# AIR STANDARD CYCLES

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2006

# ASSUMPTIONS IN AIR STANDARD CYCLES

- The gas in the engine cylinder is air (a perfect gas), i.e., it obeys the gas laws and has constant specific heat.
- The physical constants of the gas in the engine cylinder are same as those of air at moderate temperatures.
- All the compression and expansion processes are adiabatic and they take place without any internal friction.
- Heat is supplied by bringing a hot body in contact with the cylinder at appropriate points during the process. Similarly, heat is rejected by bringing a cold body in contact with the cylinder at these points.
- The cycle is considered to be closed one and the same air is used again and again to repeat the cycle.
- No chemical reaction takes place in the engine cylinder.

# OTTO CYCLE

# Thermal Efficiency of Internal Combustion Engine

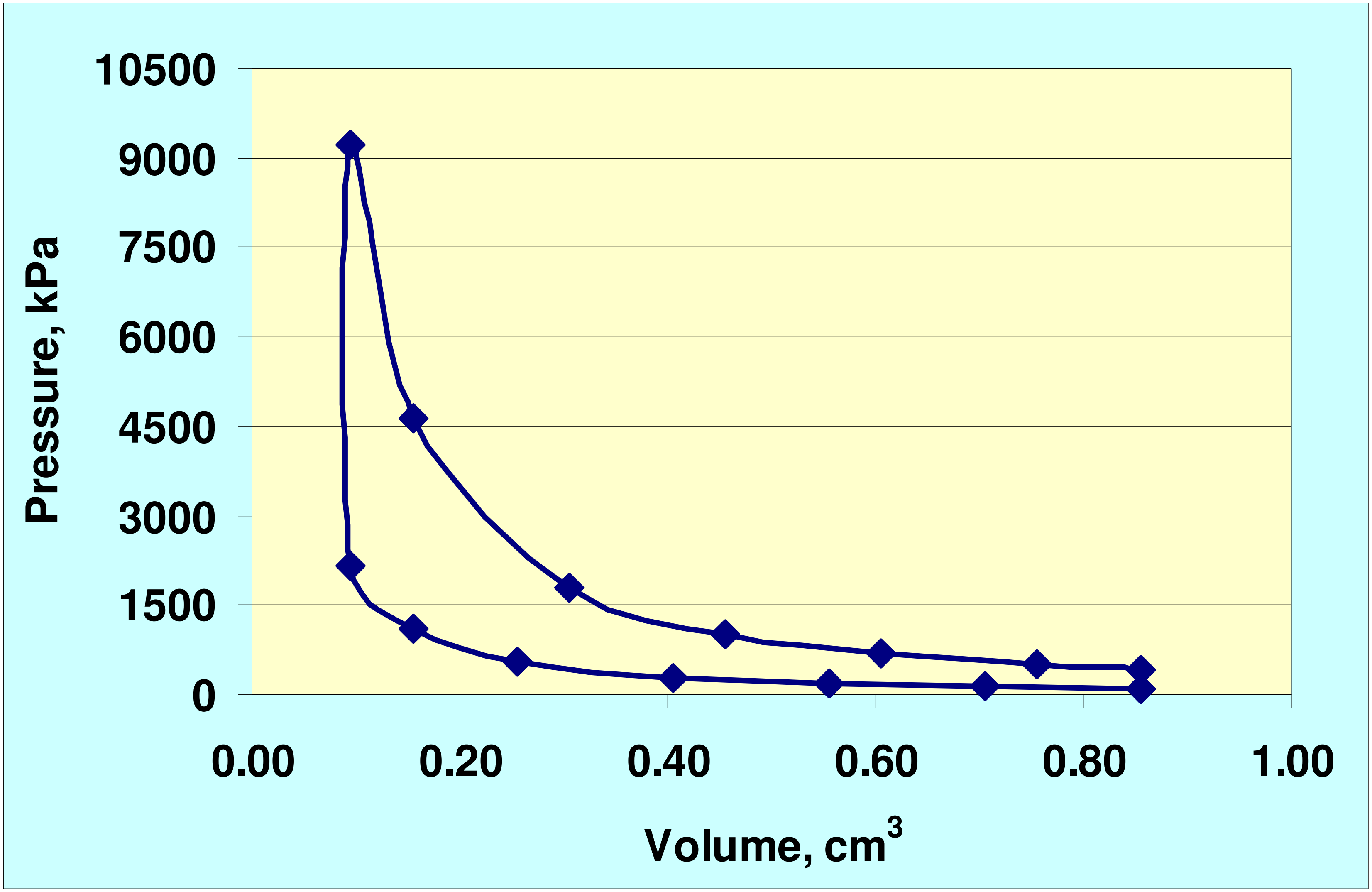
Thermal efficiency for Otto cycle:

$$\eta = 1 - \frac{1}{r_c^{\,k-1}}$$

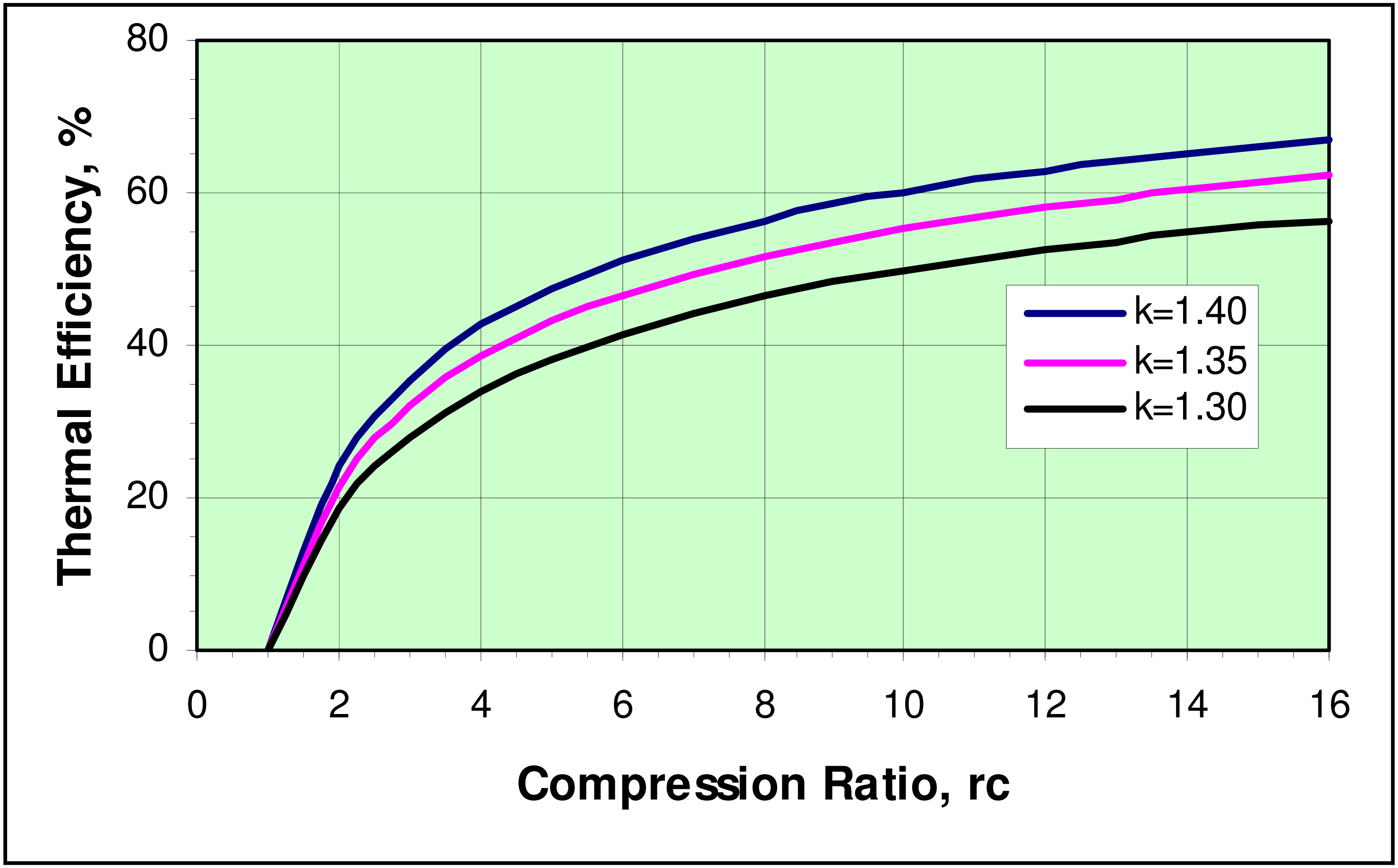
Where, compression ratio, $r_c = v_1/v_2$ and $k = c_p/c_v$

$\eta_{OTTO}$ increases with the increase of $r_c$ & $k$

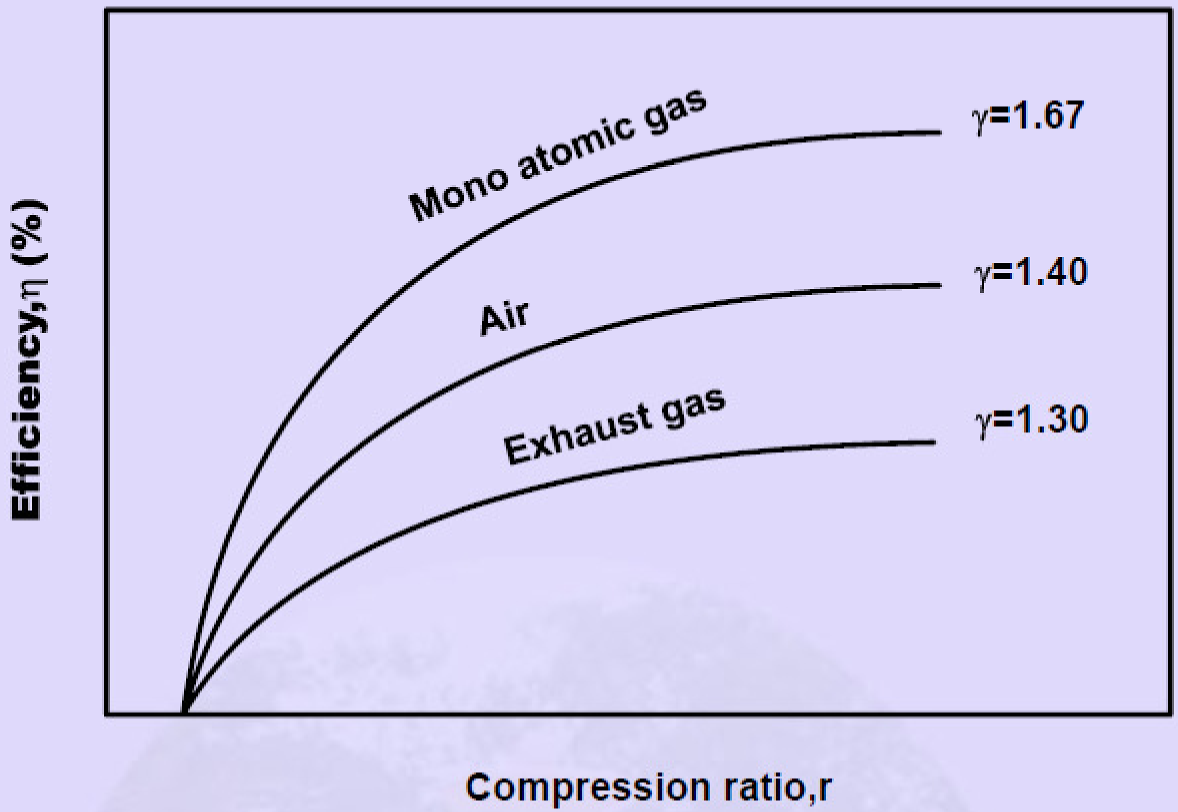
# OTTO CYCLE FOR SI ENGINE



# Thermal Efficiency of Otto Cycle



# VARIATION OF THERMAL EFFICIENCY WITH COMPRESSION RATIO




Effect of CR and $\gamma$ on efficiency for Otto cycle.

# DIESEL CYCLE

## Thermal Efficiency of Internal Combustion Engine

Thermal efficiency for Diesel cycle:

$$\eta = 1 - \frac{1}{r_c^{k-1}}\left[\frac{(\beta^k - 1)}{k(\beta - 1)}\right]$$

