Energy conversion processes and cycles

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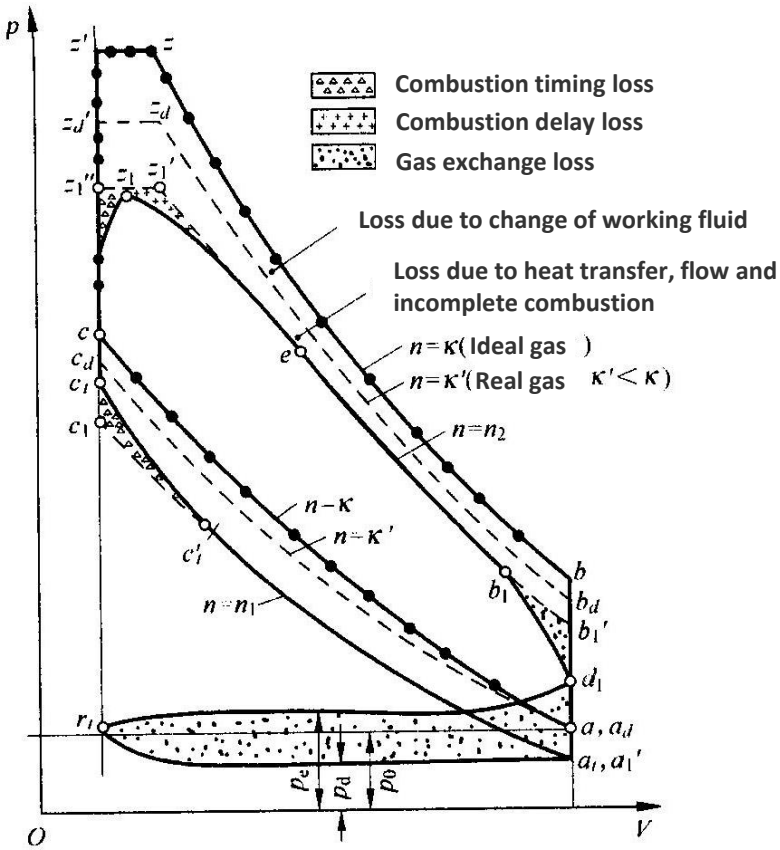
- Combined heat and power (CHP)
- Carnot vapour cycle and steam turbine cycle -- Rankine cycle
- Refrigeration and heat pump cycle
    - Reversed Carnot cycle
    - Ideal vapour-compression refrigeration cycle

Ideal cycle and real cycle

- Most of energy conversion machines/devices operate on cycles.
- The cycles are usually very complex, involving thermodynamics, fluid mechanics, heat transfer, mass transfer, combustion, and etc.
- In the subject of thermodynamics, we take simplification and assumption, focus on equilibrium or quasi-equilibrium processes
- The idealised cycles are very different from real cycles, however they still retain the general characteristics of the actual cycles they represent.
- The analysis of ideal cycle also provides a direction of improvement, and theoretical limits.



Ideal engine cycle vs real engine cycle

Definitions and analysis method

- Thermal efficiency:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}}$$

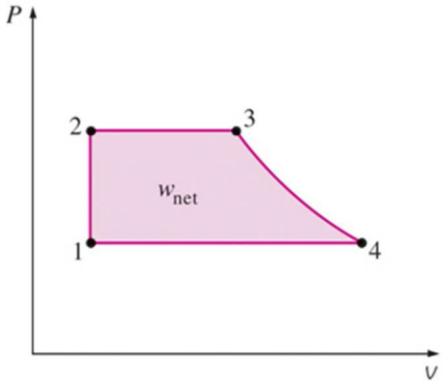
- Idealisations and simplifications
  - Friction losses neglected
  - All processes take place in a quasi-equilibrium manner, internally reversible
  - In most components, apart from heat exchangers, heat transfer is neglected.
  - Mostly, kinetic and potential energies are neglected.
- Approaches:
  - p-v diagram and T-s diagram are often used

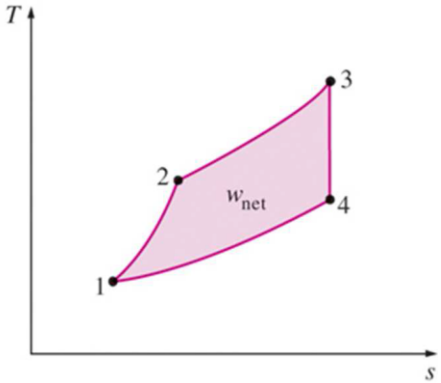
$$w = \int p dv \qquad q_R = \int T ds$$

Over a cycle, due to the first law, $w_{net} = \oint p dv = q_{net} = \oint T ds$

Definitions and analysis method

- Approaches
  - p-v and T-s diagrams





- General procedure
  - State equations

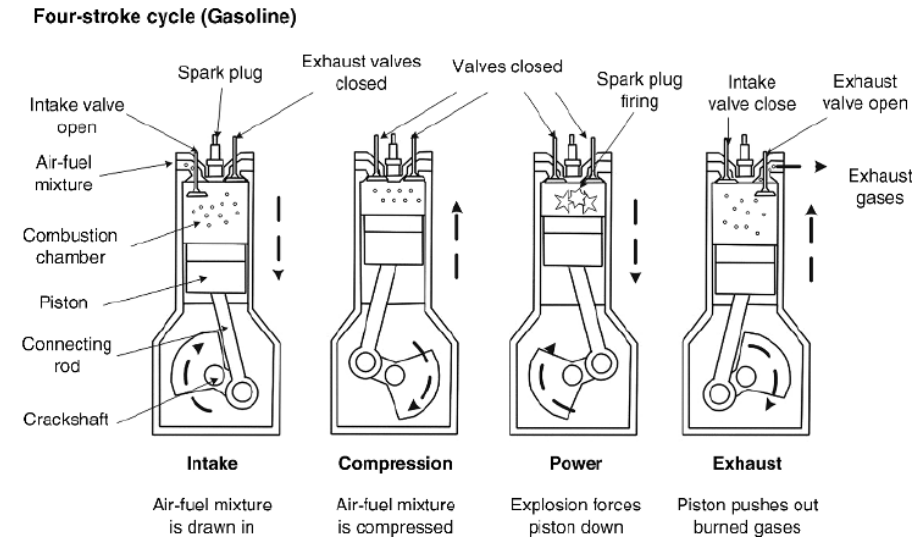
    $pV = mRT = nR_0T$ for ideal gas

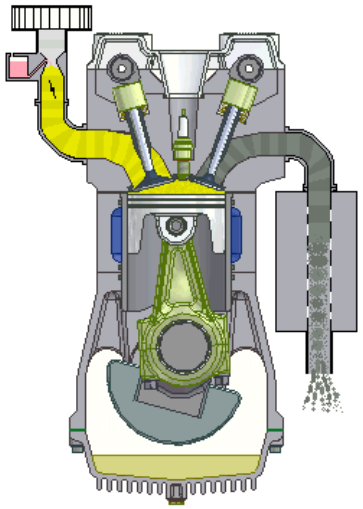
    Otherwise using steam tables
  - Process equations

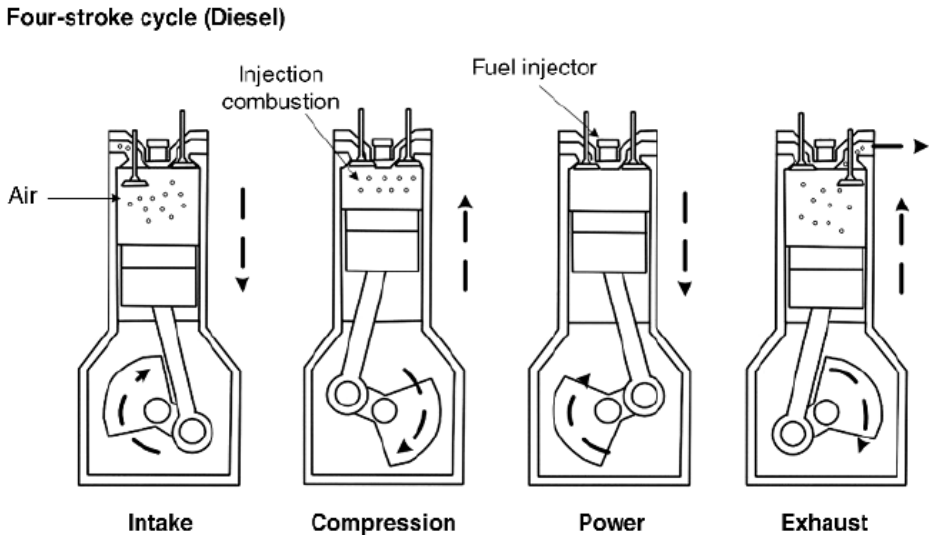
    $pv^{\gamma} = c$ … for isentropic process, and so on, for ideal gas

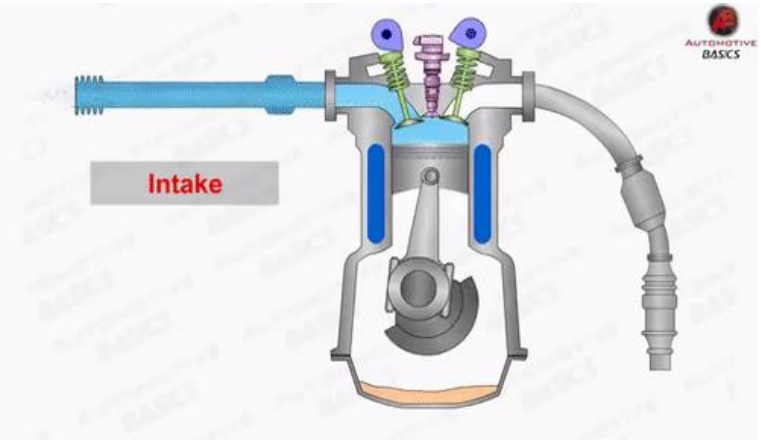
    Otherwise using steam tables following particular processes
  - States → processes → cycle → cycle performance

4-stroke internal combustion engines









An overview of IC engine cycles

Assumptions:

1. The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.
2. All the processes are internally reversible.
3. The combustion process is replaced by a heat-addition process from an external source.
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

Definitions

TDC= Top dead centre

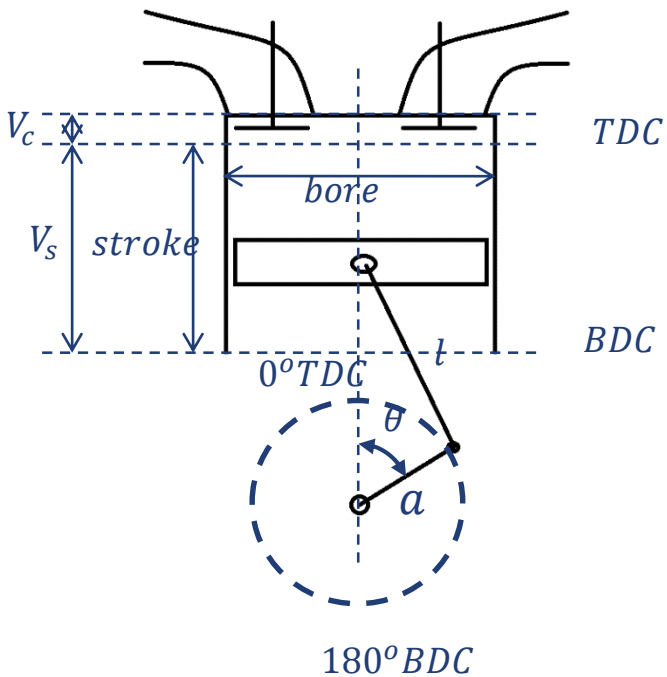
BDC= Bottom dead centre

b → bore

S → stroke

Swept volume $V_s = \frac{\pi b^2 s}{4}$

Displacement $V_d = V_s \times n$ n: cylinder number



Definition: 曲轴角度

θ: Crank angle

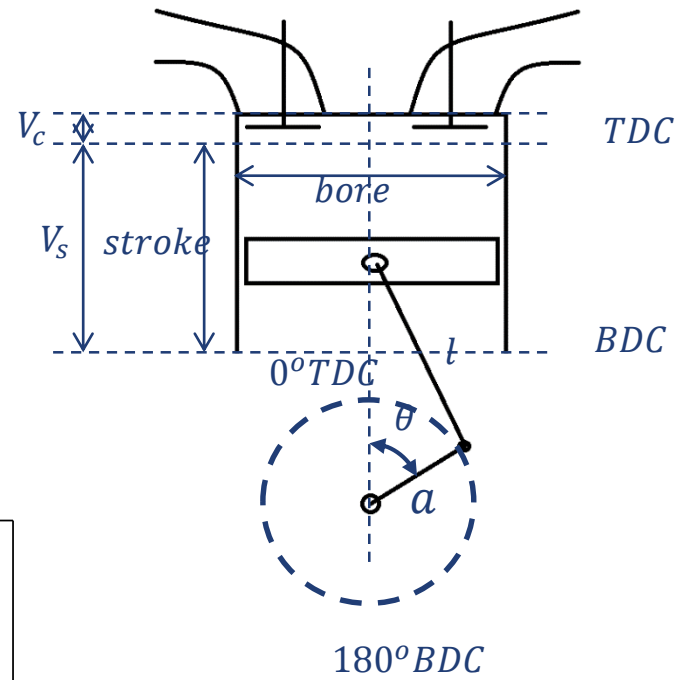
Note: for 4 stroke engines, 2 rev=720 ˚CA=1 cycle

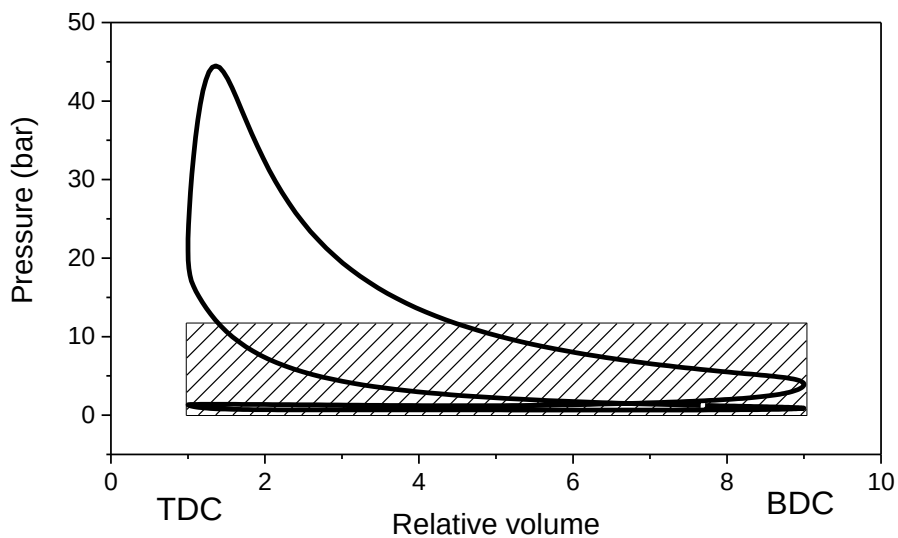
V_c: clearance volume

Compression ratio $r_c = \frac{V_s+V_c}{V_c} = 1 + \frac{V_s}{V_c}$

Mean Effective Pressure (MEP)

