Outline

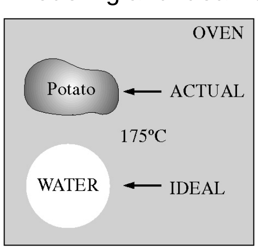
- Thermodynamic cycle analysis - intro
- Vapor power cycles
- Carnot vs. Rankine
- Ideal vs. actual

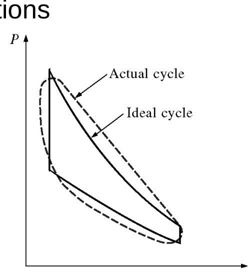
Thermodynamic cycle analysis

- Important application areas for thermodynamics include:
  - Power generation – power cycles; engines
  - Refrigeration – refrigeration cycles; refrigerators, air conditioners, and heat pumps
- Gas vs. vapor cycles (phase of working fluid)
- Closed vs. open cycles
- Heat engines: internal vs. external combustion engines (how heat is supplied to working fluid)
- We will typically seek an ideal cycle for which we can model and use to compare actual cycle performance
- See link on website for cycle animations

Idealized cycles

- Modeling and idealizations





Carnot vs. ideal cycles

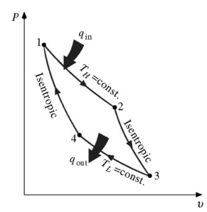
- Why not use Carnot cycle as the ideal cycle?
- Ideal cycles will be internally reversible but not externally reversible (heat transfer through finite temperature difference)
- We will: neglect friction, only consider quasi-equilibrium compression/expansion, neglect extraneous heat transfer (also KE/PE changes will be neglected except for nozzles/diffusers)
- Recall area enclosed by cycles on P-v and T-s diagrams represent net work and heat for internally reversible cycle

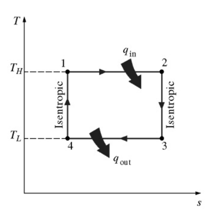
Heat engines - review

- What is the purpose of a heat engine?
- How does it operate?
- What is the thermal efficiency of a heat engine?
- What is the Carnot cycle?
- Gives maximum efficiency
- 4 reversible processes

$$\eta_{th,rev} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$= 1 - \frac{T_L(s_3 - s_4)}{T_H(s_2 - s_1)} = 1 - \frac{T_L}{T_H}$$



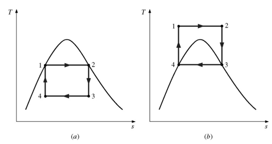


Vapor power cycles

- Vapor power cycle – working fluid is alternatively vaporized and condensed
- Steam is most common working fluid (low cost, availability, high enthalpy of vaporization)
- Heat source either coal, nuclear, or NG
- Steam goes through same cycle for all

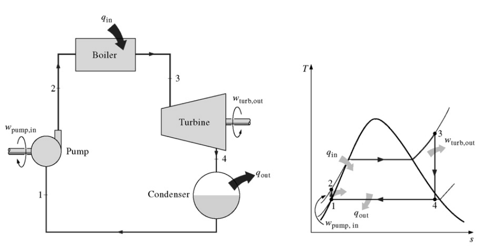
Carnot cycle as ideal vapor power cycle?

- Carnot most effiiency between two T limits
- Under wet dome 1-2, 3-4 feasible in boiler and condenser
- T<Tcr; limits efficiency
- Isentropic expansion in turbine but high moisture content is concern
- Isentropic compression involves two-phase to SL
- Two-phase compressor hard; hard to stop at SL
- Other options also difficult (high pressure compression and isothermal heat transfer in single-phase



Ideal Rankine cycle

- Superheat steam in boiler; condense completely in condenser



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Ideal Rankine cycle processes

- Internally reversible; four processes
- 1-2: Isentropic compression in a pump
- 2-3: Constant pressure heat addition in a boiler
- 3-4: Isentropic expansion in a turbine
- 4-1: Constant pressure heat rejection in a condenser
- Water enters pump as SL at 1; enters boiler as CL at 2; leaves as SHV at 3; high x SLVM enters condenser

Ideal Ranking cycle analysis

- CV around each component
- Work and heat magnitudes
- Can you derive energy balances?
- Recall working fluid is steam, so do not use ideal gas relations!!!