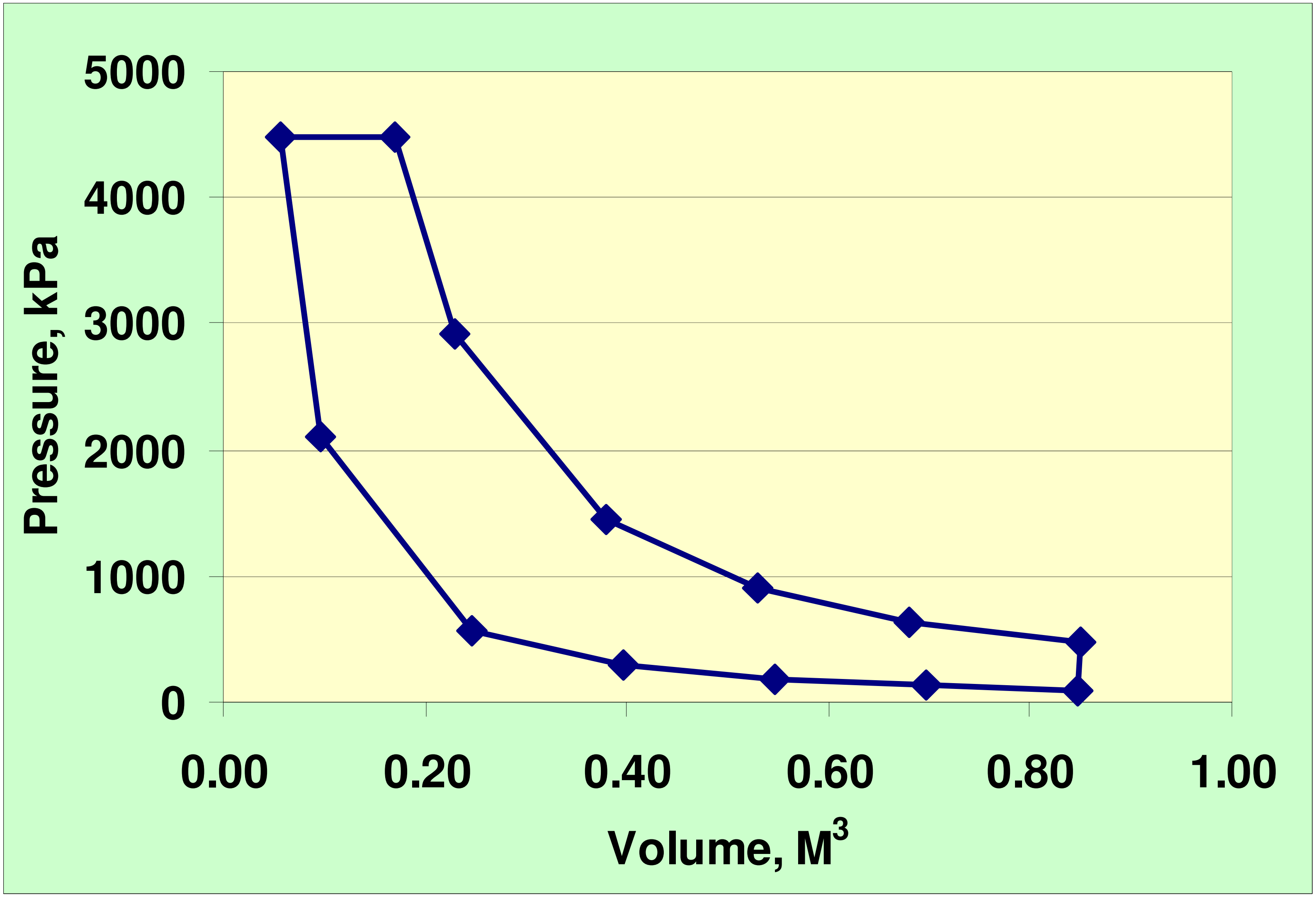
Where,

Compression ratio, $r_c = v_1/v_2$ and $k = c_p/c_v$

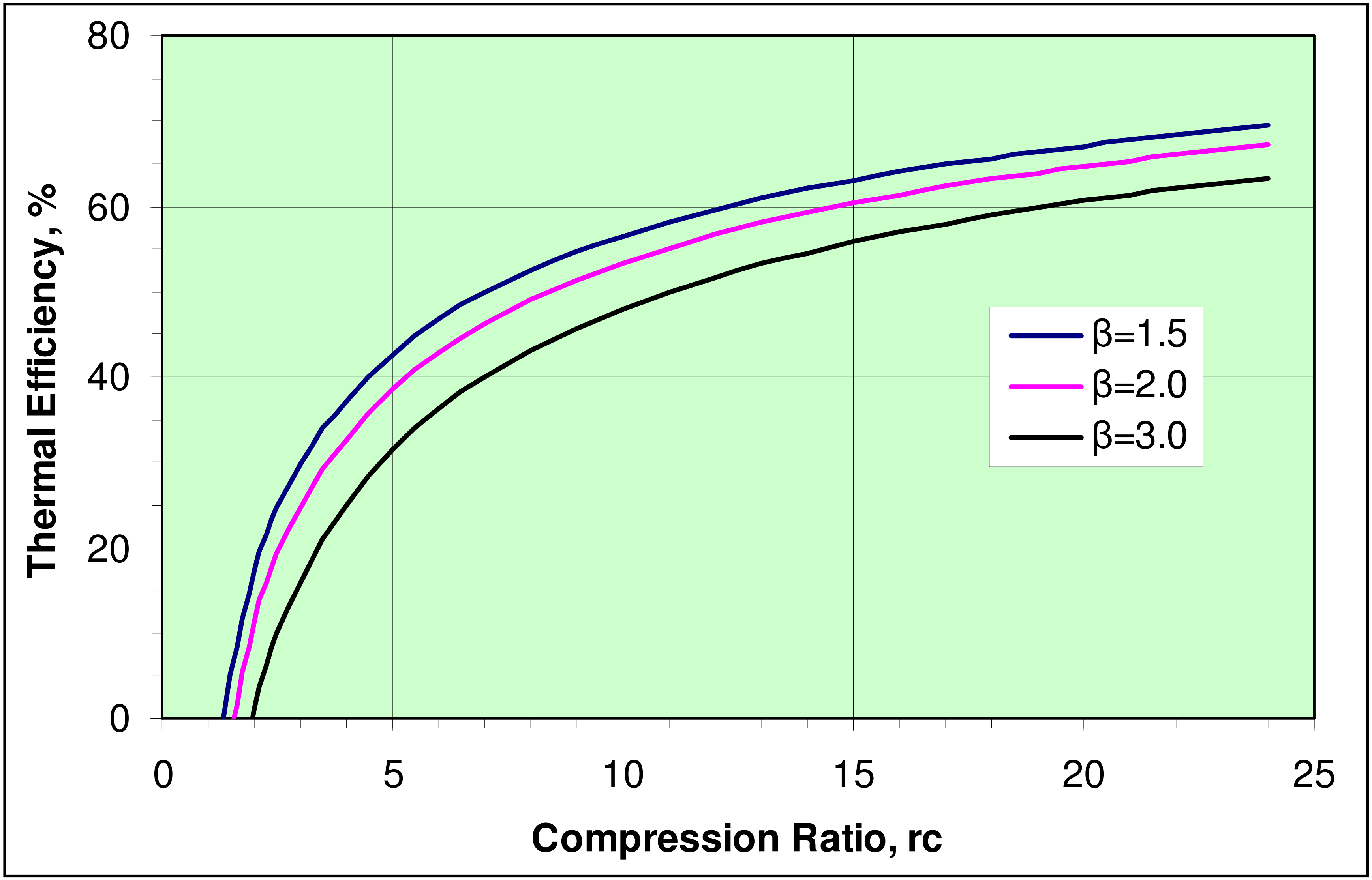
Cut off ratio, $\beta = v_3/v_2 = T_3/T_2$

The term in the bracket is always > 1. So the $\eta_{\mathbf{DIESEL}} < \eta_{\mathbf{OTTO}}$ for the same $r_c$

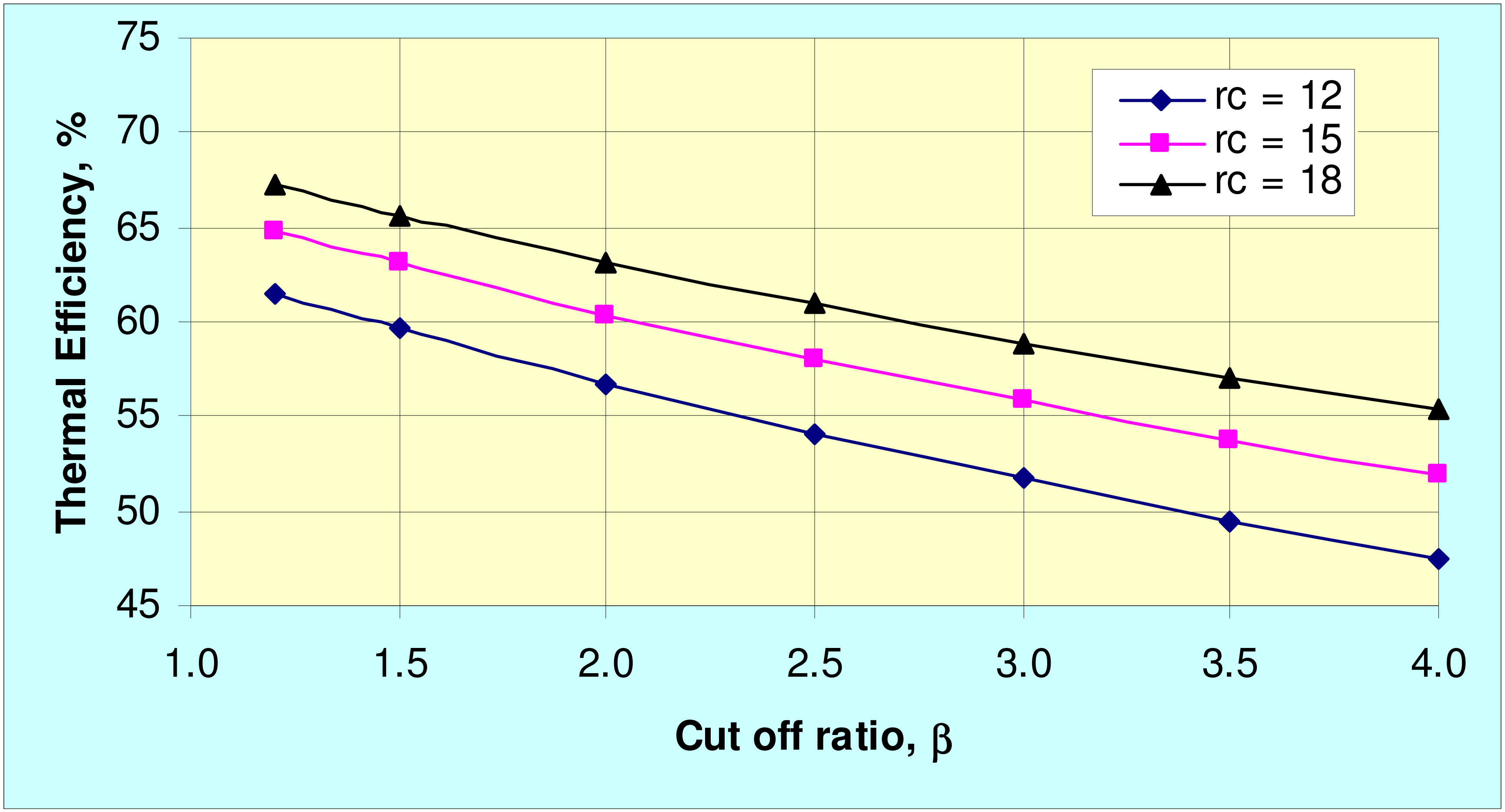
# DIESEL CYCLE FOR CI ENGINE



# Thermal Efficiency of Diesel Cycle



# THERMAL EFFICIENCY VS CUT OFF RATIO FOR DIESEL CYCLE



# DIESEL CYCLE

**Advantages:**

- Very mature and widespread technology, relatively low cost and high reliability, high efficiency (fuel economy of 200-250 g/kWh for modern engines), long life, relatively low thermal signature. Power density of the prime mover (only) is 100 to 400 W/kg over the 1-40 kW power output range. Initial cost is about $100 to 250 per kW, and operating costs are low. Can meet EPA emission standards.

**Disadvantages:**

- Can produce high noise and vibration, which increases with rotational speed.

# DUAL OR LIMITED PRESSURE CYCLE

# INTRODUCTION TO DUAL CYCLE

- Diesel cycle would operate on the more efficient higher compression ratios, while constant volume combustion of the Otto cycle would give higher efficiency for a given compression ratio.
- The modern CI engine operates to meet this by injecting the fuel much earlier in the cycle, somewhere around 20º bTDC. The first fuel then ignites late in the compression stroke, and some of the combustion occurs almost at constant volume at TDC, much like the Otto cycle.
- The last of the fuel is still being injected at TDC, and combustion of this fuel keeps the pressure high into the expansion stroke.

# ACTUAL CYCLE IN CI ENGINE

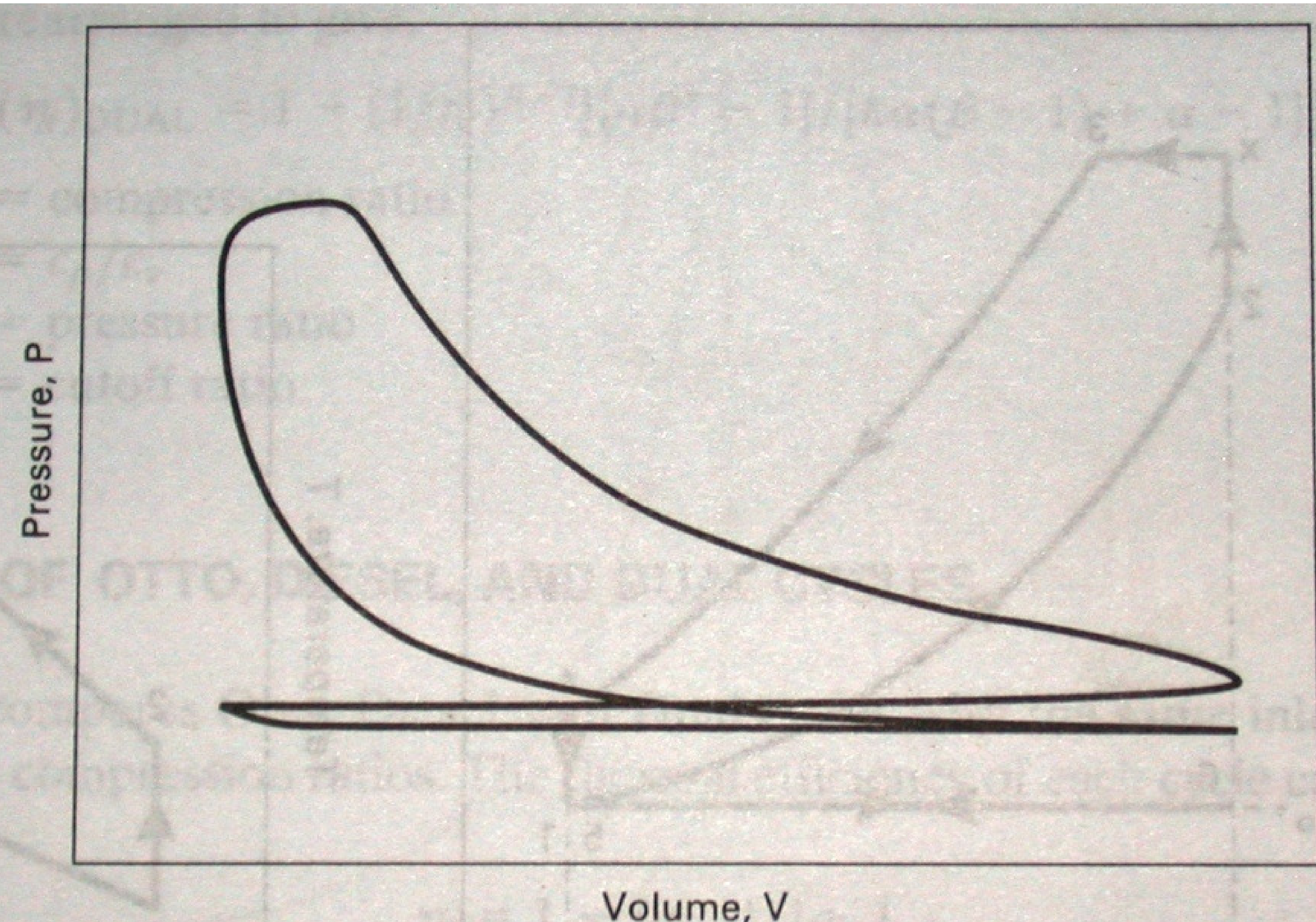
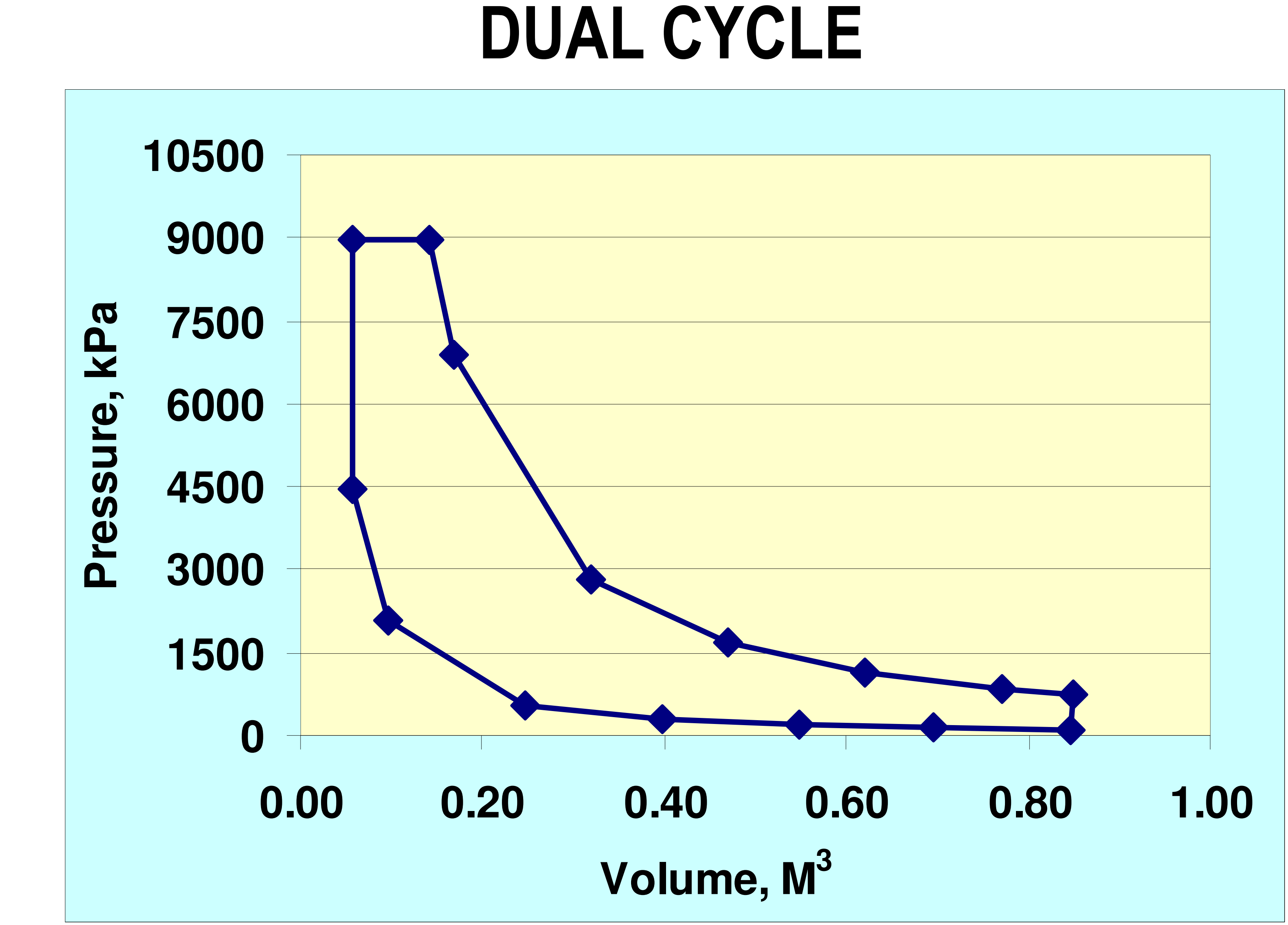