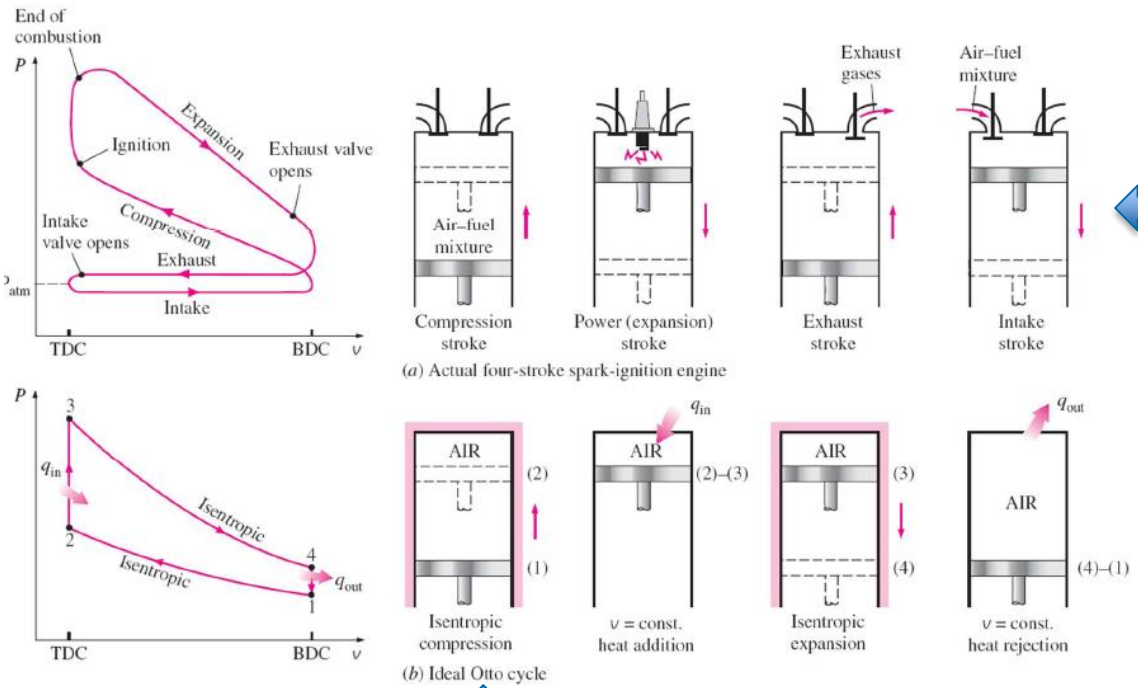
$$W_{net} = MEP\ (V_{max} - V_{min})$$





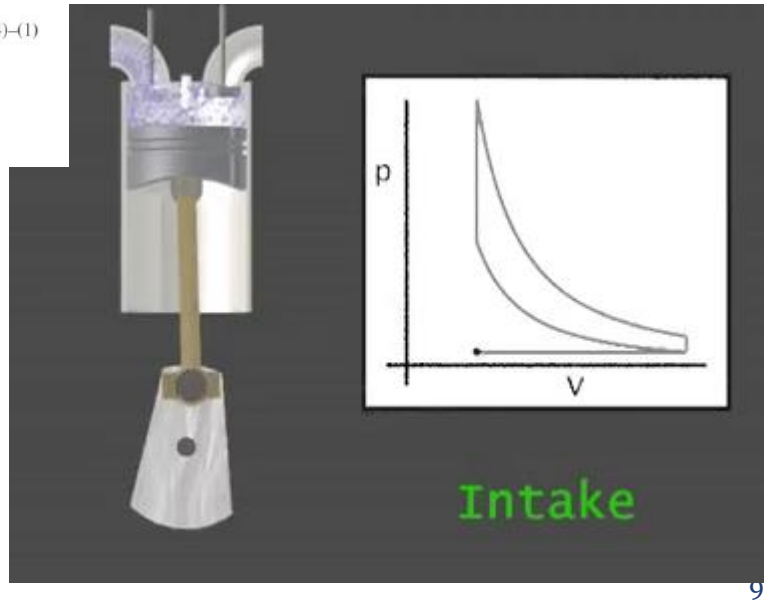
Otto cycle

Ideal cycle for gasoline engines

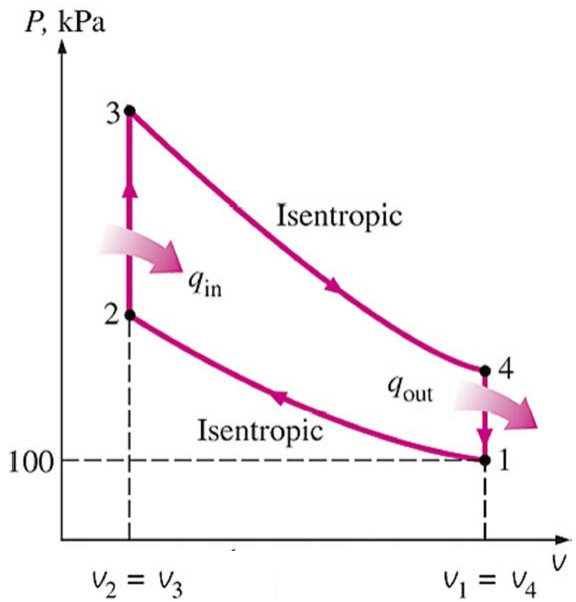


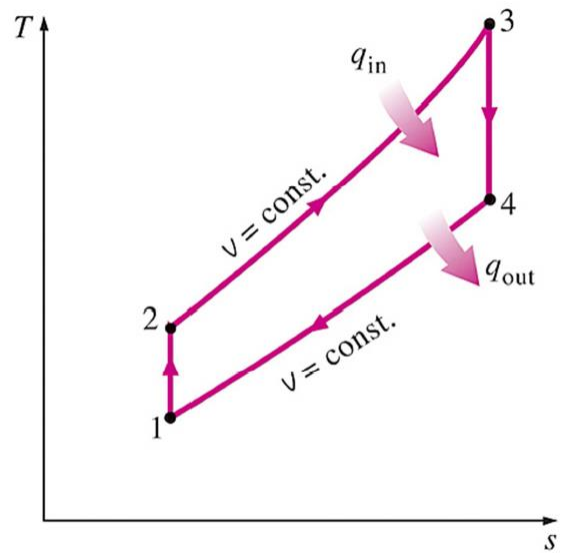
Actual cycle

Ideal cycle



Otto cycle





why T-s graph is this

Compression ratio $r_c = \frac{v_4}{v_3} = \frac{v_1}{v_2}$

1→2 isentropic compression (reversible adiabatic) $(q = 0, -w = \Delta u, pv^{\gamma} = c)$

2→3 isochoric heat addition (combustion) $(q = \Delta u, w = 0, \frac{p}{T} = c)$

3→4 isentropic expansion (reversible adiabatic)

4→1 isochoric heat rejection (exhaust)

Otto cycle efficiency

Heat addition (2→3): $q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$

Heat rejection (4→1): $q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$

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

1→2 and 3→4 isentropic process, therefore

$PV = RT$

$PV^{\lambda} = C$

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} = \left(\frac{v_3}{v_4}\right)^{\gamma-1} = \frac{T_4}{T_3} = \left(\frac{1}{r_c}\right)^{\gamma-1}$$

$PV = \frac{cT_1}{V^{\lambda-1}}$

$= \frac{P_1 V_1}{P_2 V_2} = \left(\frac{V_2}{V_1}\right)^{\lambda-1}$

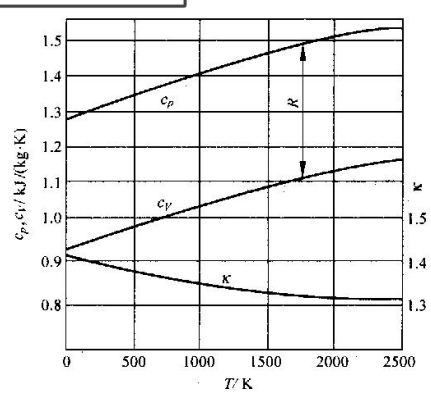
From above, $\frac{T_4}{T_1} = \frac{T_3}{T_2}$

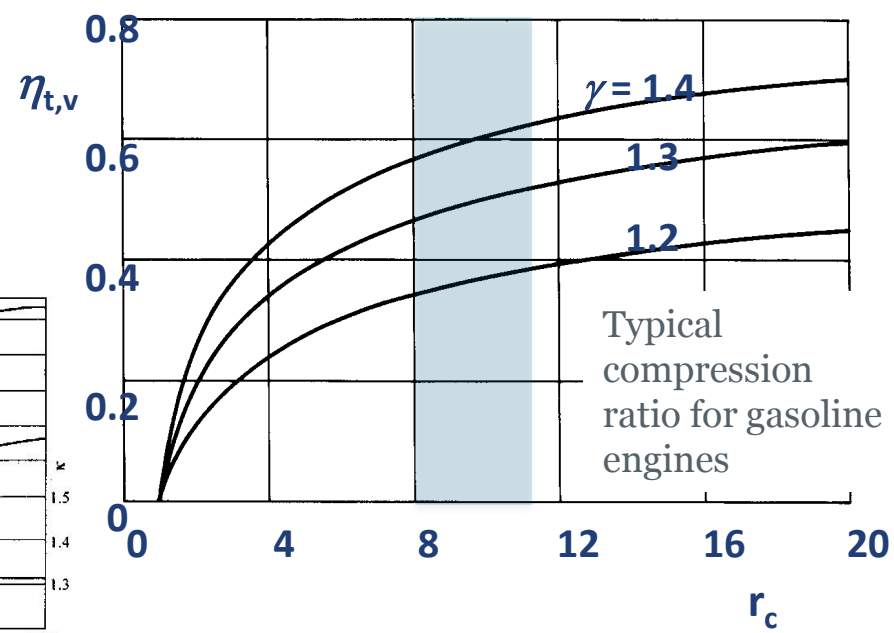
Hence efficiency

$$\boxed{\eta_{th} = 1 - \left(\frac{1}{r_c}\right)^{\gamma-1}}$$

$r_c = \frac{V_4}{V_3} = \frac{V_1}{V_2}$

$\lambda = \frac{C_p}{C_v} = 1 + \frac{R}{C_v}$





Example

For an Otto cycle described as in the following diagrams, calculate:

(a) the temperature and pressure at the end of each process

(b) the thermal efficiency $\eta = 1-\left(\frac{1}{8}\right)^{1.4} = 0.56$ $PV = RT$ $PV^{1.4} = C$

(c) the mean effective pressure $MEP = \frac{W_{net}}{V_2 - V_1}$ $T \cdot V^{0.4} = C$

(take $\gamma = 1.4$, $R = 287\ kJ/kgK$, $c_v = 720\ kJ/kgK$) $T_1 \cdot V_1^{0.4} = T_2 \cdot V_2^{0.4}$

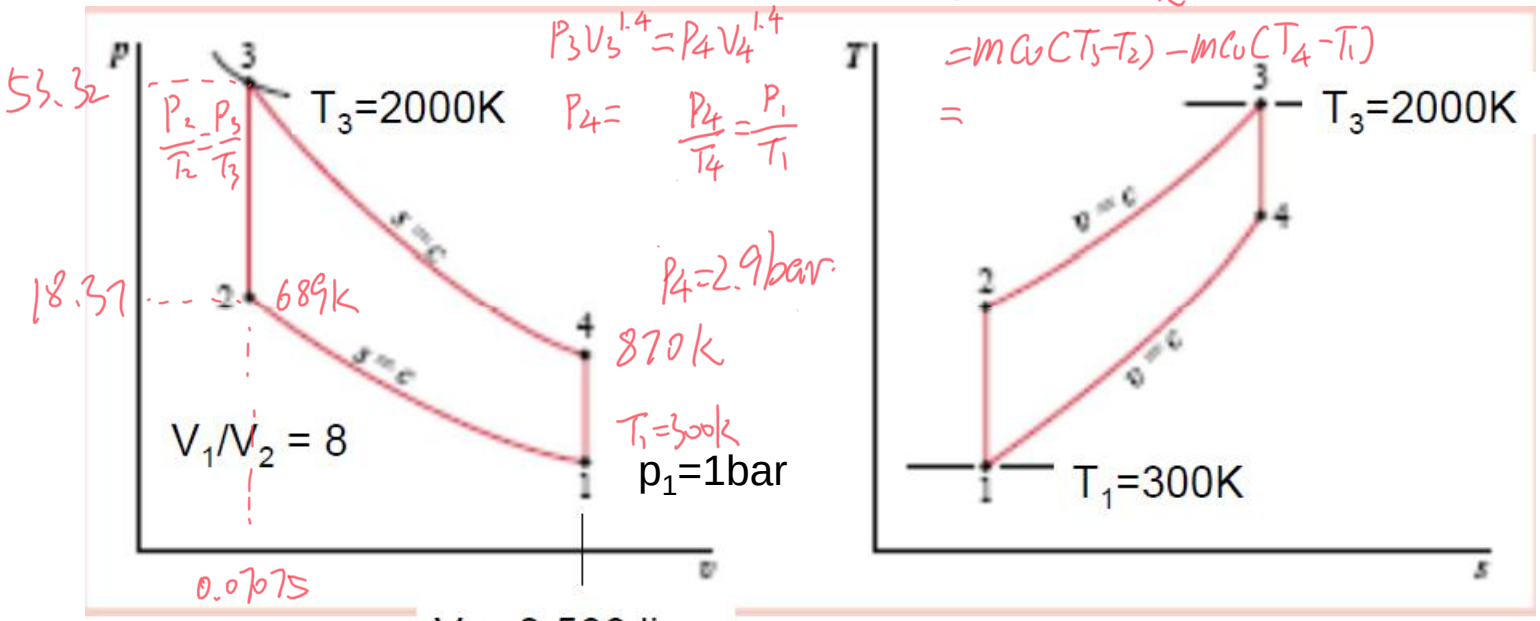
$W_{net} = Q_{in} - Q_{out}$

$P_3 V_3^{1.4} = P_4 V_4^{1.4}$

$= mC_v(T_3 - T_2) - mC_v(T_4 - T_1)$

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

$P_4 = 2.9\,bar$



$V_1 = 0.566$ li

Solution

Given conditions:

$p_1 = 1 \times 10^5; T_1 = 300; V_1 = 0.566 \times 10^{-3}; rc = \frac{V_1}{V_2} = 8;$

$T_3 = 2000; \gamma = 1.4; R = 287; cv = 720;$

Following the process:

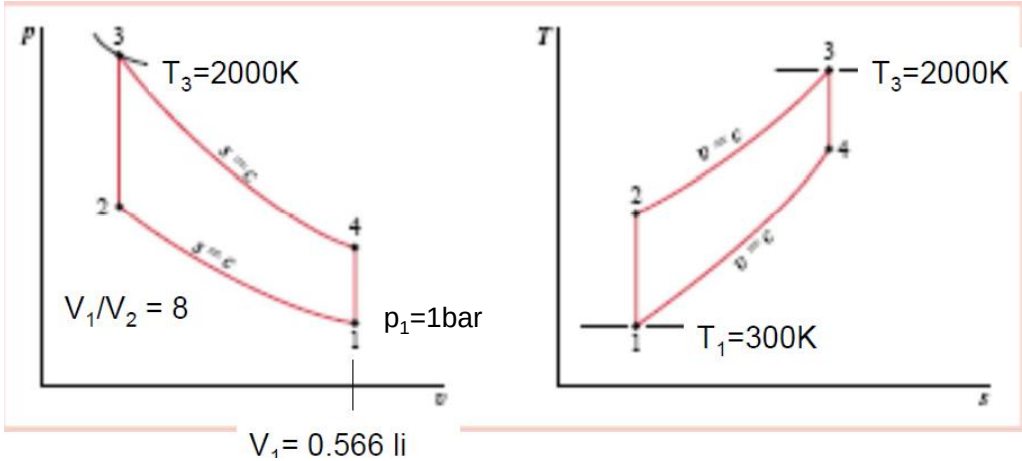
1→2 isentropic compression

$p_1 V_1^{\gamma} = p_2 V_2^{\gamma};$

$=> p_2 = 18.38 \text{ bar}$

$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1}$

$=> T_2 = 689.2 \text{ K}$



2→3 isochoric heat addition: $\frac{p_3}{T_3} = \frac{p_2}{T_2} => p_3 = 53.3 \text{ bar}$