$q - w = \Delta h \ (kJ/kg)$ for each process

$(q_{in} - q_{out}) - (w_{out} - w_{in}) = h_{inlet} - h_{outlet}$

$Pump(q=0): w_{pump,in} = h_2 - h_1 = v(P_2 - P_1)$

$h_1 = h_{f@P_1}; v \cong v_1 = v_{f@P_1}$

$Boiler(w=0): q_{in} = h_3 - h_2$

$Turbine(q=0): w_{turb,out} = h_3 - h_4$

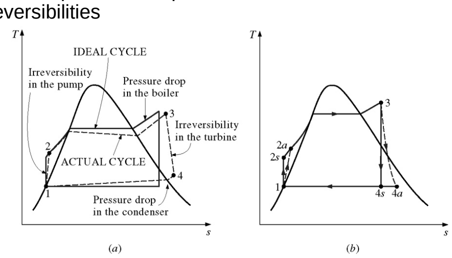
$Condenser(w=0): q_{out} = h_4 - h_1$

$\therefore \eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$

$w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$

Deviation of actual vapor power cycle from idealized ones

- Frictional pressure drop and heat loss introduce irreversibilities



- Account for via isentropic efficiencies in pump and turbine

Examples

- To be done in class

Outline

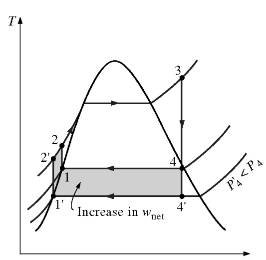
- Increasing Rankine cycle efficiency
- Examples

Increasing Rankine cycle efficiency

- Average temperature should be as high as possible during heat addition and as low as possible during heat rejection – Carnot theory
- How to achieve this?
  - Lowering condenser pressure
  - Superheating steam to high temperatures
  - Increasing boiler pressure
- Keep in mind area enclosed by cycle on T-s or P-v diagram represents net work/heat for internally reversible cycles

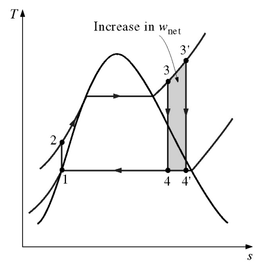
Lowering condenser pressure

- This lowers steam temperature during heat rejection
- More work input to pump and heat input, but more work output from turbine
- Overall efficiency increases due to lowering temperature during heat rejection
- Limited by saturation pressure for temperature of cooling medium; watch for air leakage; higher moisture content in turbine



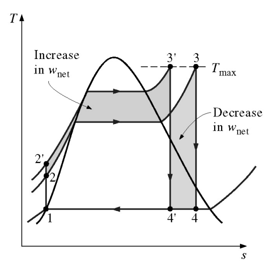
Superheating steam to high temperatures

- Increase in net work and heat input
- Overall effect is increase in efficiency due to higher T during heat addition
- Helps by decreasing moisture content in turbine
- Limited by turbine blade material (620C or 1150F); ceramics



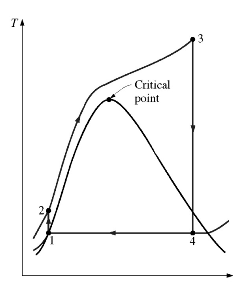
Increasing boiler pressure

- Increases average T during heat addition
- Consider fixed turbine inlet temperature
- Moisture content increases (reheat)
- 2.7MPa (400psia) in 1922 to over 30MPa (4500psia) today producing >1000MW

