\left(T_4-T_3\right)$$

$$=629.65\tfrac{kJ}{kg}$$

$$(a)\,\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{204.52}{629.65} = \underline{\underline{0.675(\,67.5\%)}}$$

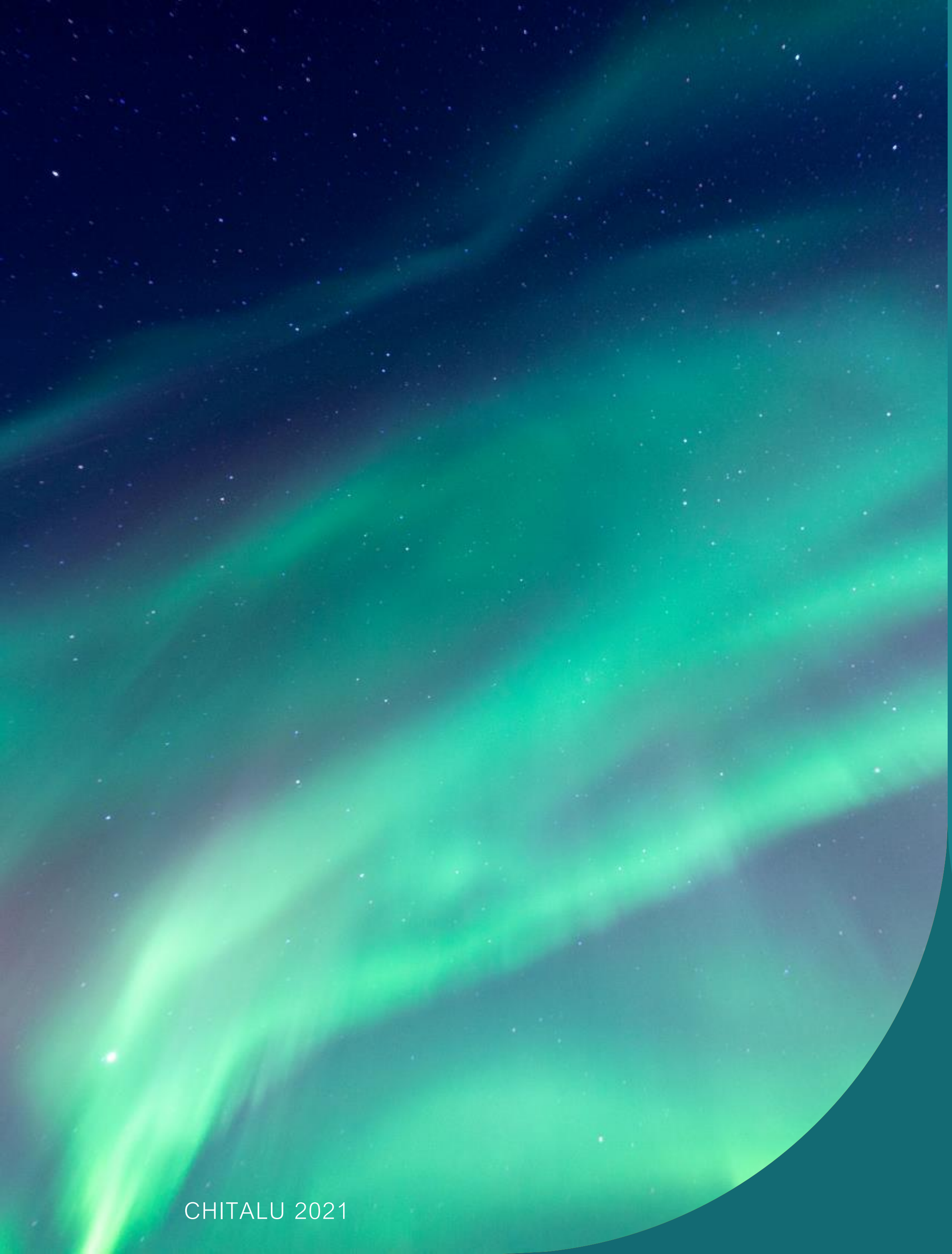
$$(b)\,MEP = \frac{W_{net}}{v_1\left(1 - \frac{1}{r}\right)}$$

$$= \frac{425.13}{0.8613\left(1 - \frac{1}{18}\right)}$$

$$= \underline{\underline{522.63\,kPa}}$$

“If your theory is found to be against the Second Law of Thermodynamics, I give you no hope; there is nothing but for it to collapse in deepest humiliation”

Arthur Eddington



Thank You

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