internal combustion engines - MAK3031

Lecture Notes

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Syllabus

MAK3031-Internal Combustion Engines

Week	Subjects
1	Definitions of internal combustion engines, Engine Classification
2	Principle of operation of engines, Introduction to engine thermodynamics
3	Ideal cycles (Otto, Diesel and Seilinger Cycles, Thermal Efficiency and Mean Indicated Pressure Expressions)
4	Ideal Cycles and Comparisons
5	Introduction to Combustion in Engines, (Minimum Oxygen and Air Intake calculation, Complete and Incomplete Combustion combustion, excess air factor, Number of moles of Change)
6	Exercises: Examples of related application cycles (Engine Thermodynamics), Application examples on Combustion (Combustion Engines)
7	Actual Engine Cycle (the differencies between the ideal and the actual engine cycle). Definition of Organic Efficiency, the combustion event, Compression and Expansion Events, Gas Exchange Event
8	Mid-term Exam
9	Volumetric Efficiency (Definition, Calculation, Factors Affecting Volumetric Efficiency)
10	(Compression Ignition and Combustion, Compression Ignition Delay, definition and factors affecting) in Diesel Engines
11	Detonation (Knock) in Engines, Otto and Diesel Engines Knocking, knocking disadvantages
12	The factors affecting detonation, octane and cetane numbers
13	The basic of Mixture Fomation in Otto Engines, Carburation (simple carburettor), Fuel Injection (Injection type, the general scheme of injection), The basics of Diesel Mixture Formation, General Injection System, the expected properties from Injection System
14	Mid-term Exam
15	Engine Power Calculation
16	Final Exam

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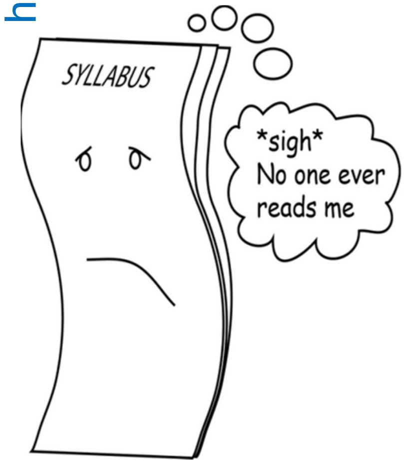
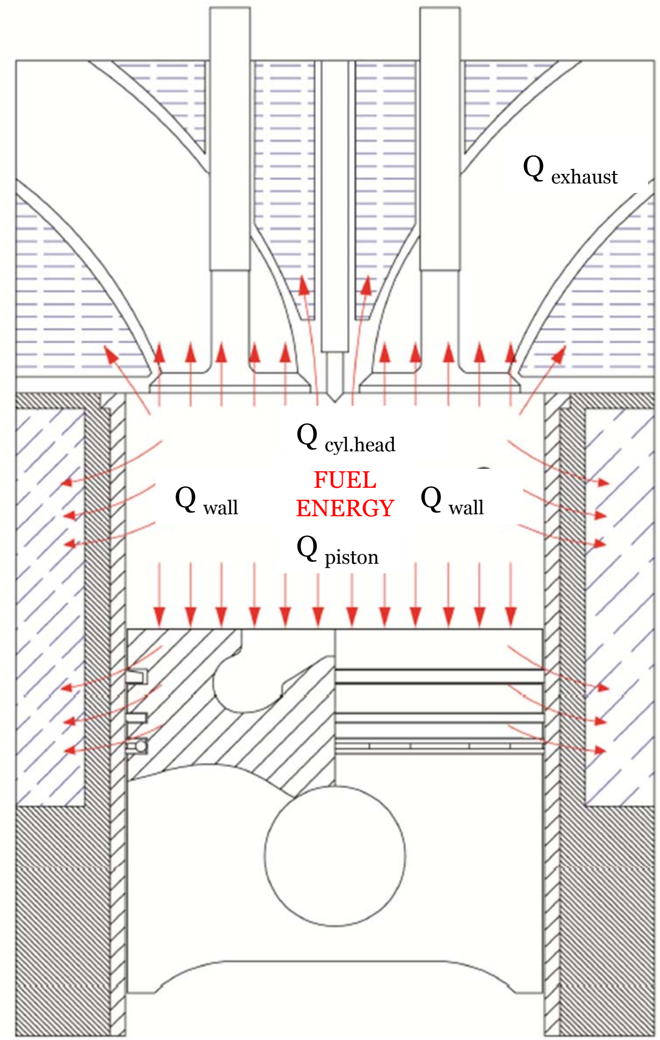
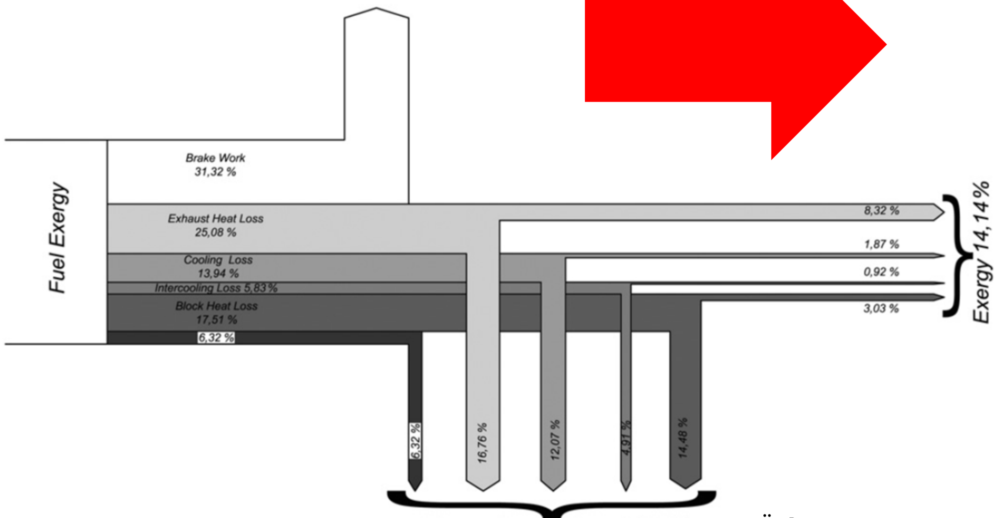


ENGINE CYCLES

ENGINE is a thermodynamic system





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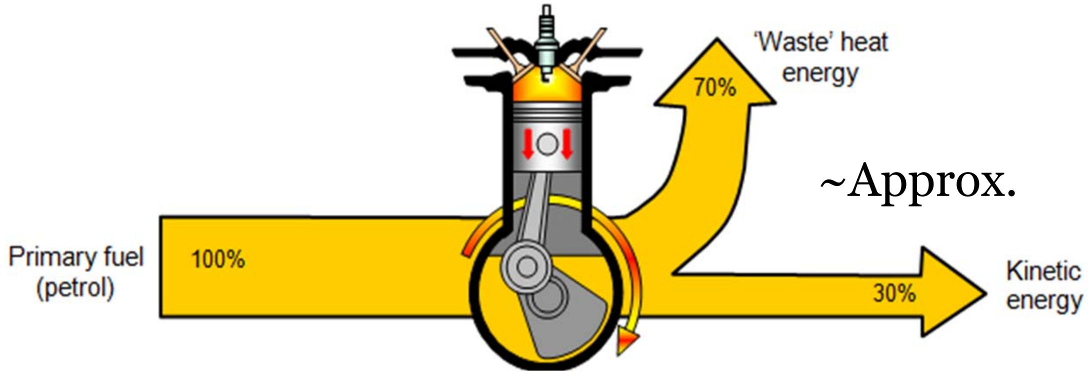


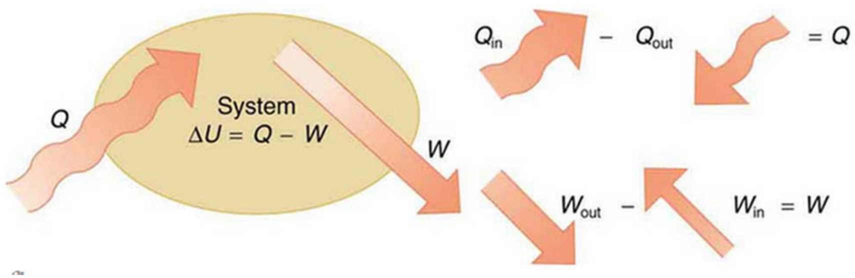
~Approx.

So , Thermodynamic LAWS are Valid.

1st LAW

First law of thermodynamics say us, energy is conserved

The first law of thermodynamics is the conservation-of-energy principle stated for a system where heat and work are the methods of transferring energy for a system in thermal equilibrium. Q represents the net heat transfer—it is the sum of all heat transfers into and out of the system. Q is positive for net heat transfer into the system. W is the total work done on and by the system. W is positive when more work is done by the system than on it. The change in the internal energy of the system, ΔU, is related to heat and work by the first law of thermodynamics, $\Delta U=Q-W$.

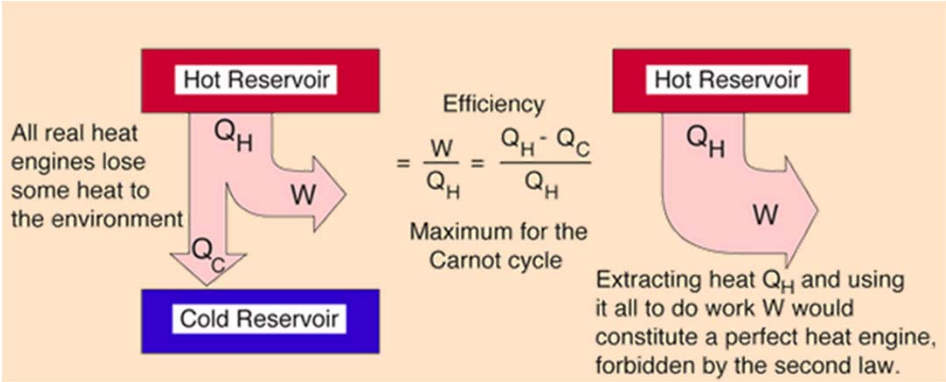


So the fuel chemical energy entering the system, is converted in to mechanical energy !!!

So , Thermodynamic LAWS are Valid.

2nd LAW

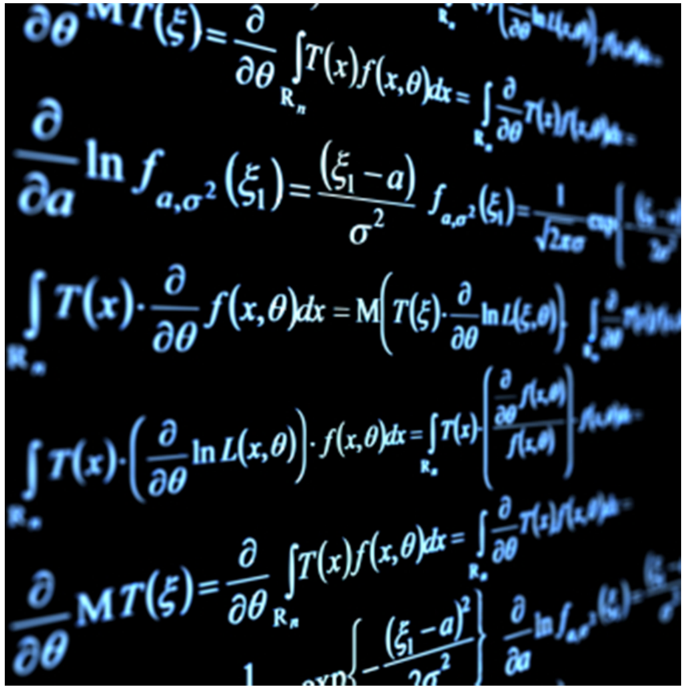
Second law of thermodynamics says for converting heat in to mechanical Energy two heat source is needed



For converting the heat energy in to mechanic energy, a closed cycle is needed between two heat source. For obtaining this cycle we need to have a working mixture

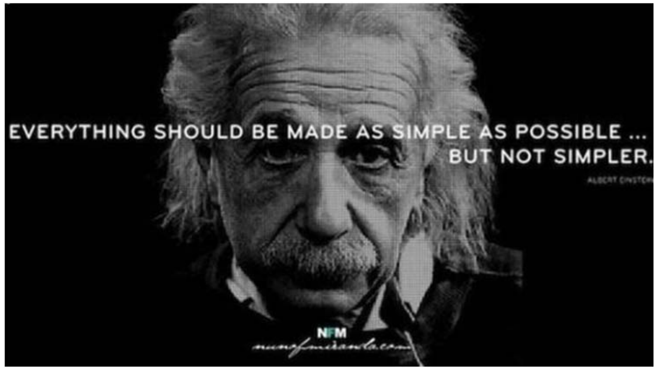
ENGINE CYCLES

Engine Cycles are open cycles with changing composition. The system is difficult to analyze.