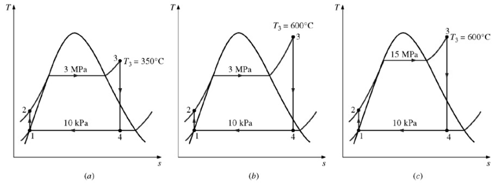


Supercritical power plant

- P>22.09MPa leads to 40/34% efficiencies for fossil/nuclear power plants
- U.S. has 112 nuclear power plants producing 21% nation's electricity
- France – 75% nuclear

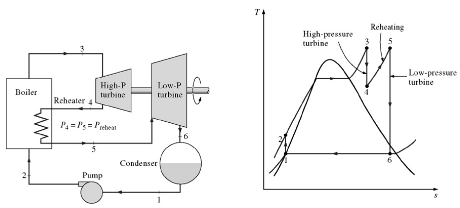


Example



Ideal reheat cycle

- How can we take advantage of increased efficiencies at higher boiler pressures without facing problem of excessive moisture at final stages of turbine?
  - Superheat steam to very high temperatures (metal limits)
  - Expand steam in stages with reheat in-between



Example

- To be done in class

Outline

- Gas power cycles
- Air standard cycles
- Otto cycle
- Diesel cycle
- Dual cycle
- Brayton cycle

Gas power cycle modeling challenges and assumptions

- Working fluid composition in gas power cycles changes both phase and composition during internal combustion (IC) process
- But since liquid fuel evaporates rapidly and air as oxidizer is mostly nitrogen (which is mostly inert) modeling the working fluid as gaseous air (ideal gas) at all times is a good approximation
- Then combustion process can be modeled as heat addition process

More gas power cycle modeling challenges and assumptions

- Most IC engines involve intake and exhaust process and so operate, while device operates on a mechanical cycle working fluid does not undergo a thermodynamic cycle
- Replacing exhaust/intake processes by heat rejection process is common assumption
- Further assumptions designed to simplify the problem include modeling all processes as internally reversible

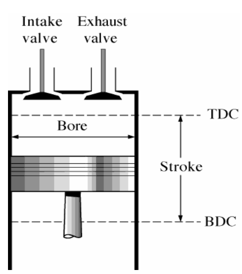
Air standard assumptions

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

If you further assume constant specific heats at room temperature then you have the so-called cold air standard assumptions (CASA)– good for quick analysis and trends

Overview of reciprocating engines

- Piston-cylinder device
- Piston reciprocates between TDC (smallest cylinder volume) and BDC (largest cylinder volume)
- Distance is stroke, s
- Piston diameter, bore, b
- Intake/exhaust valves



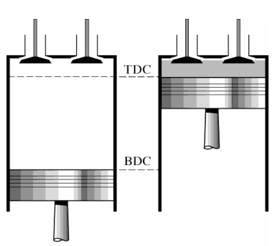
Engine volumes

- Displacement volume

$$DV = s\left(\frac{\pi d^4}{4}\right)$$

- Clearance volume, CV
- Compression ratio (volume ratio)

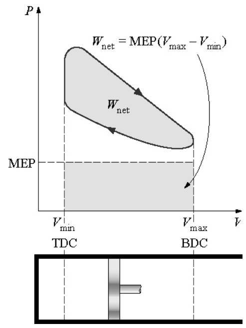
$$r \equiv \frac{DV + CV}{CV} = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$$



(*a*) Displacement volume (*b*) Clearance volume

Mean effective pressure (MEP)

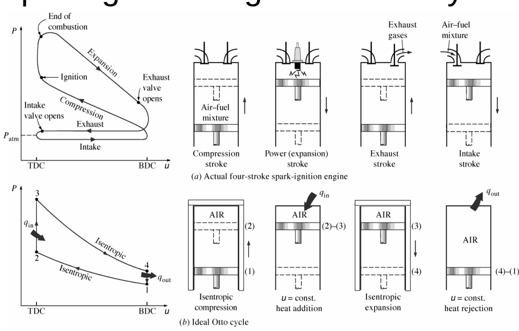
- A theoretical average pressure that if acted on the piston during the entire power stroke, would produce same amount of net work as during actual cycle



$$W_{net} = \text{MEP} \times \text{Piston area} \times \text{Stroke} = \text{MEP} \times \text{DV}$$