3→4 isentropic expansion: same as from 1→2: $T_4 = 870.6K, p_4 = 2.9\ bar$

Solution

Cyclic parameters:

$$\eta_{th} = 1 - \frac{1}{rc^{\gamma-1}} = 0.5647$$

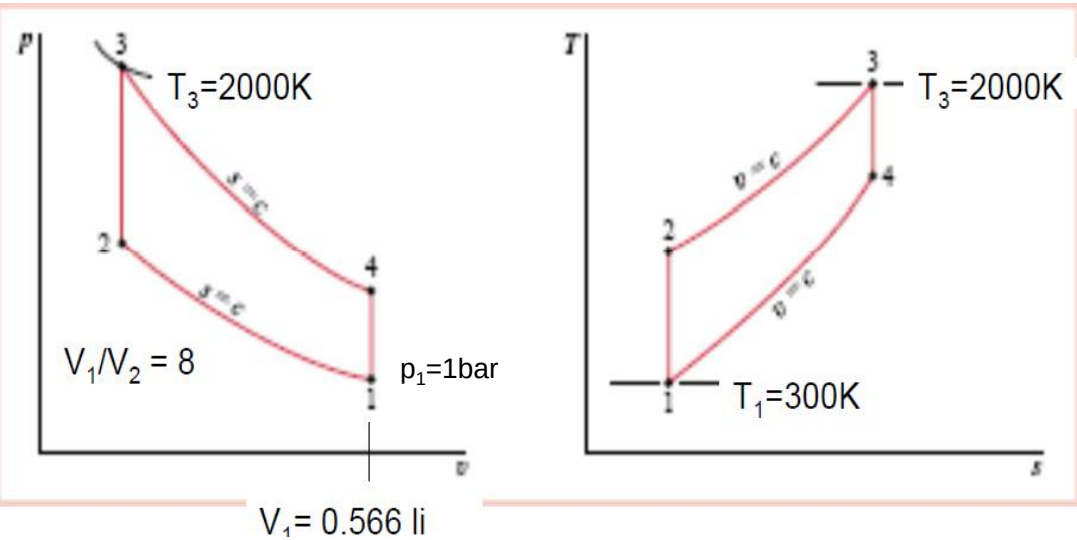
$$m = \frac{p_1 V_1}{RT_1} = 0.6574g$$

$$Q_{in} = mc_v(T_3 - T_2) = 620.4J$$

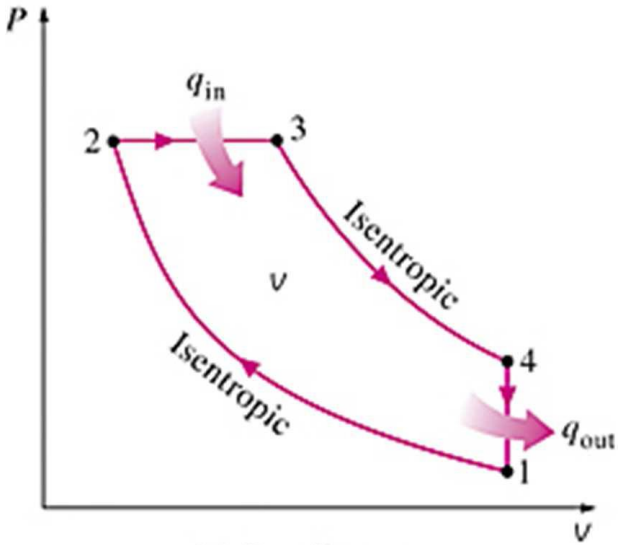
$$Q_{out} = mc_v(T_4 - T_1) = 270J$$

$$W_{net} = Q_{in} - Q_{out} = 350.4J$$

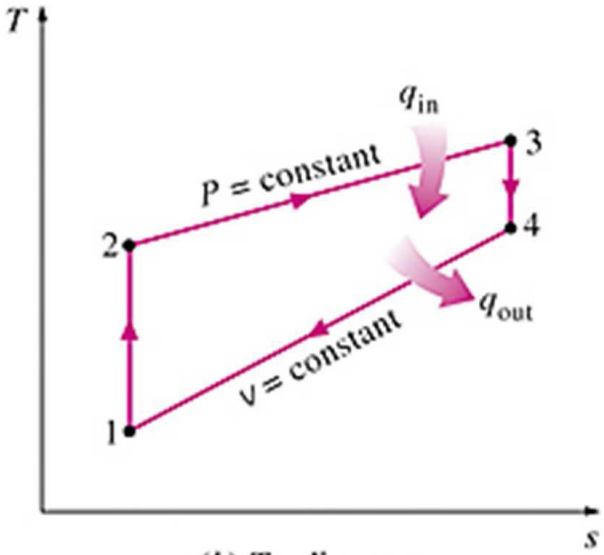
$$MEP = \frac{W_{net}}{V_1 - V_2} = 7.1bar$$



Diesel cycle



(a) P- v diagram



(b) T-s diagram

1→2 isentropic compression (reversible adiabatic) $(q = 0, -w = \Delta u, pv^{\gamma} = c)$

2→3 **isobaric** heat addition (combustion) $(q = \Delta u + w,\ \ w = p\Delta v,\ \ p = c)$

3→4 isentropic expansion (reversible adiabatic)

4→1 isochoric heat rejection (exhaust)

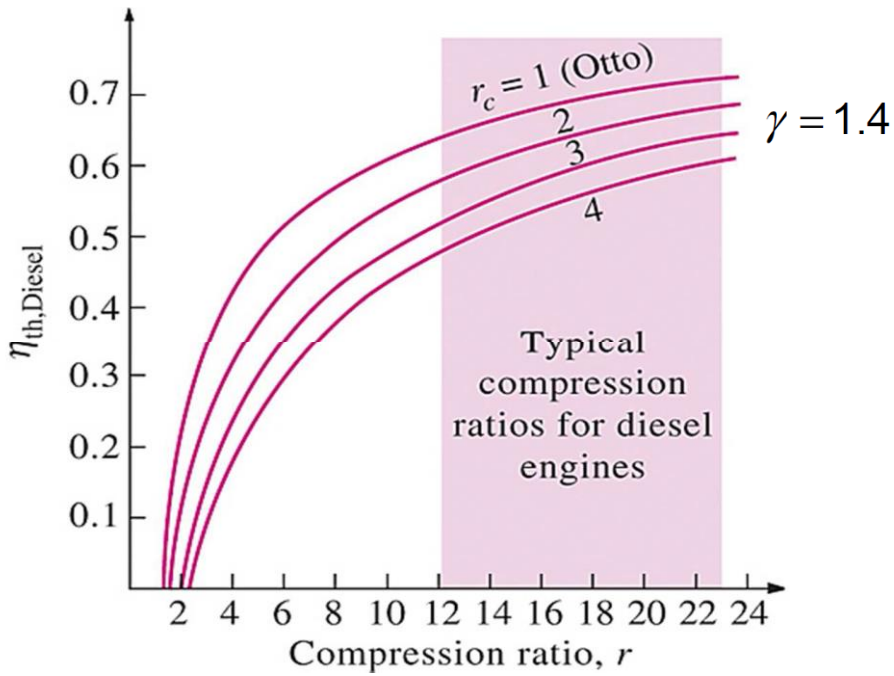
Cut-off ratio: $r_{cut-off} = V_3/V_2$

Diesel cycle efficiency

Diesel cycle efficiency:

$$\eta_{th} = 1 - \frac{1}{r_c^{\gamma-1}} \frac{1}{\gamma} \left[\frac{r_{cut-off}^{\gamma} - 1}{r_{cut-off} - 1} \right]$$

When $r_{cut-off} = 1$, Otto cycle



Diesel engines

Low speed diesel engine example

High efficiency low speed MAN 2-stroke bio-Diesel engine for power Generation

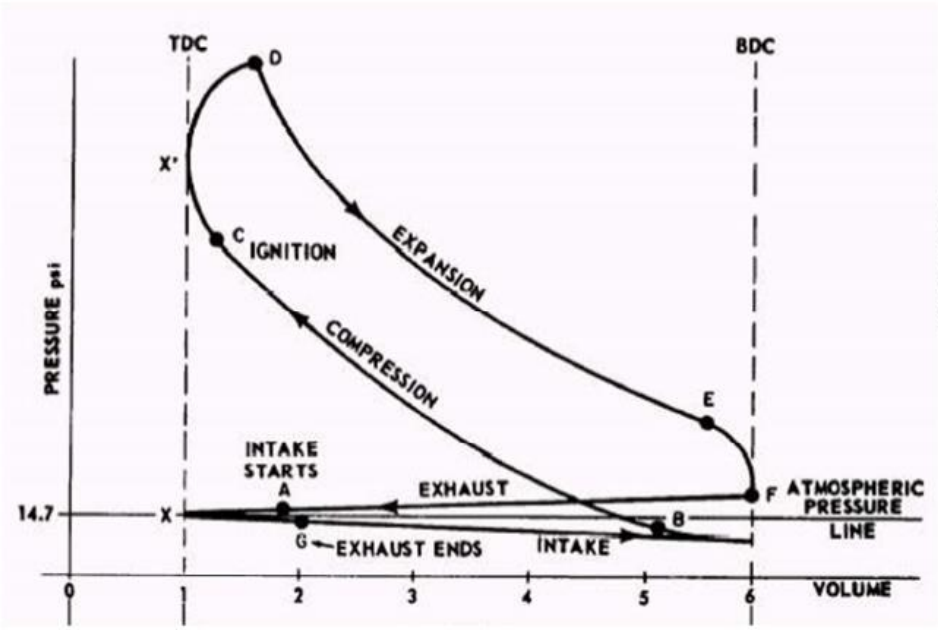
$\eta \approx 49\%$

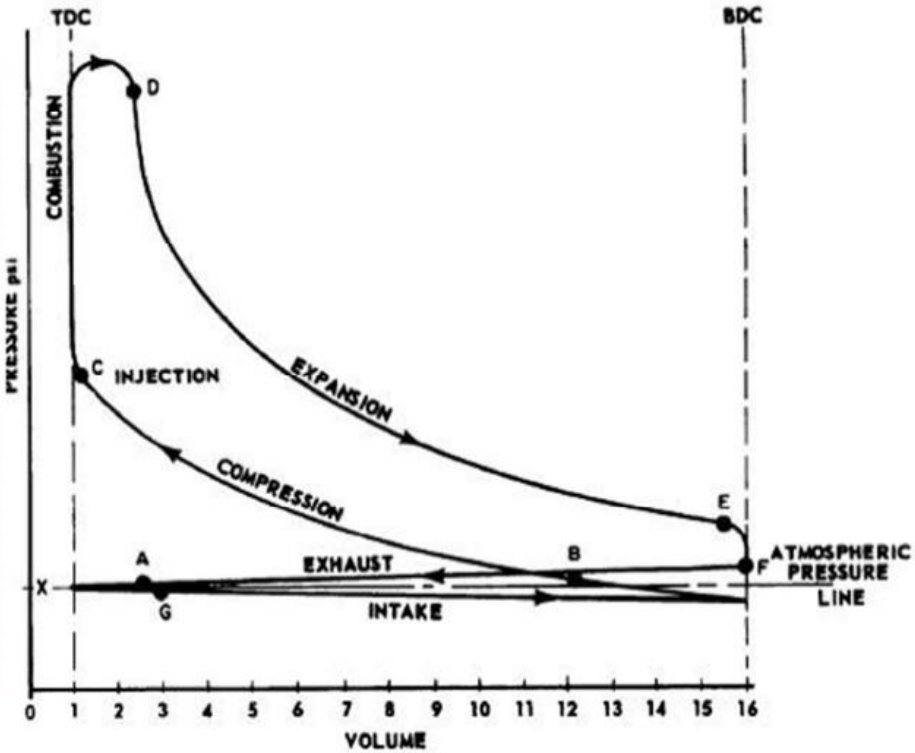
6MW at 200rpm

Dual cycle

The air-standard Otto and Diesel cycles are not perfect matches to the real IC engine cycles.

Typical p-V curves for the real cycles are shown below



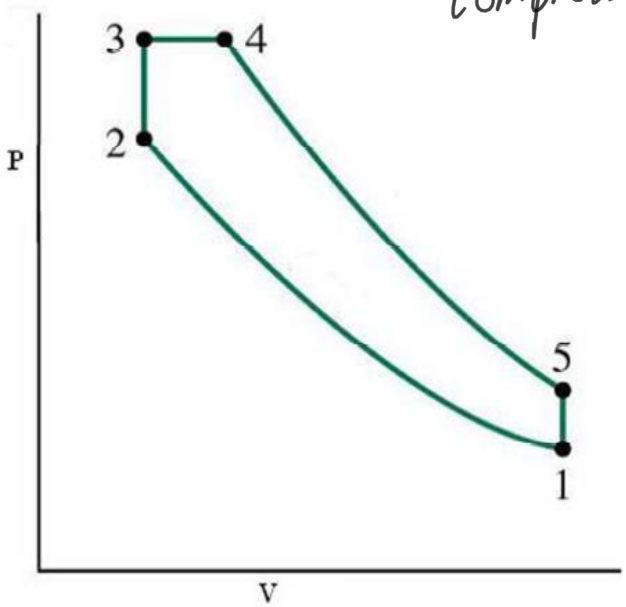


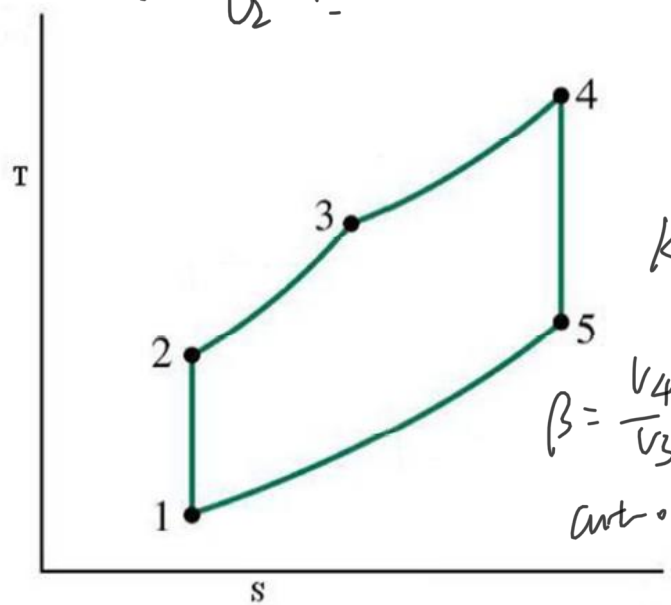
Dual cycle

By combining the idealized cycles a closer match can be obtained.

This combination is known as the dual cycle and is shown below.

Compression ratio $= \frac{V_1}{V_2}$?