**Figure 3-9** Indicator diagram of a modern CI engine operating on a four-stroke cycle.


DUAL CYCLE
10500
9000
7500
6000
4500
3000
1500
0
Pressure, kPa
0.00
0.20
0.40
0.60
0.80
1.00
Volume, M3

# Thermal Efficiency of Internal Combustion Engine

Thermal efficiency for Dual cycle:

$$\eta = 1 - \frac{1}{r_c^{k-1}}\left[\frac{(\alpha\beta^k - 1)}{(\alpha - 1) + \alpha k(\beta - 1)}\right]$$

Where,

$r_c = v_1/v_2$ and $k = c_p/c_v$

$\beta = v_3/v_2 = T_3/T_2$ and pressure ratio, $\alpha = P_3/P_2$

Since $\alpha > 1$, the $\eta_{DUAL}$ **is <** $\eta_{DIESEL}$ for the same $\beta$ and $r_c$

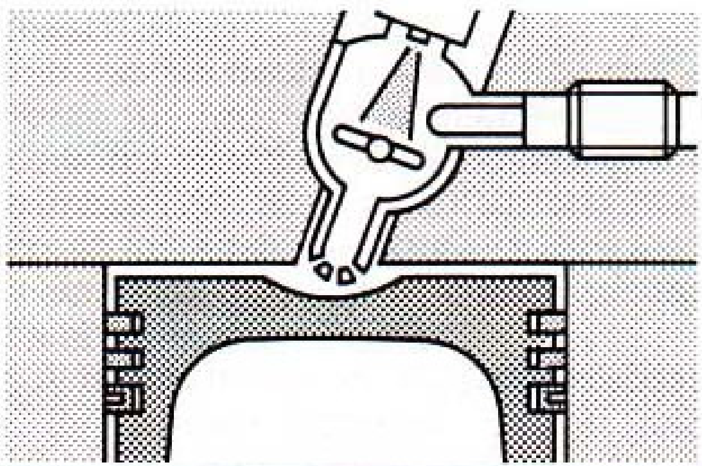
$\alpha = 1$ for Diesel Cycle and $\beta = 1$ for Otto cycle

# DUAL/LIMITED PRESSURE CYCLE

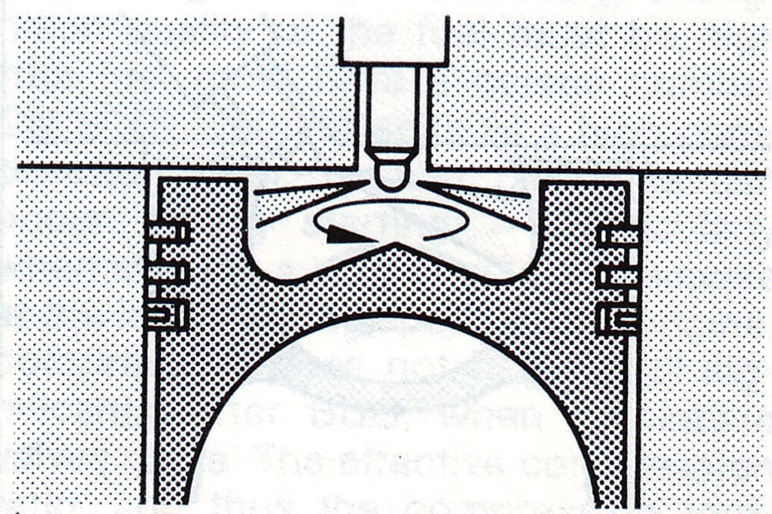
- The theoretical cycle of modern high speed diesel engine is more adequately by dual combustion or limited pressure cycle.
- In this cycle part of the heat addition takes place at constant volume and the remainder at constant pressure.
- Heat addition at constant volume tends to increase the efficiency of the cycle whereas switching over to constant pressure heat addition limits the maximum cylinder pressure.

# COMBUSTION CHAMBER FOR DIESEL ENGINE

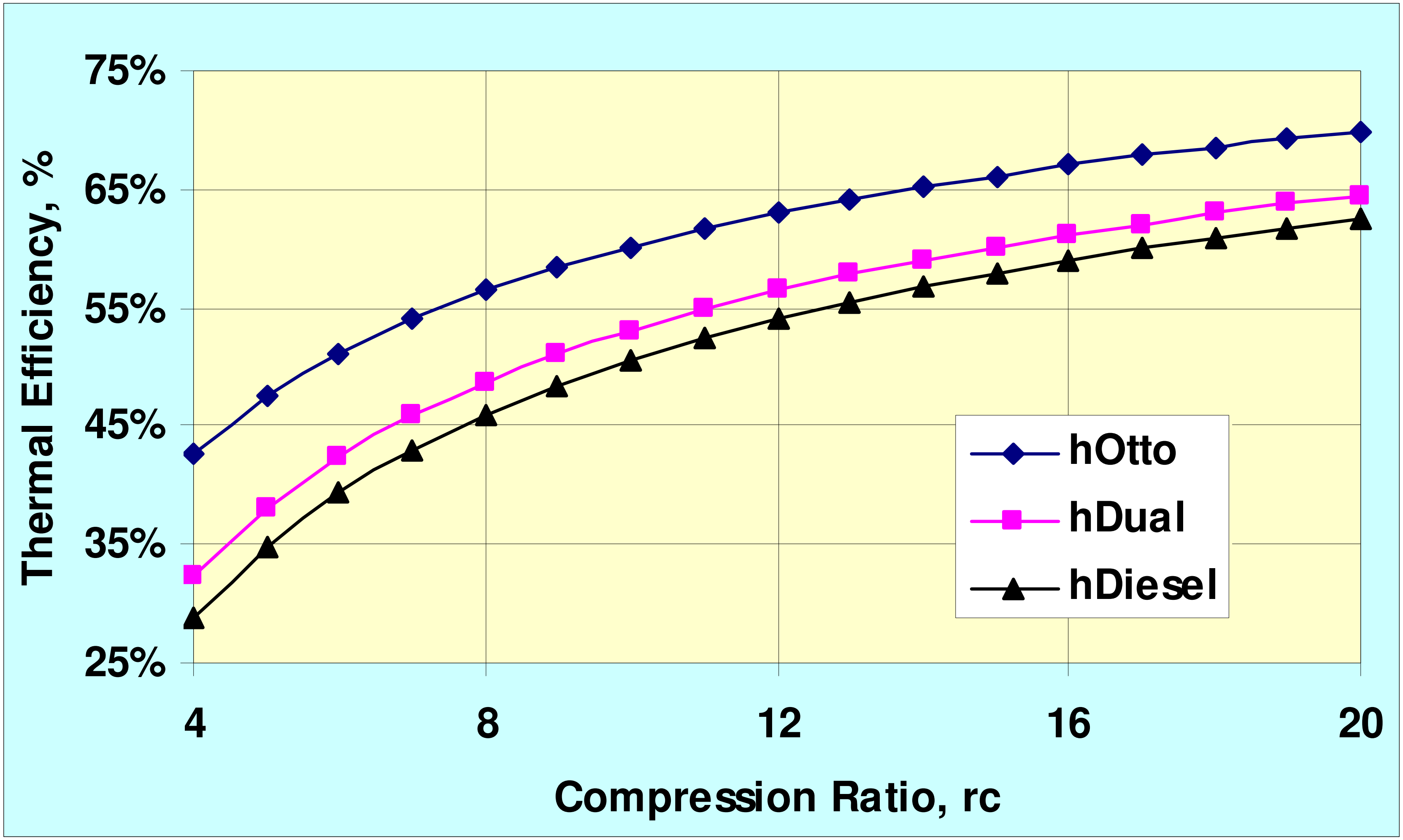
***Combustion chamber shape and nozzle location for the prechamber system.***



***Combustion chamber shape and nozzle location for the multiple-hole process with air swirling.***



# THERMAL EFFICIENCY VS COMPRESSION RATIO FOR DIFFERENT CYCLES

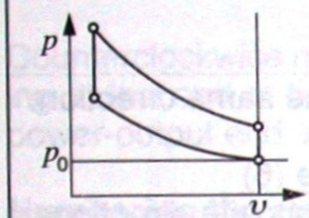
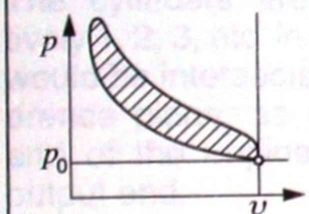
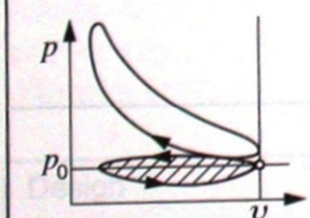


# COMPARISON OF CYCLES

- For the same compression ratio and same heat input
  - $\eta_{OTTO} > \eta_{DUAL} > \eta_{DIESEL}$
- For the peak pressure and same heat input
  - $\eta_{OTTO} < \eta_{DUAL} < \eta_{DIESEL}$
- For the peak pressure and maximum temperature
  - $\eta_{OTTO} < \eta_{DUAL} < \eta_{DIESEL}$

# OVERALL EFFICIENCY OF SI ENGINE

**Table 2. Graphic representations and definitions of the individual and overall efficiencies of the reciprocating-piston engine.**

| Pressure vs. volume diagram | Designation | Conditions | Definition | Efficiencies |
|---|---|---|---|---|
|  | Theoretical reference constant-volume cycle | Ideal gas, constant specific heat, infinitely rapid heat addition and -dissipation, etc. | $\eta_{th} = 1 - \varepsilon^{1-\varkappa}$ Theoretical or thermal efficiency | $\eta_{th}$ |
|  | Real high pressure working cycle | Wall heat losses, real gas, finitely rapid head addition and dissipation, variable specific heat | $\eta_{gHD}$ Efficiency factor of the high-pressure cycle | $\eta_i$ $\eta_e$ |
|  | Real charge cycle (4-stroke) | Flow losses, heating of the mixture or the air, etc. | $\eta_{gLW}$ Charge exchange efficiency | $\eta_g$ |
| Mechanical losses | Losses due to friction, cooling, auxiliary units | Real engine | $\eta_m$ | $\eta_m$ $\eta_m$ |

# DEFINITION OF EFFICIENCIES

- $\eta_e$ – overall efficiency includes the sum of all losses, and can thus be defined as the ratio of effective mechanical work to the mechanical work equivalent of the supplied fuel.
- $\eta_i$ – indicated efficiency is the ratio of indicated high-pressure work to the calorific content of the supplied fuel.
- $\eta_g$ – efficiency of cycle factor includes all internal losses occurring in both high-pressure and low-pressure processes.
- $\eta_m$ – mechanical efficiency defines the relationship between the effective work available at the crankshaft and the indicated work.
- $\eta_{th}$ – thermal efficiency is the efficiency of the ideal constant volume combustion cycle.

Mean effective pressure =

= work done during the cycle / Stroke volume

$$mep = \frac{Q_s - Q_r}{V_d} = \frac{workdone}{V_d}$$

Where,

$Q_s$ – energy supplied by burning of fuel

$Q_r$ – heat loss through exhaust

# TYPICAL VALUES FOR BMEP

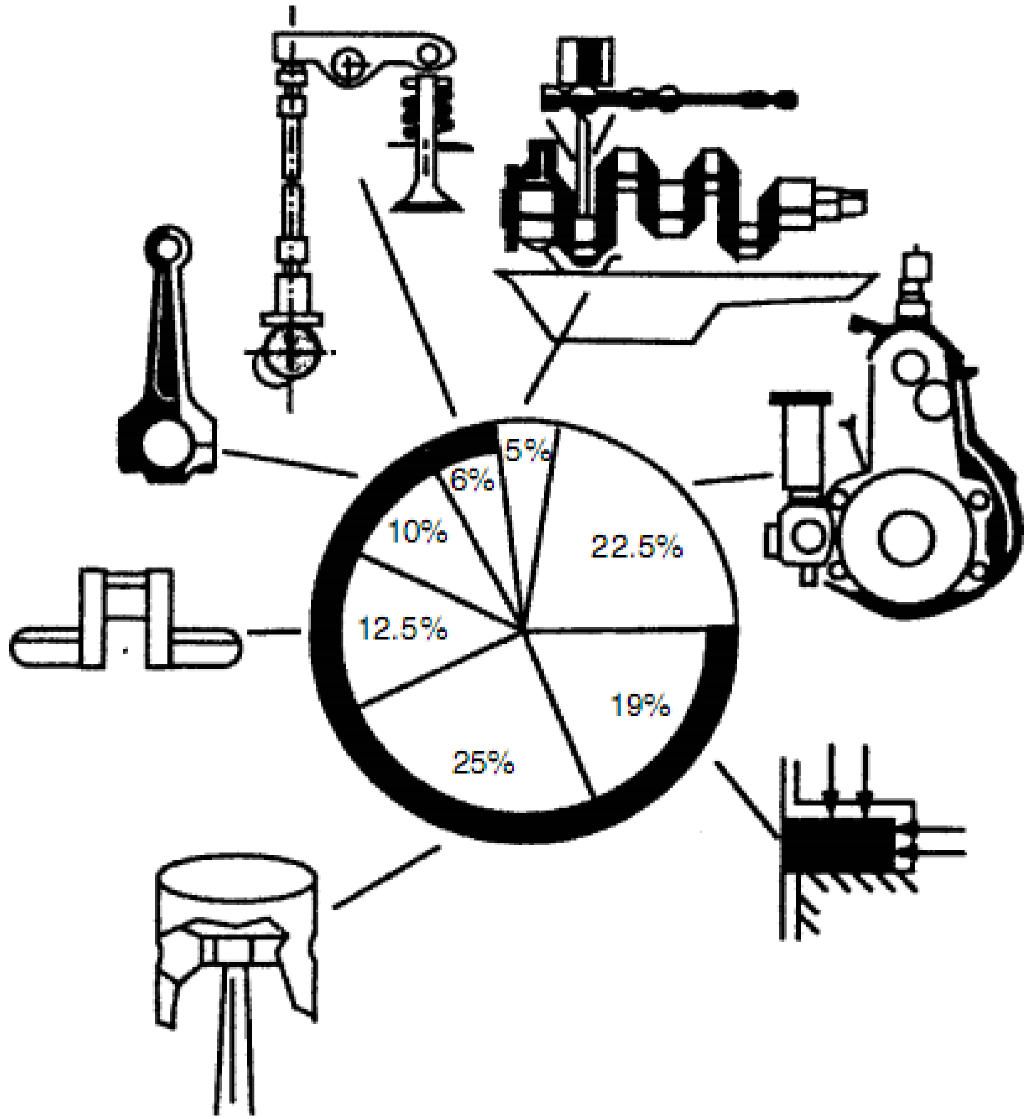
- Naturally aspirated spark-ignition engines, bmep = 850 – 1050 kPa at the engine speed where max. torque is obtained and at the rated power, bmep values are 10 to 15 percent lower.
- Turbocharged automotive SI engines the max bmep = 1250 – 1700 kPa and at max. rated power bmep is 900 – 1400 kPa.
- Naturally aspirated 4-stroke diesels the max bmep = 700 – 900 kPa and at the max. rated power = 700 kPa.
- Turbocharged 4-stroke diesel, the max bmep = 1000 – 1200 kPa and at max. rated power bmep is 850 – 950 kPa.