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

Actual and ideal spark ignition engines – Otto cycle



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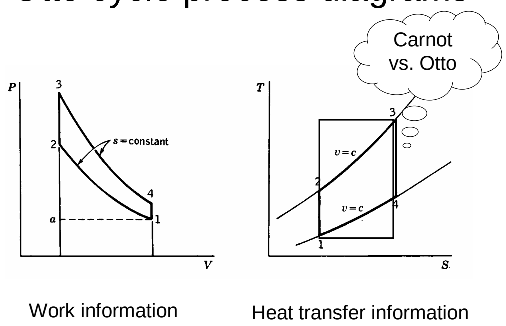
Otto cycle details

- Consists of four internally reversible processes
  - 1-2: Isentropic compression
  - 2-3: Constant-volume heat addition
  - 3-4: Isentropic expansion
  - 4-1: Constant-volume heat rejection
- P-v diagram on previous slide; T-s diagram in text (Fig. 8-15)

Otto cycle process diagrams



Otto cycle analysis

- Closed system

$q - w = \Delta u \ (kJ/kg)$ for each process

$(q_{in} - q_{out}) - (w_{out} - w_{in}) = \Delta u$

$q_{in} = u_3 - u_2 = C_v(T_3 - T_2)$

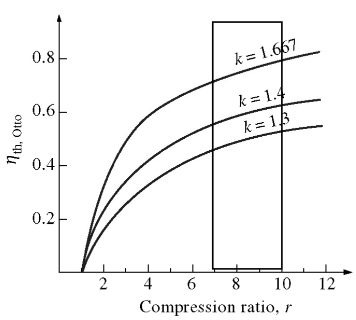
$q_{out} = u_4 - u_1 = C_v(T_4 - T_1)$

$$\therefore \eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{u_4 - u_1}{u_3 - u_2} \underset{CASA}{=} 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

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

$$\rightarrow \eta_{th,otto} = 1 - \frac{1}{r^{k-1}} \text{ with } r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$$

Thermal efficiency of Otto cycle



Examples

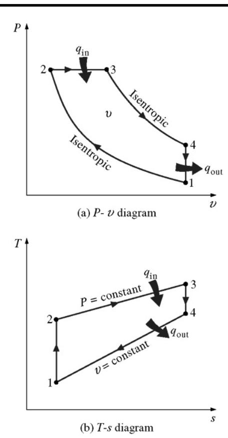
- To be done in class

Diesel cycle – ideal cycle for compression ignition engines

- SI engines
  - Air-fuel mixture compressed to below fuel autoignition temperature
  - This limits compression ration and efficiency to avoid knocking
  - Combustion initiated by a spark near TDC
  - Combustion process modeled as constant volume
- Diesel engines
  - Air compressed above fuels autoignition temperature
  - Fuel injection via liquid spray near TDC
  - Evaporates and ignites on contact with hot compressed air e.g. compression ignition
  - Combustion process modeled as constant pressure

Diesel cycle processes

- 1-2: Isentropic compression
- 2-3: Constant pressure heat addition
- 3-4: Isentropic expansion
- 4-1: Constant volume heat rejection



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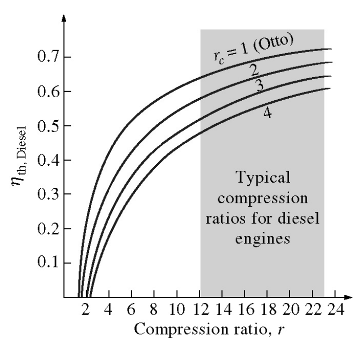
Diesel engine analysis