-Manageable
-Approximation

Air Standard Cycles



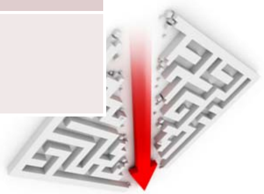
Air standard cycles differs from the actual cycles with;

NO	Differs with;
1	The gas mixture in the cylinder is treated as air for the entire cycle, and property values of air are used in the analysis. This is good approximation during the first half of the cycle, when most of the gas in the cylinder is air with only up to about ~7% fuel vapor. Even in the second half of the cycle when the gas composition mostly CO_2,H_2O, and N_2, using air properties does not create large errors in the analysis. Air will be treated as an ideal gas with constant specific heats.
2	Real open cycle is changed in to a closed cycle by assuming that gases being exhausted are fed back into the intake system. This Works with ideal air standard cycles, as both intake gases and exhaust gases are air. Closing the cycle simplifies the analysis.
3	The combustion process is replaced with a heat addition term $\boldsymbol{Q_{in}}$ of equal energy value. Air alone can not combust.
4	The open exhaust process, which carries a large amount of enthalpy out of the system, is replaced with a closed system heat rejection process $\boldsymbol{Q_{out}}$ of equal energy value.

just simplifying

Air standard cycles differs from the actual cycles with;

NO	Differs with;
5	Actual engine process are approximated with ideal process.
5.a	*The almost constant pressure intake and exhaust strokes are assumed to be constant pressure.At Wide open throttle (WOT) the intake stroke is assumed to be at pressureP0 of one atmosphere. At partially closed throttle or when supercharged, inlet pressure will be some constant value other than one atmosphere. The exhaust stroke pressure is assumed constant at one atmosphere.*
5.b	*Compression strokes and expansion strokes a re approximated with isentropic process. To be truly isentropic would require these strokes to be reversible and adiabatic. There is some friction between piston and cylinder walls but, because the surfaces are highly polished and lubricated, this friction is kept minimum an the process are close to frictionless and reversible. If this were not true, automobile engines would wear long before the 150-200 thousand miles which they now last if properly maintained. There is also fluid friction because of the gas motion with in the cylinders during these strokes. This is too minimal. Heat transfer fır any stroke will be negligibly small due to the very short time involved for that single process. Thus, and almost reversible and almost adiabatic process can quite accurately be approximated with an isentropic process.*
5.c	*The combustion process is idealized by a constant volume process(SI), a constant pressure process (CI) cycle, or a combination of both (CI dual cycle- Seilinger)*
5.d	*Exhaust blow down is approximated by a constant volume process.*
5.e	*All process are considered irreversible.*



In air standard cycles, air is considered an ideal gas such that following ideal gas relation ships can be used.

$$Pv = RT \qquad \boxed{25}$$

$$PV = mRT \qquad \boxed{26}$$

$$PV = \rho RT \qquad \boxed{27}$$

$$dh = c_p dT \qquad \boxed{28}$$

$$du = c_v dT \qquad \boxed{29}$$

$$Pv^k = constant \gg isentropic\ process \qquad \boxed{30}$$

$$Tv^{k-1} = constant \gg isentropic\ process \qquad \boxed{31}$$

$$TP^{(1-k)/k} = constant \gg isentropic\ process \qquad \boxed{32}$$

$$w_{1-2} = \frac{P_2 v_2 - P_1 v_1}{1-k} = \frac{R(T_2 - T_1)}{1-k} \gg isentropic\ work\ in\ closed\ system \qquad \boxed{33}$$

$c\sqrt{kRT} \gg$ speed of the sound $\boxed{34}$

$P = gas\ pressure\ in\ the\ cylinder$
$V = volume\ in\ the\ cylinder$
$v = specific\ volume\ of\ gas$
$R = gas\ constant\ of\ air$
$T = temperature$
$m = mass\ of\ gs\ in\ the\ cylinder$
$\rho = density$
$h = specific\ enthalpy$
$u = specific\ internal\ energy$
$c_p, c_v = specific\ heats$
$k = c_p/c_v$
$w = specific\ work$
$c = speed\ of\ the\ sound$

In addition to these, the following variables are used in this chapter for cycle analysis.

$AF = air - fuel\ ratio$
$\dot{m} = mass\ flowrate$
$q = heat\ transfer\ per\ unit\ mass\ for\ one\ cycle$
$\dot{q} = heat\ transfer\ rate\ per\ unit\ mass$
$Q = heat\ transfer\ for\ one\ cycle$
$\dot{Q} = heat\ transfer\ rate$
$Q_{HV} = heating\ value\ of\ fuel$
$r_c = compression\ ratio$
$W = work\ for\ one\ cycle$
$\dot{W} = power$
$\eta_c = combustion\ efficiency$

subscripts

$a = air$
$f = fuel$
$ex = exhaust$
$m = mixture\ of\ all\ gases$

For thermodynamic analysis specific heats of air can be treated as functions of temperature, which they are , or they can be treated as constants which simplifies calculations at slight loss of accuracy.

so

Constant specific heat analysis will be used.

Becasue of high temperatures andlarge temperature range experienced during ana engine cycle the specific heats and ratio of specific heats k do vary by fairy amount.

Low temperature end of cycle during intake and start of compression a value of k~1.4

However at the end of the combustion k~1.3 will be more accurate

So the average can be used be used for simplifying k~1.35

AIR PROPERTIES THAT WILL BE USED FOR CALCULATIONS

$$c_P = 1.108\ kJ/kgK$$

$$c_v = 0.821\ kJ/kgK$$

$$k = c_p/c_v = \frac{1.108}{0.821} = 1.35$$

$$R = c_p - c_v = 0.287\ kj/kgK$$

Air flow before it enters an engine is usually closer to standard temperature, and for these conditions a valıe of k=1.4 is correct. This would increase processes such as inlet flow in superchargers, turbochargers and carburetors and air flow through the engine radiator.