# ACTUAL CYCLES

Actual cycles differ from ideal cycles because of the following factors:

- Variation of air-fuel ratio
- Valve operation and timing
- Finite combustion time
- The actual combustion process
- Dissociation
- Exhaust blowdown
- Heat transfer
- Fluid friction

# Distribution of friction losses in an internal combustion engine



# ACTUAL CYCLE IN SI ENGINE

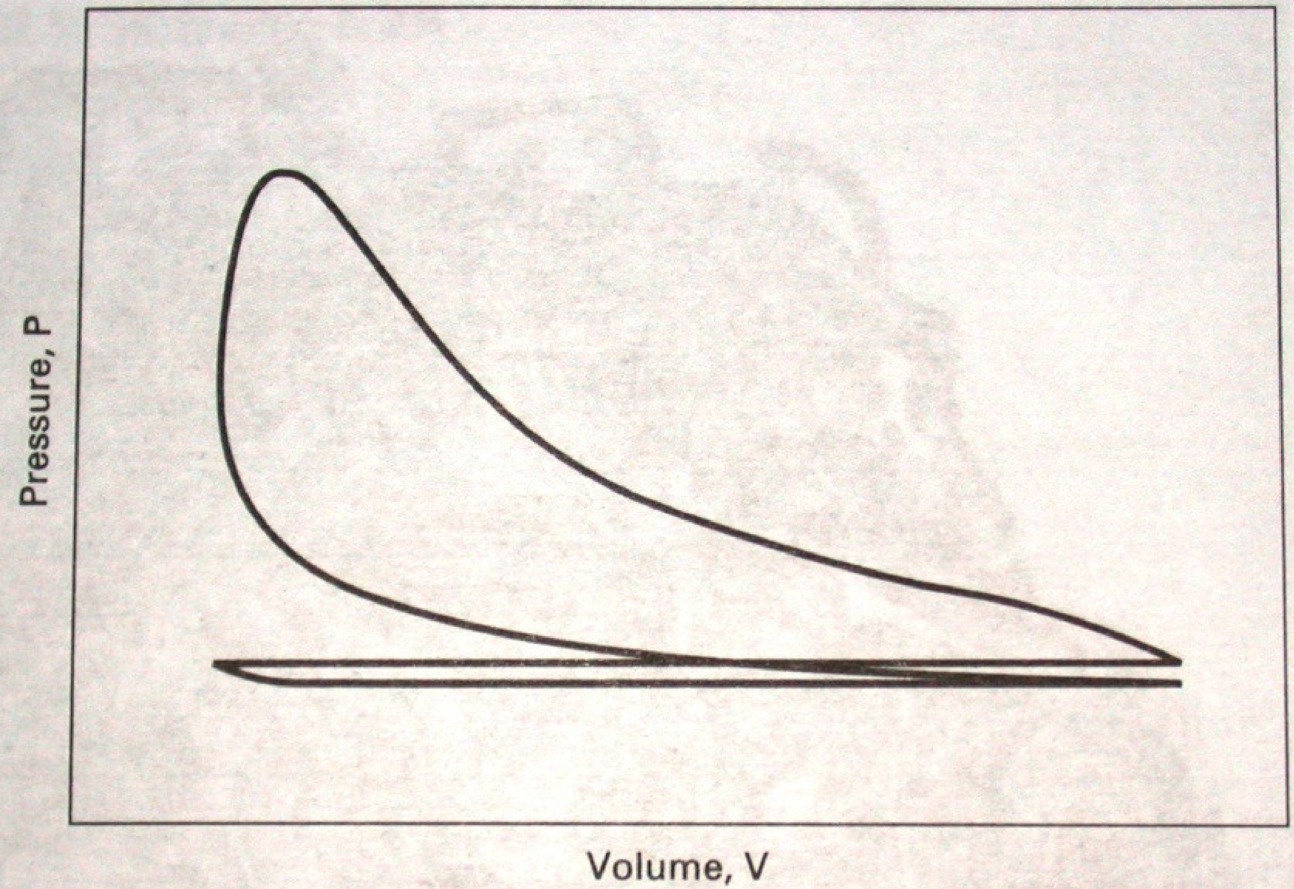
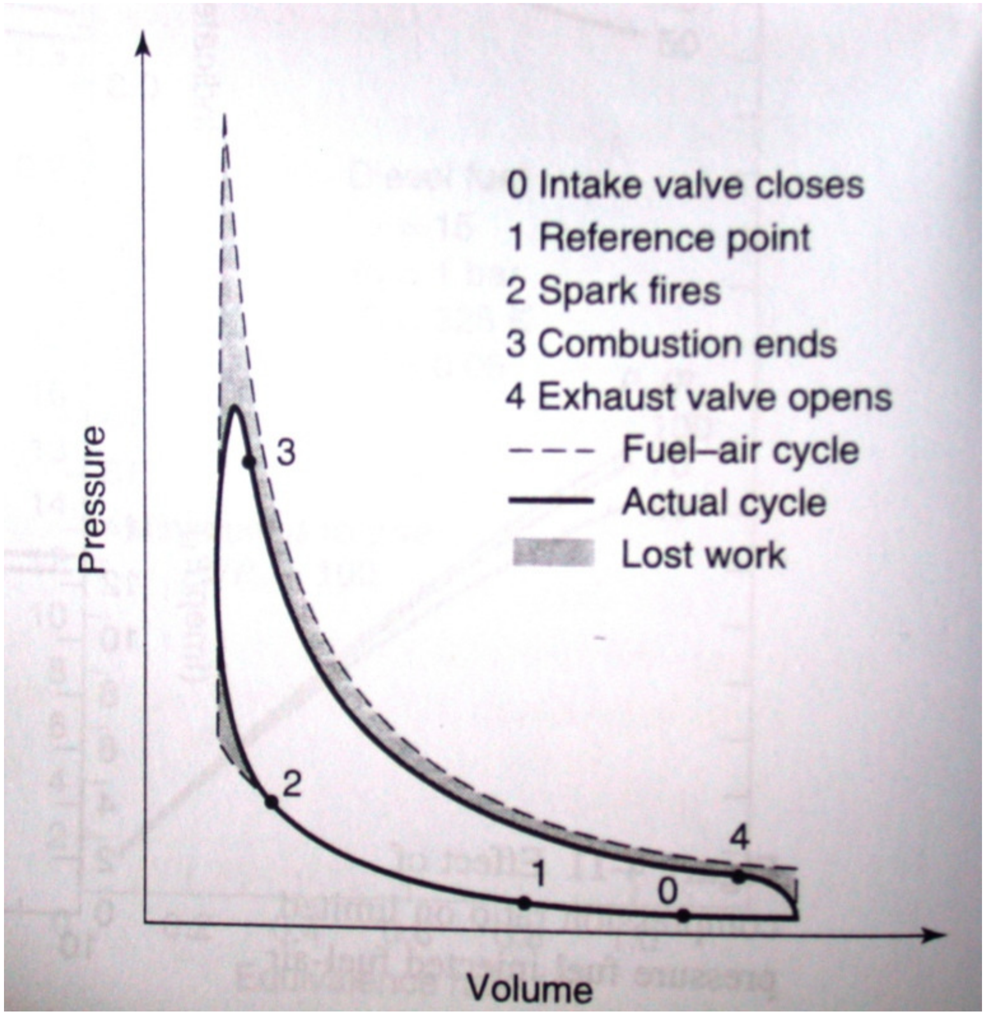



**Figure 2-6** Indicator diagram for a typical four stroke cycle SI engine. An indicator diagram plots cylinder pressure as a function of combustion chamber volume over a 720° cycle. The diagram is generated on an oscilloscope using a pressure transducer mounted in the combustion chamber and a position sensor mounted on the piston or crankshaft.

# LOST WORK




- Heat loss
- Mass loss
- Finite burn rate
- Finite blowdown rate

These losses result in the actual efficiency being less than that of equivalent fuel-air cycle by a factor ranging from 0.8 to 0.9 ($\eta/\eta_{Otto}$).

The imep is maximized slightly rich of stoichiometric, and increases with increasing compression ratio and inlet pressure.

# ACTUAL CYCLE IN CI ENGINE

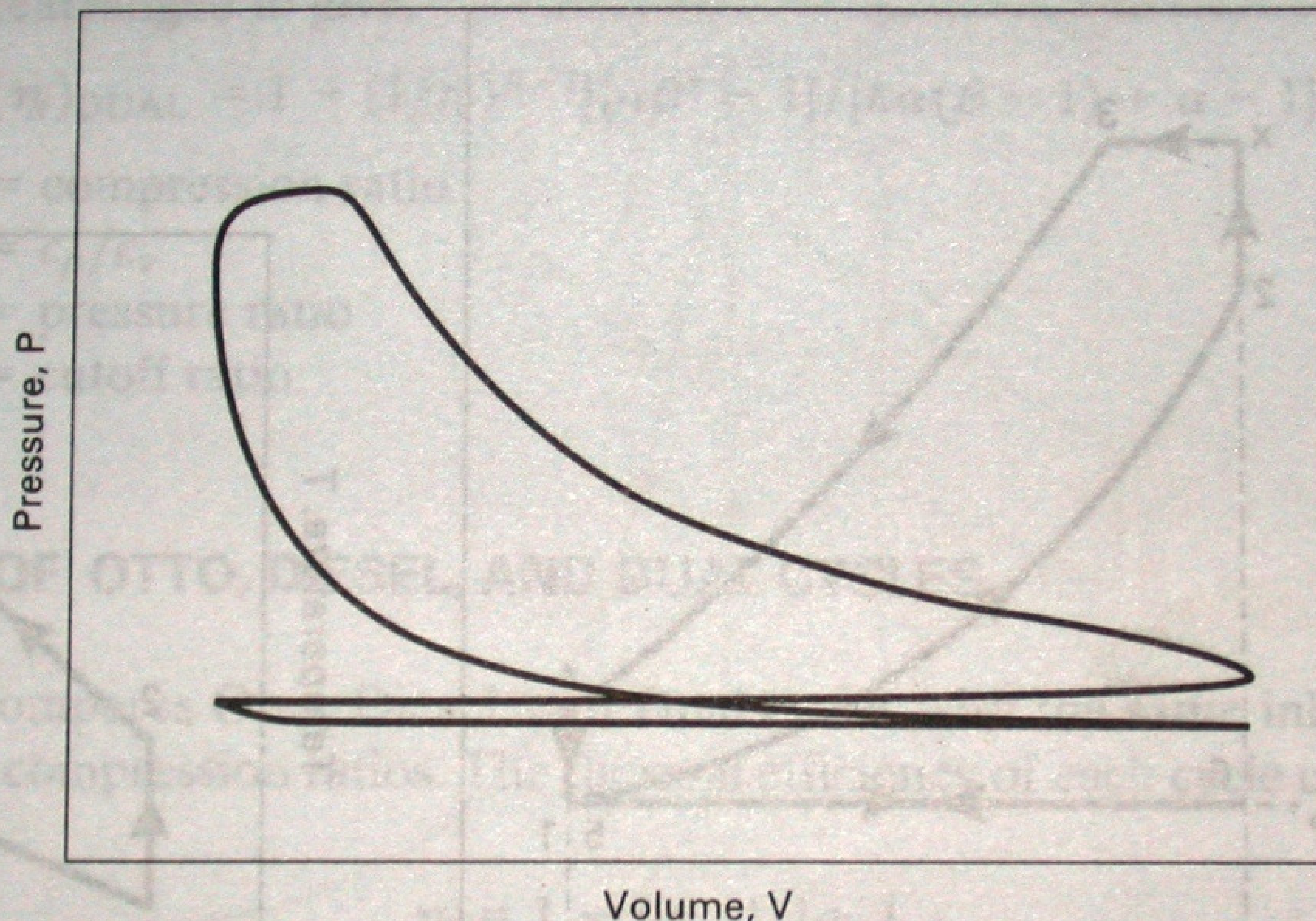
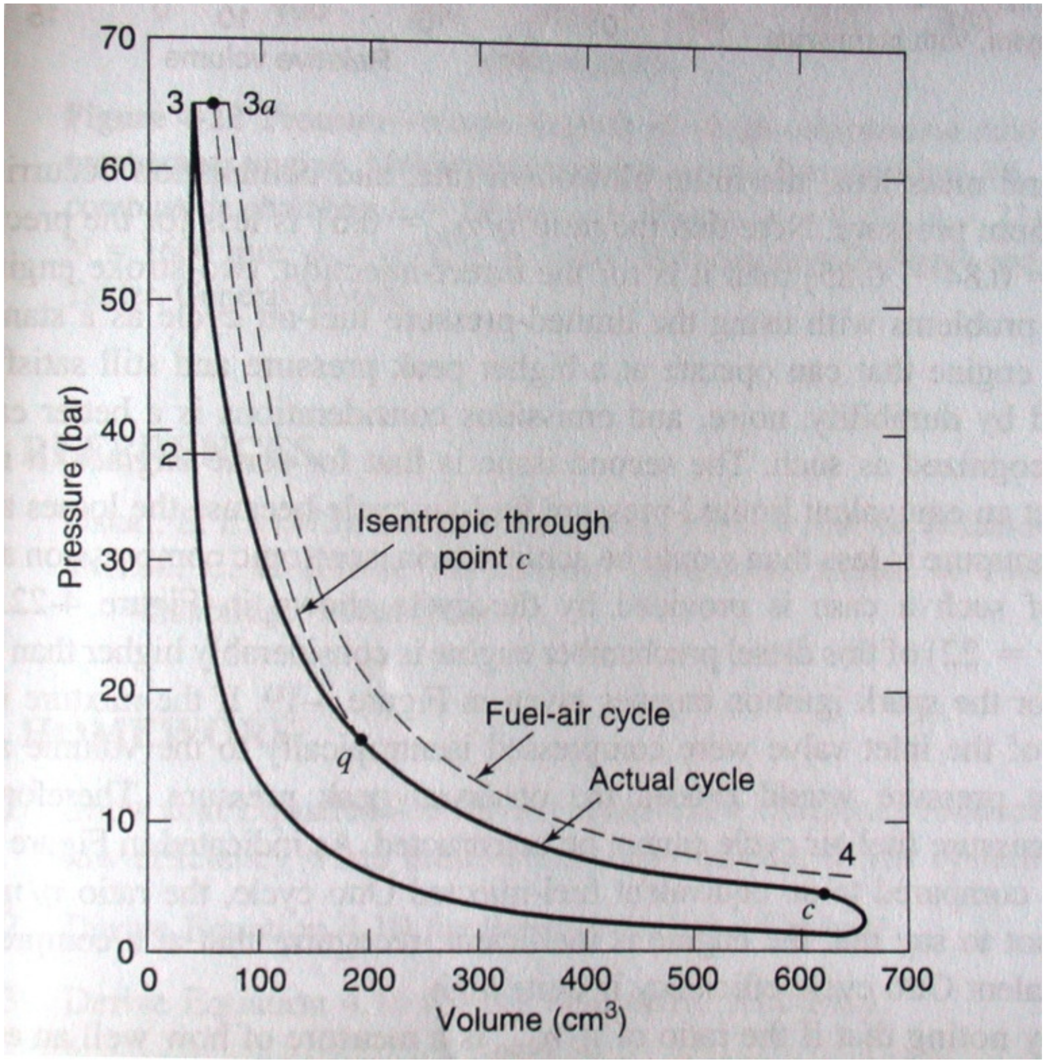