- Closed system

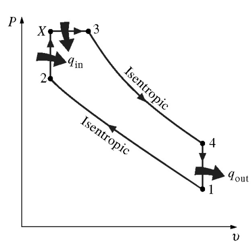
$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = C_P(T_3 - T_2)$$
$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1)$$
$$\eta_{th,diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$
$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2} \therefore \eta_{th,diesel} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right] \Rightarrow \eta_{th,otto} > \eta_{th,diesel}$$

Diesel cycle efficiency (k=1.4)

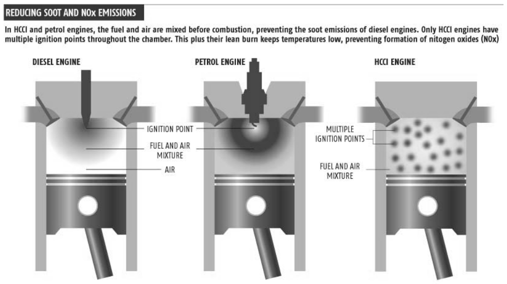


Dual cycle

- Combustion process in actual IC engines is neither constant volume nor constant pressure
- Better model is combined part constant volume/part constant pressure combustion process
- This results in dual cycle
- Adjust X to match actual cycle



New engines



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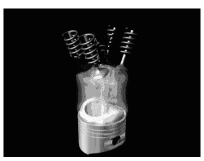
HCCI animation



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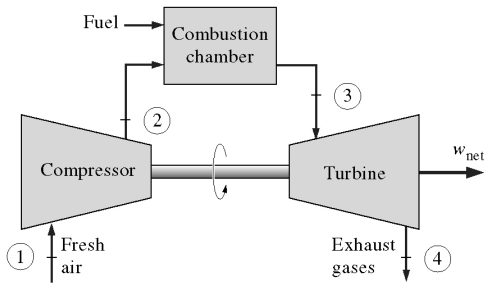
Problems

- To be done in class

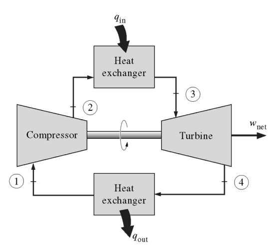
Brayton cycle – ideal cycle for gas turbine engines

- Actual open cycle http://energy.sdsu.edu/testcenter/



Brayton cycle

- Idealized closed cycle
- 1-2: Isentropic compression
- 2-3: Constant-pressure heat addition
- 3-4: Isentropic expansion
- 4-1: Constant-pressure expansion



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Brayton cycle analysis

- Steady flow CV for each component

$q - w = \Delta h \ (kJ/kg)$ for each process

$(q_{in} - q_{out}) - (w_{out} - w_{in}) = h_{inlet} - h_{outlet}$

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

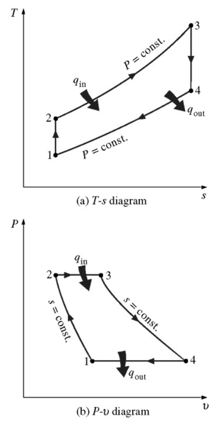
$q_{out} = h_4 - h_1 = C_P(T_4 - T_1)$

$$\therefore \eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_4 - h_1}{h_3 - h_2} \underset{CASA}{=} 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

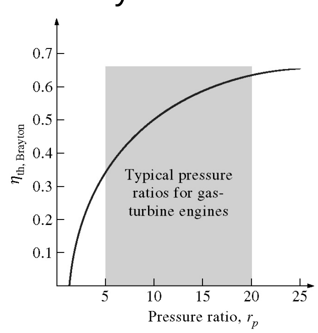
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)} = \left(\frac{P_3}{P_4}\right)^{\left(\frac{k-1}{k}\right)} = \frac{T_3}{T_4}$$

$$\rightarrow \eta_{th,brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \text{ with } r_p = \frac{P_2}{P_1} \text{ as pressure ratio}$$