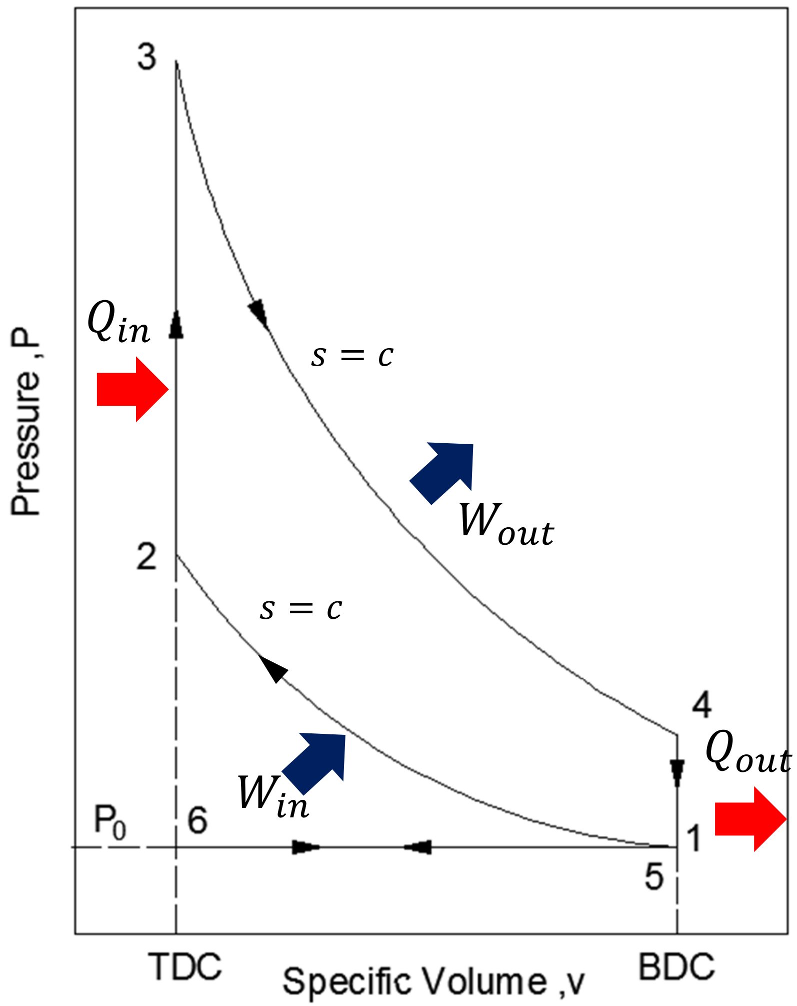
$$c_P = 1.005\ kJ/kgK$$

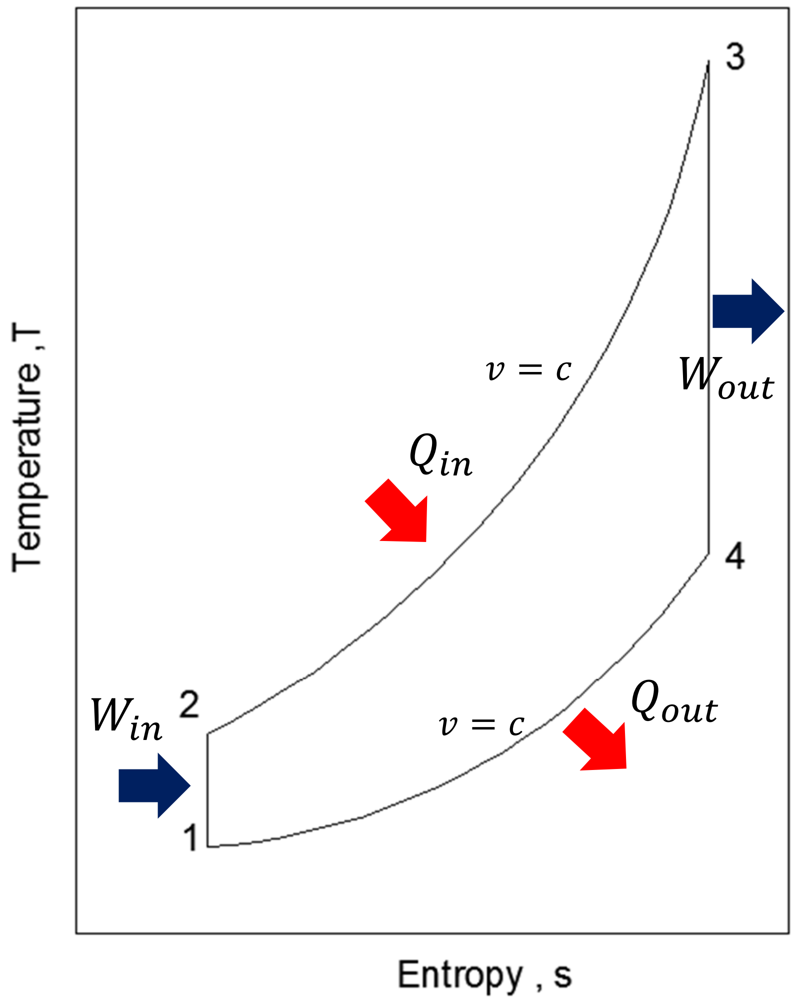
$$c_v = 0.718\ kJ/kgK$$

$$k = c_p/c_v = \frac{1.005}{0.718} = 1.4$$

$$R = c_p - c_v = 0.287\ kj/kgK$$

Air Standard - OTTO CYCLE

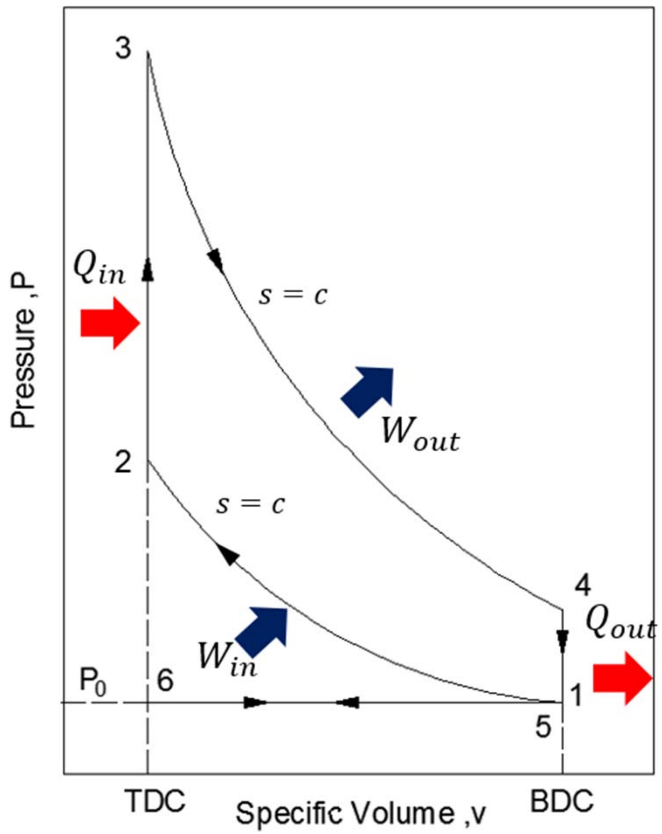




Air Standard - OTTO CYCLE

Process 6-1

constant pressure intake of air at P_o. Intake valve open and exhaust valve closed



$$P_1 = P_6 \quad \boxed{35}$$

$$P_1 = P_6 = P_0 \quad \boxed{36}$$

$$w_{6-1} = P_0(v_1 - v_6) \quad \boxed{37}$$

Air Standard - OTTO CYCLE

Process 1-2
isentropic compression stroke.
All valves closed.

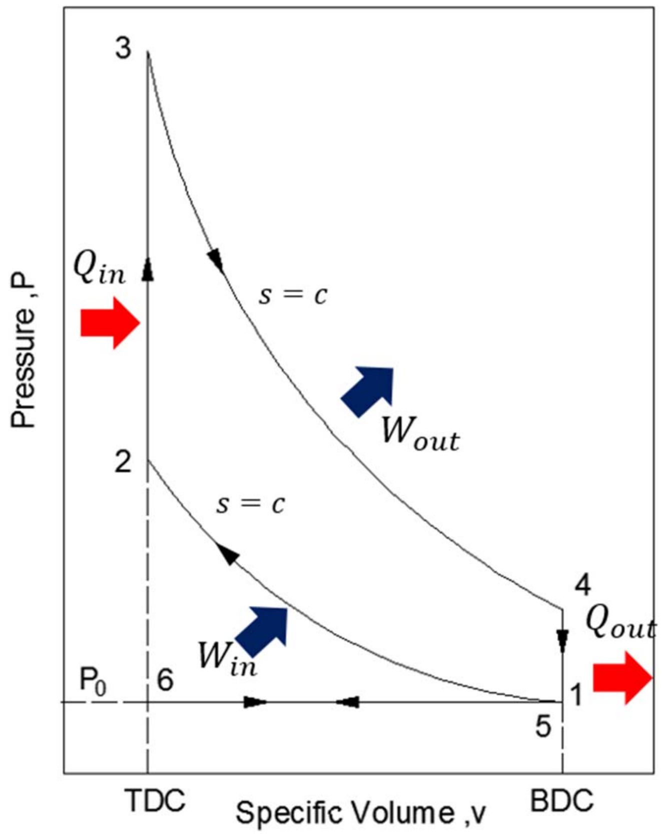
$$T_2 = T_1(\frac{v_1}{v_2})^{k-1} = T_1(\frac{V_1}{V_2})^{k-1} = T_1\varepsilon^{k-1} \quad \boxed{38}$$

$$P_2 = P_1(\frac{v_1}{v_2})^{k} = P_1(\frac{V_1}{V_2})^{k} = P_1\varepsilon^{k} \quad \boxed{39}$$

$$q_{1-2} = 0 \quad \boxed{40}$$

$$w_{1-2} = \frac{P_2v_2 - P_1v_1}{1-k} = \frac{R(T_2-T_1)}{1-k} \quad \boxed{41}$$

$$= (u_1 - u_2) = c_v(T_1 - T_2) \quad \boxed{42}$$



Air Standard - OTTO CYCLE

Process 2-3
Constant volume heat input (combustion)
All valves closed.

$$v_3 = v_2 = v_{TDC} \tag{43}$$

$$w_{2-3} = 0 \tag{44}$$

$$Q_{2-3} = Q_{in} = m_f\, Q_{HV}.\eta_c = m_f\, Q_{HV}.\eta_c \tag{45}$$

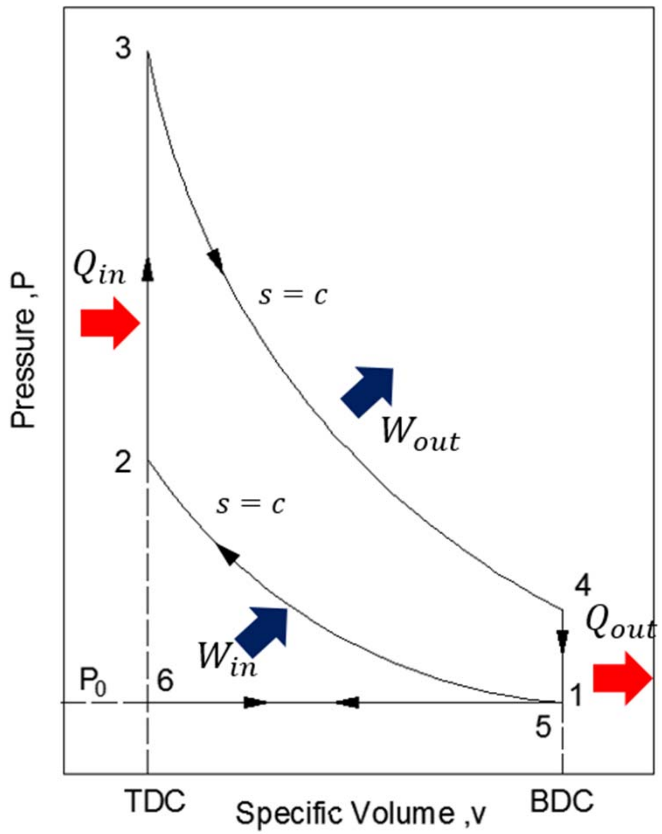
$$= (m_a + m_f)c_v\,(T_3 - T_2) \tag{46}$$

$$Q_{HV}\eta_c = (AF+1)c_v(T_3 - T_2) \tag{47}$$

$$q_{2-3} = q_{in} = c_v(T_3 - T_2) = (u_3 - u_2) \tag{48}$$

$$T_3 = T_{max} \tag{49}$$

$$P_3 = P_{max} \tag{50}$$



$\frac{P_3}{P_2} = \frac{T_3}{T_2} = \rho = \alpha \Longrightarrow$ is defined as pressure ratio.(Which defines the rise of pressure during combustion)
both $\boldsymbol{\rho}$ *and* $\boldsymbol{\alpha}$ *notation can be used for pressure ratio,* $\boldsymbol{\rho}$ ~2 ÷4

Air Standard - OTTO CYCLE

Process 3-4

Isentropic power or expansion stroke.
All valves closed.

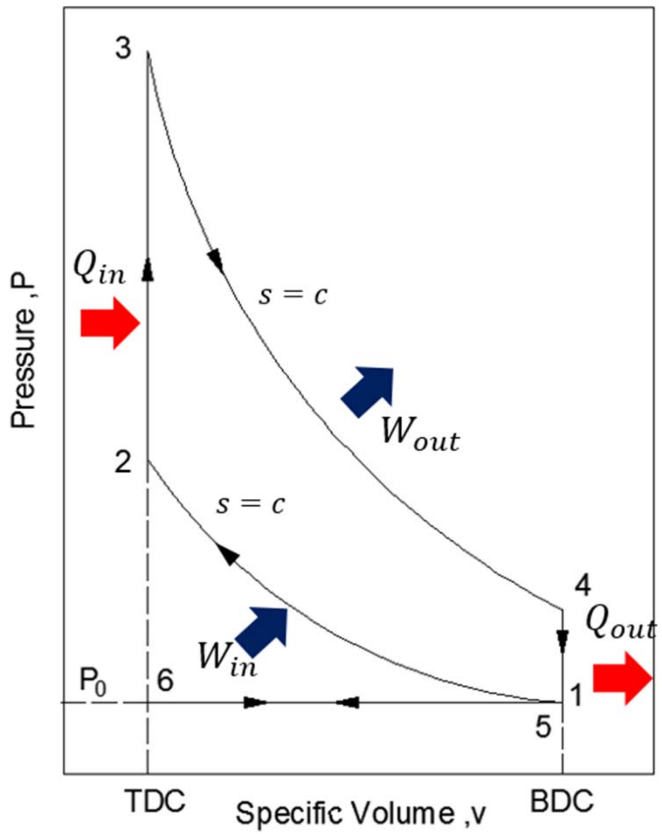
$$q_{3-4} = 0 \quad \boxed{51}$$

$$T_4 = T_3(\frac{v_3}{v_4})^{k-1} = T_3(\frac{V_3}{V_4})^{k-1} = T_3/(\varepsilon^{k-1}) \quad \boxed{52}$$

$$P_4 = P_3(\frac{v_3}{v_4})^{k} = P_3(\frac{V_3}{V_4})^{k} = P_3/(\varepsilon^{k}) \quad \boxed{53}$$