**Figure 3-9** Indicator diagram of a modern CI engine operating on a four-stroke cycle.

# LOST WORK




- Heat loss
- Mass loss
- Finite blowdown rate
- Combustion occurring at less than the maximum pressure

The figure shows the cylinder pressures in a diesel pre-chamber engine compared with equivalent limited pressure fuel-air cycle.

# MAIN DIFFERENCE BETWEEN ACTUAL AND THEORETICAL CYCLES

- Compression and expansion are not friction less adiabatic processes. A certain amount of friction is always present and there is considerable heat transfer between the gases and cylinder wall.
- Combustion does not occur either at constant volume or at constant pressure.
- The thermodynamics properties of the gases after combustion are different than those of the fuel-air mixture before combustion.
- The combustion may be incomplete.
- The specific heats of the working fluid are not constant but increases with temperature.

## MAIN DIFFERENCE BETWEEN ACTUAL AND THEORETICAL CYCLES

- The cylinder pressure during exhaust process is higher than the atmosphere. As a result, more work has to be done by the piston on the gases to expel them out of the cylinder, than work done by the gases on the piston during the intake stroke.
- This difference in work, called pumping work, is represented by the pumping loop shown by hatched are.
- Note that this work is negative and represents loss of work called pumping loss.

# CONCLUSIONS

- The thermal efficiency of actual Otto cycle in SI engine is about 25-30%.
- Similarly the thermal efficiency of actual Diesel cycle in CI engine is 30-35%
- Desirable increase in thermal efficiency of IC engine very much depends on material, air-fuel ratio, ignition timing, fuel injection timing, residual gases in the cylinder etc.

# REVIEW PROPBLEMS

1. An air standard Otto cycle has a compression ratio of 8 and temperature and pressure at the beginning of compression are 20°C and 1 bar respectively. The constant volume heat addition is 1800 kJ/kg. Calculate the maximum temperature and pressure for the cycle and the temperature at the end of the expansion process. What is the efficiency and mean effective pressure of the cycle. Take $c_v$=0.718 kJ/kg.K and k=1.4
2. The initial condition of air in an Otto cycle is 288K and 1.05 bar. The air compressed isentropically to a pressure 13 bar. Then heat is added at constant volume till the pressure becomes 35 bar. Calculate:
   a) Compression ratio
   b) Air standard efficiency
   c) Mean effective pressure of the cycle

# REVIEW PROPBLEMS

1. In an air-standard Diesel cycle, the compression ratio is 16, the cylinder bore is 200 mm and the stroke is 300 mm. Compression begins at 1 bar and 27°C. The cut-off takes place at 8 percent of the stroke. Determine:
   a) The pressure, the volume and the temperature at all salient points [$V_1$=0.010048m$^3$, $V_2$=0.000628m$^3$, m=0.01167 kg, $p_2$= 48.5 bar, $T_2$= 909.4K]
   b) The cut-off ratio [2.2]
   c) The air standard efficiency [60.4%]
   d) The mean effective pressure [8.21 bar]

# Review question on diesel cycle

- Given:
  a) Cylinder bore = 200 mm and stroke = 250 mm
  b) $R_c = 18$ and $\beta = 2$
  c) $P_1 = 0.1$ MPa and $T_1 = 300$K

- Find:
  a) T and P in state 2,3 & 4
  b) Thermal efficiency ($\eta$), work done and mep

# REVIEW PROPBLEMS

1. An air-standard Dual cycle has a compression ratio of 15, and the compression begins at 1 bar, 27°C. The maximum pressure is limited to 60 bar. The heat transferred to air at constant volume is twice that at constant pressure. Calculate:
   a) the pressures and temperatures at the cardinal points of the cycle, [$T_2$=886K, $p_2$=44.3 bar, $T_3$=1200K, $T_4$=1312K, $\beta$=1.093, $T_5$=460K, W=223 kJ/kg
   b) the cycle efficiency, [66%] and
   c) the mean effective pressure of the cycle. [2.77 bar]