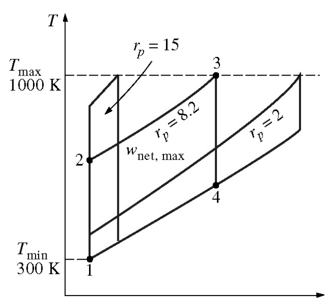
P-v and T-s diagrams



Thermal efficiency for Brayton cycle



Effect of pressure ratio on net work for fixed min/max temperatures



Problems

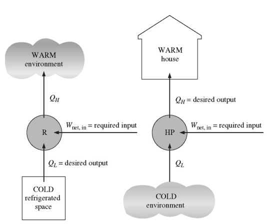
- To be done in class

Outline

- Refrigeration cycles
- Vapor compression refrigeration cycle

Objective of refrigerator/heat pump

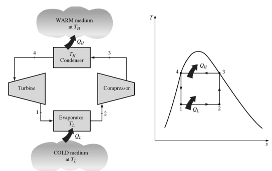
- Refrigerators
- Refrigerants
- Heat pump
- COP
- Cooling capacity is rate of heat removal from refrigerated space
- Measured in tons (1 ton (2000lbm) of liquid water at 0C into ice at 0C in 24h is 1 ton cooling capacity)
- 1 ton of refrigeration is 211kJ/min or 200Btu/min



(*a*) Refrigerator (*b*) Heat pump

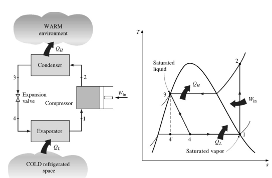
Carnot cycle as ideal cycle?

- Reversed Carnot cycle as ideal
- Gives max COP between T limits
- 1-2, 3-4 ok
- 2-3 hard to compress SLVM
- 4-1 expansion of high-moisture content refrigerant



Ideal vapor compression refrigeration (VCR) cycle

- Vaporize refrigerant completely before compression
- Replace turbine with throttling device (expansion valve or capillary tube)
- Gives ideal VCR cycle



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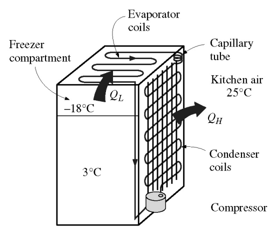
Ideal VCR cycle processes

- 1-2: Isentropic compression in compressor
- 2-3: Constant-pressure heat rejection in a condenser
- 3-4: Throttling in an expansion device
- 4-1: Constant-pressure heat absorption in evaporator
- WF enters compressor as SV at 1; exits at higher T than surrounding medium; enters condenser as SHV at 2 and leaves as SL at 3 due to heat rejection to surroundings, etc.

Household refrigerator

- Tubes in freezer compartment where heat is absorbed by refrigerant serve as evaporator
- Coils behind where heat is dissipated to surrounding air serve as condenser

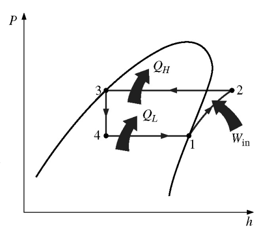


Property diagrams

- 3 of 4 processes are straight lines
- Why?
- Is ideal VCR cycle internally reversible?

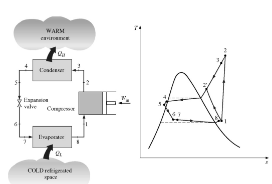
$$(q_{in} - q_{out}) - (w_{out} - w_{in}) = h_{inlet} - h_{outlet}$$

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$



Actual VCR cycle

- SH at compressor inlet
- Pressure drops in lines
- Entropy may increase or decrease in compressor due to heat loss
- Subcool liquid before entering throttling valve (increase cooling capacity)



Examples