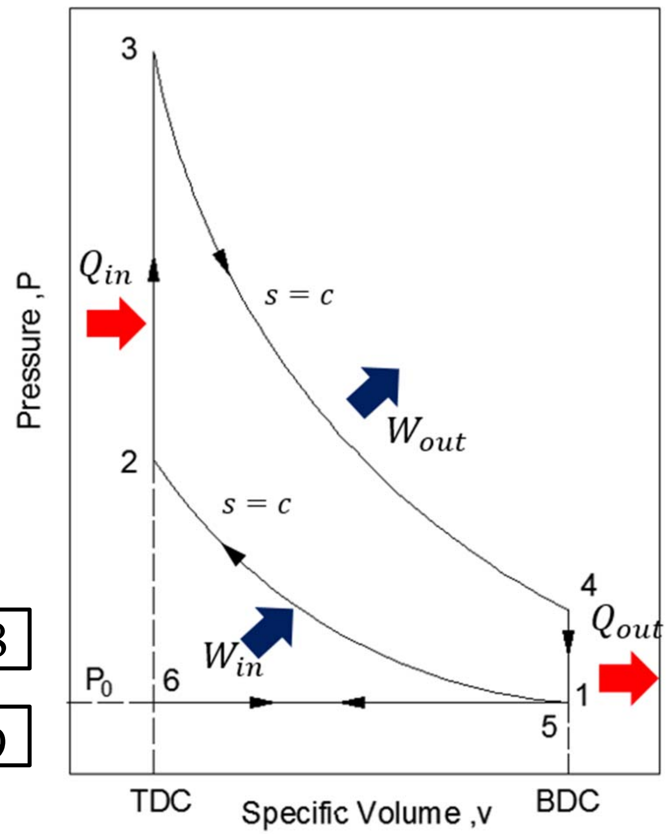
$$w_{3-4} = \frac{P_4 v_4 - P_3 v_2}{1-k} = \frac{R(T_4 - T_3)}{1-k} \quad \boxed{54}$$

$$= (u_3 - u_4) = c_v(T_3 - T_4) \quad \boxed{55}$$



Air Standard - OTTO CYCLE

Process 4-5 & Process 4-1
Constant volume heat rejection (exhaust blow down) .
Exhaust valve open and intake valve closed



$$v_4 = v_5 = v_1 = v_{BDC} \quad \boxed{56}$$

$$w_{4-5} = 0 \quad \boxed{57}$$

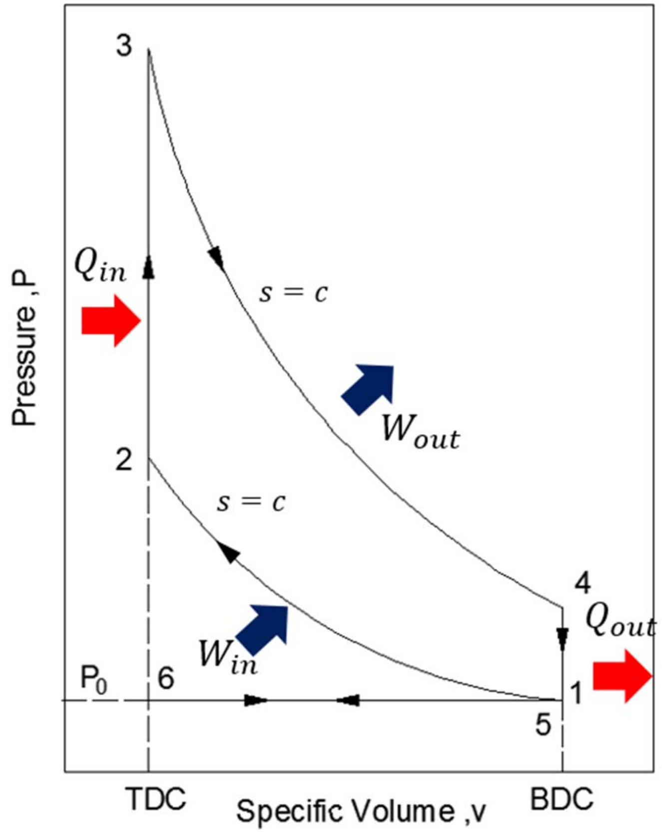
$$Q_{4-5} = Q_{out} = m_m\, c_v.(T_5 - T_4) = m_m\, c_v.(T_1 - T_4) \quad \boxed{58}$$

$$q_{4-5} = Q_{out} = c_v.(T_5 - T_4) = (u_5 - u_4) = c_v.(T_1 - T_4) \quad \boxed{59}$$

Air Standard - OTTO CYCLE

Process 5-6

constant pressure intake of air at P_o. Intake valve open and exhaust valve closed



$$P_5 = P_6 = P_0 \qquad \boxed{60}$$

$$w_{5-6} = P_0(v_6 - v_5) \qquad \boxed{61}$$

$$w_{5-6} = P_0(v_6 - v_1) \qquad \boxed{62}$$

Thermal Efficiency of OTTO CYCLE

$$\eta_{t_{OTTO}} = \frac{|w_{net}|}{|q_{in}|} = 1 - \left(\frac{|q_{out}|}{|q_{in}|}\right) \quad \boxed{63} \quad \Rightarrow \quad \eta_{t_{OTTO}} = 1 - \left[\frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)}\right] \quad \boxed{64} \quad \Rightarrow \quad \eta_{t_{OTTO}} = 1 - \left[\frac{(T_4 - T_1)}{(T_3 - T_2)}\right] \quad \boxed{65}$$

Only cycle temperatures need to be known to determine thermal efficiency. This can be simplified further applying ideal gas relationships for the isentropic compression and expansion strokes and recognizing that $v_1 = v_4 \; and \;\; v_2 = v_3$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{v_4}{v_3}\right)^{k-1} = \left(\frac{T_3}{T_4}\right) \quad \boxed{66}$$

Rearranging the temperature terms gives:

$$\left(\frac{T_4}{T_1}\right) = \left(\frac{T_3}{T_2}\right) \quad \boxed{67}$$

Equation 63-65 can be rearranged as:

$$\eta_{t_{OTTO}} = 1 - \left(\frac{T_1}{T_2}\right)\left\{\frac{[(T_4 - T_1) - 1]}{[(T_3 - T_2) - 1]}\right\} \quad \boxed{68} \quad \Rightarrow$$

Via usage Equation 67 gives:

$$\eta_{t_{OTTO}} = 1 - \left(\frac{T_1}{T_2}\right) \quad \boxed{69}$$

Thermal Efficiency of OTTO CYCLE

Combining equation 38 and 69 gives:

$$\eta_{t_{OTTO}} = 1 - \left[\frac{1}{\left(\frac{v_1}{v_2}\right)^{k-1}}\right] \quad \boxed{70}$$

With $\frac{v_1}{v_2} = \varepsilon$ (compression ratio)

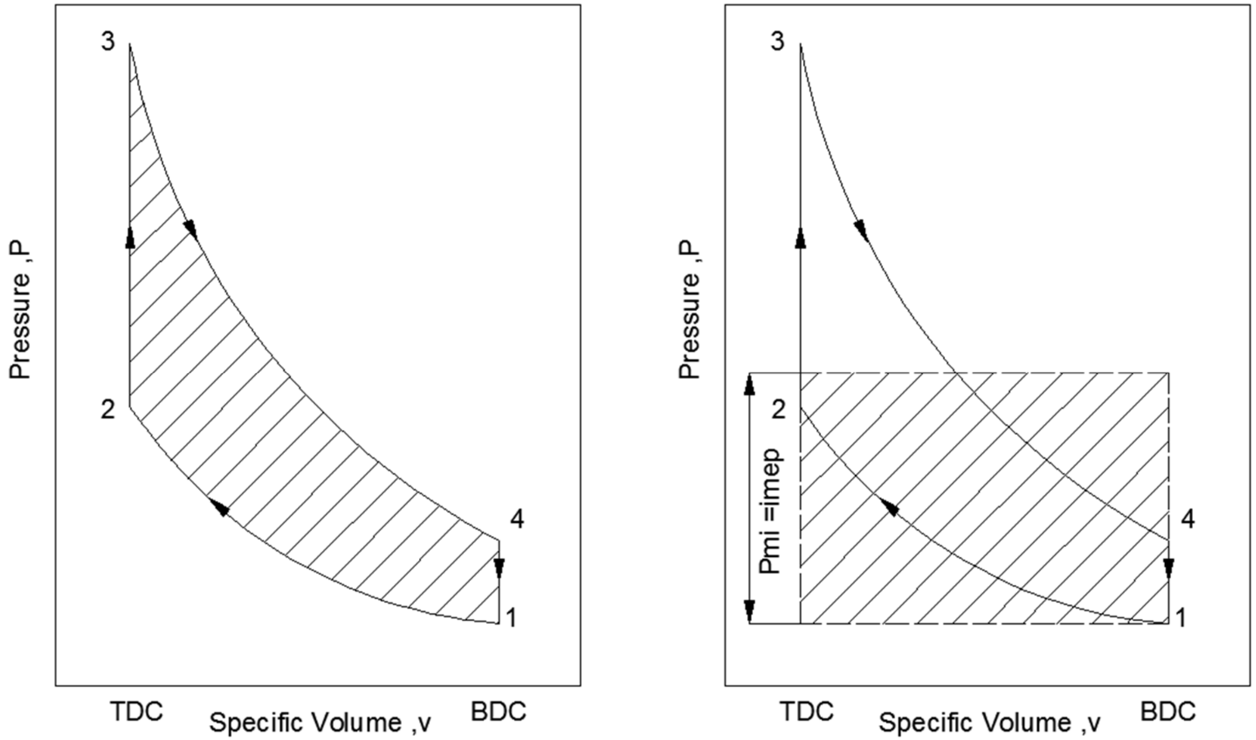
$$\eta_{t_{OTTO}} = 1 - \left[\frac{1}{(\varepsilon)^{k-1}}\right] \quad \boxed{71}$$

Only the compression ratio is needed to determine the thermal efficiency of the Otto cycle at WOT. As the compression ratio goes up, the thermal efficiency goes up. This efficiency is the indicated thermal efficiency, as the heat transfer values are those to and from the air within the combustion chamber.

Mean Indicated Pressure of OTTO CYCLE

$imep = P_{mi} = w_i / \Delta v$

Both P_{mi} and imep notation is used for defining indicated mean effective pressure



Both w_i or $L_ç$ can be used for defining the indicated work.

The diagram area which corresponds to w_i or $L_ç$ can be changed with a rectangular area which its base is V1-V2=Vh

Mean Indicated Pressure of OTTO CYCLE

imep= $P_{mi} = \frac{w_i}{V_1 - V_2}\left(\frac{N.m}{m^3} = \frac{N}{m^2}\right)$

⇨ $\boldsymbol{imep}$ (indicated mean effective pressure) or $\boldsymbol{P_{mi}}$ (P mean indicated) shows the specific work
⇨ $\boldsymbol{imep}$ or $\boldsymbol{P_{mi}}$ describes the work done by unit volume

It is seen from the equation the unit of specific work is equal to a pressure unit (N/m²) that this pressure acts on piston from V1 to V2. The community commonly uses 'the mean indicated pressure' term instead of specific work.

imep= $\frac{w_i}{V_1 - V_2}$

$$w_i = Q_1 - Q_2$$

$$\eta_t = \frac{Q_1 - Q_2}{Q_1} = \frac{w_i}{Q_1}$$

$$w_i = \eta_t Q_1$$

$$Q_1 = mc_v(T_3 - T_2) = mc_v T_2\left(\frac{T_3}{T_2} - 1\right) = mc_v T_2(\rho - 1)$$

Mean Indicated Pressure of OTTO CYCLE

$$\text{imep} = P_{mi} = \frac{\eta_t Q_1}{V_1 - V_2} = \eta_t \frac{m c_v T_2 (\rho - 1)}{V_2\left(\frac{V_1}{V_2} - 1\right)} = \eta_t \frac{m c_v T_1 \varepsilon^{k-1} (\rho - 1)}{\frac{V_1}{\varepsilon}(\varepsilon - 1)}$$