$k = \gamma = \frac{C_p}{C_v}$

$\beta = \frac{V_4}{V_3}$ [diseal]

cut-off ratio

1→2 isentropic compression (reversible adiabatic)

2→3 **isochoric** heat addition (combustion)

3→4 **isobaric** heat addition (combustion)

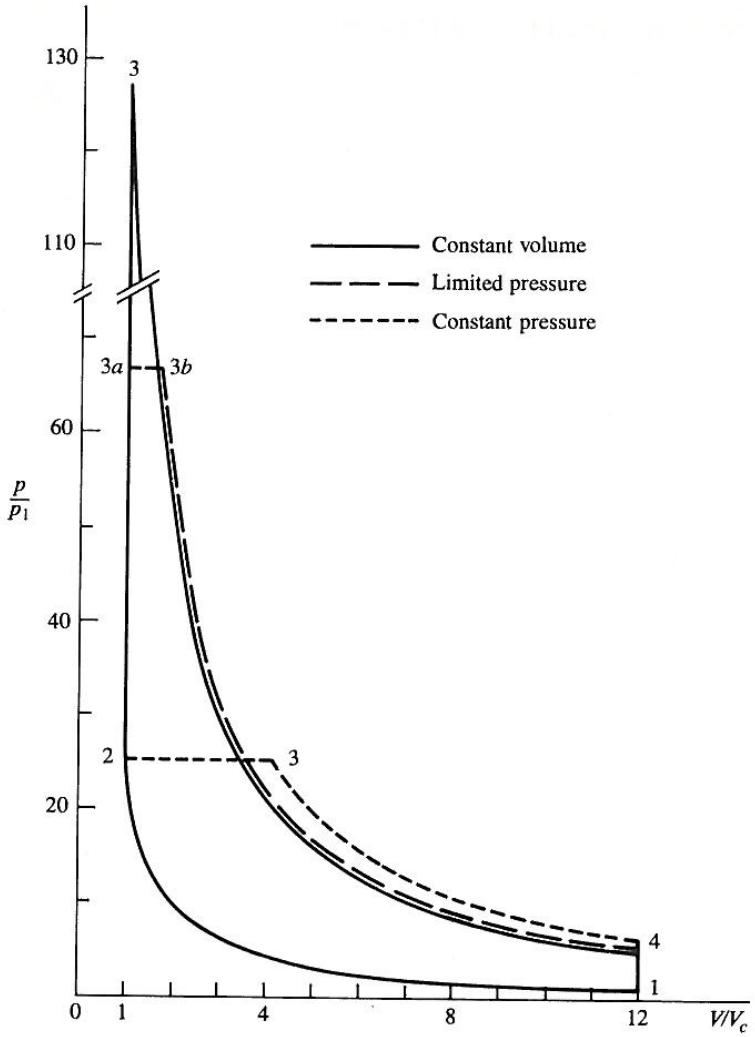
4→5 isentropic expansion (reversible adiabatic)

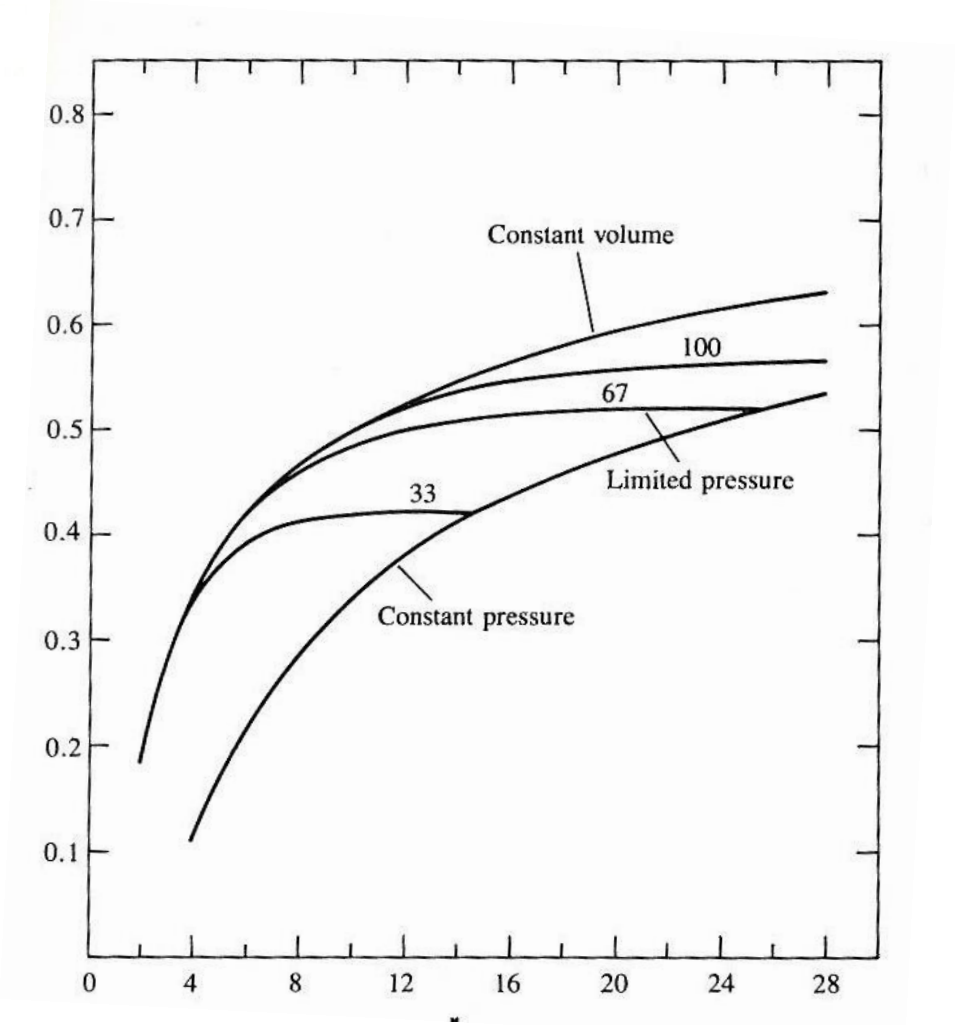
5→1 isochoric heat rejection (exhaust)

$$\boldsymbol{\eta_t = 1 - \frac{1}{{r_c}^{\kappa-1}}\left[\frac{\alpha\beta^{\kappa} - 1}{\alpha\kappa(\beta - 1) + \alpha - 1}\right]}$$

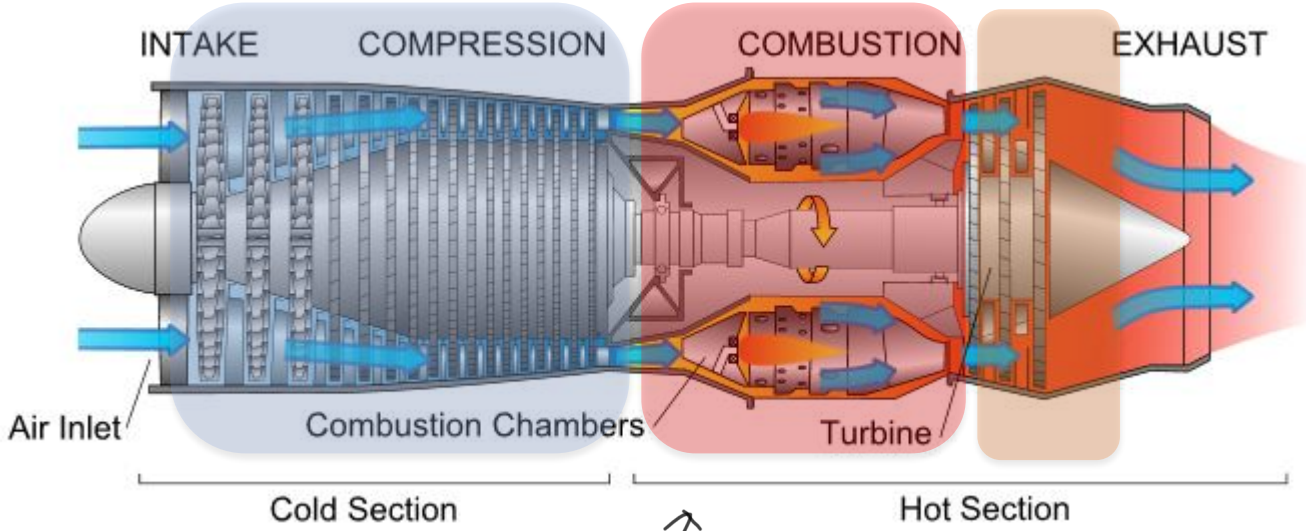
Where $\alpha = \frac{p_3}{p_2}$, $\beta = \frac{V_4}{V_3}$

Comparison of IC engine cycles

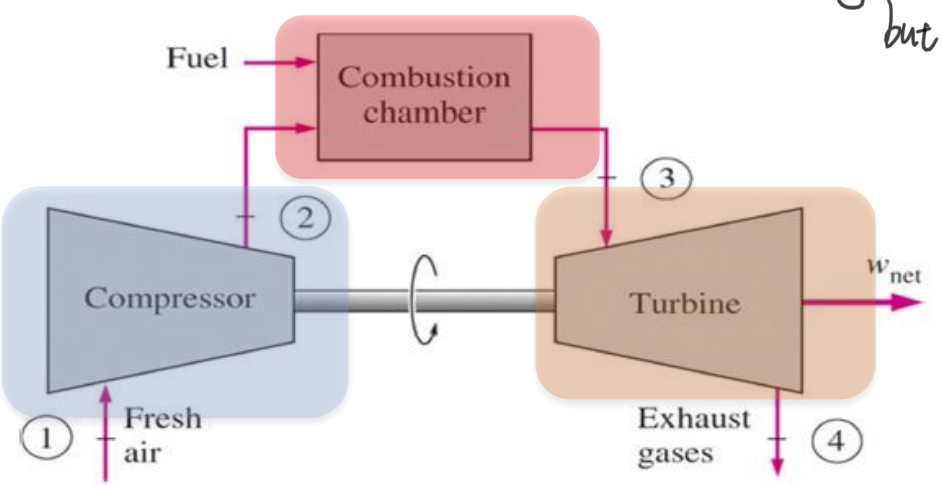




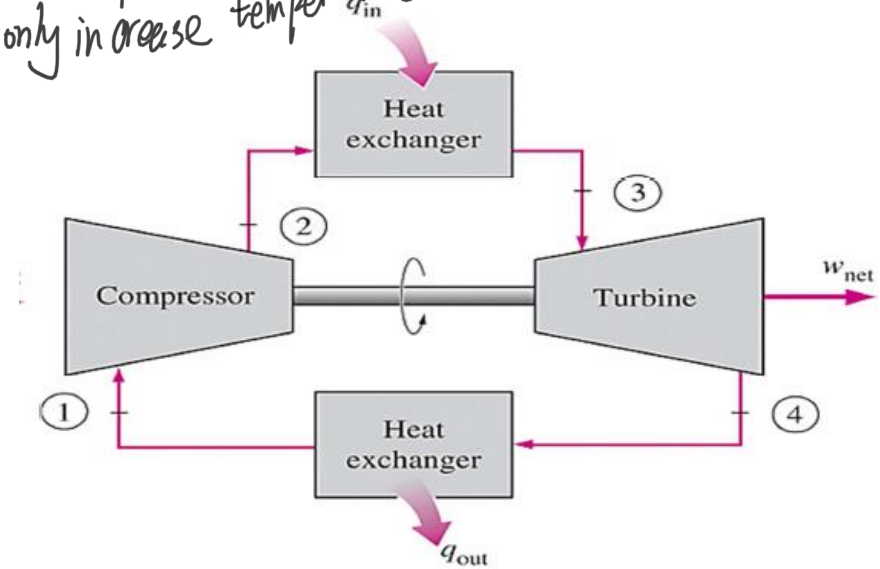
Gas turbine cycle-Brayton-Joule cycle



why constant pressure
but only increase temperature

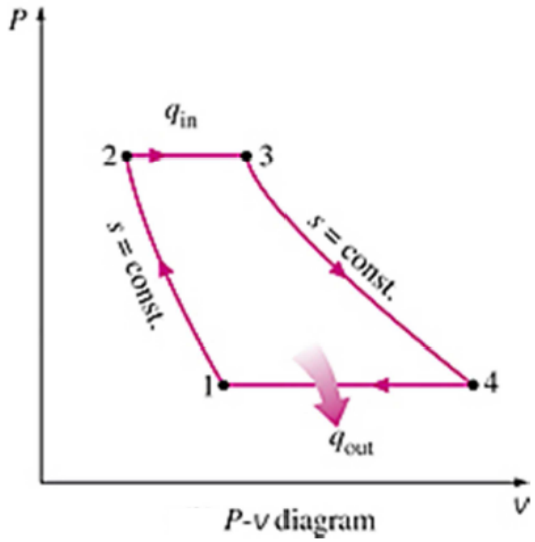


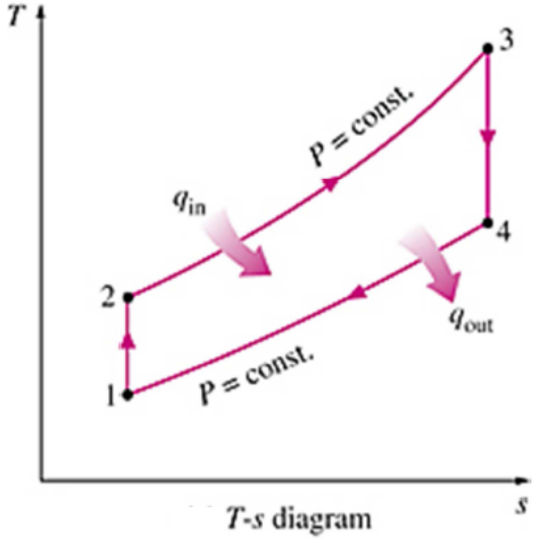
Open gas-turbine cycle



Idealized closed gas-turbine cycle

Brayton-Joule cycle





1→2 isentropic compression

2→3 isobaric heat addition (combustion)

3→4 isentropic expansion

4→1 isobaric heat rejection (exhaust)

Pressure ratio: $r_p = \frac{p_2}{p_1} = \frac{p_3}{p_4}$

Brayton-Joule cycle efficiency

In this cycle, we adopt SFEE as air flows around the system

$$q_{in} = q_{23} = h_3 - h_2 = c_p(T_3 - T_2)$$

$$q_{out} = q_{41} = h_4 - h_1 = c_p(T_4 - T_1)$$

why same

Efficiency: $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}$$

As $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}, \quad \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$ ← ideal gas

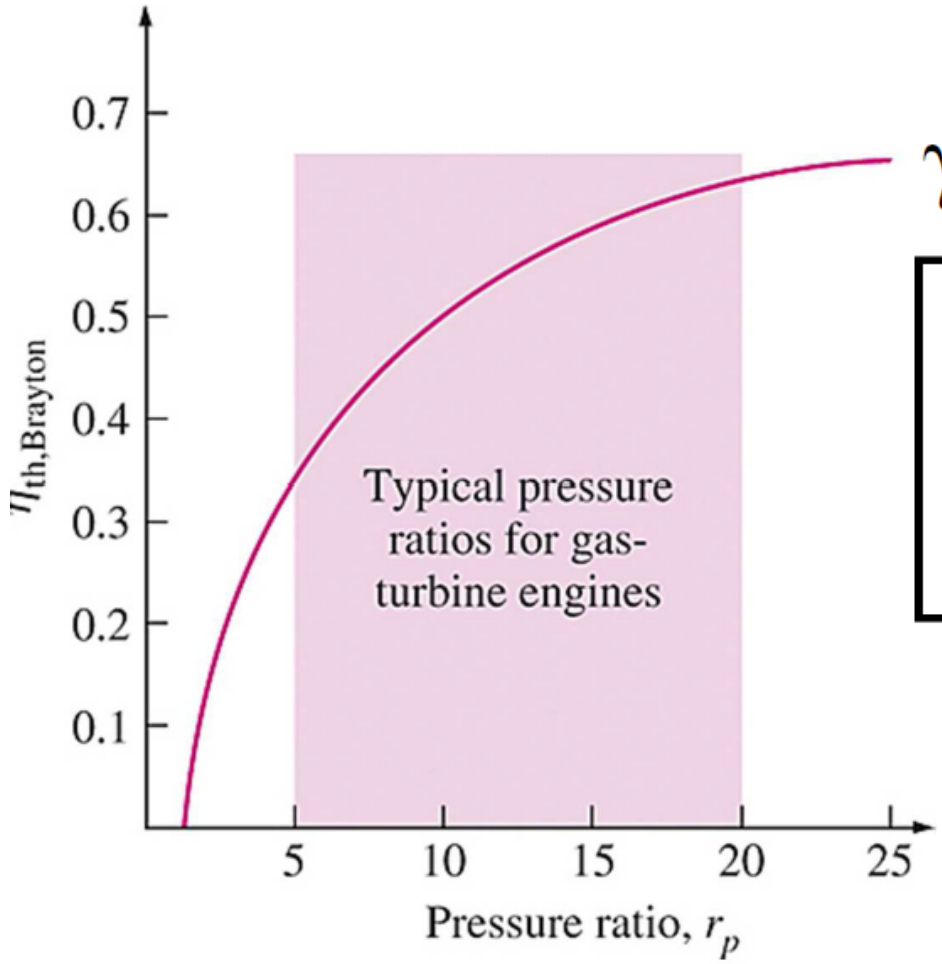
And $$p_1 = p_4, \qquad p_2 = p_3, \qquad r_p = \frac{p_2}{p_1}$$ pressure ratio

Hence $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

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

Thus $$\boldsymbol{\eta_{th} = 1 - \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}}}$$

