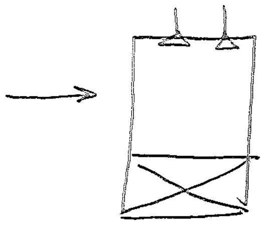
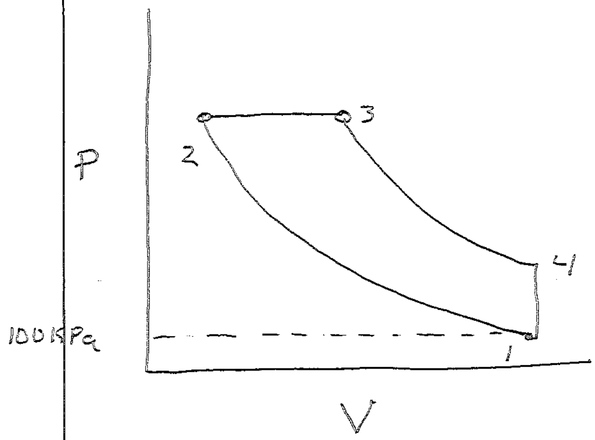


4
BDC

Open Exhaust $\longrightarrow$ 0

Return to TDC and push exhaust out

For analysis, assume piston expands during fuel injection such that P stays constant.



Compression ratio $= \dfrac{V_1}{V_2}$

Cut-Off ratio $= \dfrac{V_3}{V_2}$

(volume ratio between TDC and point where fuel shuts off).

q_{in}: $2 \to 3$ $\qquad q_{in} = \begin{Bmatrix} u_3 - u_2 \\ h_3 - h_2 \end{Bmatrix}$?

q_{out}: $4 \to 1$

<u>Why?</u>

$$Q - W = \underbrace{\sum m_e h_e - \sum m_i h_i}_{=0} + \left[m_2 u_2 - m_1 u_1\right]_{cv}$$

$$m_1 = m_2 = m$$

$q - w = u_2 - u_1$ but $w \neq 0$, $\quad w = \int_2^3 P dV = P_2 (v_3 - v_2)$

$$\Rightarrow q_{in} - P_2(v_3 - v_2) = u_2 - u_1 \quad \Rightarrow q_{in} = \underbrace{u_3 + P_3 v_3}_{h_3} - \underbrace{(u_2 + P_2 v_2)}_{h_2}$$

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

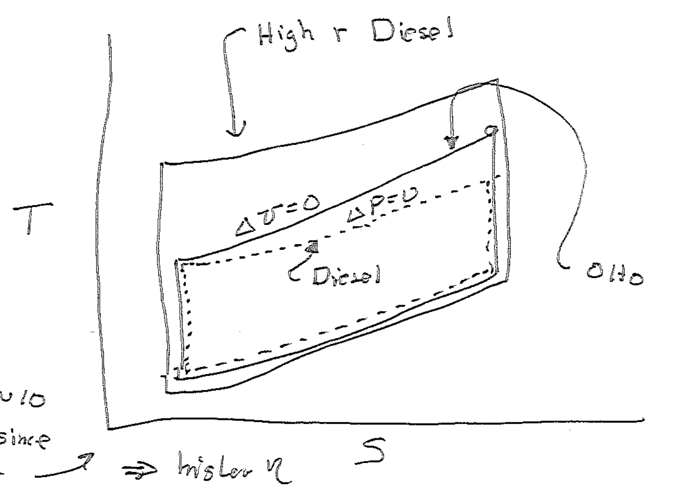
$$q_{out} \stackrel{?}{=} u_4 - u_1$$

<u>Compare OHo and Diesel.</u>

Looks like at same r, OHo η > Diesel η due to higher average T for q_{in},

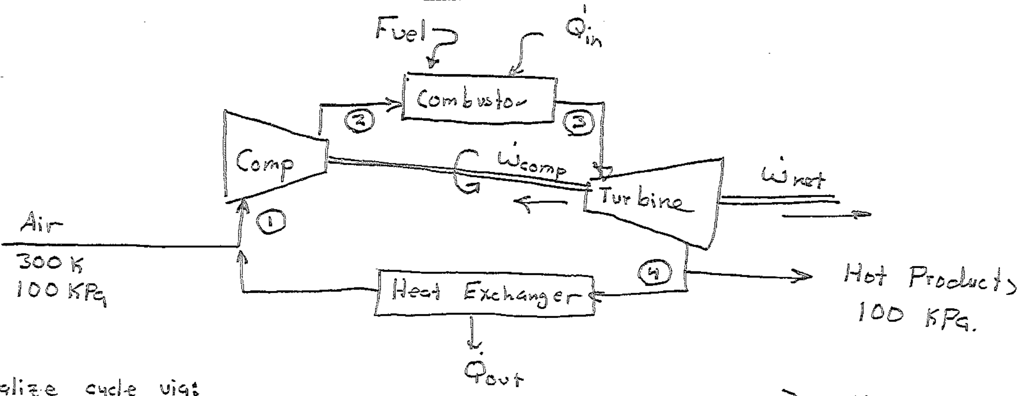
So why use Diesel?

$\Longrightarrow$ OHo r limited to ~10
Diesel can go to >25 since it just compresses air $\Rightarrow$ higher η



Cycle	r	r_c	η
Otto	9.5	—	59
Diesel	25	2.2	67

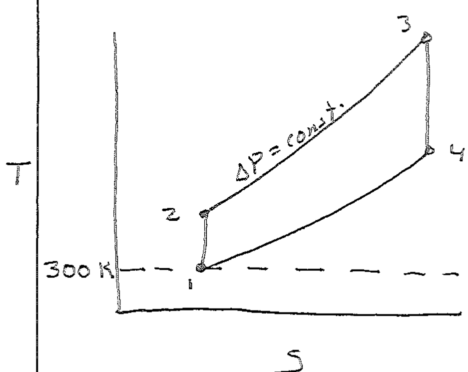
Now to Gas Turbines — $\boxed{\text{Brayton Cycle}}$



Idealize cycle via:

1. Replace exhaust with virtual heat exchanger to close cycle.
2. Replace fuel with $\dot{Q}_{in}$

} Not as flakey as Otto
- Nuclear - heated He cycles
- Indirectly - Fired gas turbine.



$$\eta = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

?

$$\dot{Q}_{in} = \dot{m}\begin{bmatrix} u_3 - u_2 \\ h_3 - h_2 \end{bmatrix} \checkmark \implies \text{Flow system.}$$

Main parameter is $r_p = \frac{P_2}{P_1}$

(now a pressure ratio — in Otto it was a volume).

Let $r_p = 8$, $T_3 = 1650K$, air standard assumptions.

① → ② ? How to do it?

$$\left[\ \frac{P_2}{P_1} = \frac{P_{r,2}}{P_{r,1}} \checkmark \qquad \frac{v_2}{v_1} = \frac{v_{r,2}}{v_{r,1}} \qquad \frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}\ \right] ?$$

$$P_{r,2} = P_{r,1}\left[\frac{P_2}{P_1}\right] = 1.386[8] = 11.09$$

Air, 300K

$T_2 = 540$ K

$h_2 = 544.14$ KJ/Kg

All work, but only the P-isentropic relation has only one unknown.

③ $T_3 = 1650K \longrightarrow h_3 = 1818.7$ KJ/Kg

Get ④ via isentropic: $P_{r,4} = P_{r,3}\left[\frac{P_4}{P_3}\right] = 902.25\left[\frac{1}{8}\right] = 112.78 \xrightarrow{\text{Air}} T_4 = 997$ K

Air, 1650 K

$h_4 = 1042.9 \frac{KJ}{Kg}$

$$w_{net} = q_{in} - q_{out} = [h_3 - h_2] - [h_4 - h_1] = 531.85 \text{ KJ/Kg}$$

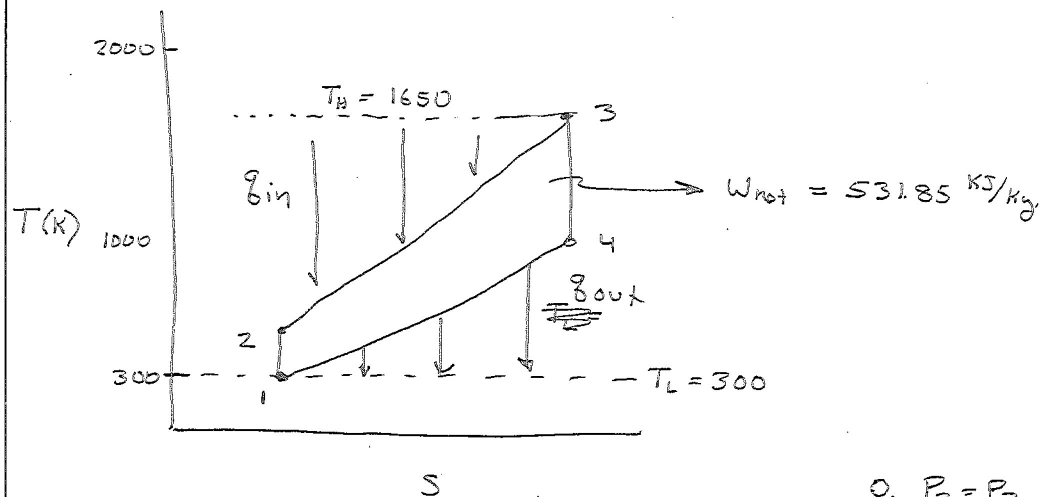
$$\eta = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_4 - h_1}{h_3 - h_2} = 41.7\%$$

Oct. 15

Carnot between same two limits: $\eta = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{1650} = 81.8\%$

$\Rightarrow$ lots of room for improvement.