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

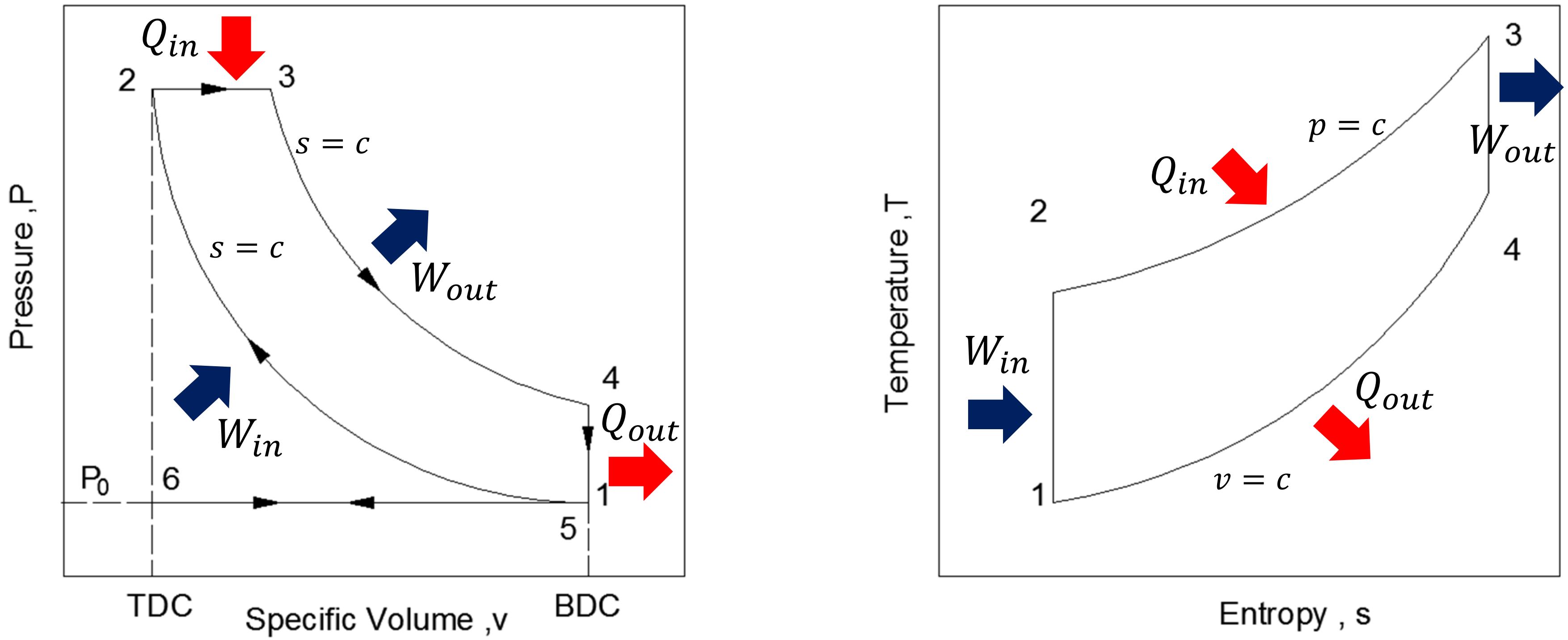
$$\left.\begin{array}{l} \dfrac{c_p}{c_v} = k \\ \\ c_p - c_v = R \end{array}\right\} \quad \frac{c_p}{c_v} = k = \frac{R + c_v}{c_v} \Rightarrow k c_v - c_v = R \Rightarrow c_v = \frac{R}{k - 1}$$

$$imep = \eta_t \frac{\frac{P_1 V_1}{T_1 R} \frac{R}{k-1} T_1 \varepsilon^{k-1} (\rho - 1)}{\frac{V_1}{\varepsilon}(\varepsilon - 1)}$$

$$imep = \eta_t \frac{P_1}{R} \frac{R}{k-1} \frac{\varepsilon^{k-1} (\rho - 1)}{\frac{1}{\varepsilon}(\varepsilon - 1)}$$

$$imep = \eta_t \frac{P_1}{k-1} \frac{\varepsilon^{k}}{(\varepsilon - 1)} (\rho - 1) \qquad \boldsymbol{\rho} \sim 2 \div 4$$

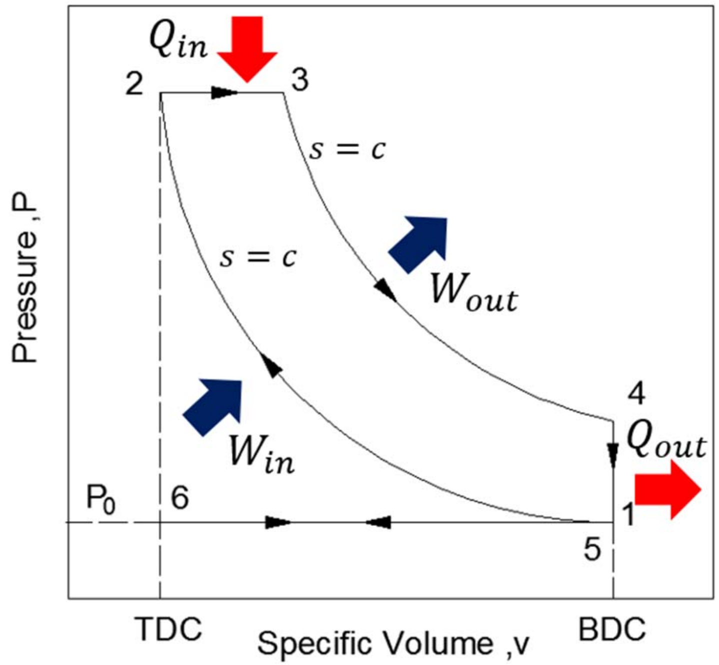
Air Standard - DIESEL CYCLE



Air Standard - DIESEL CYCLE

Process 6-1

constant pressure intake of air at P_o. Intake valve open and exhaust valve closed

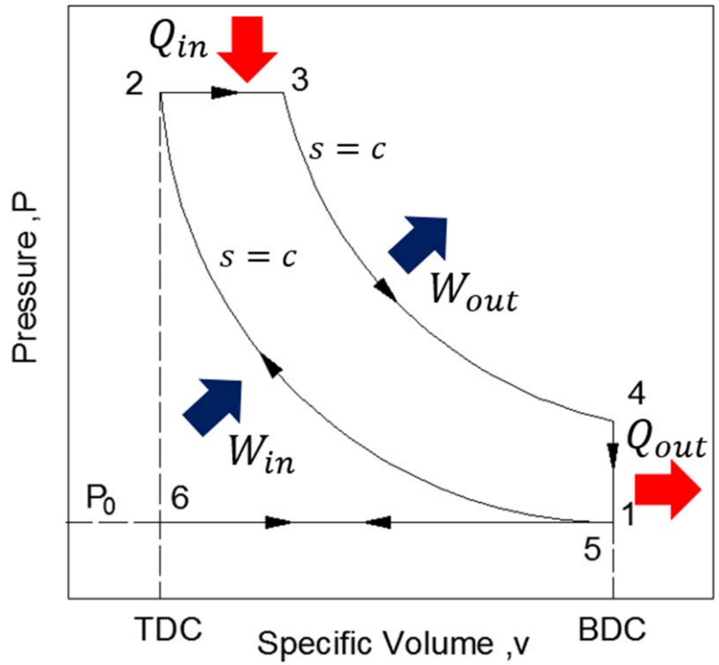


$$P_1 = P_6 = P_0 \quad \boxed{72}$$

$$w_{6-1} = P_0(v_1 - v_6) \quad \boxed{73}$$

Air Standard - DIESEL CYCLE

Process 1-2
isentropic compression stroke.
All valves closed.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = T_1\varepsilon^{k-1} \qquad \boxed{74}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1\left(\frac{V_1}{V_2}\right)^{k} = P_1\varepsilon^{k} \qquad \boxed{75}$$

$$q_{1-2} = 0 \qquad \boxed{76}$$

$$w_{1-2} = \frac{P_2v_2 - P_1v_1}{1-k} = \frac{R(T_2 - T_1)}{1-k} \qquad \boxed{77}$$

$$= (u_1 - u_2) = c_v(T_1 - T_2) \qquad \boxed{78}$$

Air Standard- DIESEL CYCLE

Process 2-3

Constant pressure heat input(combustion).
All valves closed.

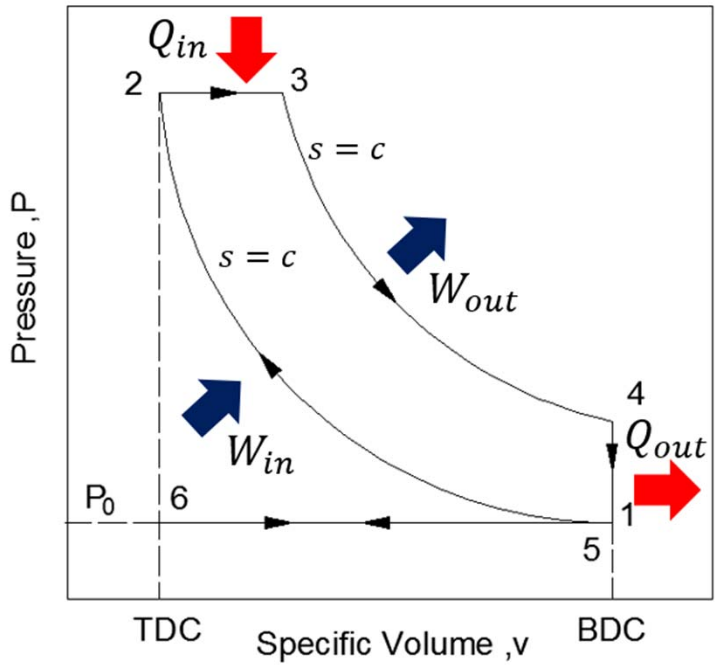
$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_p (T_3 - T_2) = (m_a + m_f) c_p (T_3 - T_2)$$

$$Q_{HV} \eta_c = (AF + 1)\, c_p (T_3 - T_2)$$

$$q_{2-3} = q_{in} = c_p (T_3 - T_2) = h_3 - h_2$$

$$w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2 (v_3 - v_2)$$

$$T_3 = T_{max}$$

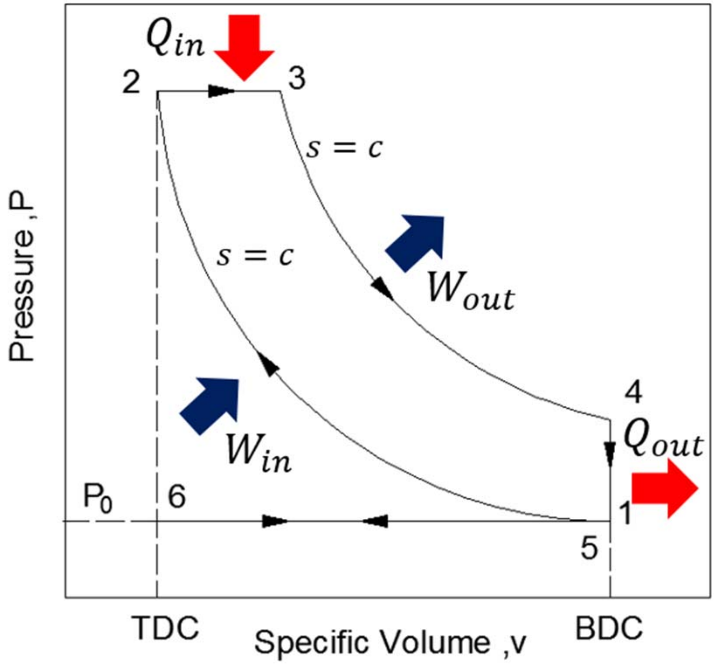


Air Standard - DIESEL CYCLE

Cutoff Ratio

Is defined as the changein volume that occurs during combustion , given as a ratio.

$$\varepsilon_g = \beta = \frac{V_3}{V_2} = \frac{v_3}{v_2} = \frac{T_3}{T_2}$$



$$\varepsilon_g = \beta \Longrightarrow Both\ two\ notation\ is\ used\ for\ cutoff\ \ ratio$$

Air Standard - DIESEL CYCLE

Process 3-4

Isentropic power or expansion stroke.
All valves closed.