Brayton-Joule cycle efficiency



$\gamma = 1.4$

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

Brayton-Joule cycle – effect of temperature limits

When operating between fixed temperature limits, the maximum net work is

$$\frac{\dot{W}_{net}}{\dot{m}} = w_{net} = w_{out} - w_{in}$$

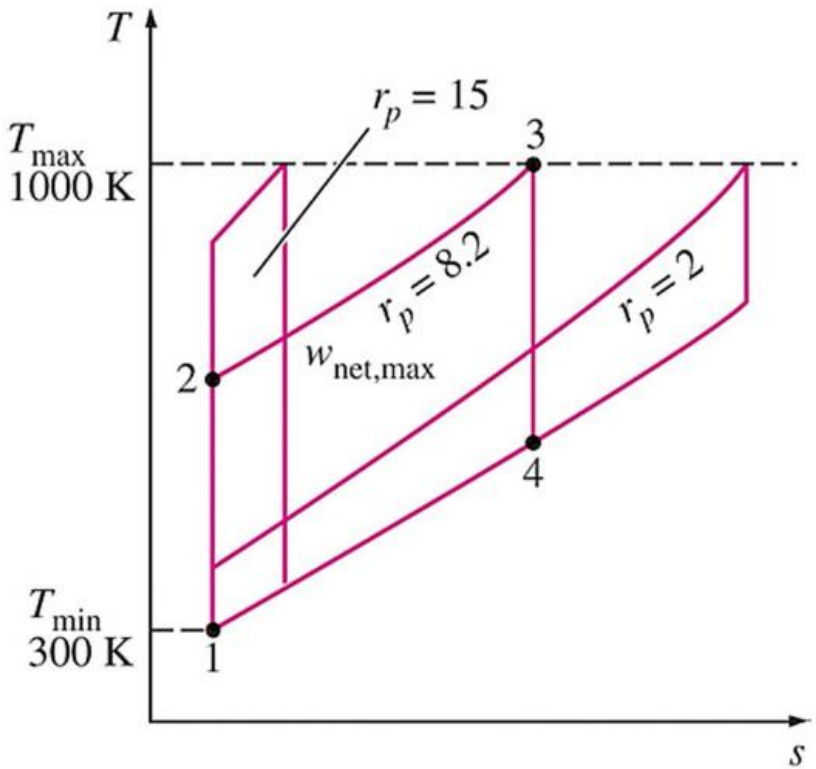
$$= c_p[(T_3 - T_4) - (T_2 - T_1)]$$

$$= c_p T_1 \left[\left(\frac{T_3}{T_1} - \frac{T_4}{T_1}\right) - \left(\frac{T_2}{T_1} - 1\right)\right]$$

$$= c_p T_1 \left[\left(\frac{T_3}{T_1} - \frac{T_3}{T_2}\right) - \left(\frac{T_2}{T_1} - 1\right)\right]$$

$$= c_p T_1 \left[\frac{T_3}{T_1}\left(1 - \frac{T_1}{T_2}\right) - \left(\frac{T_2}{T_1} - 1\right)\right]$$

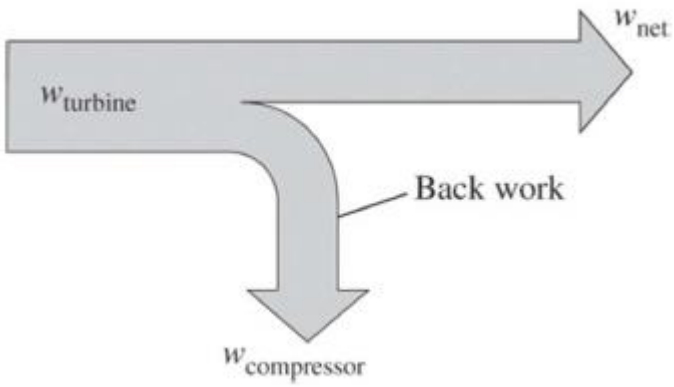
$$= c_p T_1 \left[\frac{T_3}{T_1}\left(1 - \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}}\right) - \left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right)\right]$$



This is a maximum when $\frac{T_3}{T_1} = r_p^{2(\gamma-1)/\gamma}$ or $r_p = \sqrt{\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{\gamma-1}}}$

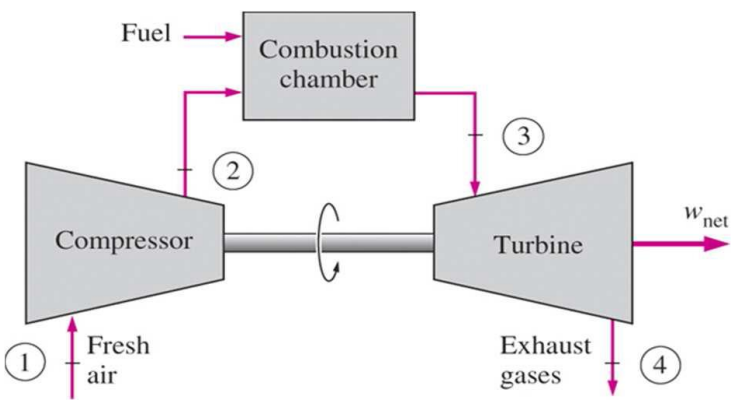
Work ratio or back work ratio

Back Work is the amount of work that is absorbed on the turbine drive shaft by the compressor.



Back-work ratio

$$= \frac{W_{compressor}}{W_{turbine}} \approx 50\% \qquad \text{typ}$$



OR …work ratio

$$= \frac{W_{net}}{W_{turbine}} = \frac{W_{turb} - W_{comp}}{W_{turb}}$$

$$= 1 - (\text{back work ratio})$$

Actual gas turbine cycles

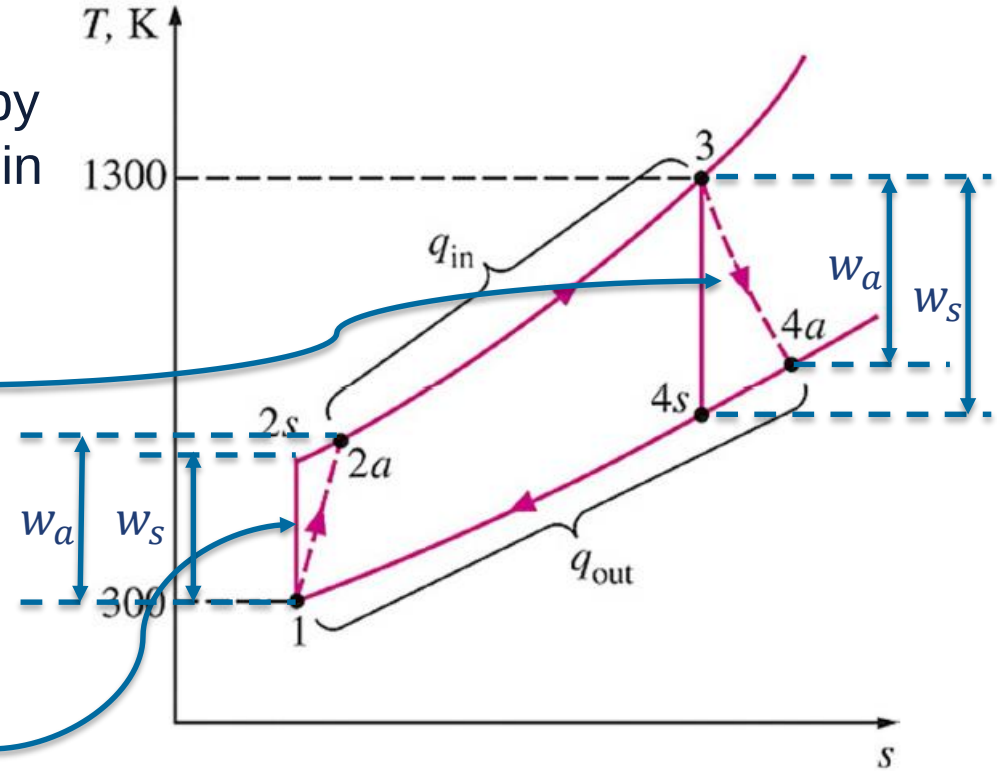
In actual gas turbine cycles, entropy increases caused by irreversibility in both compressor and turbine.

Isentropic turbine efficiency:

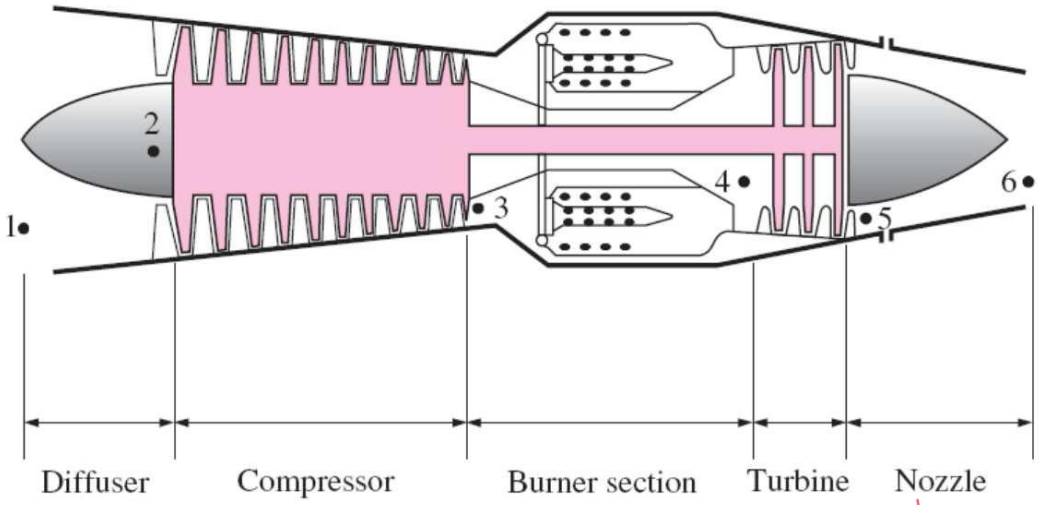
$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

Isentropic compressor efficiency:

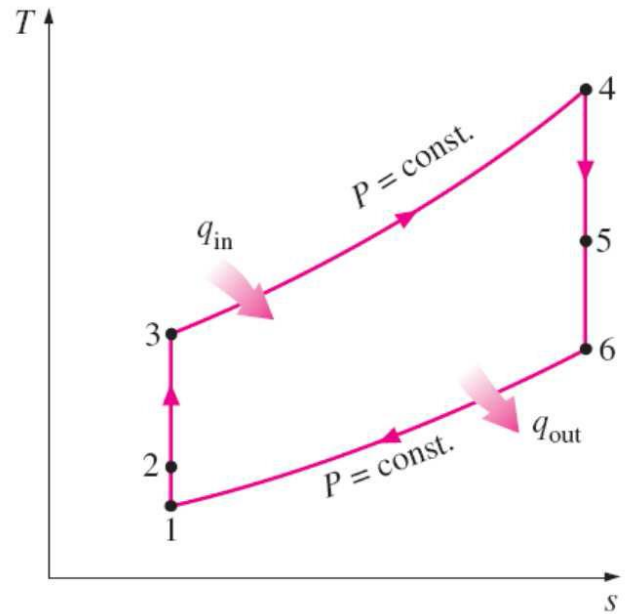
$$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$



Ideal jet propulsion cycle



what which of them works



In a jet engine, the "useful" power is the high speed jet flow from the engine;

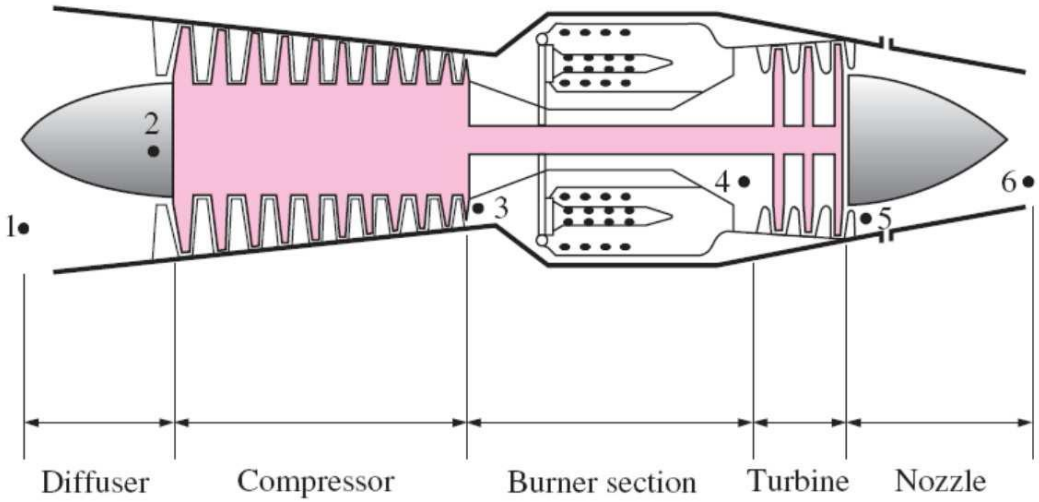
Intake (diffuser): $h_1 + \frac{c_1^2}{2} = h_2 + \frac{c_2^2}{2}$

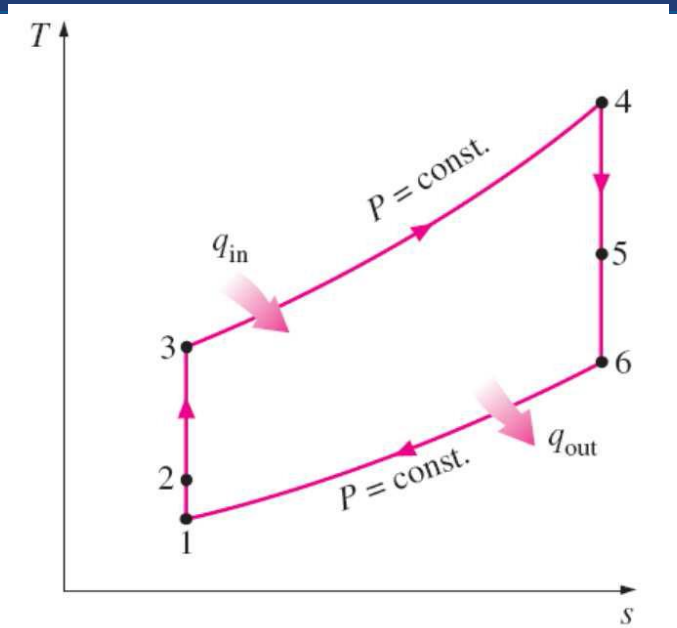
Exit (nozzle): $h_5 + \frac{c_5^2}{2} = h_6 + \frac{c_6^2}{2}$

And $w_{23} = w_{45}$ (compressor work=turbine work)$= \Delta h = c_p \Delta T$

Therefore $T_3 - T_2 = T_4 - T_5$

Jet engine cycle





Thrust $T = \dot{m}(c_6 - c_1)$ note this is thrust magnitude and ignores fuel flow

Assuming pressure $p_1 = p_6 = 0$ (gauge)

Propulsion power $\dot{W}_p = Tc_1$ where c_1 is aircraft velocity

Propulsion efficiency $\eta_p = \dot{W}_p / \dot{Q}_{in}$

All other aspect the same as Brayton-Joule cycle

Example

Air enters the compressor of an ideal air-standard Brayton cycle at 100 kPa, 300 K, with a volumetric flow rate of 5 m^3/s. The compressor pressure ratio is 10. The turbine inlet temperature is 1400 K.

Determine …

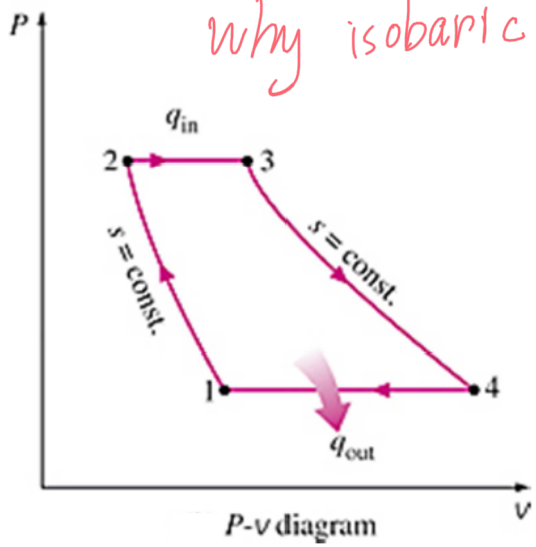
(a) the thermal efficiency of the cycle,

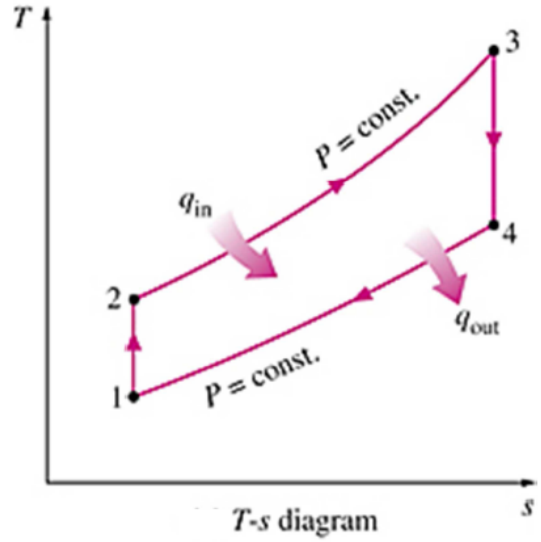
(b) the back work ratio,

(c) the net power developed, in kW.