Plot to scale and look at irreversibility:



$$i_{2-3} = T_0\left[s_3 - s_2 + \frac{q_R}{T_R}\right] = T_0\left[s_3^\circ - s_2^\circ - \cancel{R\ln\frac{P_3}{P_2}} + \frac{q_R}{T_R}\right] \qquad 0,\ P_3 = P_2$$

$$q_R = h_3 - h_2 = 1274.56 \text{ KJ/Kg}$$

$$i_{2-3} = [300\text{ K}]\left[3.5612 - 2.29906 + \frac{(-1274.56)}{1650}\right] = 146.9 \text{ KJ/Kg}$$

$$i_{4-1} = 364.06 \text{ KJ/Kg} \quad \Rightarrow \quad \frac{w_{net} + i_{2\to3} + i_{4\to1}}{q_{in}} = 81.4\% \,?$$

(matching Carnot)

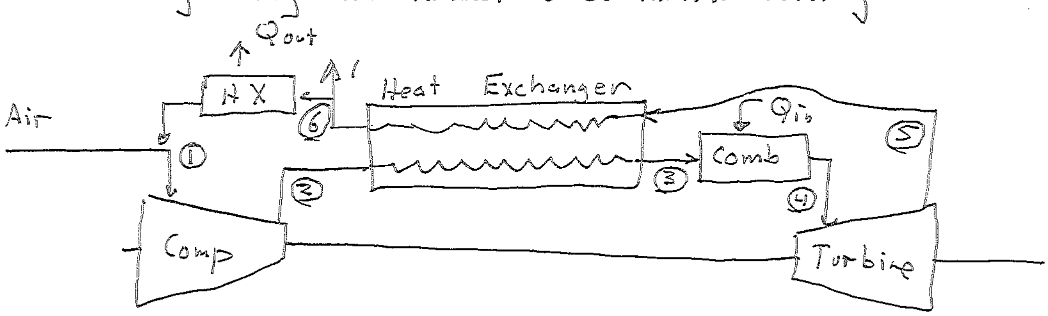
Can help hot end via higher r_p and T_3.

$\longrightarrow$ $T_{3,max} \cong 1700$ K due to turbine blade materials problems.

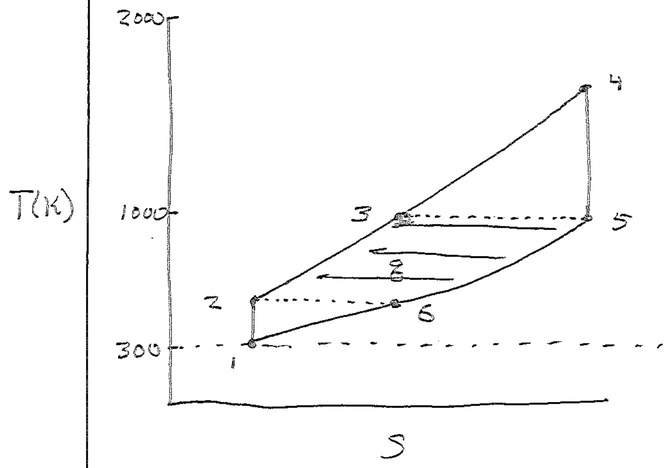
Most i is in exhaust: $\longrightarrow$ How to address

- One approach is to run a steam cycle off of the exhaust [combined cycle]. $\longrightarrow$ Later.
- Another approach.
  - $\longrightarrow$ Exhaust leaves at 997 K.
  - $\longrightarrow$ Initial q_{in} is added to gas at 540 K. } Doesn't make sense ?

$\Rightarrow$ Try using hot exhaust to do initial heating instead of fuel.



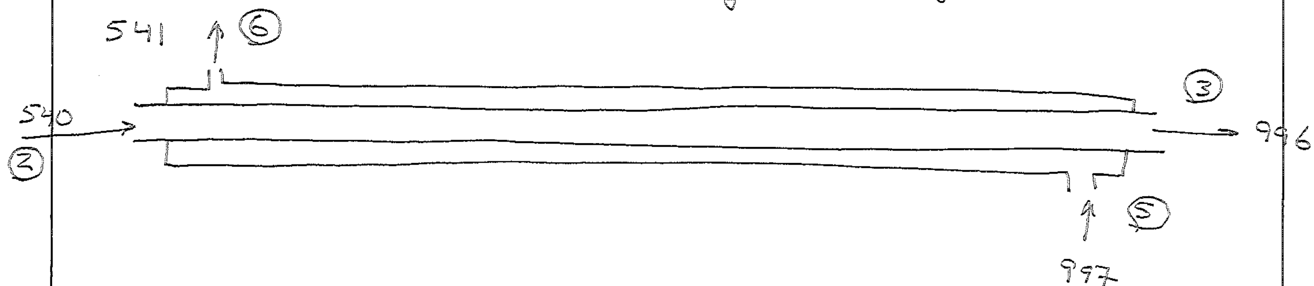
Now take scale system diagram



Point	T(K)	h (KJ/Kg)
1	300	300.19
2	540	544.19
3	→ 997	→ 1047.7
4	1650	1818.7
5	997	1042.7
6	→ 540	→ 544.19

Now locate (3)   Can $T_3 > T_5$?   No
If $T_3 < T_5$ have q across ΔT → irreversibility.
Best case is $T_3 \cong T_5$

Same argument holds for $T_6 \cong T_2$
How can this be? Look at long heat exchanger.



⇒ At each cross-section, stream (5)→(6) is 1 K hotter than stream (2)→(3) so heat flow occurs, but with very low ΔT.
⇒ Requires huge exchanger → high capital cost.

$$\Rightarrow \quad h_3 = h_5$$
$$h_6 = h_2$$

W/O Regeneration

So: $q_{in} \overset{?}{=} h_4 - h_3 = 776$ KJ/Kg $h_4 - h_2 = 1274.6$

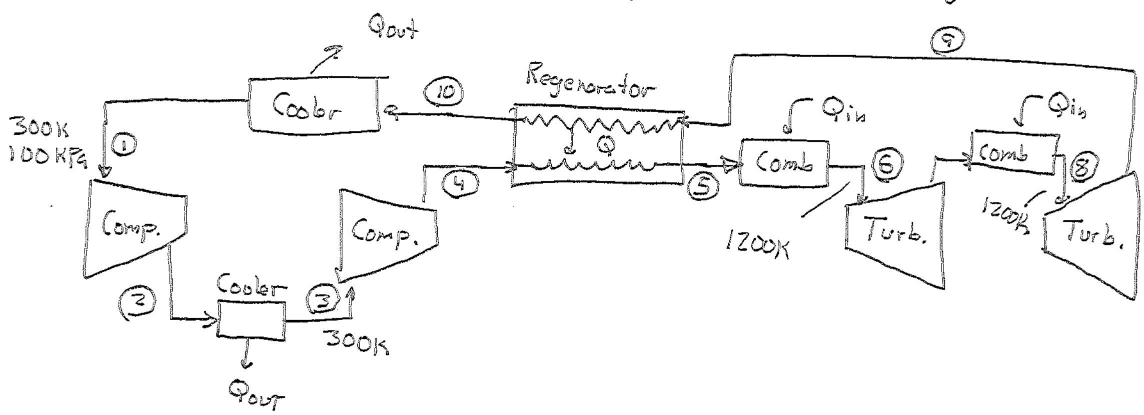
$q_{out} \overset{?}{=} h_6 - h_1 = 532.0$ KJ/Kg $h_5 - h_1 = 742.7$