# *Engine Thermodynamic Analysis*

## *Ideal Otto Cycle*

Glenn Research Center

$C_v$ = Specific Heat constant volume

$\gamma$ = Specific Heat Ratio

p = pressure

T = Temperature

V = Volume

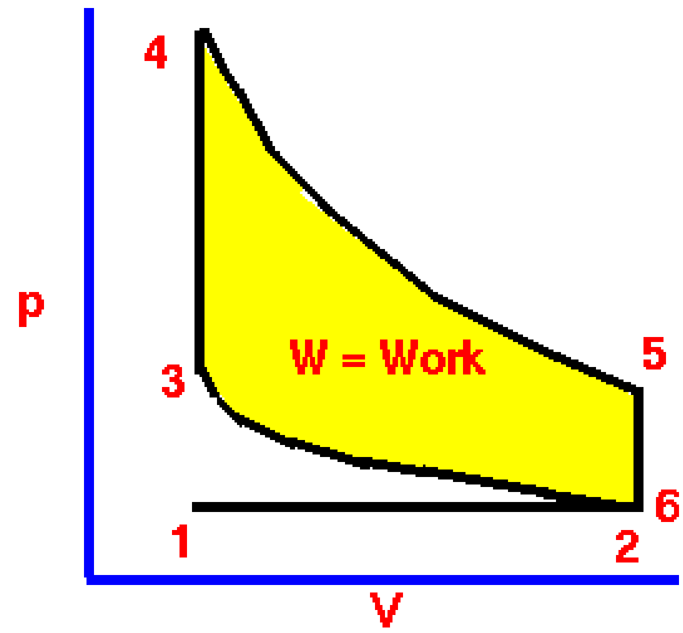
f = fuel / air ratio

Q = Fuel heating value

c p s = cycles per second

P = Power

$V_2 / V_3 = r$ = Compression Ratio




**Compression Stroke :**

$T_3 / T_2 = r^{\gamma - 1}$

$p_3 / p_2 = r^{\gamma}$

**Combustion :**

$T_4 = T_3 + f\,Q / c_v$

$p_4 = p_3 (T_4 / T_3)$

**Power Stroke :**

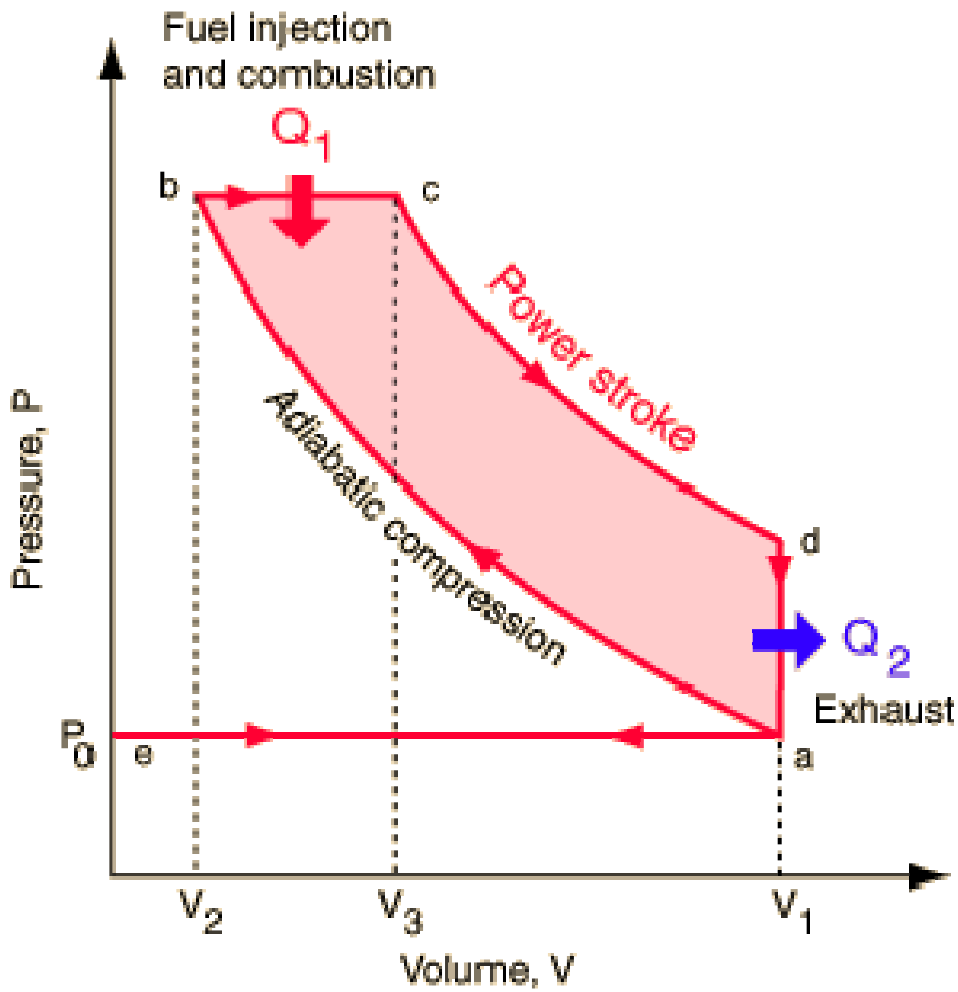
$T_5 / T_4 = r^{1 - \gamma}$

$p_5 / p_4 = r^{-\gamma}$

Work per cycle : $W = c_v [ (T_4 - T_3) - (T_5 - T_2) ]$

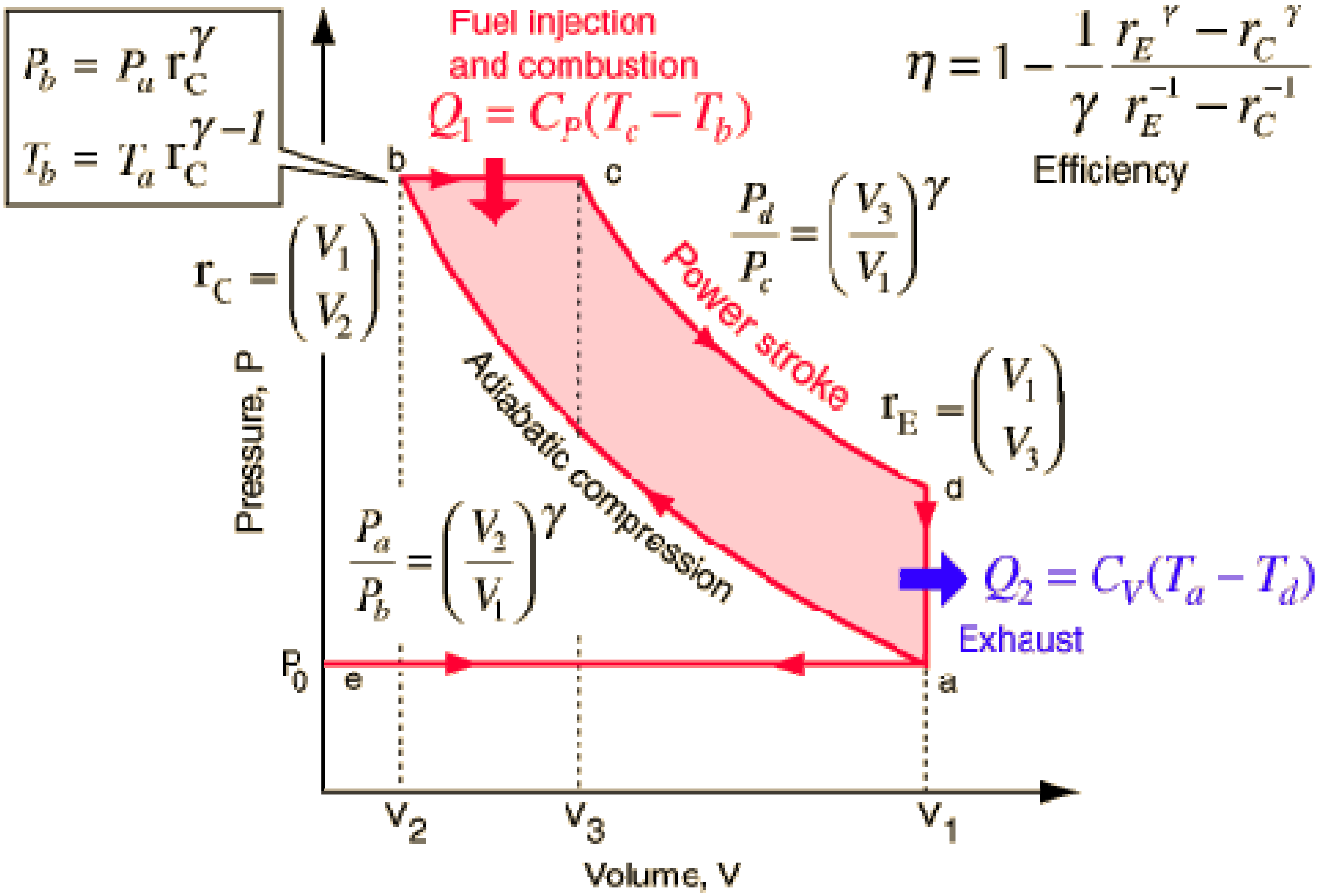
Engine Power : $P = W \text{ cps}$

# Diesel cycle



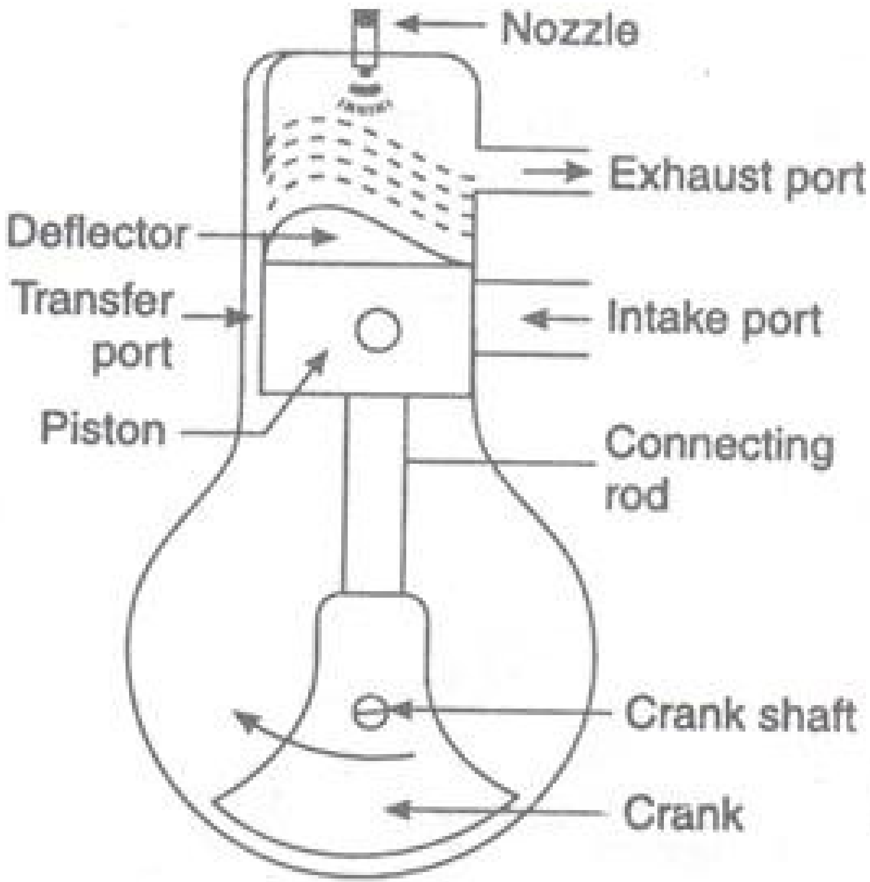

http://hyperphysics.phy-astr.gsu.edu/hbase/thermo/diesel.html

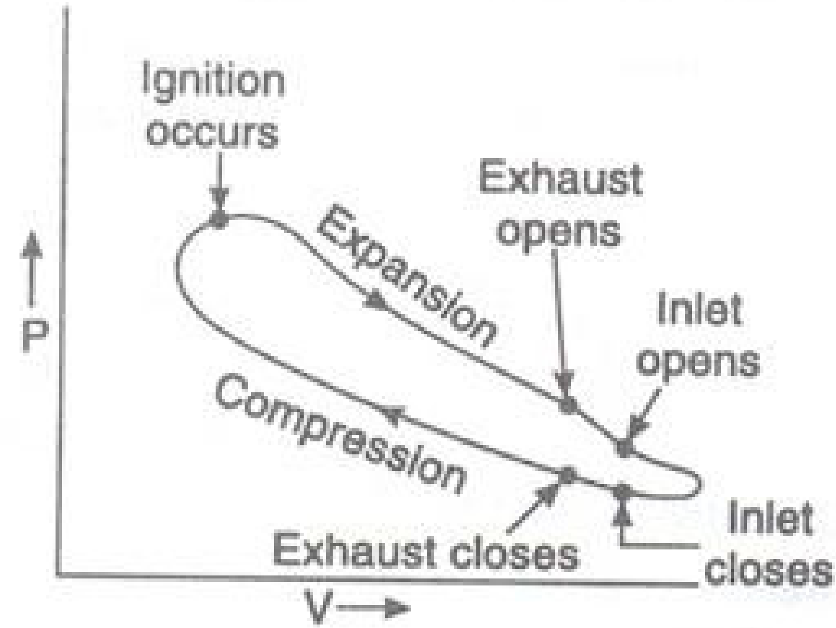
# DIESEL ENGINE CYCLE




http://hyperphysics.phy-astr.gsu.edu/hbase/thermo/diesel.html

# 2 stroke working cycle

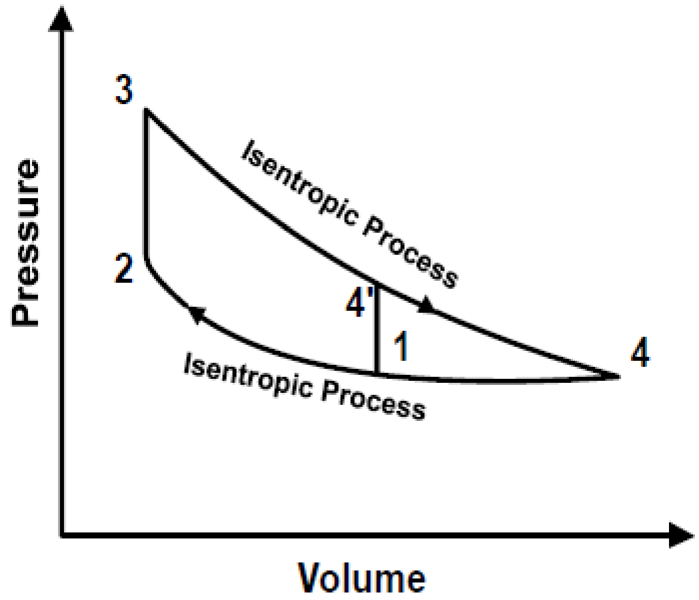


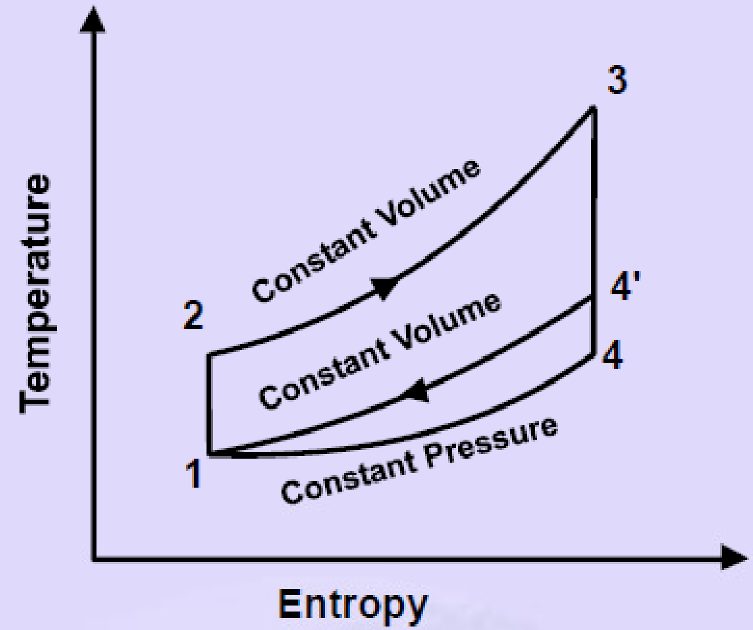

Indicator diagram for 2-stroke C.I. engine

# ATKINSON CYCLE

# Atkinson cycle on p-v and T-s diagrams




(a)




(b)

Process 1-2: Reversible adiabatic compression ($v_1$ to $v_2$).

Process 2-3: Constant volume heat addition.

Process 3-4: Reversible adiabatic expansion ($v_3$ to $v_4$).

Process 4-1: Constant pressure heat rejection.

$$\eta_{th} = 1 - \frac{1}{r^{\gamma - 1}} \frac{\gamma (r_v - 1)}{(r_p - 1)}$$

$$r_v = \frac{T_4}{T_3} = \frac{V_4}{V_1} \;\&\; r_p = \frac{T_3}{T_2} = \frac{P_3}{P_2}$$

# Thermal efficiency of Atkinson cycle

The Atkinson cycle is a variation on the Otto cycle which effectively increased the engine's expansion ratio compared with the compression ratio by using a complex crankshaft linkage. This enables the exhaust stroke to be longer than the induction stroke and hence the swept volumes are different. The greater expansion allows more energy to be extracted from the fuel charge and allows the engine to run cooler. It provides better efficiency at the expense of power density.

$$\text{Heat supplied} = C_v (T_3 - T_2)$$

$$\text{Heat rejected} = C_p (T_4 - T_1)$$

$$\text{Net workdone} = C_v (T_3 - T_2) - C_p (T_4 - T_1)$$

$$\eta_{th} = \frac{C_v (T_3 - T_2) - C_p (T_4 - T_1)}{C_v (T_3 - T_2)}$$

$$= 1 - \frac{\gamma (T_4 - T_1)}{(T_3 - T_2)}$$

$$\text{Let, } r = \frac{v_1}{v_2} = CR$$

$$T_2 = T_1 \, r^{\gamma - 1}$$

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} = r_p = \text{Pressure ratio}$$

$$T_3 = T_2\, r_p = T_1\, r^{\gamma - 1}\, r_p$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{P_3}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{P_3}{P_2}.\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} = \left(r_p.\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} = r_p^{\frac{\gamma - 1}{\gamma}}.\, r^{\gamma-1}$$

since,

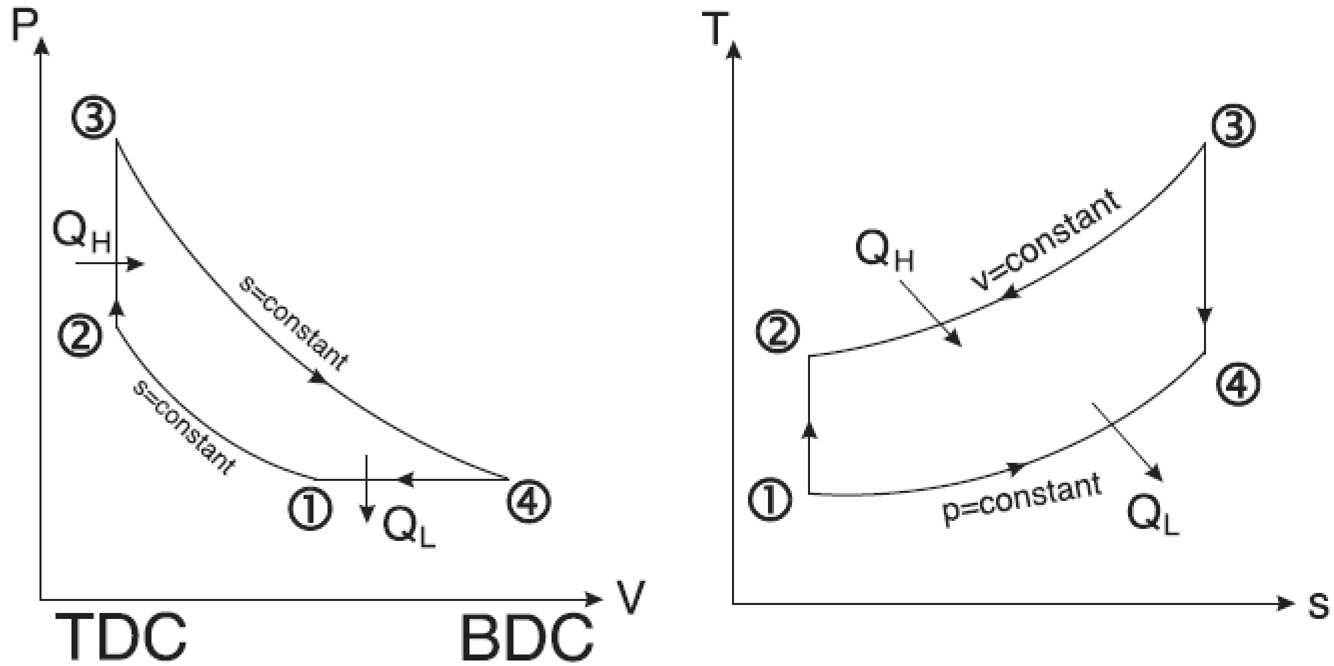
$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r^{\gamma}$$

and

$$T_4 = \frac{T_3}{r_p^{\frac{\gamma - 1}{\gamma}}\; r^{\gamma - 1}} = \frac{T_1\, r_p\, r^{\gamma - 1}}{r_p^{\frac{\gamma - 1}{\gamma}}\; r^{\gamma - 1}} = T_1\, r_p^{\frac{1}{\gamma}}$$

$$\eta_{th} = 1 - \gamma\left[\frac{r_p^{\frac{1}{\gamma}} - 1}{\left(r_p - 1\right) r^{\gamma - 1}}\right]$$

http://www.nptel.iitm.ac.in/courses/IIT-MADRAS/Applied_Thermodynamics/Module_4/9_Atkinson_cycle.pdf




The Atkinson Cycle

$$r = \frac{V_1}{V_2} = compression\ ratio$$

$$r_\alpha = \frac{V_4}{V_3} = \text{expansion ratio (larger than the compression ratio)}$$

$$\eta_{th} = 1 - \gamma^{1-\gamma}\frac{\left[\dfrac{r_\alpha}{r} - 1\right]}{\left[\dfrac{r_\alpha^\gamma}{r^\gamma} - 1\right]} = 1 - \gamma\left[\frac{r_\alpha - r}{r_\alpha^\gamma - r^\gamma}\right]$$

Similar to Otto cycle but with constant pressure heat rejection that allows for a higher expansion ratio (more work extraction) compared to the compression ratio and in turn a higher cycle efficiency.