Solution

why isobaric





Known: $p_1 = 100\ kPa,\ \ T_1 = 300\ K,\ \ \dot{V}_1 = 5\,\frac{m^3}{s},\ \ \ \frac{p_2}{p_1} = 10,\ \ \ T_3 = 1400\ K,$

$R = 287 J/kgK, \gamma = 1.4$

1→ 2 isentropic compression

ideal gas

$pv^{\gamma} = c$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \Longrightarrow T_2 = 579.2\ K$$

Solution

3→4 isentropic expansion

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \Longrightarrow T_4 = 725.1\ K$$

$$q_{in} = c_p(T_3 - T_2), \qquad q_{out} = c_p(T_4 - T_1)$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 48.2\%$$

$$w_{in} = w_{12} = c_p(T_2 - T_1)$$

$$w_{out} = w_{34} = c_p(T_3 - T_4)$$

Back work ratio= $\frac{T_2 - T_1}{T_3 - T_4} = 41.37\%$

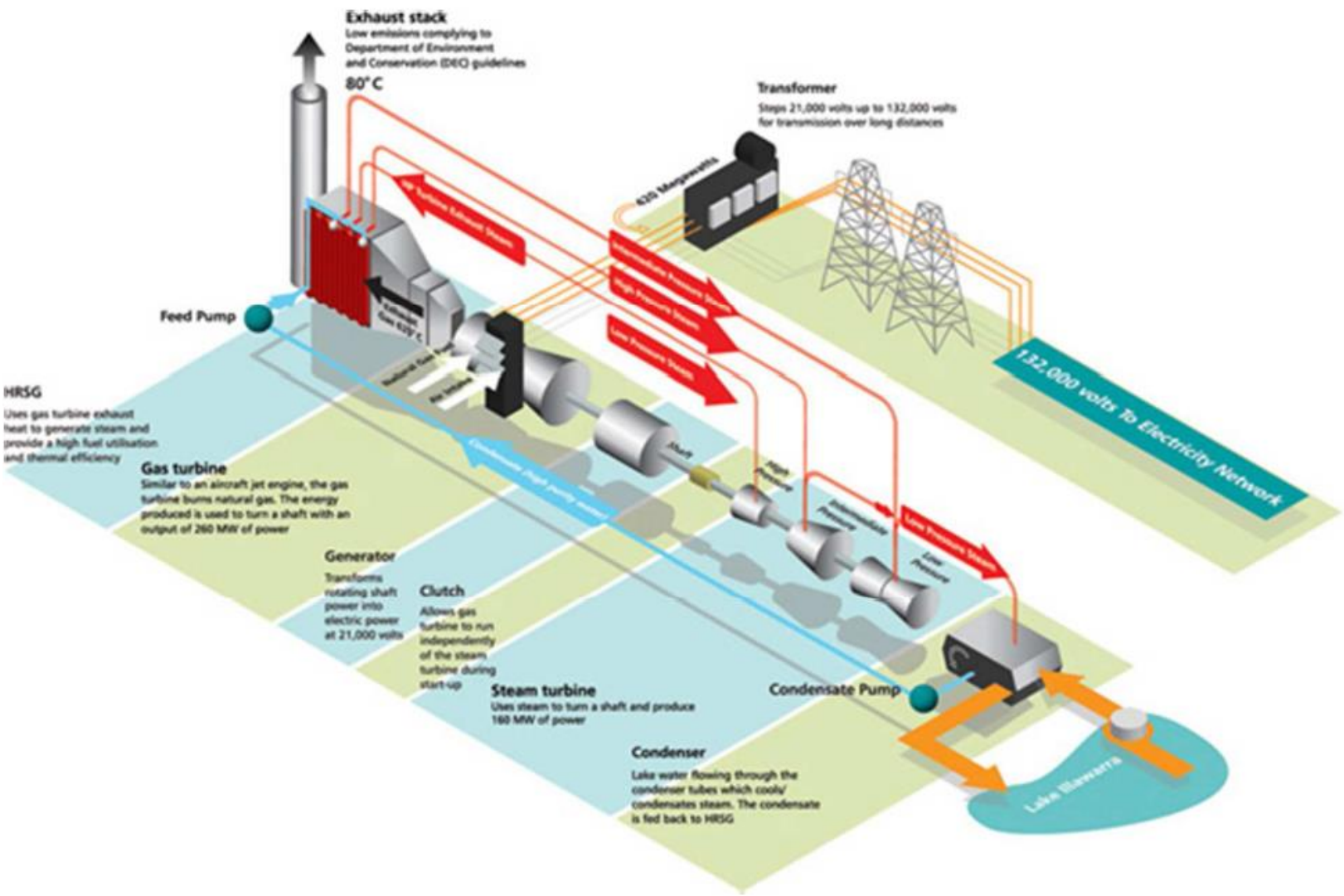
$\frac{c_p}{c_v} = \gamma$ $\quad c_p - c_v = R$

$$c_p = \frac{R\gamma}{\gamma - 1} = 1005\ J/kgK$$

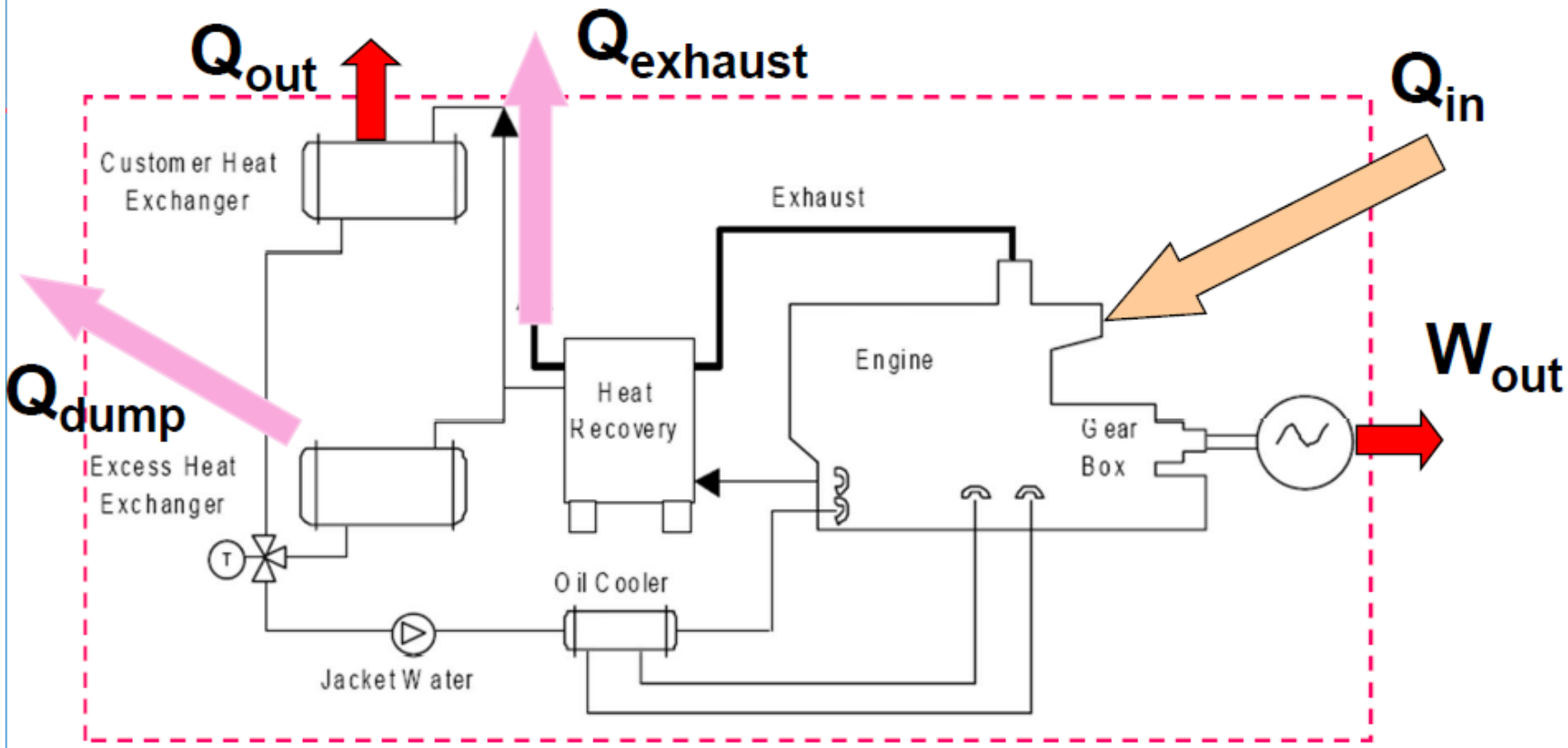
$$\dot{m} = \frac{p_1 \dot{V}_1}{RT_1} = 5.807 kg/s$$

$$\dot{W} = \dot{m}\eta_{th} q_{in} = \dot{m}\eta_{th} c_p(T_3 - T_2) = 2.308 \times 10^6 W$$

Combined heat and power (CHP)



CHP (Cogeneration)



$$\eta = \frac{W_{out} + Q_{out}}{Q_{in}} \approx 70 - 80\%$$

Q: what is the view of exergy analysis?

The Carnot vapour cycle

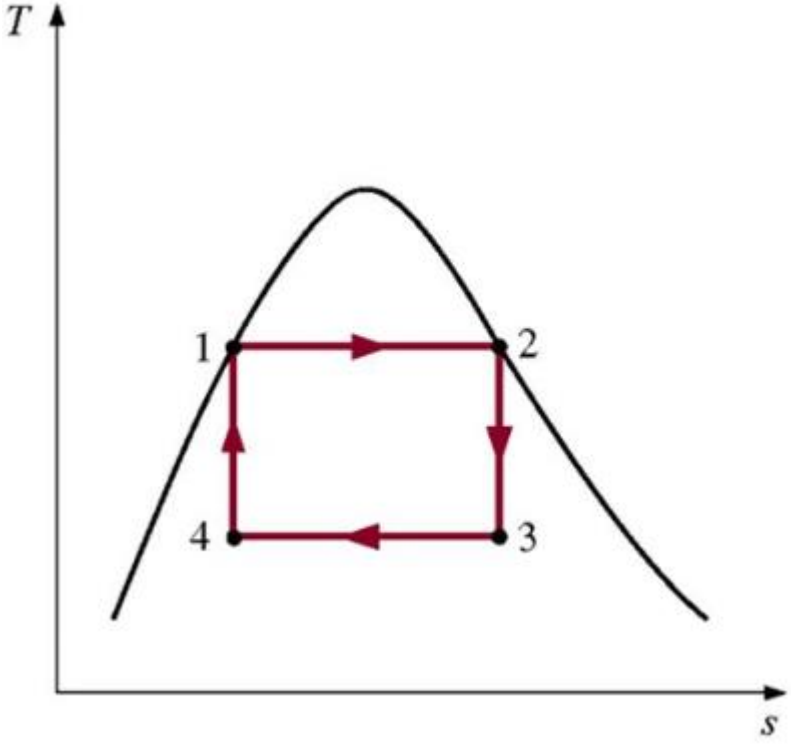
Consider a steady-flow Carnot cycle within the saturation dome of steam, and this allow the fluid to be evaporated and condensed in an isothermal and also isentropic way (during phase change) Constant T ⇒ Constant P

1→2 isothermal/isobaric boiling

2→3 reversible adiabatic (isentropic) expansion

3→4 isothermal/isobaric condensation

4→1 isentropic compression



This is an ideal cycle not implemented but give a fundamental idea leads to the next cycle- Rankine cycle

Rankine cycle (steam turbine cycle)

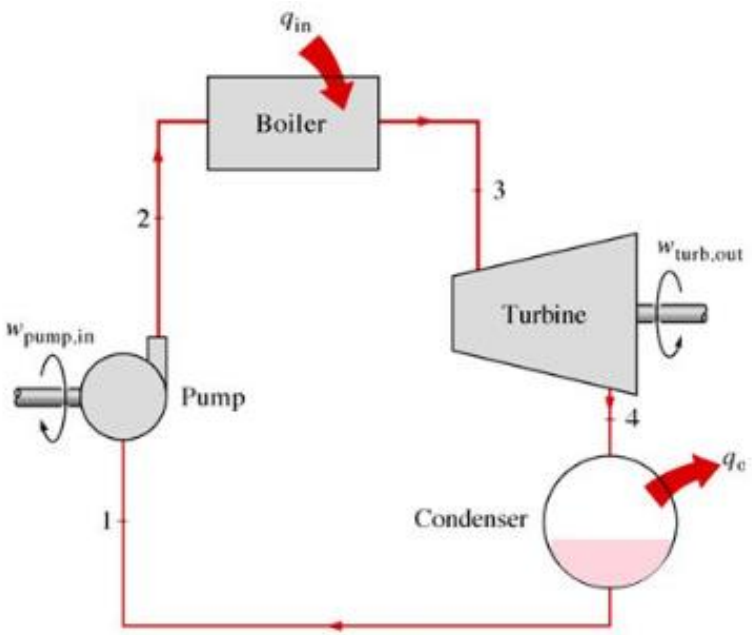
Rankine cycle is widely used as a basic cycle of vapour power plant. The processes are:

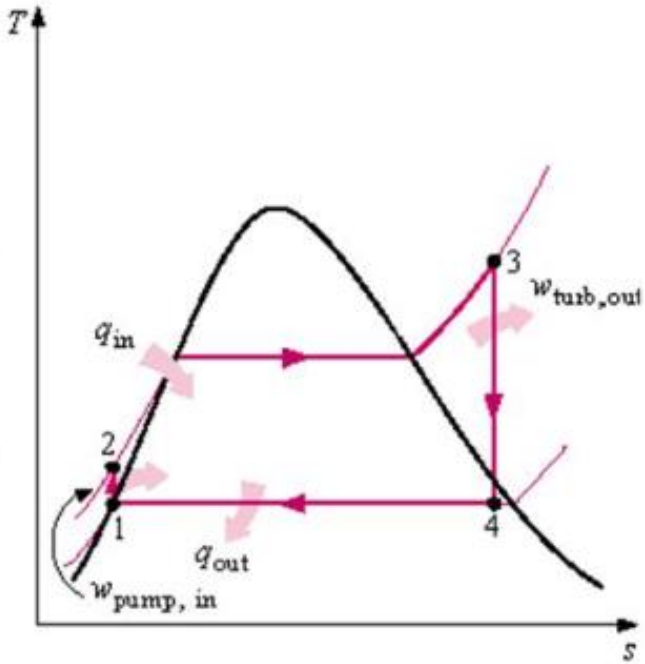
1→2 Isentropic compression in a pump

2→3 Constant pressure heat addition in a boiler/superheater

3→4 Isentropic expansion in a turbine

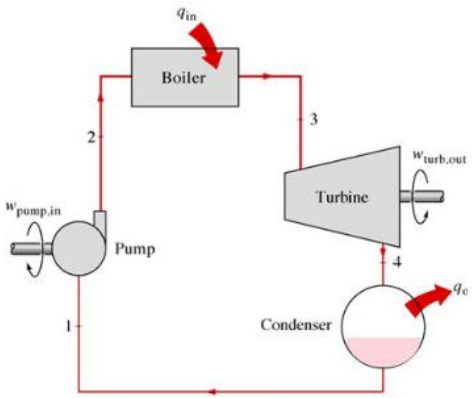
4→1 Constant pressure heat rejection in a condenser