$W_{net} = q_{in} - q_{out} = 532.0$ 532

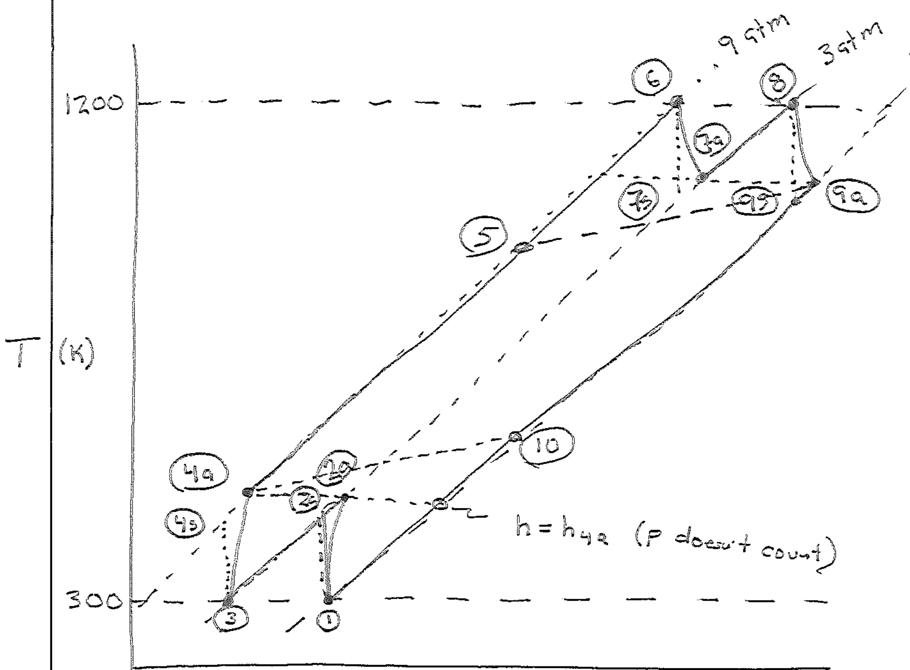
$\eta = 1 - \dfrac{q_{out}}{q_{in}} = 68.6\%$ 41.7%

This η would not be possible due to
1. Compressor and turbines not reversible
2. Heat exchangers require finite ΔT
   ⇒ Not leaking heat q
   ⇒ Rather ΔT → lost work potential.

Another approach is to use dual-stage compression with intercooling and
" " expansion " reheating.



$r_p = 3$ for each stage
$\eta_c = 0.85$
$\eta_T = 0.90$
$\varepsilon_{regen} = 0.75$ (will define when we get to it).
Air standard.



Station	h (kJ/kg)
1	300.19
2s	411.26
2a	430.86
3	~~300.19~~ 1
4s	2s
4a	2a
5	842.34
6	1277.79
7s	946.36
7a	979.5
8	6
9s	7s
9a	7a
10	568.0

1 → 2 Isentropic compression: $P_{r,2} = P_{r,1}\left[\frac{P_2}{P_1}\right] = 1.386\,[3] = 4.158 \longrightarrow h_{2,s} = 411.26$
Air, 300K

Compressor $\eta \Rightarrow$ ~~$\eta_c = \frac{W_s}{W_a}$~~ = "Real compressors require more w for a given ΔP than ideal compressors"

$$\Rightarrow \eta_c = \frac{W_s}{W_a} = \frac{h_{2,s} - h_1}{h_{2,a} - h_1} \Rightarrow 0.85 = \frac{411.26 - 300.19}{h_{2,a} - 300.19}$$

$\Rightarrow h_{2,a} = 430.86 \ \mathrm{kJ/kg}$ [now forget $h_{2,s}$]

Cooler takes us where? Back to 300 K.

$P_3 = 300$ KPa, $T_3 = 300$K $\Big\}$ $h_3 = 300.19$ KJ/Kg, $r_p = 3$, $\eta_c = 0.85$ $\Big\}$ Same as stage 1, except for $P_3 = 300$ KPa.

$P_{r,4} = P_{r,3}\left[P_4/P_3\right]$ = Same as stage 1 (300K, 3)

$\eta_c = \dfrac{h_{4,s} - h_3}{h_{4,a} - h_3}$ Same as stage 1

$\Rightarrow h_{4,s} = h_{2,s}$
$h_{4,a} = h_{2,a}$.

Skip 5

Oct. 17

$T_6 = 1200$ K $\longrightarrow h_6 = 1277.79$ KJ/Kg

To 7 $\Rightarrow$ Isentropic expansion: $P_{r,7} = P_{r,6}\left[\frac{P_7}{P_6}\right] = 238\left[\frac{1}{3}\right] = 79.33 \xrightarrow{Air} h_{7,s}$ 946.36 KJ/Kg

1200 K Air

Turbine η: "Real turbines deliver less work for a given ΔP than actual turbines"

$$\eta_T = \frac{W_a}{W_s} = \frac{h_6 - h_{7,a}}{h_6 - h_{7,s}} \Rightarrow 0.9 = \frac{1277.79 - h_{7,a}}{1277.79 - 946.36} \Rightarrow h_{7,a} = 979.5 \text{ KJ/Kg}$$

Again, $6 \to 7s$, $7s \to 7a$
Same as $8 \to 9s$, $9s \to 9a$

Now finish with regenerator:

- In absence of regeneration, where would q_{in} occur.

$$q_{in} = (h_6 - h_{4,a}) + (h_8 - h_{7a})$$

- Next, add an ideal regenerator ($\Delta T = 0$) How much heat is moved? (add dashes).

$$q_{regen,\ ideal} = h_{9,a} - h_{4,a} \quad \text{(doesn't depend on P!)}$$

- But a real heat exchanger needs a finite ΔT to drive the heat flow. [Add dashed lines and numbers]

- Logical definition of "Regenerator Effectiveness" is

$$\varepsilon_{regen} = \frac{\text{actual heat transferred}}{\text{max heat transfer if } \Delta T = 0} = \frac{h_5 - h_{4a}}{h_{9a} - h_{4a}}$$

$$0.75 = \frac{h_5 - 430.86}{979.5 - 430.86} \Longrightarrow h_5 = 842.34 \text{ KJ/Kg}$$

Last: Get h_{10} from 1st law on exchanger.

$$h_{9a} - h_{10} = h_5 - h_{4a} \Rightarrow 979.5 - h_{10} = 842.34 - 430.86$$
$$\Rightarrow h_{10} = 568.0 \ \text{KJ/kg}$$

$$\dot{q}_{in} \overset{?}{=} [h_8 - h_{7a}] + [h_6 - h_5] = 733.76 \ \text{KJ/kg}$$

$$q_{out} \overset{?}{=} [h_{10} - h_1] + [h_{2,a} - h_3] = 398.48 \ \text{KJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 335.28 \ \text{KJ/kg}$$

$$\eta = 1 - \frac{q_{out}}{q_{in}} = 45.7\% \qquad (55.3\% \text{ if } \eta_c = \eta_T = 1)$$

An aside: i for turbines $\neq 0$ now. Lets compare.

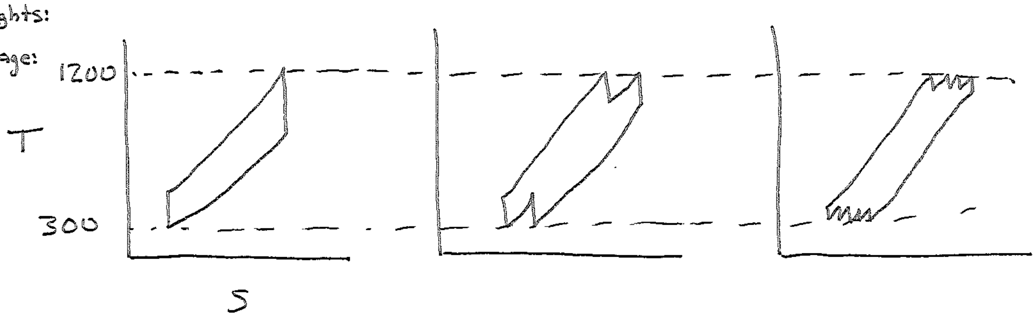
$$i_{6\to 7a} = T_0 \left[s^\circ_{7a} - s^\circ_6 - R \ln \frac{P_{7a}}{P_6} + \cancel{\frac{q_R}{T_R}} \right]^0$$
$$= [300\ \text{K}]\left[2.89914 - 3.17888 - \frac{8.314}{29} \ln\left[\tfrac{1}{3}\right]\right] = 10.67 \ \text{KJ/kg}$$

$$i_{10\to 1} = \cancel{75.85\ \text{KJ/kg}} \ [300\ \text{K}]\left[1.70203 - 2.3419 + \frac{568 - 300.19}{300}\right]$$
$$i_{10\to 1} = 75.85 \ \text{KJ/kg}$$

So exhaust heat transfer is still the biggest problem.

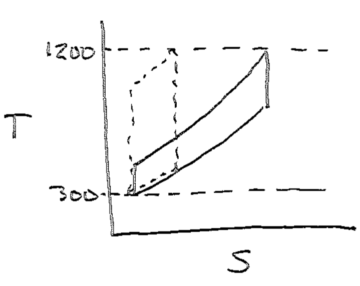
Some thoughts:

① Multistage:



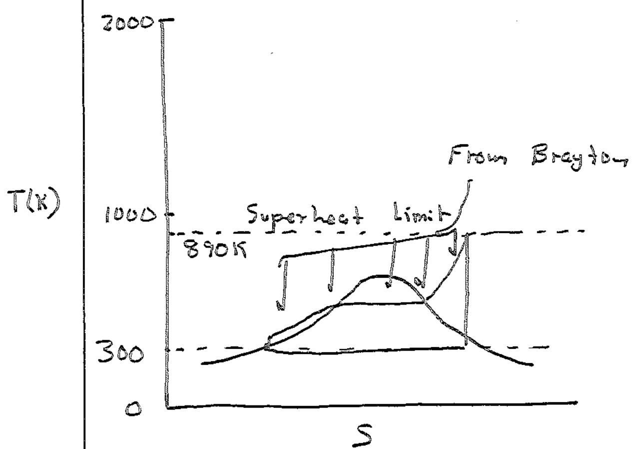
- Note that final T is hot
  $\Rightarrow$ Need to fix with regenerator.
- If you do, what is ΔT for q_{in}, q_{out}? $\Delta T \doteq 0$.
  $\Rightarrow$ Get Carnot η for 1200 $\longrightarrow$ 300 K.
  $\Rightarrow$ Complexity is up,
  $\Rightarrow$ This has lost importance to combined cycles.