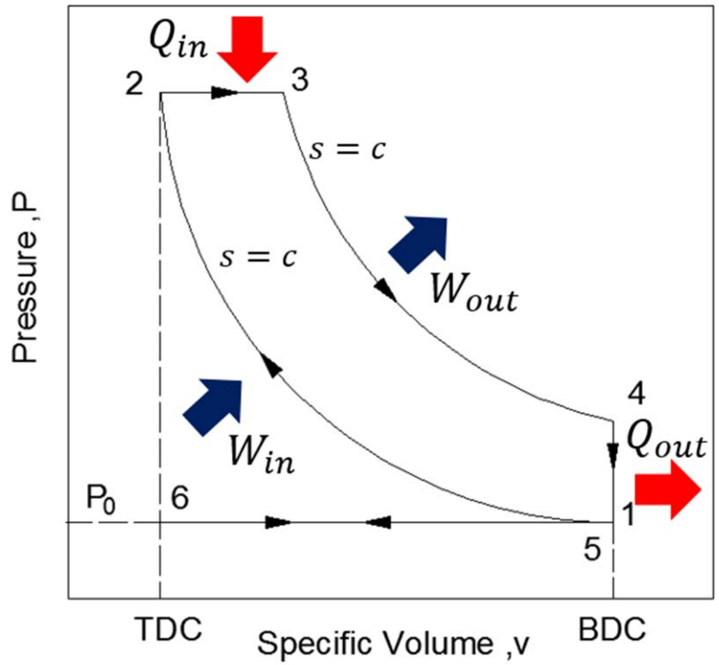
$$q_{3-4} = 0$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{V_3}{V_4}\right)^{k-1}$$

$$P_4 = P_3\left(\frac{v_3}{v_4}\right)^{k} = P_3\left(\frac{V_3}{V_4}\right)^{k}$$

$$w_{3-4} = \frac{P_4 v_4 - P_3 v_3}{1-k} = \frac{R(T_4 - T_3)}{1-k}$$

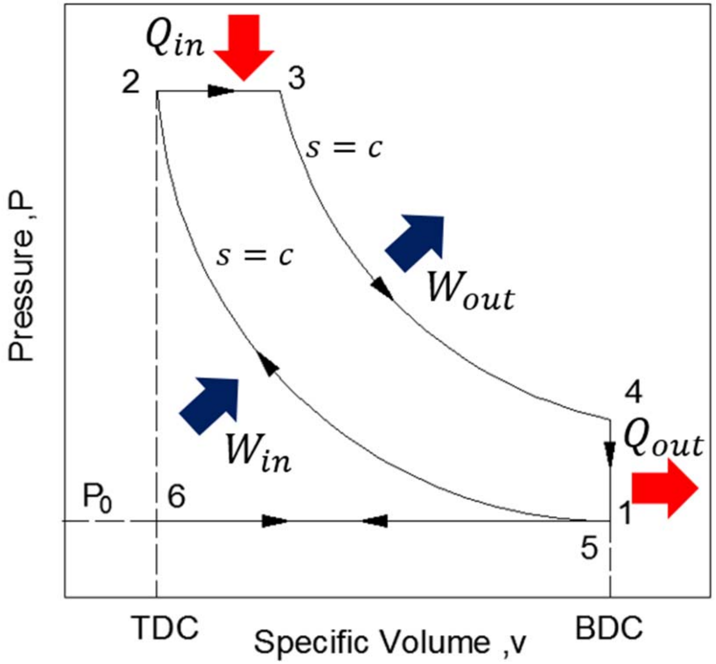
$$= (u_3 - u_4) = c_v(T_3 - T_4)$$



Air Standard - DIESEL CYCLE

Process 4-5

Constant volume heat rejection (exhaust blowdown) .
Exhaust valve open and intake valve closed



$$v_4 = v_5 = v_1 = v_{BDC}$$

$$w_{4-5} = 0$$

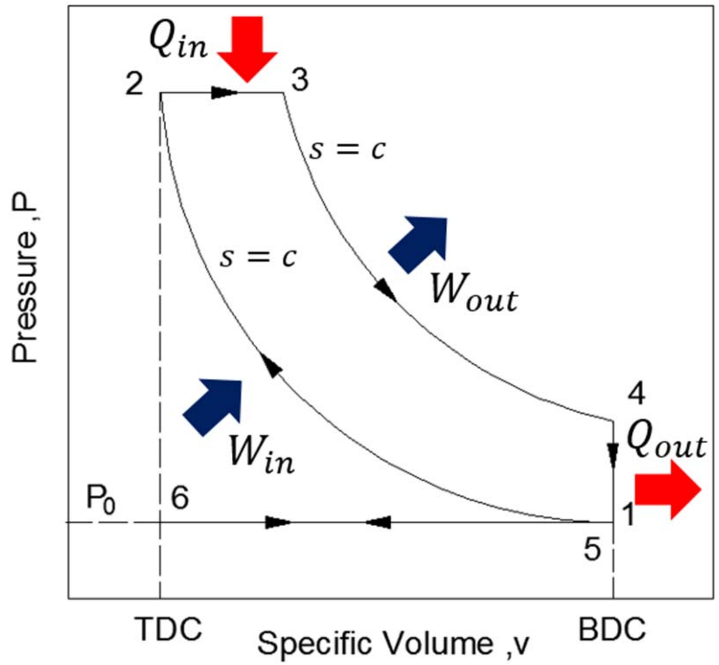
$$Q_{4-5} = Q_{out} = m_m\, c_v.(T_5 - T_4) = m_m\, c_v.(T_1 - T_4)$$

$$q_{4-5} = Q_{out} = c_v.(T_5 - T_4) = (u_5 - u_4) = c_v.(T_1 - T_4)$$

Air Standard - DIESEL CYCLE

Process 5-6

constant pressure intake of air at P_o. Intake valve open and exhaust valve closed



$$P_5 = P_6 = P_0$$

$$w_{5-6} = P_0(v_6 - v_5) = P_0\ (v_6 - v_1)$$

Thermal Efficiency of DIESEL CYCLE

$$\eta_{t_{DIESEL}} = \frac{|w_{net}|}{|q_{in}|} = 1 - \left(\frac{|q_{out}|}{|q_{in}|}\right) \Rightarrow \eta_{t_{DIESEL}} = 1 - \left[\frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)}\right] \Rightarrow \eta_{t_{OTTO}} = 1 - \left[\frac{(T_4 - T_1)}{k(T_3 - T_2)}\right]$$

With rearrangement of equation XX diesel thermal efficiency is obtained as.

$$\eta_{t_{DIESEL}} = 1 - \left(\frac{1}{\varepsilon}\right)^{k-1} \left\{\frac{({\varepsilon_g}^k - 1)}{k(\varepsilon_g - 1)}\right\}$$

Comparing these two efficiency, it is seen that for a given certain compression ratio the thermal efficiency of otto cycle is greater than the diesel cycle.

$$\eta_{t_{OTTO}} = 1 - \left[\frac{1}{(\varepsilon)^{k-1}}\right] \qquad \eta_{t_{DIESEL}} = 1 - \left(\frac{1}{\varepsilon}\right)^{k-1} \boxed{\left\{\frac{({\varepsilon_g}^k - 1)}{k(\varepsilon_g - 1)}\right\}}$$

Greater than >1

However the SI engine CR~ 8-12 and CI engine CR~18-22

Mean Indicated Pressure of DIESEL CYCLE

$$\boxed{imep = P_{mi} = w_i / \Delta v}$$

Both P_{mi} and imep notation is used for defining indicated mean effective pressure

Both w_i or $L_ç$ can be used for defining the indicated work.

The diagram area which corresponds to w_i or $L_ç$ can be changed with a rectangular area which its base is V1-V2=Vh

Mean Indicated Pressure of DIESEL CYCLE

imep= $\frac{w_i}{V_1-V_2}$

$$w_i = Q_1 - Q_2$$

$$\eta_t = \frac{Q_1 - Q_2}{Q_1} = \frac{w_i}{Q_1}$$

$$w_i = \eta_t Q_1 = \eta_t c_p (T_3 - T_2) = \eta_t c_p T_2 \left(\frac{T_3}{T_2} - 1\right)$$

$$\left.\begin{array}{l} \dfrac{c_p}{c_v} = k \\ \\ c_p - c_v = R \end{array}\right\} \quad c_p(1 - c_v/c_r) = \mathrm{R} \quad \Rightarrow \quad c_p = \frac{R}{1 - \dfrac{1}{k}} = \frac{k.R}{k-1}$$

$$T_2 = T_1 \varepsilon^{k-1}$$

$$\frac{T_3}{T_2} = \varepsilon_g$$

Mean Indicated Pressure of DIESEL CYCLE

$$w_i = \eta_t \frac{kR}{k-1} T_1 \varepsilon^{k-1}(\varepsilon_g - 1)$$

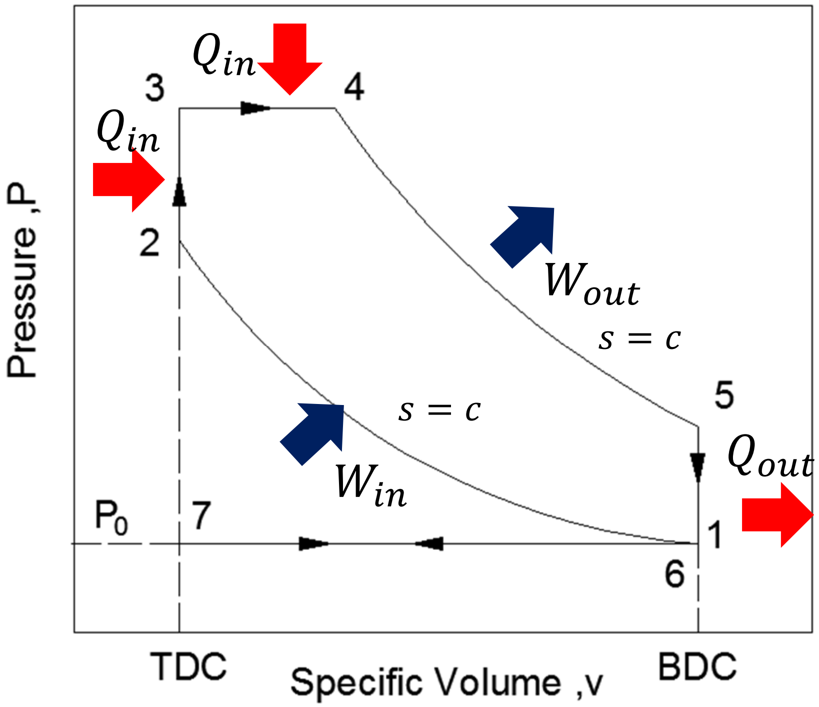
$$V_H = V_1 - V_2 = V_2\left(\frac{V_1}{V_2} - 1\right) = \frac{V_1}{\varepsilon}(\varepsilon - 1) = \frac{\varepsilon}{V_1(\varepsilon - 1)}$$