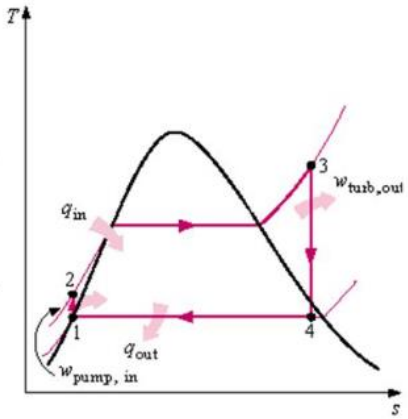




Rankine cycle analysis

This involves using the steam table





1→2 $\quad q = 0, w_{pump} = h_2 - h_1$

As liquid is incompressible

$(v_1 = v_2 = v_{f@p_1})$ $\neq v_{f@p_2}$

$w_{pump} = v_1(p_2 - p_1)$

2→3 isobaric process thus $w_{23} = 0$

$$q_{in} = h_3 - h_2$$

3→4 turbine $\quad q_{34} = 0, \quad w_{turbine} = h_3 - h_4$

4→1 condenser $w_{41} = 0, \quad q_{out} = h_4 - h_1$

Thermal efficiency $\quad \eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$

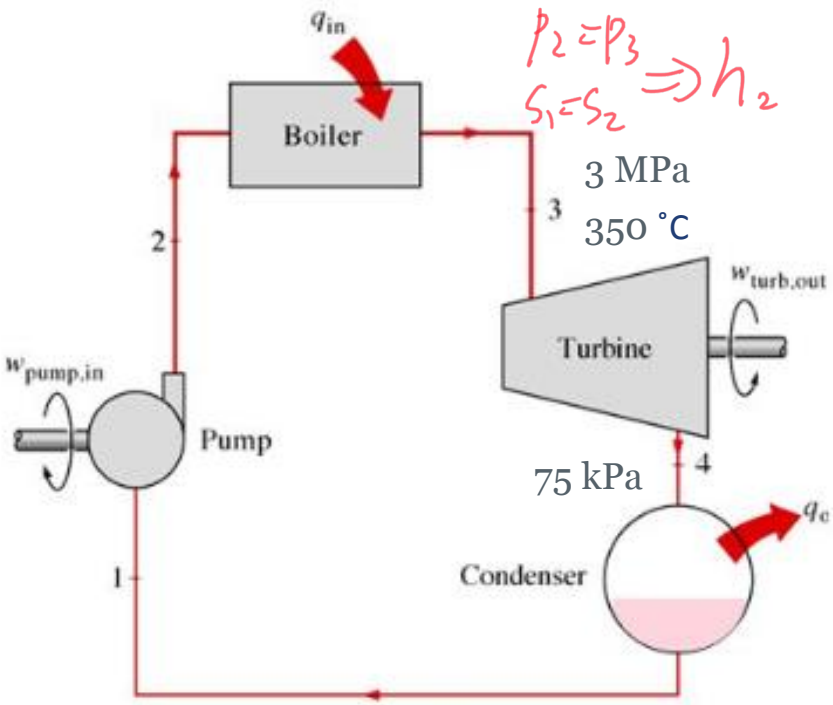
Work output $\quad w_{net} = q_{in} - q_{out} = w_{turbine} - w_{pump}$

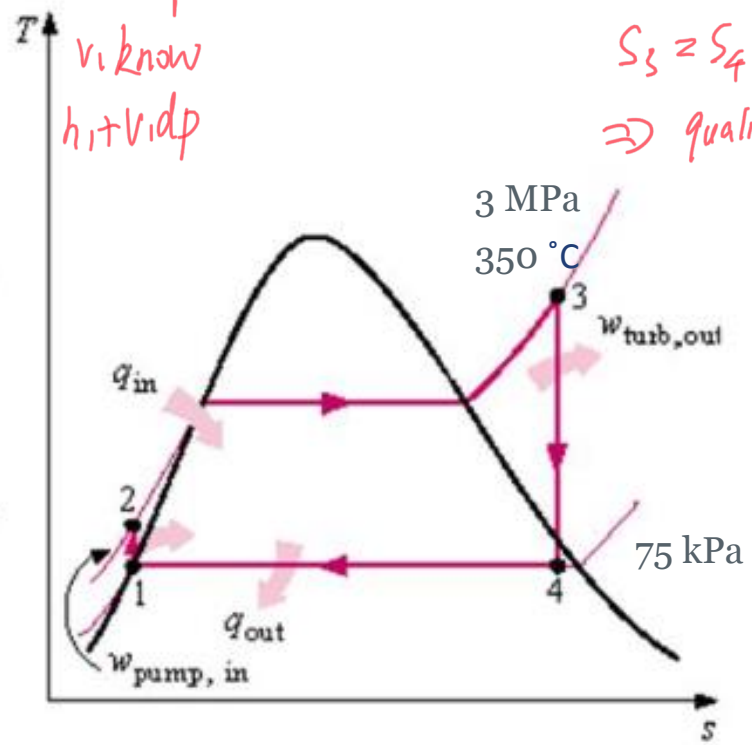
Example

Consider a steam power plant operating on the simple ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350 ˚C and is condensed in the condenser at a pressure of 75 kPa. Determine the thermal efficiency of the cycle.

1-2
with subcool water table
or
$v\,dp$
$P_2 = P_3$
$S_1 = S_2 \Rightarrow h_2$
v_1 know
$h_1 + v_1 dp$

3-4
$S_3 = S_4$
$\Rightarrow$ quality out 4





Solution

@ state 1:

$p_1 = 75\ kPa$, saturated liquid

$h_1 = h_{f@0.75bar} = 384\ kJ/kg$

$v_1 = v_{f@0.75bar} \approx 0.001\ m^3/kg$

@state 1:

$p_2 = 30\ bar$, subcooled liquid

$w_{pump} = v_1 \Delta p = 0.001(30 - 0.75)10^5 \approx 3\ \mathrm{kJ/kg}$

$h_2 = h_1 + w_{pump} = 387\ kJ/kg$

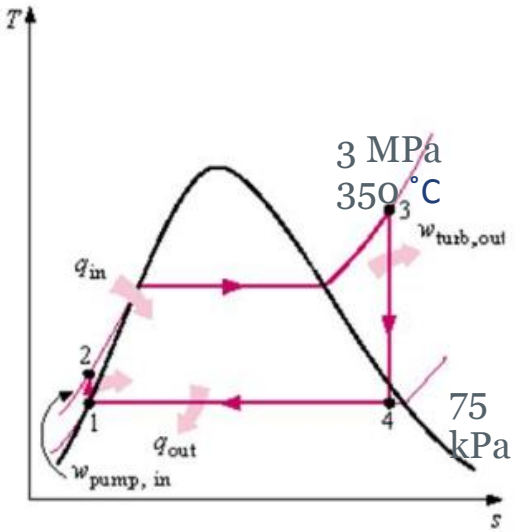
@state 3:

$p_3 = 30\ bar \quad T_3 = 350°\mathrm{C}$ superheated steam

$h_3 = 3117\ kJ/kg \quad s_3 = 6.744\ kJ/kgK$

@state 4:

$p_4 = 0.75\ bar, s_4 = s_3 = 6.744\ kJ/kgK$, wet steam



$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.744 - 1.213}{6.243} = 0.886$$

$$h_4 = h_{f@0.75bar}(1 - x_4) + h_{g@0.75bar}x_4 = 2403\ kJ/kg$$

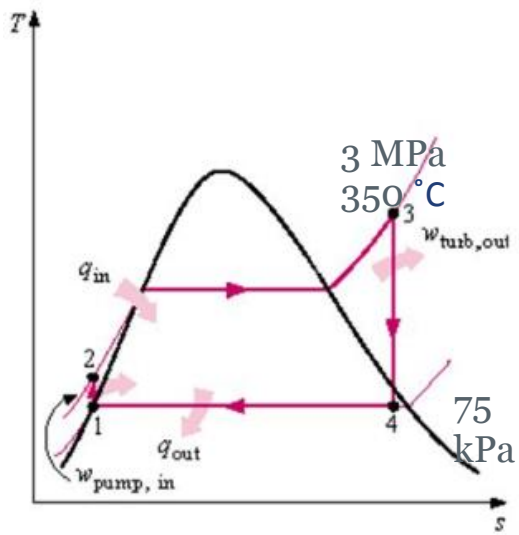
Solution

Thus

$$q_{in} = h_3 - h_2 = 2728.6\ kJ/kg$$
$$q_{out} = h_4 - h_1 = 2018.6\ kJ/kg$$
$$\eta_{th} = 1 - \frac{h_{out}}{h_{in}} = 0.26 = 26\%$$

The net work output

$$w_{net} = w_{turbine} - w_{pump} = h_3 - h_4 - w_{pump} = 714 - 3 = 711\ kJ/kg$$



Extended reading

- Actual steam power cycle
  - Pressure drop across boiler/superheater and condenser
  - Irreversibility (entropy generation/exergy destruction) across pump/turbine
- Parameters that affects the Rankine cycle efficiency
  - Condenser pressure/temperature
  - Steam superheated temperature
  - Boiler pressure
- Reheat Rankine cycle

pump → boiler → turbine → condenser

Read:

9-11 to 9-13 in the book

https://en.wikipedia.org/wiki/Rankine_cycle

Refrigeration and heat pump cycle

Unlike power cycles with a purpose of generating power, refrigerators and heat pumps serve a purpose of transferring heat from low temperature objects to high temperature objects.

These processes will not happen spontaneously!

- For refrigerators, the task is to remove heat from low temperature objects
- For heat pumps, the task is to add heat to the high temperature objects

Define

effective work

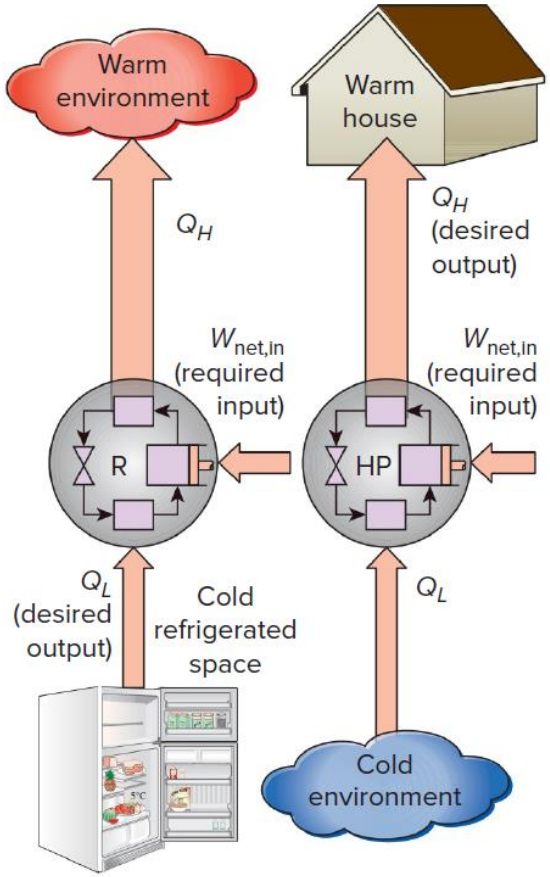
Coefficient of Refr. $\boldsymbol{COR} = \dfrac{\boldsymbol{Cooling\ effect}}{\boldsymbol{work\ input}} = \dfrac{\boldsymbol{Q_L}}{\boldsymbol{W_{net,in}}}$

Coefficient of Perf. $\boldsymbol{COP} = \dfrac{\boldsymbol{Heating\ effect}}{\boldsymbol{work\ input}} = \dfrac{\boldsymbol{Q_H}}{\boldsymbol{W_{net,in}}}$

From the 1st Law: $Q_H = Q_L + W_{net,in}$

Therefore: $\boldsymbol{COP} = \boldsymbol{COR} + \boldsymbol{1}$

Note: sometimes COP is used for refrigeration as well.



(a) Refrigerator (b) Heat pump

The reversed Carnot cycle

The Carnot power cycle introduced before is completely reversible.

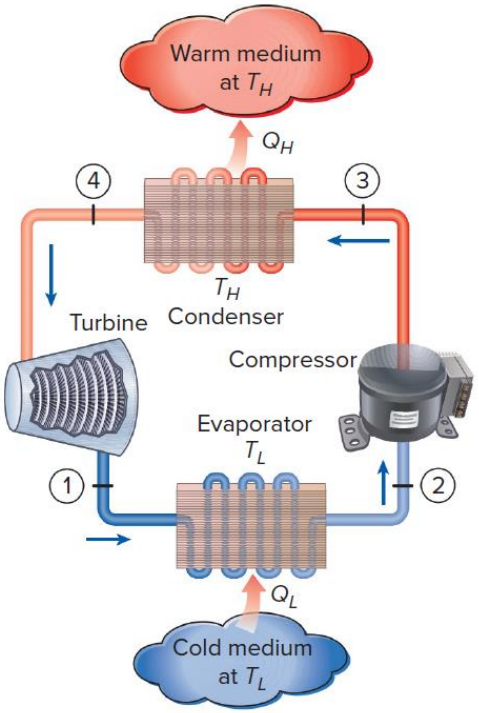
The cycle on the right is a reversed Carnot cycle that serves the purpose of removing heat from low temperature objects and rejection heat into high temperature objects (refrigerator or heat pump).

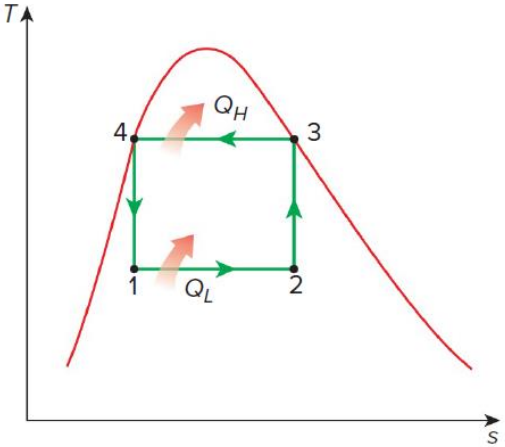
It is called Carnot refrigerator of Carnot heat pump.

$$COR_{Carnot} = \frac{1}{\frac{T_H}{T_L} - 1}$$

Or

$$COP_{Carnot} = \frac{1}{1 - \frac{T_L}{T_H}}$$





Example

Derive the Carnot COR and COP

$$COR = \frac{Q_L}{W_{net}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{\frac{Q_H}{Q_L} - 1}$$

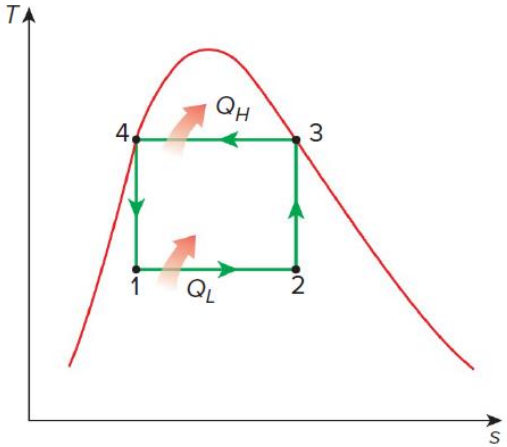
Remember for Carnot cycle $\frac{Q_H}{Q_L} = \frac{\mathrm{T_H}}{\mathrm{T_L}}$