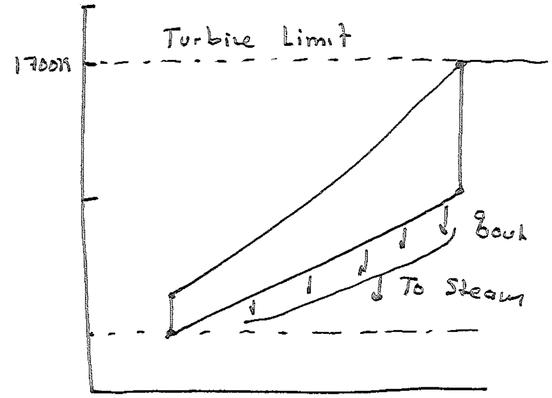
② High r_p: Have discussed before



Higher η, but
$\dot{m}$ for say 200 MW
goes up $\Rightarrow$ Huge plant.

Intro. to Combined Cycle.

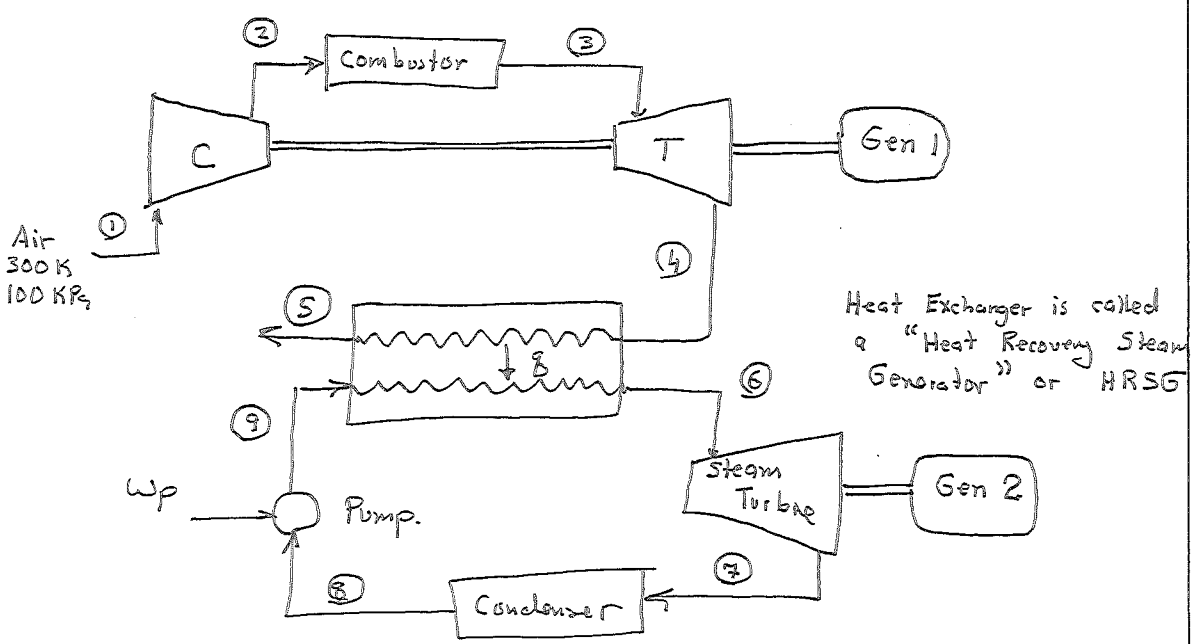




- Rankine is weak because ΔT between heat source ~~and~~ (flame) and steam is large.
- Good because Q_{out} is across small ΔT

- Brayton is weak because exhaust T is high.
- But q_{in} is at higher average ΔT.

⇒ <u>But</u>: Brayton exhaust is hotter (≈1000 K) than T_{max} in Rankine (890 K).
⇒ Use Brayton exhaust to run Rankine.



<u>Key points:</u>

- Both cycles are physically separate in each has own $\dot{m}_a$, $\dot{m}_s$.
- What is $\dot{Q}_{in} \stackrel{?}{=} \dot{m}_a (h_3 - h_2)$
- What is $\dot{Q}_{out} \stackrel{?}{=} \dot{m}_a (h_5 - h_1) + \dot{m}_s (h_7 - h_8)$

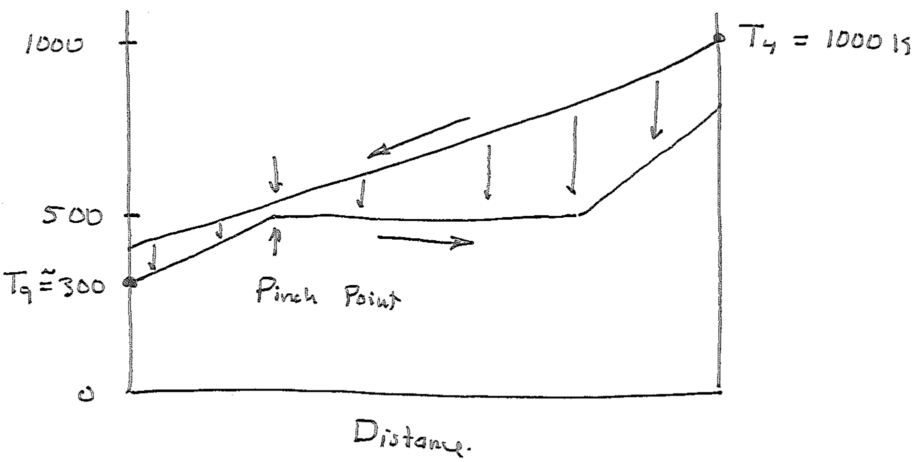
$$\dot{W}_{net} = \dot{Q}_{in} - \dot{Q}_{out}$$

$$\eta = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

Oct 17

Think about some design issues:

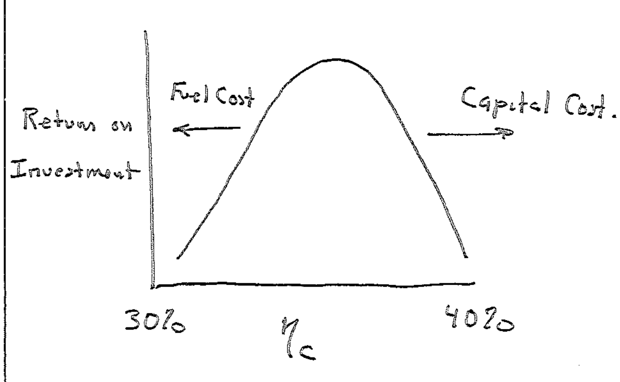
- High or low r_p? Not too high because of low T_4.
- Regeneration? No – heat not lost, but goes to Rankine.
- HRSG:



- Hard to get a low i because HRSG has a big ΔT built in.
- "Pinch Point" is a limit.
  ⟹ Advanced designs make use of complex systems with multiple pressure levels. (ME 430).
  ⟹ Handout on web shows one example.

Historically, power generation is mostly Rankine; Why?
⟶ Can handle dirty fuels (coal).
⟶ Early gas turbines were very inefficient ⟶ Fuel Hogs!
  - Low turbine inlet -T : New materials and aerodynamic film cooling
  - Bad η_c, η_T : Modern CFD design codes.

Rankine $\eta \cong 36-37\%$



Very mature technology.

Opening of Canadian and Rocky Mt. gas fields drove prices down.

Coal – $1.5/10^6 Btu.
Gas – $2.4/10^6 Btu.
~~Cheap gas ...~~

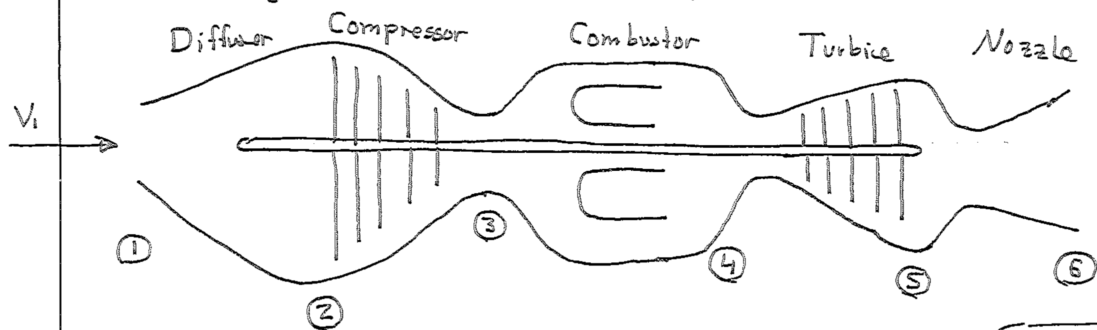
⟹ Combined Cycle because
1) Although coal cheaper, gas easier to use and requires no pollutant cleanup (SO_2, NO, particulates).
2) $\eta_{cc} \cong 60\%$
3) Fast, modular construction.