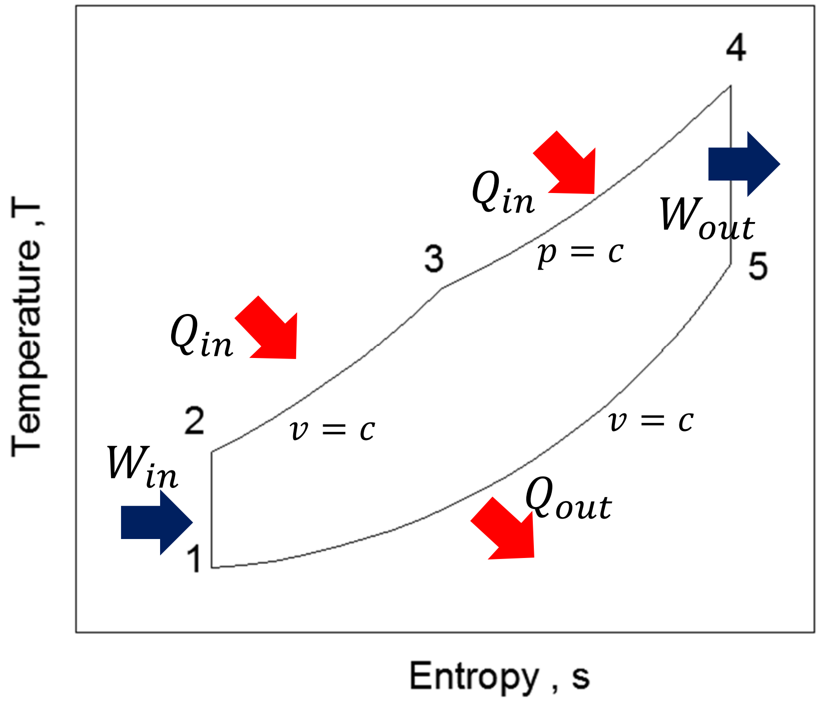
$$imep = \eta_t \frac{kR}{k-1} \frac{T_1}{V_1} \frac{\varepsilon^{k-1}\varepsilon(\varepsilon_g - 1)}{(\varepsilon - 1)}$$

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

$$imep = \eta_t \frac{k}{k-1} \frac{P_1\, \varepsilon^{k-1}(\varepsilon_g - 1)}{(\varepsilon - 1)}$$

Air Standard - SEILIGER CYCLE



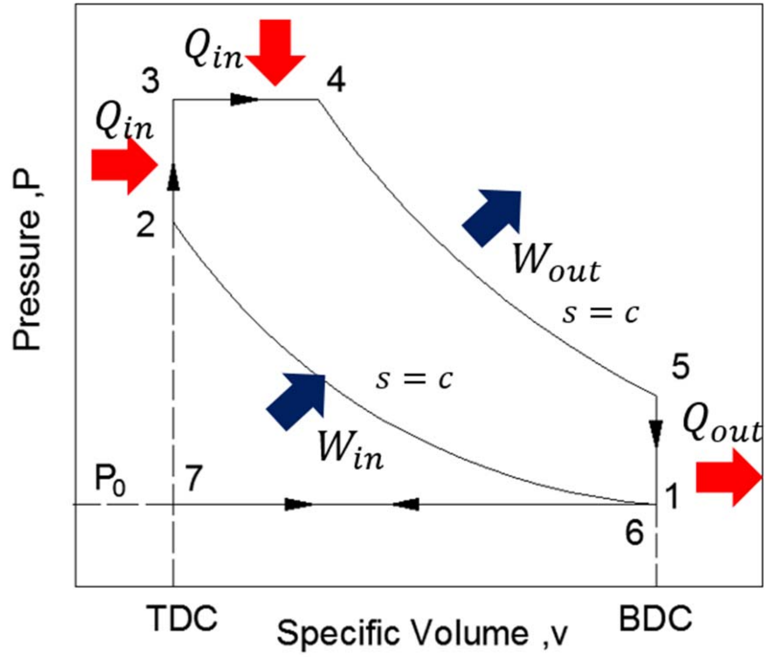


Air Standard - SEILINGER CYCLE

dual cycle

Process 7-1

constant pressure intake of air at P_o. Intake valve open and exhaust valve closed



$$P_1 = P_7 = P_0$$

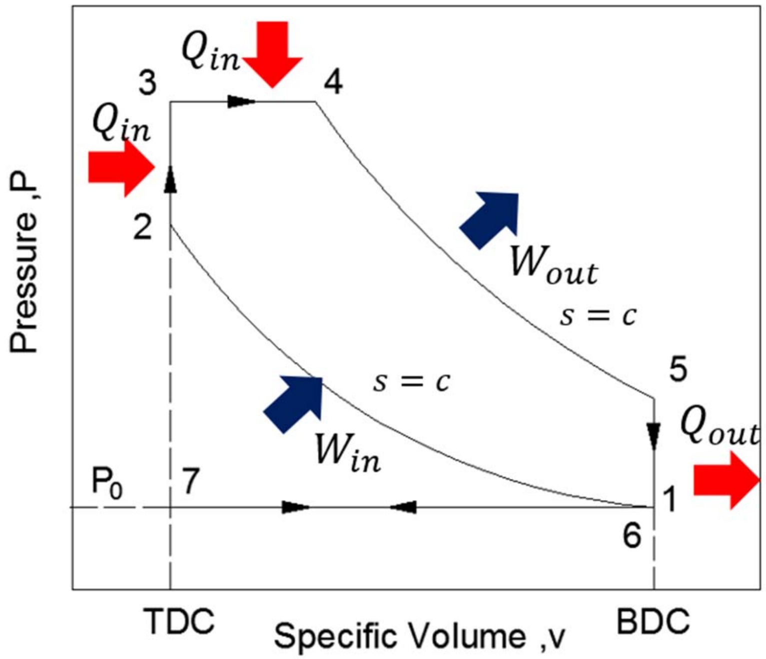
$$w_{7-1} = P_0(v_1 - v_7)$$

Air Standard - SEILINGER CYCLE

dual cycle

Process 1-2

isentropic compression stroke.
All valves closed.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = T_1\varepsilon^{k-1}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1\left(\frac{V_1}{V_2}\right)^{k} = P_1\varepsilon^{k}$$

$$q_{1-2} = 0$$

$$w_{1-2} = \frac{P_2v_2 - P_1v_1}{1-k} = \frac{R(T_2 - T_1)}{1-k}$$

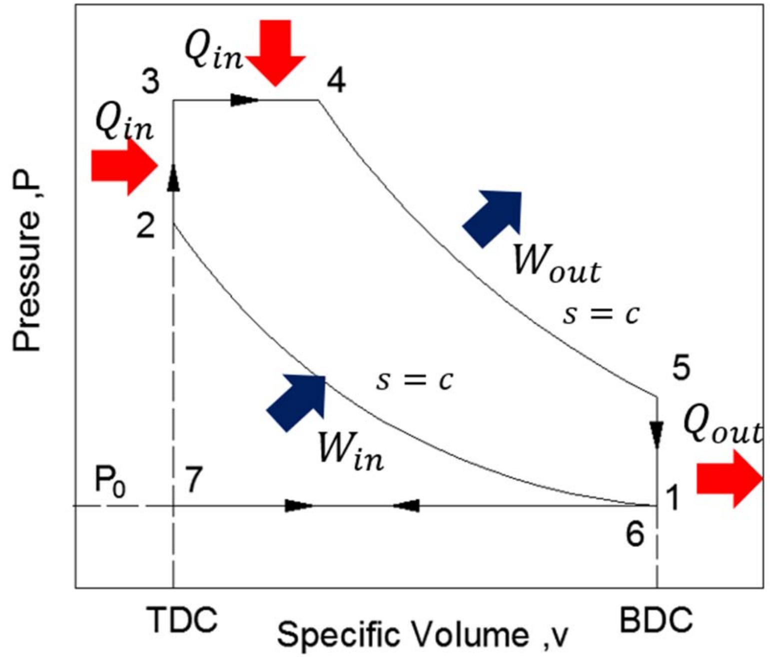
$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

Air Standart - SEILINGER CYCLE

dual cycle

Process 2-3

Constant volume heat input (combustion)
All valves closed.



$$v_3 = v_2 = v_{TDC}$$

$$w_{2-3} = 0$$

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

$$q_{2-3} = c_v(T_3 - T_2) = (u_3 - u_2)$$

$$P_3 = P_4 = P_{max}$$

$\dfrac{P_3}{P_2} = \dfrac{T_3}{T_2} = \rho = \alpha \Longrightarrow$ is defined as pressure ratio.(Which defines the rise of pressure during combustion)

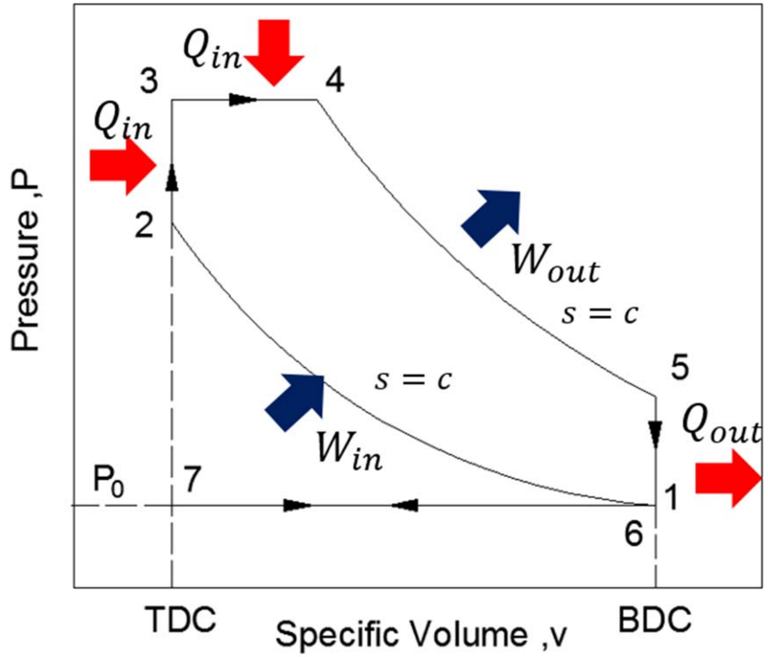
$both\ \boldsymbol{\rho}\ \ and\ \ \boldsymbol{\alpha}\ \ notation\ can\ be\ used\ for\ pressure\ ratio$

Air Standart - SEILINGER CYCLE

dual cycle

Process 3-4

Constant pressure heat input(combustion).
All valves closed.



$$P_4 = P_3 = P_{max}$$

$$Q_{3-4} = m_m c_p(T_4 - T_3) = (m_a + m_f)c_p(T_4 - T_3)$$

$$q_{3-4} = c_p(T_3 - T_4) = h_4 - h_3$$

$$w_{3-4} = q_{3-4} - (u_4 - u_3) = P_3(v_4 - v_3) = P_4(v_4 - v_3)$$

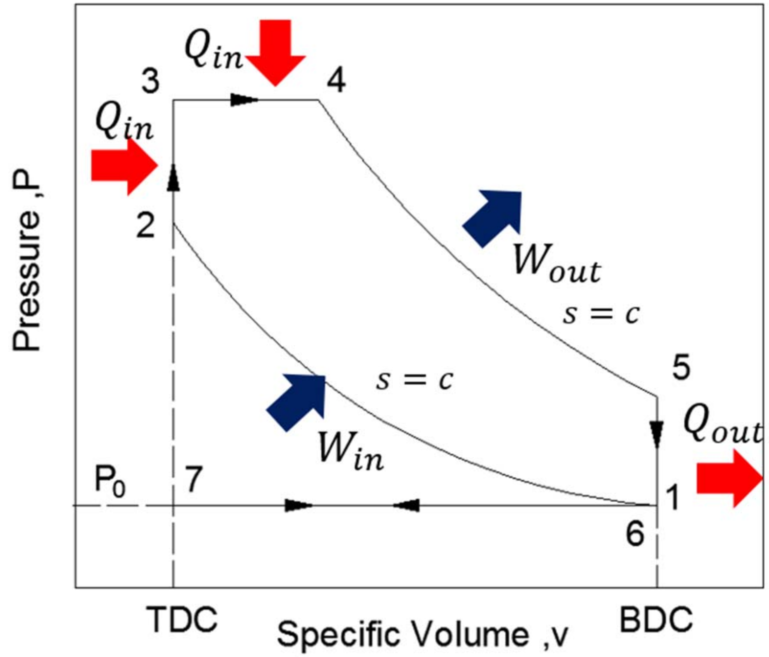
$$T_4 = T_{max}$$

Air Standard- SEILINGER CYCLE

dual cycle

Cutoff Ratio:

Is defined as the change in volume that occurs during combustion , given as a ratio.