# ATKINSON CYCLE

**Advantages**

- Reduces pumping losses in the engine by using a more efficient combustion cycle
- Reduces the amount of fuel-air to be compressed, resulting in greater fuel efficiency
- Allows the expansion stroke to be longer than the compression stroke, improving thermal efficiency

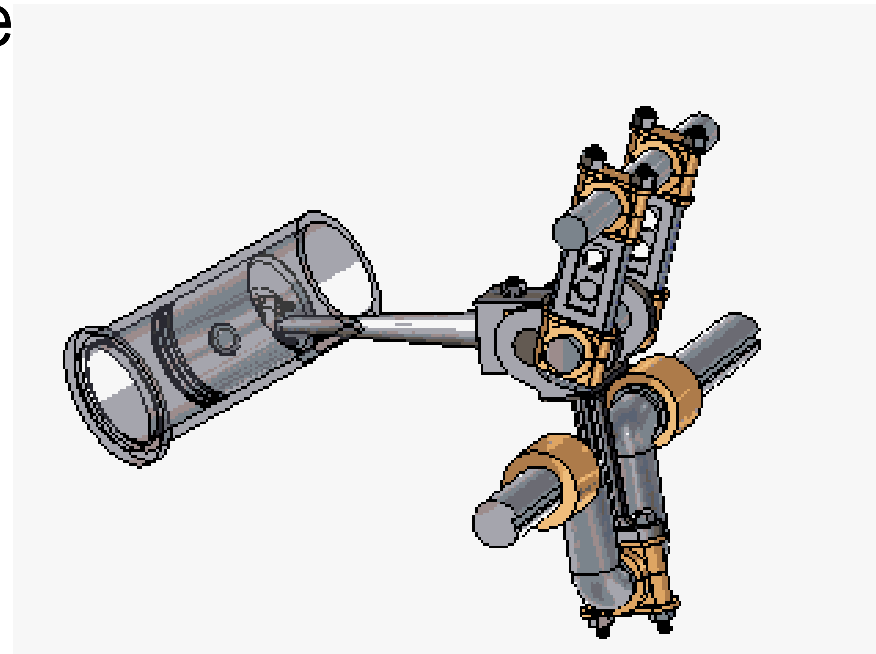
**Disadvantages**

- Requires a more complex linkage system
- Some engine power is lost due to backflow of fuel-air mixture

**Atkinson cycle is used in hybrid vehicles**

# Atkinson-Cycle Engines

- Operates on a four-stroke cycle
- **Intake, compression, combustion, and exhaust strokes occur in a one revolution of crankshaft**
- Combustion stroke is longer than compression stroke allowing more expansion of combustion gases
- **Greater efficiency when compared to Otto engines**



**The Atkinson Cycle Engine**

# Atkinson-Cycle Engines Continued

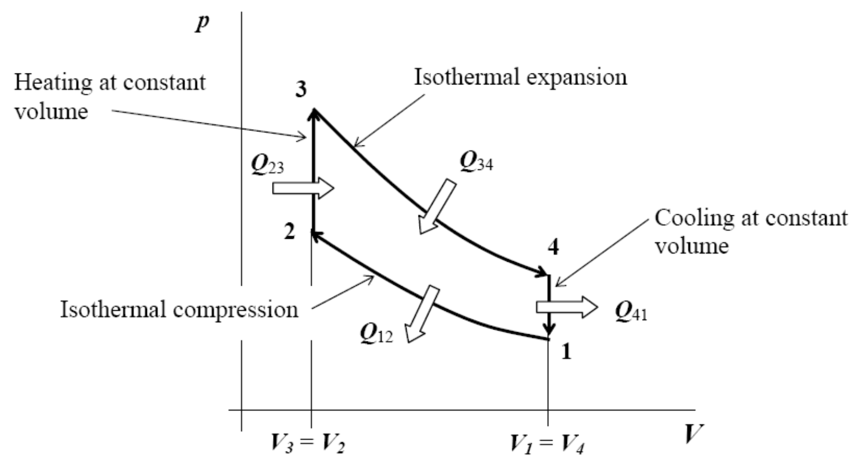
- Biggest disadvantage is reduction in power density (power/unit volume) arising from the reduction in air intake.
- Atkinson cycle engine can be supplemented with electric motor to provide more power if needed.
- Electric motors can be used in combination or independent of Atkinson cycle engines to provide the desired power output most efficiently.

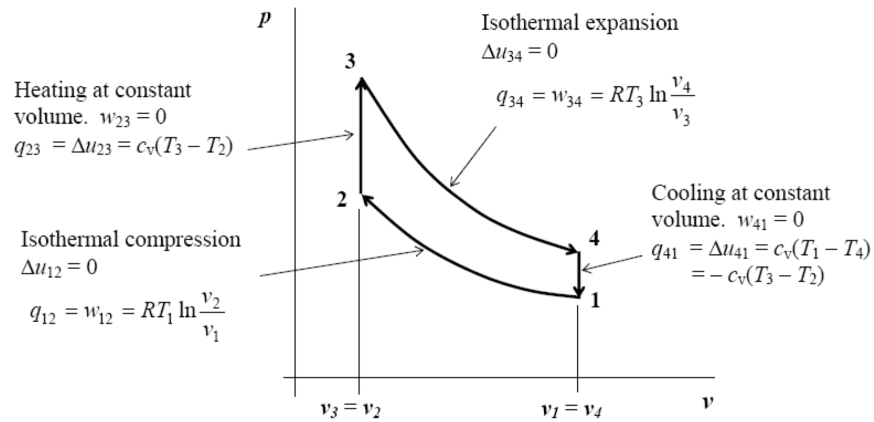
# STIRLING CYCLE

# The Stirling Cycle

## Process breakdown

Stirling cycle.




Recall that $v_3 = v_2$ and $v_4 = v_1$, also $v_1 > v_2$

$$w_{net} = RT_1 \ln\frac{v_2}{v_1} + RT_3 \ln\frac{v_1}{v_2} = -RT_1 \ln\frac{v_1}{v_2} + RT_3 \ln\frac{v_1}{v_2}$$

$$w_{net} = R(T_3 - T_1)\ln\frac{v_1}{v_2}$$

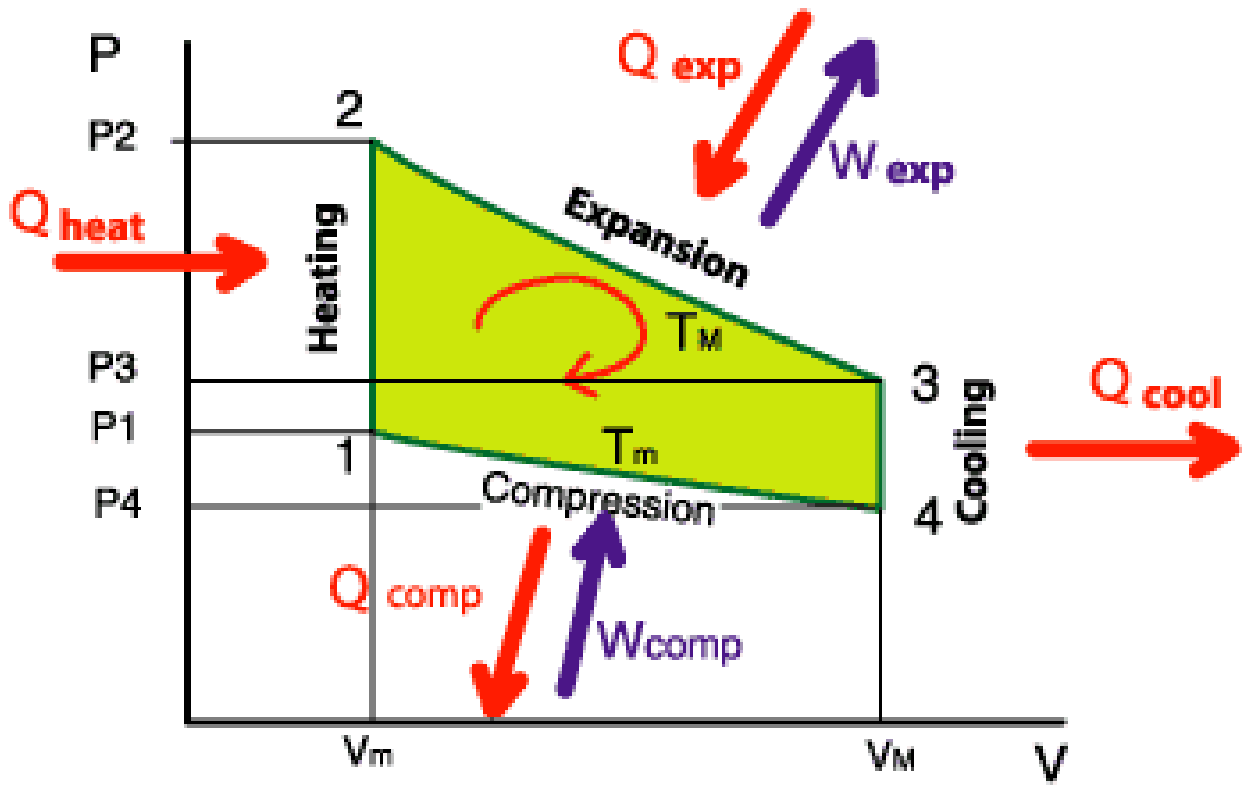
For stirling engine with regeneration

$$\eta = \frac{W_{net}}{Q_{34}} = \frac{R(T_3 - T_2)\ln\frac{V_4}{V_3}}{RT_3 \ln\frac{V_4}{V_3}} = \frac{T_3 - T_2}{T_3} = 1 - \frac{T_1}{T_3} = 1 - \frac{T_C}{T_H}$$

For stirling engine without regeneration

$$\eta = \frac{R(T_3 - T_1)\ln\frac{V_1}{V_2}}{c_V(T_3 - T_1) + RT_3 \ln\frac{V_1}{V_2}}$$

# Stirling cycle




Source: http://www.robertstirlingengine.com/theory.php#thermo

# Stirling cycle

According to the good old principle: "nothing is lost, nothing is created, everything is transformed" we can approach the energy exchanges during a cycle of Stirling: any loss of calorific energy during a cycle is equal to the recovered mechanical energy during this same cycle.

On the diagram opposite, we see that we provide in calorific energy: Qexp + Qheat
However, we recover: Qcool + Qcomp
About the mechanical energy, we recover Wexp but we provide Wcomp
The overall balance becomes, by stipulating that the heat energy lost was fully transformed into mechanical energy:

**Qexp + Qheat - Qcool - Qcomp = Wexp - Wcomp**

This leads us to speak about mechanical efficiency: it is the ratio of recovered mechanical energy (the engine goal, in fact) to the heat energy that we must provide:

**Efficiency = ( Wexp - Wcomp ) / (Qexp + Qheat )**

NB : as we can see it in the page “the regenerator”, it is not necessary to provide Qheat if a regenerator is installed. Indeed, at this time there, we recover Qcool

If we refer to the page “the principles”, we are able to show that the efficiency may be expressed according to the temperatures (expressed in Kelvin) of the heat source and of the cold source, according to the following formula:

**Efficiency = $1 - T_m / T_M$**

# STIRLING CYCLE

**Advantages:**

- Potentially high thermal efficiency. Fairly insensitive to fuel or fuel quality. No lubricants needed; the working fluid can support bearing loads.

**Disadvantages:**

- High cost, which may decrease as less expensive materials are used and more units are produced. Low power density and low power-to-weight ratio. Working fluid leakage and sealing.

# Numerical example

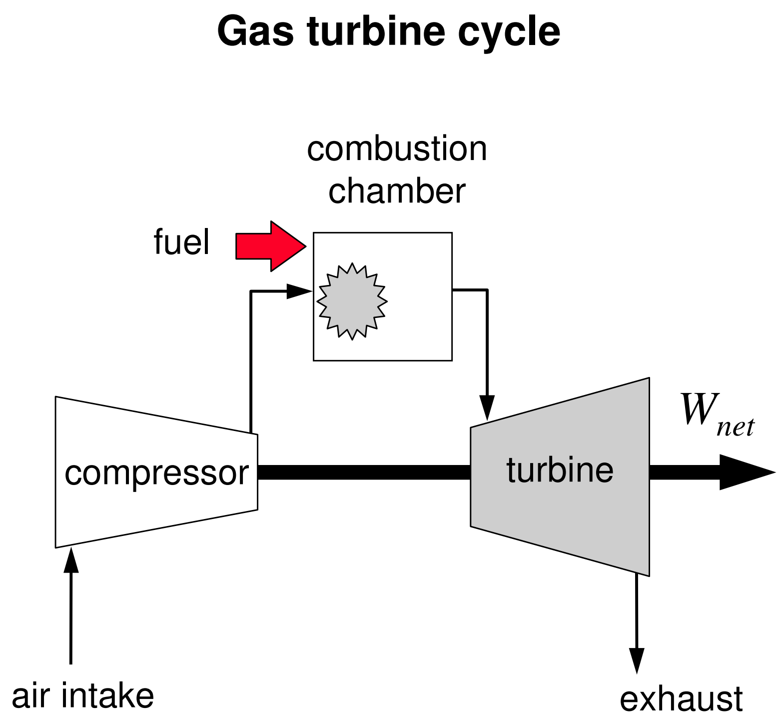
- Consider an ideal Stirling-cycle engine in which the state at the beginning of the isothermal compression process is 100 kPa, 25°C, the compression ratio is 6, and the maximum temperature in the cycle is 1100°C. Calculate the maximum cycle pressure and the thermal efficiency of the cycle with and without regenerators.
- Ans: 2763 kPa, 0.374, 0.783

Source: http://nptel.iitm.ac.in/courses/101101001/downloads/lec20-problem-solution.pdf

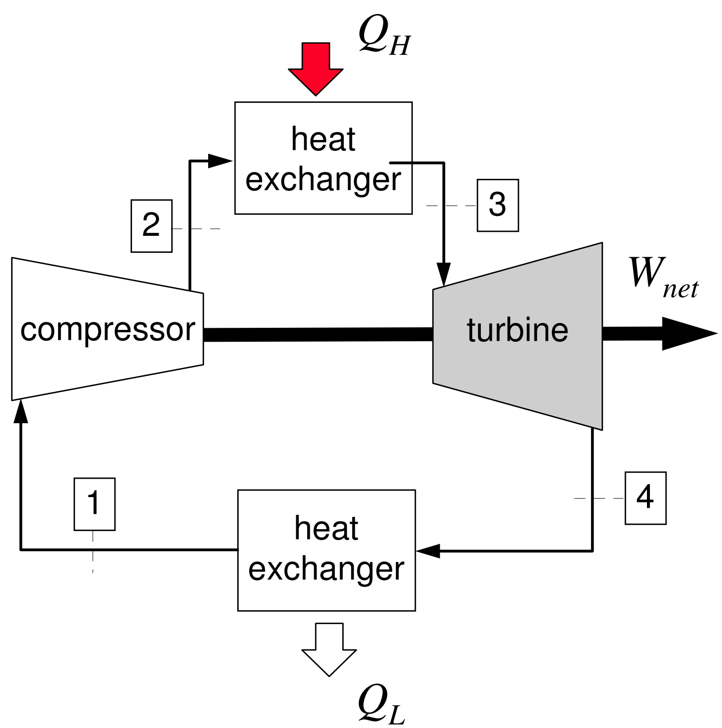
# BRAYTON CYCLE

# BRAYTON CYCLE

Open Brayton cycle




Closed Brayton cycle



# EFFICIENCY OF A BRAYTON CYCLE

## Efficiency of a Brayton cycle

1st law for this cycle: $W = Q_H - Q_L$

energy conversion efficiency is:

$$\eta = \frac{\text{useful work}}{\text{heat input}} = \frac{W}{Q_H} = \frac{Q_H - Q_L}{Q_H}$$

$$\eta = 1 - \frac{Q_L}{Q_H} = 1 - \frac{mC_P(T_4 - T_1)}{mC_P(T_3 - T_2)}$$

$$\eta = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

## Efficiency of a Brayton cycle (cont.)

for an isentropic process:

$$P_1/P_2 = (V_2/V_1)^k$$

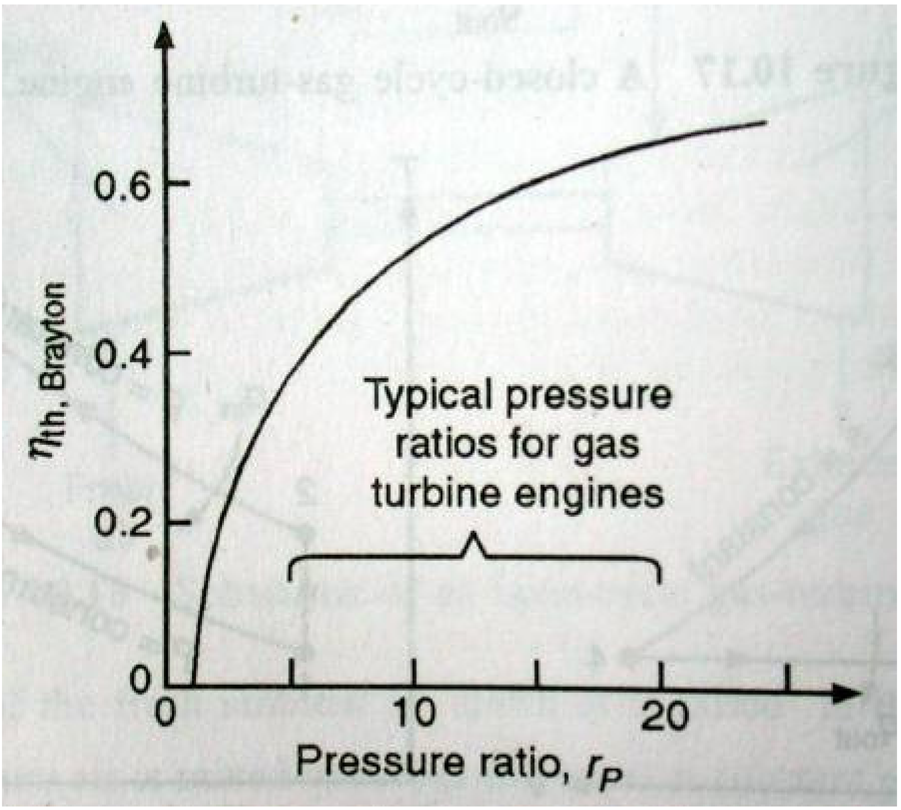
in case of an ideal gas: $P_1V_1/P_2V_2 = T_1/T_2$