Blip last winter $\longrightarrow$ Gas to \$10/10^6 Btu. Now down to <3\$. } PNW is tying its future to gas.

Jet Propulsion Cycles

A gas turbine that produces thrust instead of net shaft work:

- Let engine sit still and flow air past it.



Liner: 600 mph, 30,000 ft.
$T_1 = 220K$ (-64°F)
$P_1 = 40$ KPa
$V_1 = 270$ m/s

Key differences from land-based gas turbine.

$\rightarrow$ Turbine only generates enough power to drive compressor - No extra.
$\rightarrow$ This means $P_5 > P_{atm}$
$\rightarrow$ Use this pressure to drive flow through nozzle $\longrightarrow$ thrust!

So we need to retain kinetic energy terms: $q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$ $\boxed{\text{Input}}$

$\boxed{\text{Diffuser}}$ Converts KE of air to enthalpy by slowing it isentropically (like 1st stage of compression).

$$\cancel{q}^{0} - \cancel{w}^{0} = h_2 - h_1 + \frac{\overbrace{V_2^2}^{\text{negligible, but not } = 0.} - V_1^2}{2} \Rightarrow h_2 = h_1 + \frac{V_1^2}{2}$$

$$h_2 = 219.17 \frac{KJ}{Kg} + \left[\frac{270\ m}{s}\right]^2 \Bigg| \frac{1}{2} \Bigg| \frac{KJ/Kg}{1000\ m^2/s^2} = 256.42\ KJ/Kg.$$
(256 K = 0°F).

$$\text{Get } P_2 = P_1\left[\frac{P_{r,2}}{P_{r,1}}\right] = [40\ KPa]\left[\frac{\overset{256K}{0.801}}{\underset{220K}{0.469}}\right] = 68.3\ KPa$$

$\boxed{\text{Compressor}}$ Nothing new here. $\overset{let}{\Longrightarrow}$ $r_p = 25$ (usually high for aero engines.

$$P_{r,3} = P_{r,2}\left[\frac{P_3}{P_2}\right] = [0.801][25] = 20.03 \longrightarrow \begin{array}{l} T_3 = 632\ K \\ h_3 = 641.1\ KJ/Kg \\ P_3 = 25P_2 = 1708\ KPa. \end{array}$$

$$W_c = h_3 - h_2 = 384.7\ KJ/Kg.$$

$\boxed{\text{Combustor}}$ Constant P heat input: Add as much as turbine blades will handle $T_4 = 1700K$; $h_4 = 1880.1$ KJ/Kg

$\boxed{\text{Turbine}}$ Isentropic expansion, but only to point where $w_c = w_T$

$$w_c = w_T = h_4 - h_5 \Rightarrow 384.7 = 1880.1 - h_5 \Rightarrow h_5 = 1495.4 \tfrac{KJ}{Kg}$$

$$T_5 = 1380.3\ K.$$

$$P_5 = P_4\left[\frac{P_{r,5}}{P_{r,4}}\right] = \left[1708\ KPa\right]\left[\frac{\overset{1380.3K}{428.5}}{\underset{1700K}{1025}}\right] = 714\ KPa$$

$\boxed{\text{Nozzle}}$ Drop pressure to atmospheric isentropically $\longrightarrow$ Get V_6

$$\cancel{q}^{\,0} - \cancel{w}^{\,0} = h_6 - h_5 + \frac{V_6^2 - \cancel{V_5^2}}{2} \quad \text{negligible}$$

$\Rightarrow V_6 = \left[2(h_5 - h_6)\right]^{1/2}$; But how to get h_6

$\Rightarrow$ Known ΔP & assume $\Delta S = 0$

Oct. 19

$$P_{r,6} = P_{r,5}\left[\frac{P_6}{P_5}\right] = \left[428.3\right]\left[\frac{40}{714}\right] = 24.01 \Longrightarrow h_6 = 667.49 \tfrac{KJ}{Kg}$$

$$T_6 = 666.6\ K$$

$$V_6 = \left[\frac{2 \,|\, 1495.4 - 667.49\ KJ \,|\, 1000\ m^2/s^2}{\,|\, Kg \,|\, KJ/Kg}\right]^{1/2} = \underline{1279\ m/s}$$

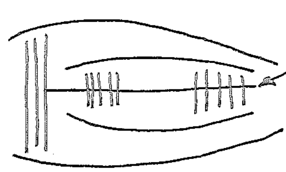
(supersonic, shocks, noisy).

$$\text{Thrust} = \text{Force} = \dot{m}(V_6 - V_1) = \left[1\ Kg/s\right]\left[1279 - 270\ m/s\right] = 1009\ N/_{Kg/s}$$

@150 lb, you put 670 N on your chair, so you would need 0.66 Kg/s to levitate!

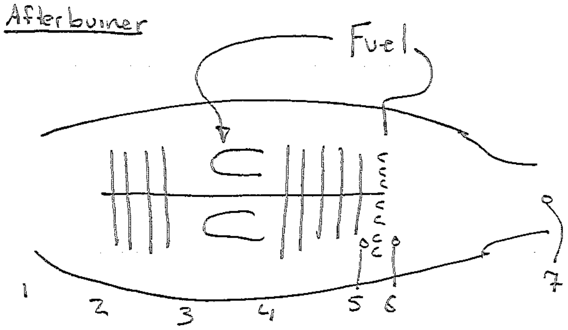
Two augmentations:

① Turbofan



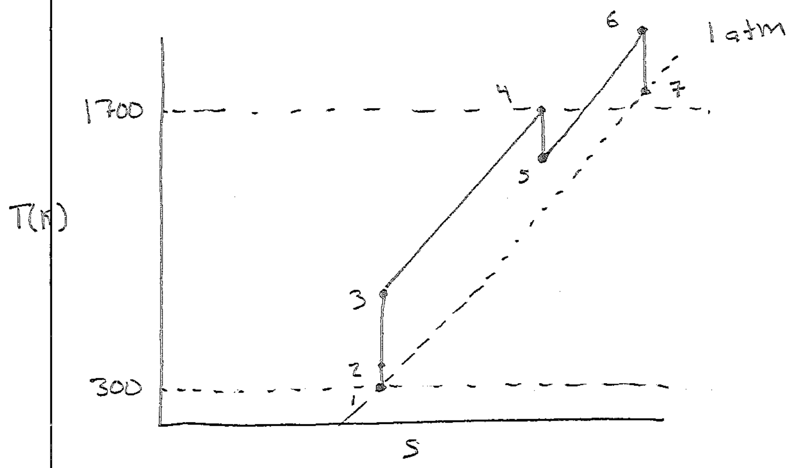
- Use extra shaft work to blow air around outside.
  - $\longrightarrow$ Better η
  - $\longrightarrow$ Thrust $\sim \dot{m} V_6$
    - $\Rightarrow \dot{m}$ higher, V_6 lower
      - $\Rightarrow$ No shocks, quieter engine

② Afterburner



- Add more heat after turbine
  - $\longrightarrow$ Expand gas
  - $\longrightarrow$ Higher velocity
  - $\longrightarrow$ More thrust.

Look @ T-s:



• But: Heat added after partial expansion @ lower T

• Also, exhaust T is hotter.

⟹ Very poor η.

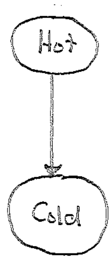
⟹ Use briefly when you need high thrust, not otherwise.

• Thermodynamically better to add extra fuel to combustor, but this will ruin blades.

Chapter 9 - Refrigeration Cycles

The Basic Problem.

2nd Law lets heat go from hot to cold



• This is irreversible, and entails a loss of work potential.

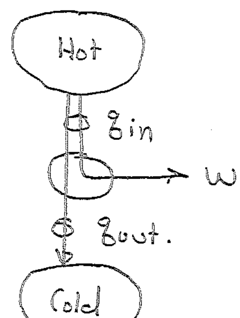
• Same thermal energy @ T_H is more valuable than that @ T_L

$$dS = \frac{\delta q}{T}\Big)_{rev.}$$

Higher T, lower s, more order.

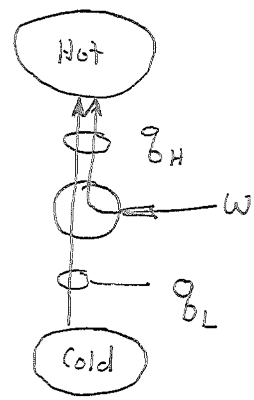
• Heat Engine grabs some of thermal energy as work.

• Rejects rest to cold.



In a perfect cycle, the loss in "value" of q_{out} going from hot → cold exactly balances the gain in value of $q_{in} - q_{out}$ being converted from thermal energy at T_H → W.