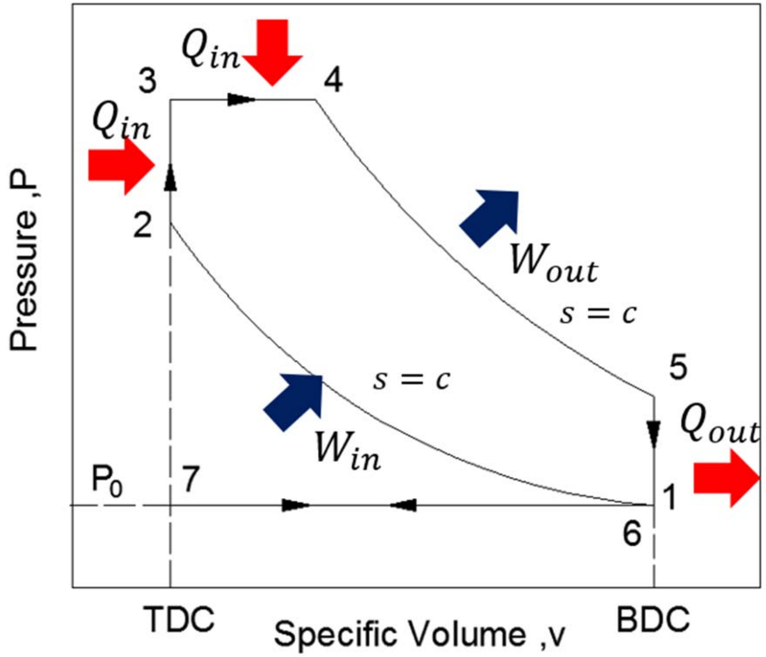
$$\varepsilon_g = \beta = \frac{V_4}{V_3} = \frac{v_4}{v_3} = \frac{T_4}{T_3}$$

$$\varepsilon_g = \beta \Longrightarrow Both\ two\ notation\ is\ used\ for\ cutoff\ \ ratio$$

Air Standard - SEILINGER CYCLE

dual cycle

Heat IN



$$Q_{in} = Q_{2-3} + Q_{3-4} = m_f Q_{HV} \eta_c$$

$$q_{in} = q_{2-3} + q_{3-4} = (u_3 - u_2) + (h_4 - h_3)$$

Air Standard - SEILINGER CYCLE

Process 4-5

Isentropic power or expansion stroke.
All valves closed.

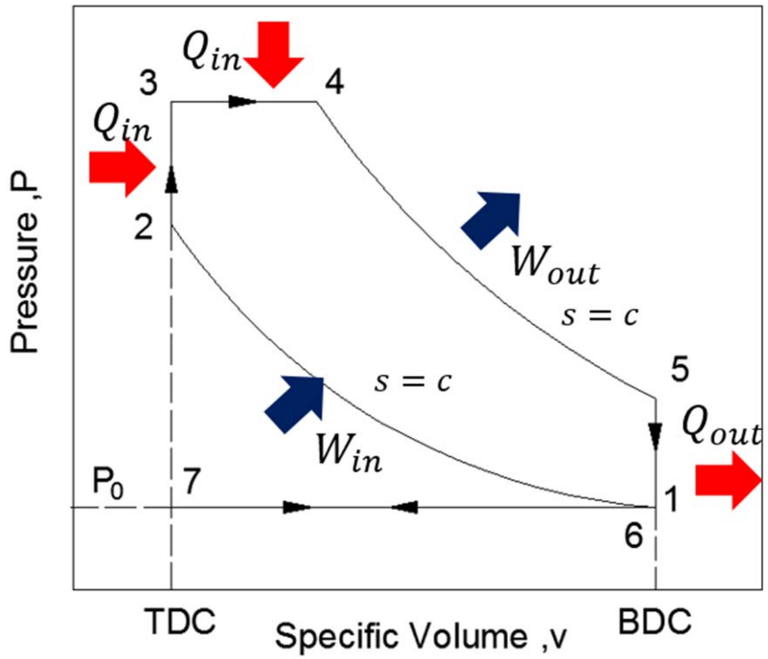
$$q_{4-5} = 0$$

$$T_5 = T_4\left(\frac{v_4}{v_5}\right)^{k-1} = T_4\left(\frac{V_4}{V_5}\right)^{k-1} = T_4\left(\frac{\varepsilon_g}{\varepsilon}\right)^{k-1}$$

$$P_5 = P_5\left(\frac{v_4}{v_5}\right)^{k} = P_5\left(\frac{V_4}{V_5}\right)^{k} = P_4\left(\frac{\varepsilon_g}{\varepsilon}\right)^{k-1}$$

$$w_{4-5} = \frac{P_5 v_5 - P_4 v_4}{1-k} = \frac{R(T_5 - T_4)}{1-k}$$

$$= (u_5 - u_4) = c_v(T_5 - T_4)$$

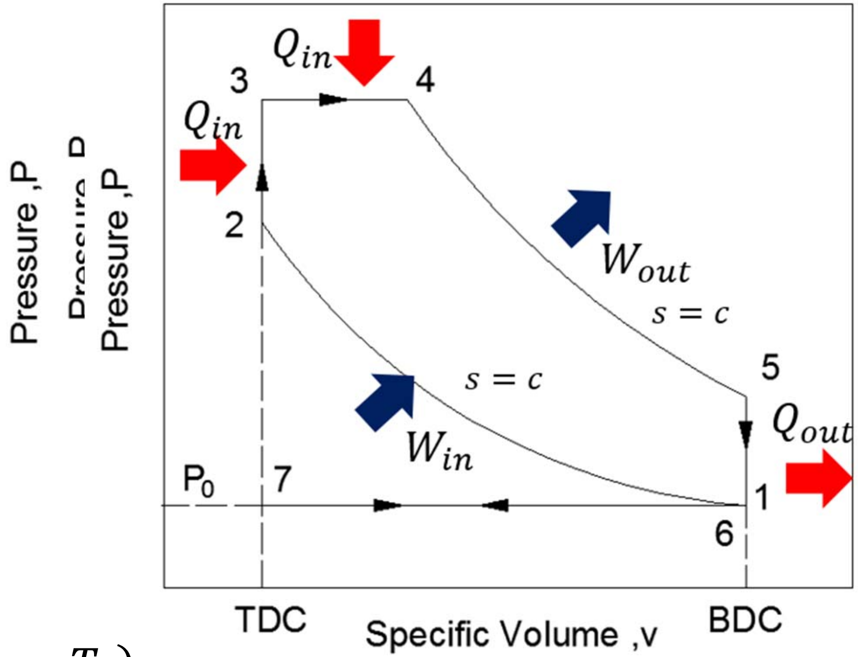


Air Standard - SEILINGER CYCLE

Process 5-1 & Process 5-1

Isentropic power or expansion stroke.

All valves closed.



$$v_5 = v_6 = v_1 = v_{BDC}$$

$$w_{5-1} = 0$$

$$Q_{5-1} = Q_{out} = m_m\, c_v.(T_6 - T_5) = m_m\, c_v.(T_1 - T_5)$$

$$q_{5-1} = Q_{out} = c_v.(T_6 - T_5) = (u_6 - u_5) = c_v.(T_1 - T_5)$$

Thermal Efficiency of Air Standard - SEILINGER CYCLE

- dual cycle

$$\eta_{t_{SEILINGER}} = \frac{|w_{net}|}{|q_{in}|} = 1 - \left(\frac{|q_{out}|}{|q_{in}|}\right) \Rightarrow \eta_{t_{SEILINGER}} = 1 - \left[\frac{c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}\right]$$

$$\eta_{t_{SEILINGER}} = 1 - \left[\frac{(T_5 - T_1)}{(T_3 - T_2) + k(T_4 - T_3)}\right]$$

Equation can be rearranged to :

$$\eta_{t_{SEILINGER}} = 1 - \left(\frac{1}{\varepsilon}\right)^{k-1} \left\{\frac{(\rho {\varepsilon_g}^k - 1)}{k\rho\left(\varepsilon_g - 1\right) + \rho - 1}\right\}$$

Mean Indicated Pressure of SEILINGER CYCLE

imep= $\frac{w_i}{V_1 - V_2}$

$$w_i = Q_1 - Q_2$$

$$\eta_t = \frac{Q_1 - Q_2}{Q_1} = \frac{w_i}{Q_1}$$

$$Q_1 = mc_v(T_3 - T_2) + mc_p(T_4 - T_3)$$

$$Q_1 = mc_v\left[T_2\left(\frac{T_3}{T_2} - 1\right) + \frac{c_p}{c_v}T_3\left(\frac{T_4}{T_3} - 1\right)\right]$$

$$Q_1 = mc_vT_2\left[\rho - 1 + k\,\frac{T_3}{T_2}\left(\varepsilon_g - 1\right)\right]$$

$$Q_1 = m\frac{R}{k-1}T_1\varepsilon^{k-1}\left[\rho - 1 + k\,\rho\left(\varepsilon_g - 1\right)\right]$$

Mean Indicated Pressure of SEILINGER CYCLE

$$V_H = V_1 - V_2 = V_2\left(\frac{V_1}{V_2} - 1\right) = \frac{V_1}{\varepsilon}(\varepsilon - 1) = \frac{\varepsilon}{V_1(\varepsilon - 1)}$$

$$imep = \frac{w_i}{V_H} = \eta_t \frac{mR}{k-1}\frac{T_1}{V_1}\frac{\varepsilon.\varepsilon^{k-1}}{\varepsilon-1}\left[\rho - 1 + k\rho(\varepsilon_g - 1)\right]$$

$$\frac{mT_1}{V_1} = \frac{P_1}{R}$$

$$imep = \eta_t \frac{R}{k-1}\frac{P_1}{R}\frac{\varepsilon^k}{\varepsilon-1}\left[\rho - 1 + k\rho(\varepsilon_g - 1)\right]$$

$$imep = \eta_t \frac{P_1}{k-1}\frac{\varepsilon^k}{\varepsilon-1}\left[\rho - 1 + k\rho(\varepsilon_g - 1)\right]$$

References of Week 3 Lecture Notes

- *Engineering Fundementals Of Internal Combustion Engines.*, William J.Pulkrabek., Prentice Hall.Inc.-1997
- *Internal Combustion Engines Lecture Notes*, Prof.Dr.Orhan DENİZ.-YTU-2008
- https://www.boundless.com

Thank You

Dr.Orkun ÖZENER