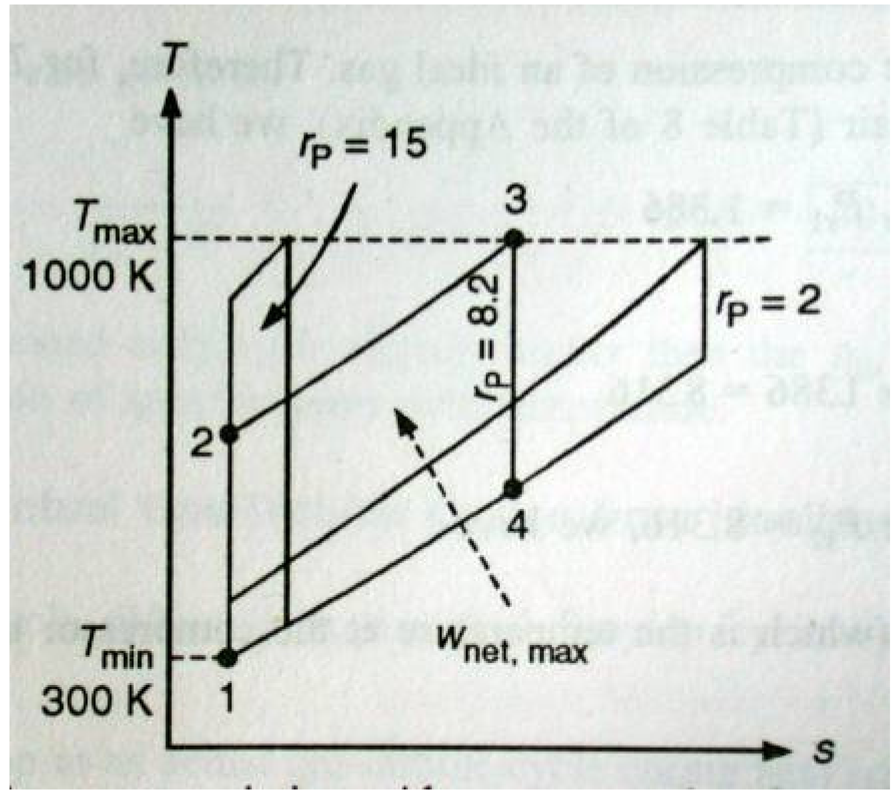
$$\left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}} = \frac{P_2}{P_1} = \frac{P_3}{P_4} = \left(\frac{T_3}{T_4}\right)^{\frac{k}{k-1}}$$

$$\frac{T_3}{T_2} = \frac{T_4}{T_1}$$

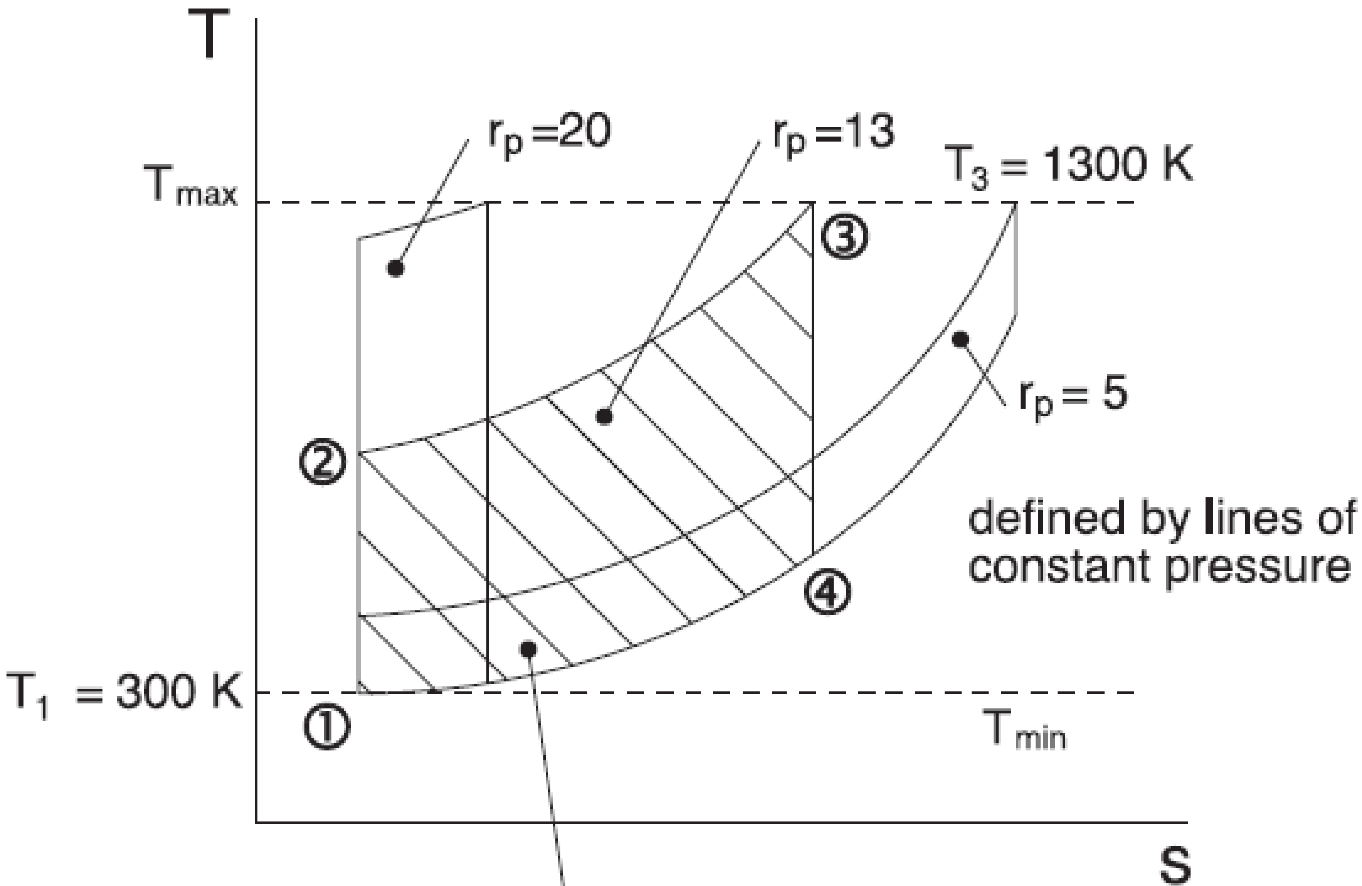
$$\eta = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

# Efficiency vs pressure ratio



# Max net work



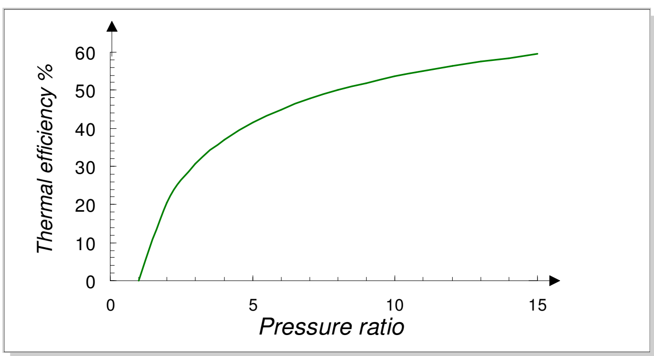

$$W_{net,max} \Rightarrow \quad r_p = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{k}{2k-2}}$$

# EFFICIENCY (cont.)

Efficiency of a Brayton cycle (cont.)

$$\eta = 1 - \frac{1}{\left(P_2 / P_1\right)^{\frac{k-1}{k}}}$$

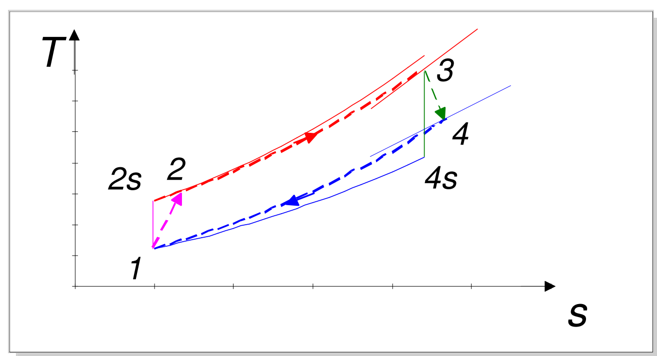
isentropic pressure ratio




The actual gas turbine process

$$\eta_{comp} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

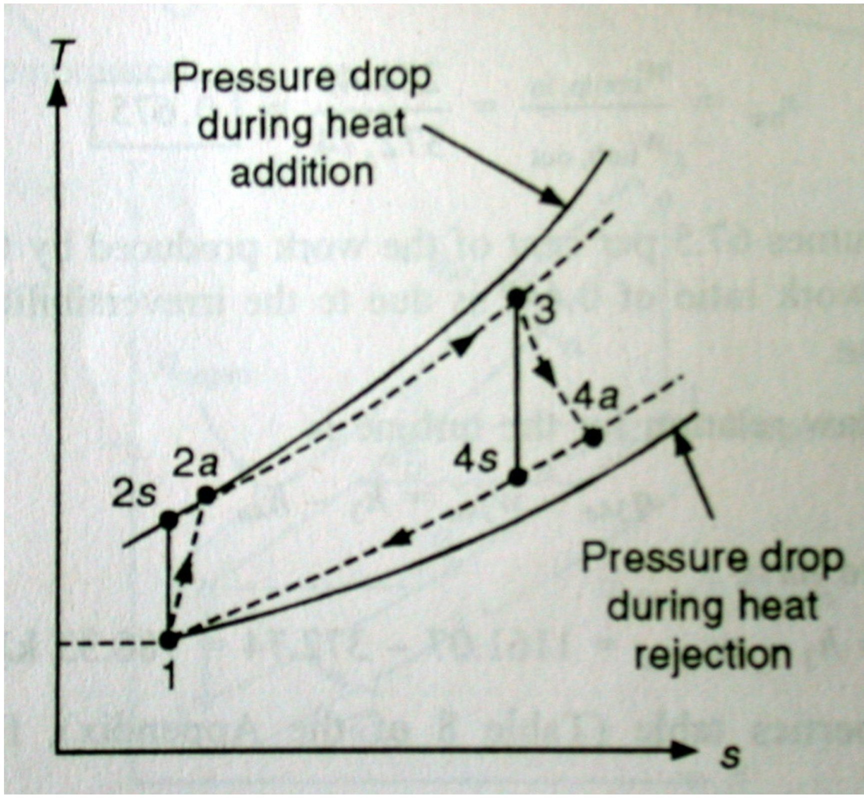
$$\eta_{turb} = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

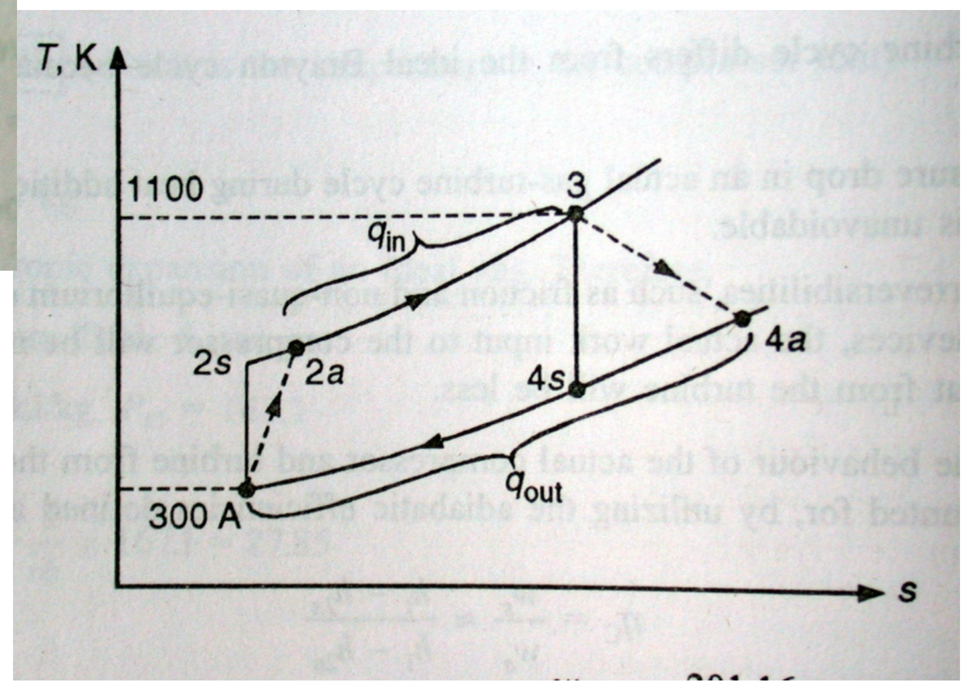


# DEVIATION OF ACTUAL GAS-TURBINE CYCLE FROM IDEALIZED CYCLE

The actual gas turbine cycle differs from the ideal Brayton cycle because of the following reasons:

- Some pressure drop in an actual gas turbine cycle during heat addition and heat rejection processes is unavoidable.
- Owing to irreversibility, such as friction and non-quasi-equlibrium operation conditions of actual devices, the actual work input ot the compressor will be more, and the actual work output from the turbine will be less.


T
Pressure drop during heat addition
3
4a
2a
4s
2s
Pressure drop during heat rejection
1
s


T, K
1100
3
$q_{in}$
4a
2s
2a
4s
$q_{out}$
300 A
s

# BRAYTON CYCLE

**Advantages:**

- These continuous flow machines offer a high power density, and a respectable efficiency at full load on the order of 30% for the open cycle, and even higher for the closed. Brayton cycles are fairly insensitive to fuel quality. The sizes considered for this study also have a distinct advantage over other methods in that the rotating components can be supported by air bearings in any orientation. Thus, no auxiliary lubricating fluid or oil is needed.

**Disadvantage:**

- The primary disadvantage is cost. Although expensive alloys are now used, the cost will decrease as less expensive materials are used and more units are produced.