Reverse the heat engine



[Increase in value of q_L going from T_L to T_H]

= [Decrease in value of W becoming thermal energy at T_H]

	Cold	Hot
Refrigerator:	Icebox	Environment
Heat Pump	Environment	Warm House

Efficiencies:

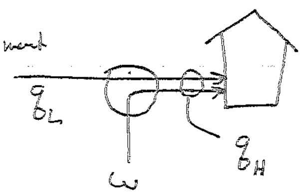
$$\text{Coefficient of performance} = \beta = \frac{\text{What we want}}{\text{What we pay for}} = \frac{q_L}{W}$$

$$= \frac{q_L}{q_H - q_L} = \frac{1}{\frac{q_H - q_L}{q_L}} = \frac{1}{\frac{q_H}{q_L} - 1} \underset{\text{Only}}{\overset{\text{Carnot}}{=}} \frac{1}{\frac{T_H}{T_L} - 1}$$

For home refrigerator: $\left.\begin{array}{l} T_L = 0°F \text{ icebox} \\ T_H = 70°F \text{ room} \end{array}\right\} \beta = 6.57$

→ 100 Watts of electricity pulls
→ 657 Watts of heat from coldspace, and delivers
→ 757 Watts of heat to environment.

Heat Pump:



$$\text{Coefficient of Performance} = \gamma = \frac{\text{What we want}}{\text{What we pay for}} = \frac{q_H}{W} = \frac{q_H}{q_H - q_L} = \frac{1}{\frac{q_H - q_L}{q_H}}$$

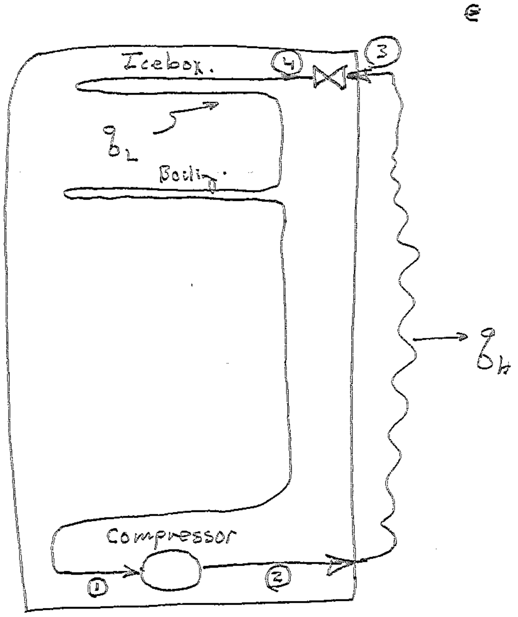
$$= \frac{1}{1 - \frac{q_L}{q_H}} \overset{\text{Carnot}}{=} \frac{1}{1 - \frac{T_L}{T_H}}$$

$\left.\begin{array}{l} \text{Environment} = 40°F \\ \text{House} = 70°F \end{array}\right\} \gamma = 17.67$

→ 100 Watts of power yields
→ 1767 Watts of heating

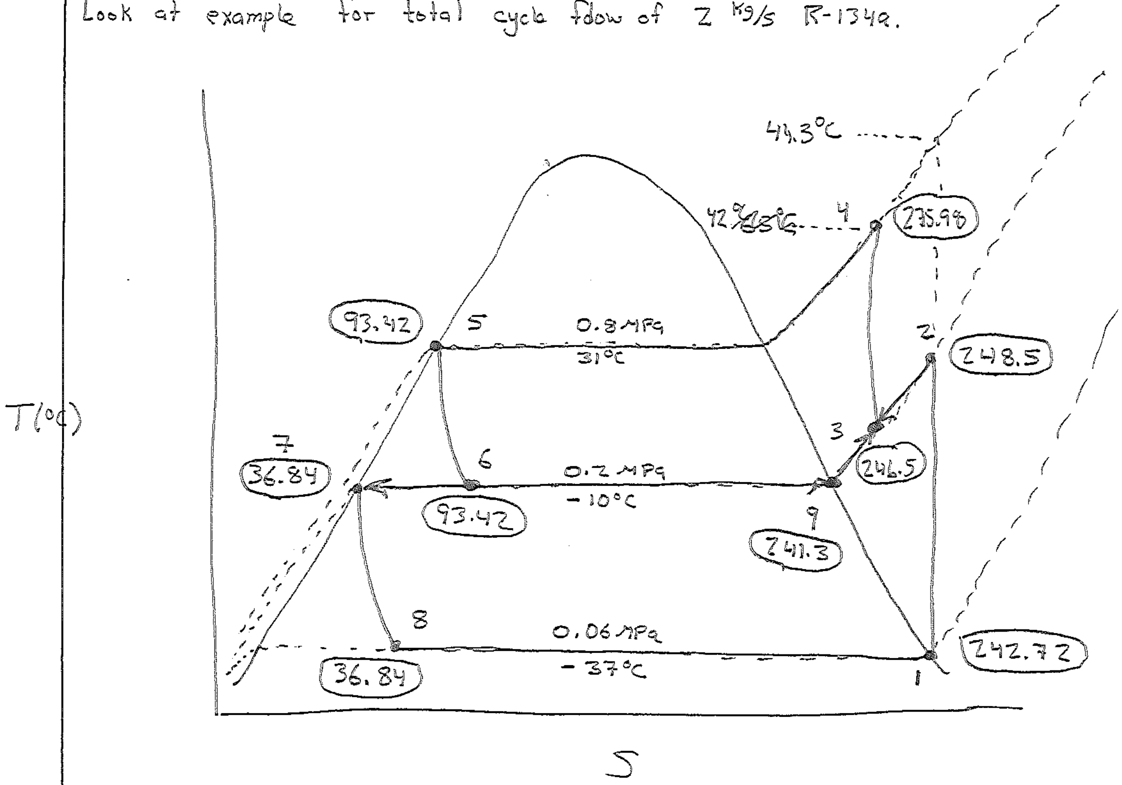
Compare with resistance heaters
→ 100 W power
→ 100 W of heat.

Basic Vapor Compression Cycle:



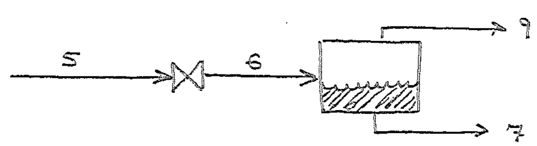
- Spray an aerosol can long enough and it gets cold due to boiling inside] Propane.
- Icebox contains low-P boiling refrigerant.
  ⇒ Product is cold vapor
- Compress the vapor, make it hot.
- Send out back, let hot vapor condense to ~~warm liquid~~ room-T liquid.
- Expand liquid through throttle to start boiling at low-T.

Look at example for total cycle flow of 2 kg/s R-134a.



Oct. 23

Get h_3 by mixing streams 2 and 9. Focus on flash chamber.
(need $\dot{m}$'s of each).



Since all is adiabatic, flash chamber just separates mix from throttle, it doesn't change it.

$$\Rightarrow \quad \frac{\dot{m}_9}{\dot{m}_6} = X_6 \; , \quad \frac{\dot{m}_7}{\dot{m}_6} = 1 - X_6$$

How to get X_6? $\quad X_6 = \dfrac{h_6 - h_f}{h_{fg}} = \dfrac{93.42 - 36.84}{204.46} = 0.28$

Now do first law on mixer: $\quad \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_9 h_9$

$$\Rightarrow h_3 = \underbrace{\frac{\dot{m}_2}{\dot{m}_3}}_{1-X_6} h_2 + \underbrace{\frac{\dot{m}_9}{\dot{m}_3}}_{X_6} h_9 = [1 - 0.28][248.5] + [0.28][241.3] =$$

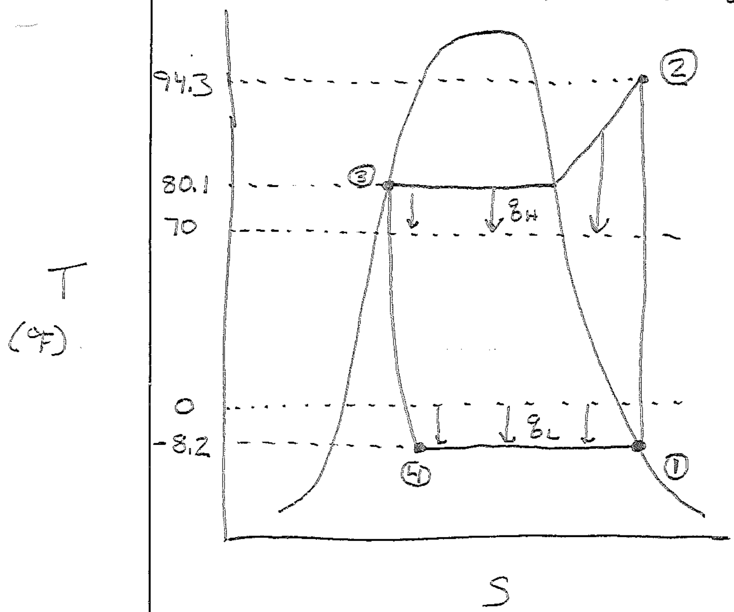
$\Rightarrow h_3 = 246.5$ kJ/kg

$S_3 = 0.9447$ kJ/kg-K

$$\left. \begin{array}{l} P_4 = 0.8 \text{ MPa} \\ S_4 = S_3 \Rightarrow \end{array} \right\} \Rightarrow h_4 = 275.98 \text{ kJ/kg}$$

Ideal Cycle: R-134a.