Therefore

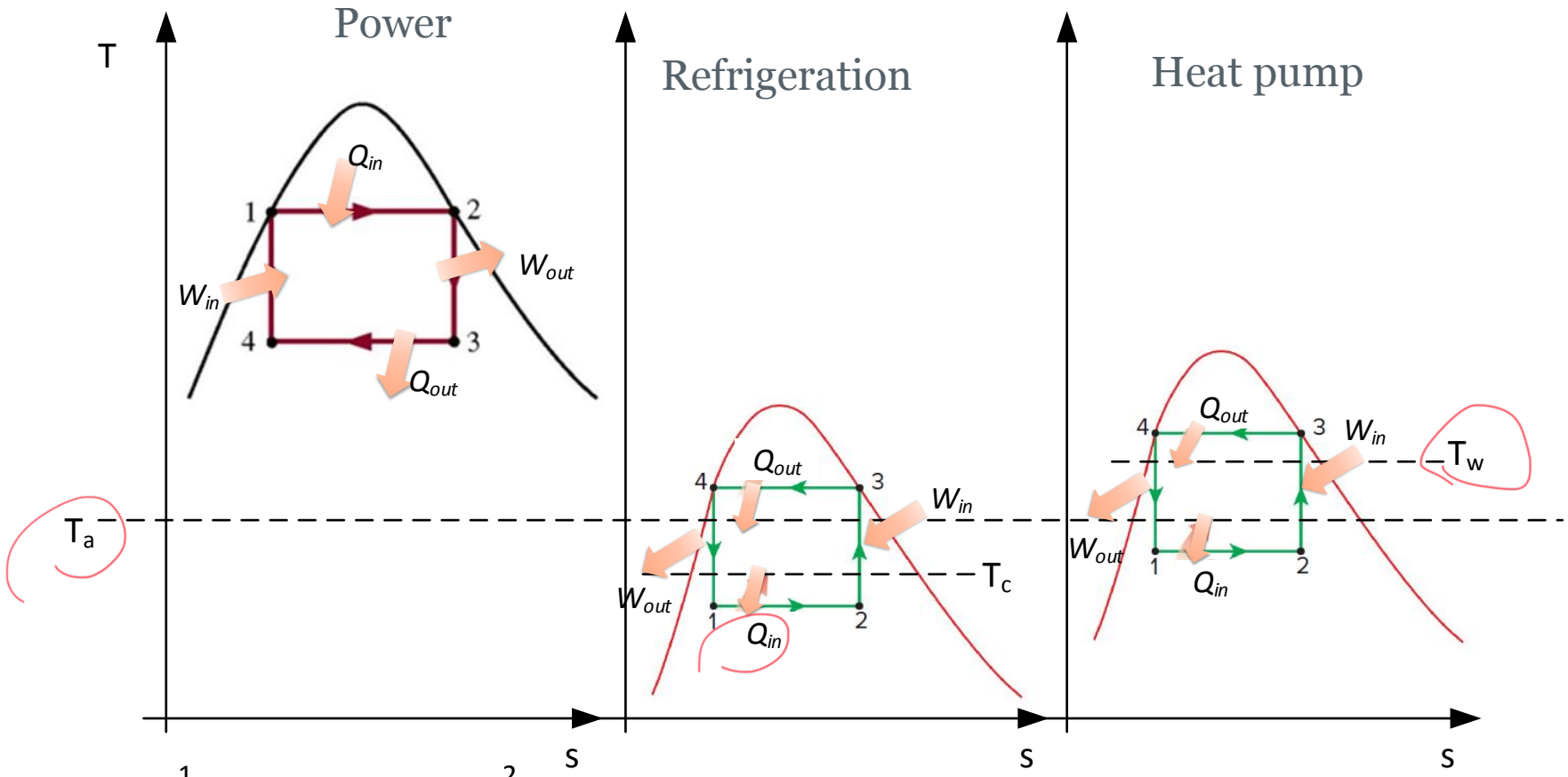
$$COR = \frac{1}{\frac{Q_H}{Q_L} - 1} = \frac{1}{\frac{\mathrm{T_H}}{\mathrm{T_L}} - 1}$$

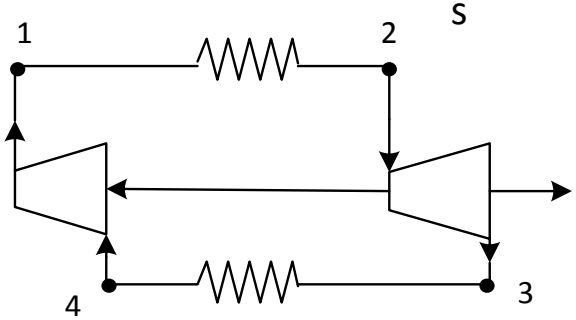
Exactly the same way

$$COP = \frac{1}{1 - \frac{Q_L}{Q_H}} = \frac{1}{1 - \frac{T_L}{T_H}}$$

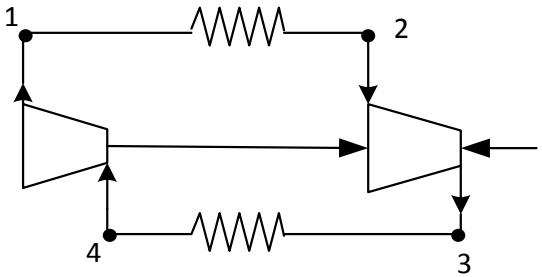


Carnot power cycle, refrigeration and heat pump cycles





Power cycle



Refrigeration and heat pump cycle

The ideal vapour-compression refrigeration cycle

The ideal reversed Carnot cycle is impractical to implement.

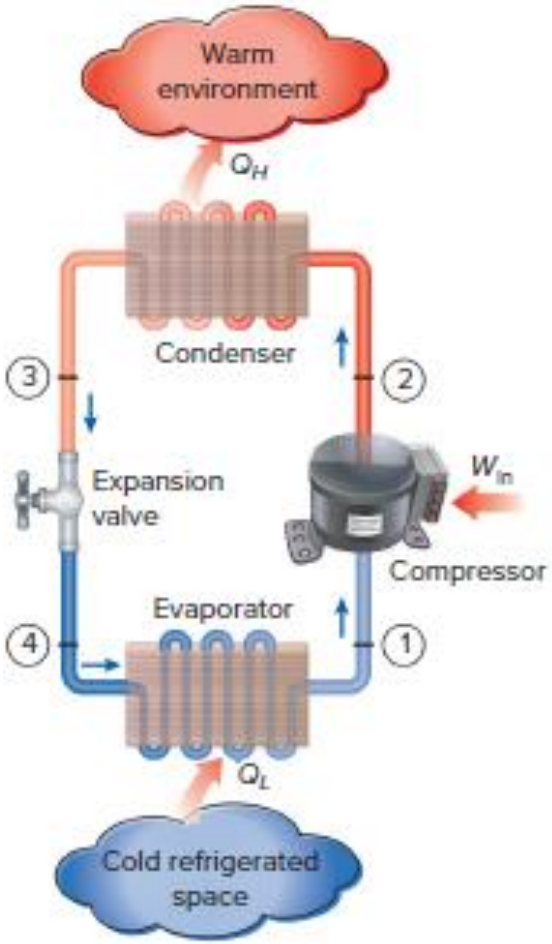
Ideal vapour-compression refrigeration cycle is a replacement.

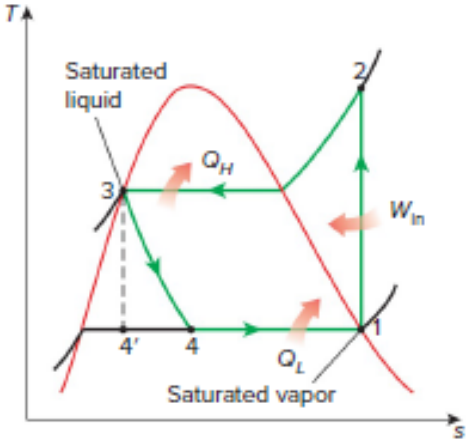
1→2 Isentropic compression in a compressor (w_{in})

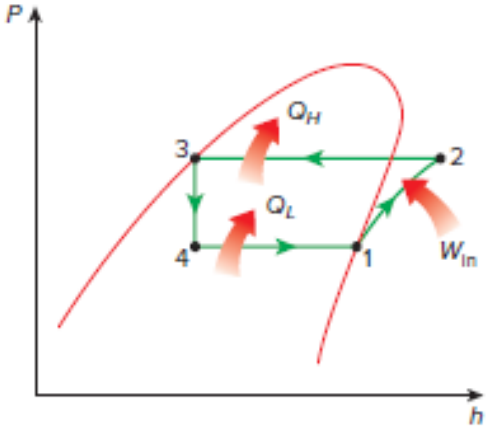
2→3 Constant-pressure heat rejection in a condenser (q_{out})

3→4 Throttling in an expansion device ($h_3=h_4$)

4→1 Constant-pressure heat absorption in an evaporator (q_{in})



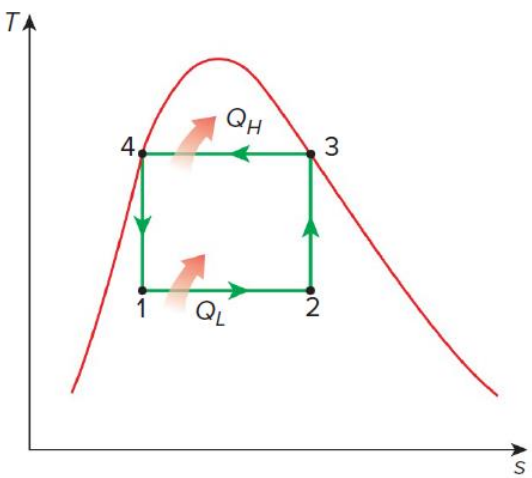


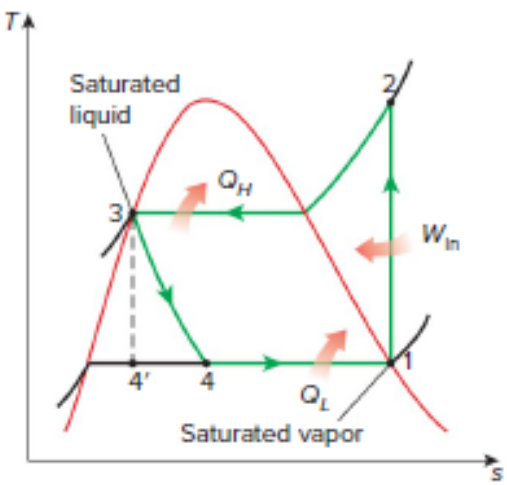


Major difference

Between the ideal vapour compression refrigeration cycle and the reversed Carnot cycle

1. 1→2 compression begins from saturated vapour, whereas in the reversed Carnot cycle, it ends at saturated vapour
2. 2→3 for i.v.c.r cycle, condensing starts from superheated vapour, but they are both isobaric
3. 3→4 instead of having a turbine, i.v.c.r cycle has a throttle (isenthalpic process, enthalpy constant) as the work out from this process is limited
4. 4→1 both isobaric heat addition but i.v.c.r ends at saturated vapour, which makes it much easier to control





Ideal vapour compression refrigeration cycle

Quantitative analysis

$$w_{in} = h_2 - h_1 \qquad (1 \rightarrow 2)$$

$$q_{out} = h_2 - h_3 \qquad (2 \rightarrow 3)$$

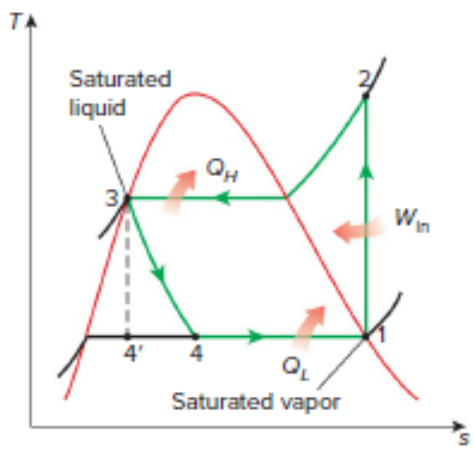
$$q_{in} = h_1 - h_4 \qquad (4 \rightarrow 1)$$

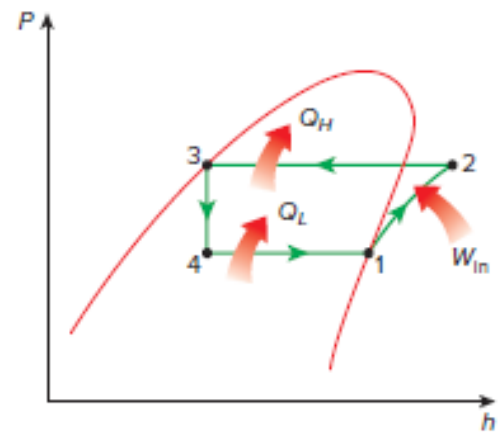
$$w_{out} = h_4 - h_3 = 0 \qquad (3 \rightarrow 4)$$

Therefore

$$COR = \frac{q_L}{w_{in}} = \frac{q_{in}}{w_{in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

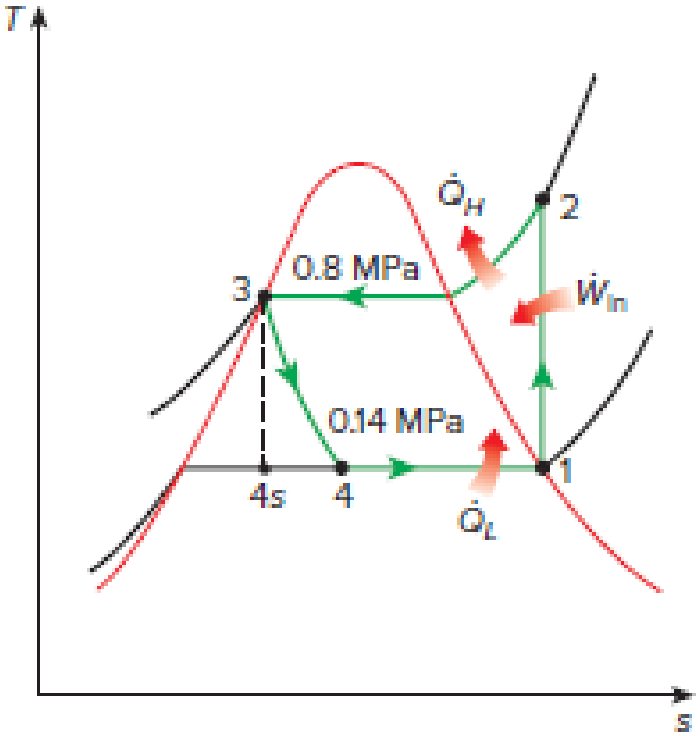
$$COP = \frac{q_H}{w_{in}} = \frac{q_{out}}{w_{in}} = \frac{h_2 - h_3}{h_2 - h_1}$$





Example

A refrigerator uses refrigerant R134a as the working fluid and operates on an ideal vapour-compression refrigeration cycle between 0.14 and 0.8 MPa. If the mass flow rate of the refrigerant is 0.05 kg/s, determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the rate of heat rejection to the environment, and (*c*) the COP of the refrigerator.



Solution

Need to use property tables

@state 1 $p_1 = 0.14\ MPa, x_1 = 1 \rightarrow h_1 = 239.2\ kJ/kg\ ,\ \ s_1 = 0.94467\ kJ/kgK$

@state 2 $p_2 = 0.8\ MPa, s_2 = s_1 \rightarrow h_2 = 275.4\ kJ/kg$

@state 3 $p_3 = p_2 = 0.8\ MPa, x_3 = 0 \rightarrow h_3 = 95.48\ kJ/kg$

@state 4 $p_4 = p_1 = 0.14\ MPa, \mathrm{h}_4 \approx h_3 = 95.48\ kJ/kg$

$$q_{out} = q_H = h_2 - h_3 = 179.9\ kJ/kg$$
$$\dot{Q}_{out} = \dot{m} q_{out} = 9\ kW$$
$$\dot{Q}_{in} = \dot{m} q_{in} = \dot{m}(h_1 - h_4) = 7.19\ kW$$
$$\dot{W}_{in} = \dot{m} w_{in} = \dot{m}(h_2 - h_1) = 1.91\ kW$$
$$COR = \frac{\dot{Q}_{in}}{\dot{W}_{in}} = 3.97$$
$$COP = \frac{\dot{Q}_{out}}{\dot{W}_{in}} = 4.97 = COR + 1$$

Summary

- Basic considerations in power cycle analysis (understanding)
- Internal combustion engine cycles (understanding and calculation)
    - Otto cycle
    - Diesel cycle
    - Dual cycle
- Gas turbine cycle -- Brayton-Joule cycle and actual gas turbine cycle (understanding and calculation)
- Combined heat and power (CHP) (understanding)
- Carnot vapour cycle and steam turbine cycle -- Rankine cycle (understanding and calculation)
- Refrigeration and heat pump cycle (understanding and calculation)
    - Reversed Carnot cycle
    - Ideal vapour-compression refrigeration cycle