$P_H = 0.7$ MPa, $P_L = 0.12$ MPa, $\dot{m} = 0.1$ kg/s } Get $\dot{W}$, $\dot{Q}_H$, $\dot{Q}_L$, β



Point	h (kJ/kg)	S (kJ/kg-K)
1	233.86	0.9354
2	270.22	"
3	86.78	0.3242
4	"	0.3489

① Sat. Vapor, $P = 0.12$ MPa $\Rightarrow$ $h_1 = 233.86$ kJ/kg
$S_1 = 0.9354$ kJ/kg-K
$T_1 = -22.36°C = -8.2°F$

② Before we used $w_p = v_1(P_2 - P_1)$ • Will this work? • Why not?

Isentropic compression $P_2 = 0.7$ MPa, $S_2 = S_1$ $\xrightarrow[\text{Superheat}]{134a}$ $h_2 = 270.22$ kJ/kg, $T_2 = 34.6°C = 94.3$ F

③ Cool at constant P to a saturated liquid. $P = 0.7$ MPa.
$\Rightarrow$ $h_3 = 86.78$ kJ/kg
$S_3 = 0.3242$ kJ/kg-K
$T_3 = 26.72°C = 80.1°F$

④ Expansion valve $\longrightarrow$ Throttle to 0.12 MPa. $\cancel{\dot{Q}}^{0} - \cancel{\dot{W}}^{0} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$
$T_4 = -8.2°F$ $\dot{m}_e = \dot{m}_i \Rightarrow h_e = h_i$

Get $S_4 \Rightarrow$ $X_4 = \frac{h_4 - h_f}{h_{fg}} = \frac{86.78 - 21.32}{212.54} = 0.308$

$S_4 = S_f + X_4 S_{fg} = 0.0879 + [0.308][0.9354 - 0.0879] = 0.3489$ kJ/kg-K

$\boxed{\text{Add icebox, ambient T's now}}$

Note that cycle is bigger than needed $\longrightarrow$ more work needed to move required q_L!

$\dot{W}_{in} = \dot{m}[h_2 - h_1] = 3.64$ KW (36.4 kJ/kg)

$\dot{Q}_H = \dot{m}[h_2 - h_3] = 18.3$ KW (183 kJ/kg)

$\dot{Q}_L = \dot{m}[h_1 - h_4] = 14.7$ KW (147 kJ/kg)

Oct 22

$\beta = \frac{\dot{Q}_L}{\dot{W}} = \frac{14.7}{3.64} = 4.04$ For Carnot $0°F \to 70°F$ $\beta = 6.57$

So where do irreversibilities occur, and which are biggest?

$i_{1\to2} = 0$

$i_{2\to3} = T_0\left[(s_3 - s_2) + \frac{q_R}{T_R}\right] = [\overset{70°F}{294\,K}]\left[(0.3242 - 0.9354) + \frac{+183}{294}\right] = 3.31 \frac{kJ}{kg}$ $0.331\ kW$

$i_{3\to4} = T_0\left[(s_4 - s_3) + \cancel{\frac{q_R}{T_R}}^{\,0}\right] = 7.26\ kJ/kg = 0.726\ kW$

$i_{4\to1} = T_0\left[(s_1 - s_4) + \frac{q_R}{T_R}\right] = [294\,K]\left[(.9354 - .3489) + \frac{-147}{255}\right] = 2.95\ kJ/kg = 0.295\ kW$

$$\beta_{max} = \frac{\dot{Q}_L}{\dot{W} - \sum i's} = \frac{14.7}{3.64 - 0.331 - 0.726 - 0.295} = \underline{\underline{6.42}}$$

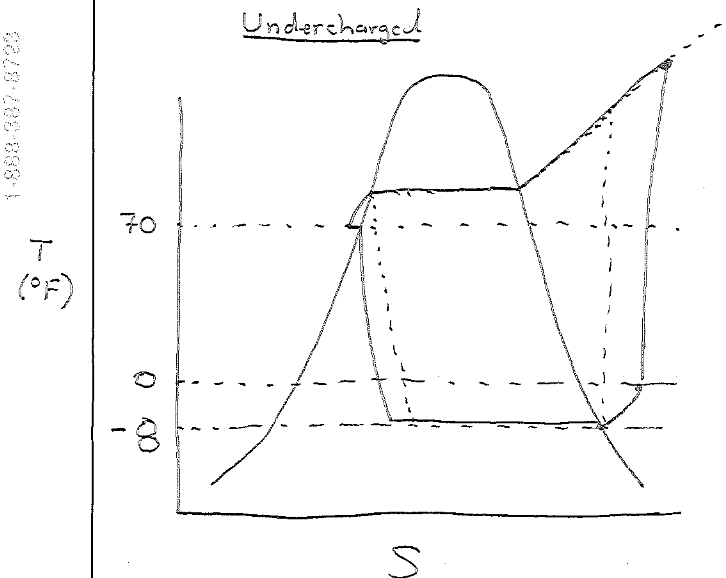
(= Carnot?)

How to improve?

Icebox: Larger surface area.
Condenser: 1) Don't fully evaporate so you stay in dome.
2) Larger area.
Throttle: Replace with turbine, But 1) Cost of turbine
2) Hard to expand liquid efficiently
3) Mechanical energy recovery ($\to$ compressor).

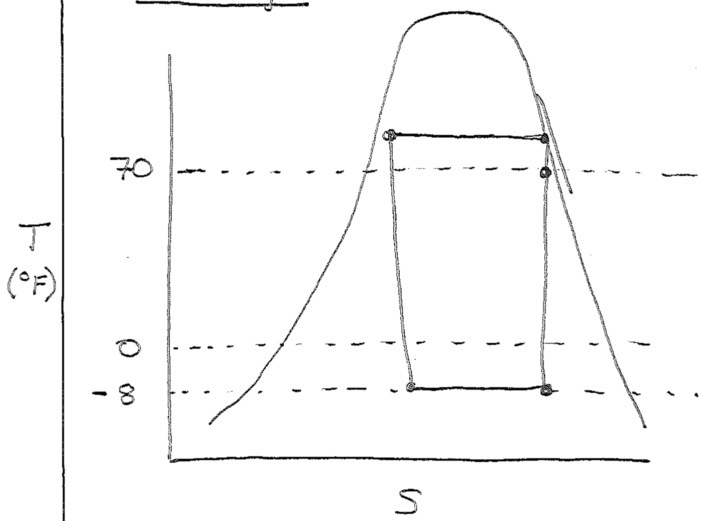
Look at real systems (two versions)

Undercharged



- Evaporator: All refrigerant evaporates and vapor reaches ice box temperature.
- Compressor: Non reversible ($s\uparrow$) and not adiabatic ($s\downarrow$) so cannot tell.
- Condenser: Condenses all liquid and takes T to ambient.
- Throttle starts on compressed liquid ($\Delta T \cong 0$).

Overcharged



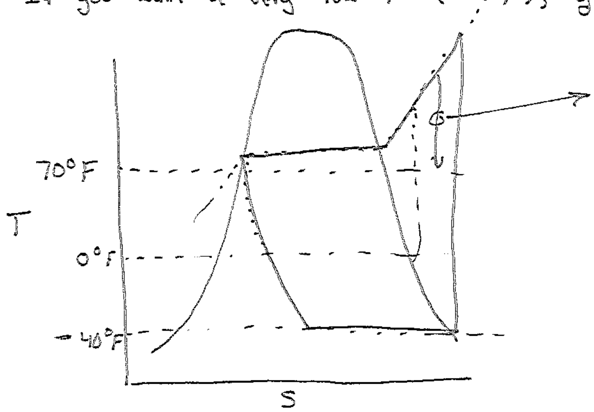
- Evaporator: Not all refrigerant evaporates.
- Compressor: Same as above
- Condenser: Not all vapor condenses

⟹ Closer to rectangle.
⟹ Like Carnot.
⟹ Higher efficiency.

⟹ Note: Recharge car AC until T into evaporator = T out.

- Another practical observation: Auto A/C charge gets low and you notice it cycles on and off. Why?
  1. Low charge reduces P_{max}, P_{min}.
  2. Evap < 0°C ⟹ Form frost plug
  3. P-sensor on low-p side shuts system down to avoid freezup.

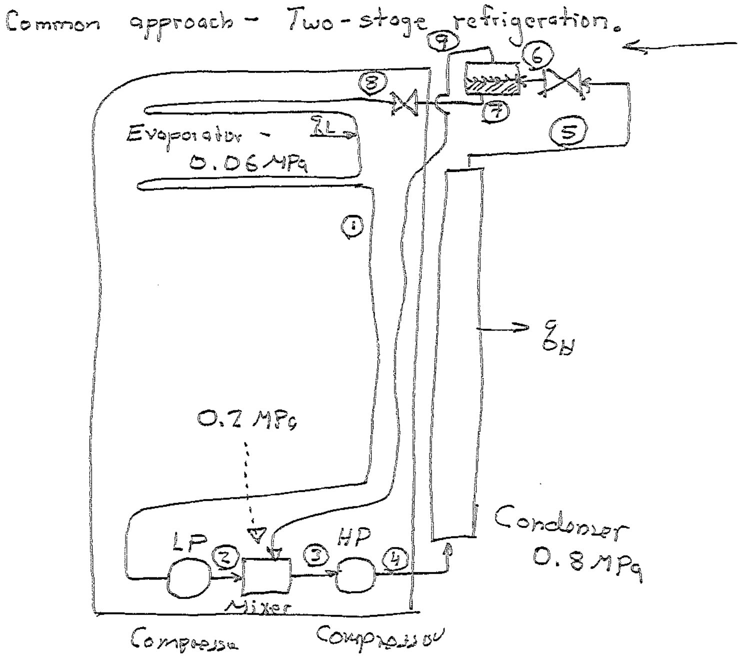
If you want a very low-T (~40°F), you just need to run condenser at low-P.

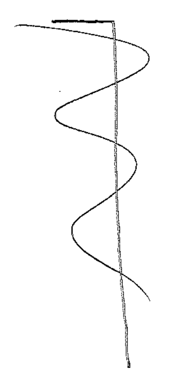


→ New q_H is across a larger ΔT. Need to reduce to get less $\dot{\imath}$ and higher β.

Common approach - Two-stage refrigeration.

Flash Chamber (Separates vapor and liquid)





Idea is to partially expand
→ Pass only liquid to final expansion
→ Send vapor (which doesn't help icebox) to cool charge between compressors (like an internal intercooler)

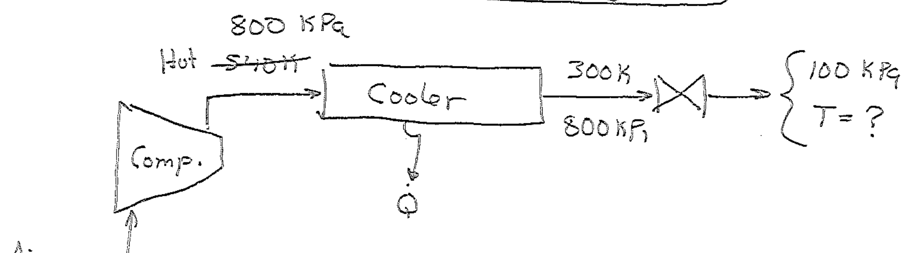
$$\dot{Q}_H = \dot{m}_4 (h_4 - h_5) = (2\ ^{kg}/_{s})(275.98 - 93.42) = 365.1 \text{ kW}$$

$$\dot{Q}_L = \dot{m}_7 (h_1 - h_8) = (2\ ^{kg}/_{s})(1-0.28)(224.72 - 36.84) = 270.6 \text{ kW}$$

$$\dot{W} = \dot{Q}_H - \dot{Q}_L = 94.55 \text{ kW} \; ; \quad \beta = \frac{\dot{Q}_L}{\dot{W}} = 2.86$$

(Equivalent simple cycle = 2.45)

Gas Refrigeration

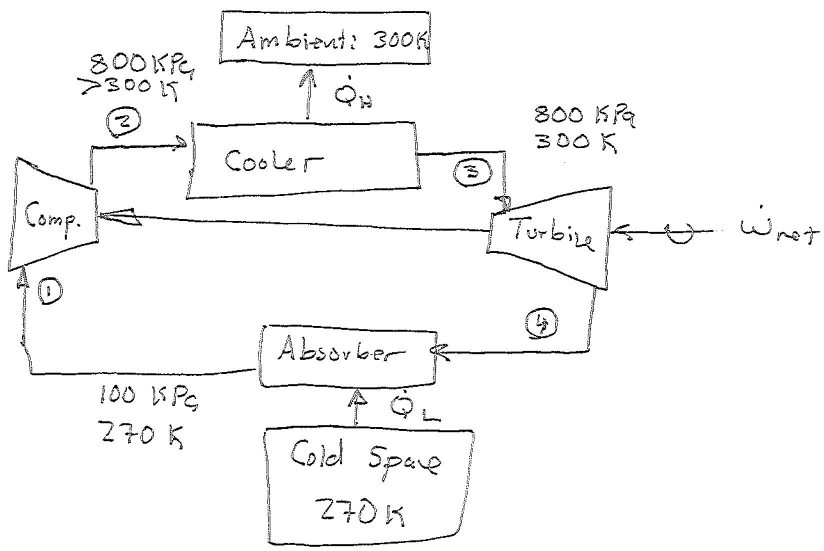


$h_{in} = 300.19\ ^{kJ}/_{kg}$ @ 300 K

$h_{out} = 300.19\ ^{kJ}/_{kg} \longrightarrow T_{out} = 300\text{K}$!

(No pressure effect on h !).

So you need to use a turbine:



Look at performance with <u>cold air assumptions</u>.

$$T_2 = T_1 \left[{}^{P_2}/_{P_1} \right]^{k-1/k} = (270\text{ K})(8)^{0.4/1.4} = 489 \text{ K}.$$

$$T_4 = T_3 \left[{}^{P_4}/_{P_3} \right]^{k-1/k} = (300\text{K})(1/8)^{0.4/1.4} = 166 \text{ K}.$$

$$q_H = C_p (T_2 - T_3) = (1.005)(489 - 300) = 190\ ^{kJ}/_{kg}.$$

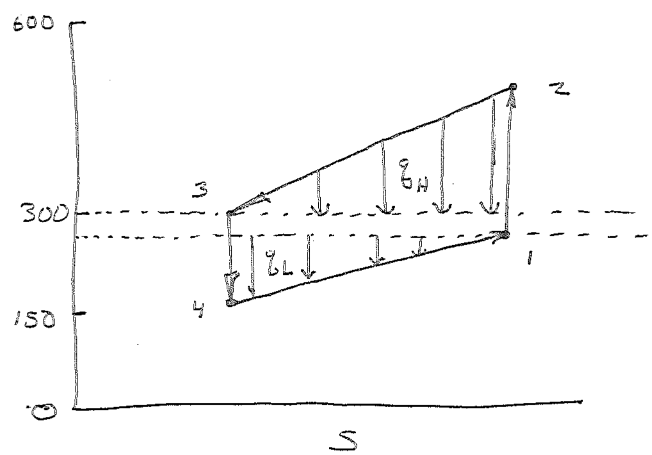
$$q_L = C_p (T_1 - T_4) = (1.005)(270 - 166) = 105\ ^{kJ}/_{kg}.$$

$$W_{net} = q_H - q_L = 190 - 105 = 85\ ^{kJ}/_{kg} \quad \text{(going in !)}$$

$$\beta = \frac{q_L}{W_{net}} = \frac{105}{85} = 1.24 \quad \text{(really bad !)} \qquad \beta_{Carnot} = \frac{1}{\frac{300}{270} - 1} = 9$$

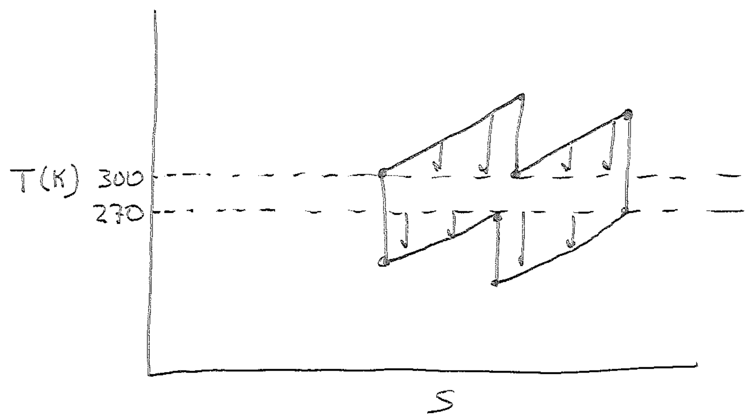
(even with perfect comp. & turbine)

Why? What can we do?



• Big ΔT for q_H and q_L
• Lots of i.

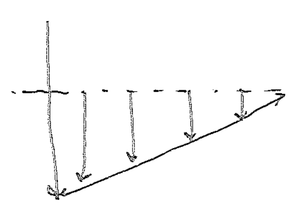
What to do? • Compress in two stages with an intercooler. ~~Reheat in~~
• Use a two-stage absorber.



$\Rightarrow \boxed{\beta = 2.89}$!

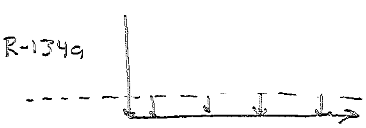
The key difference is:

Air



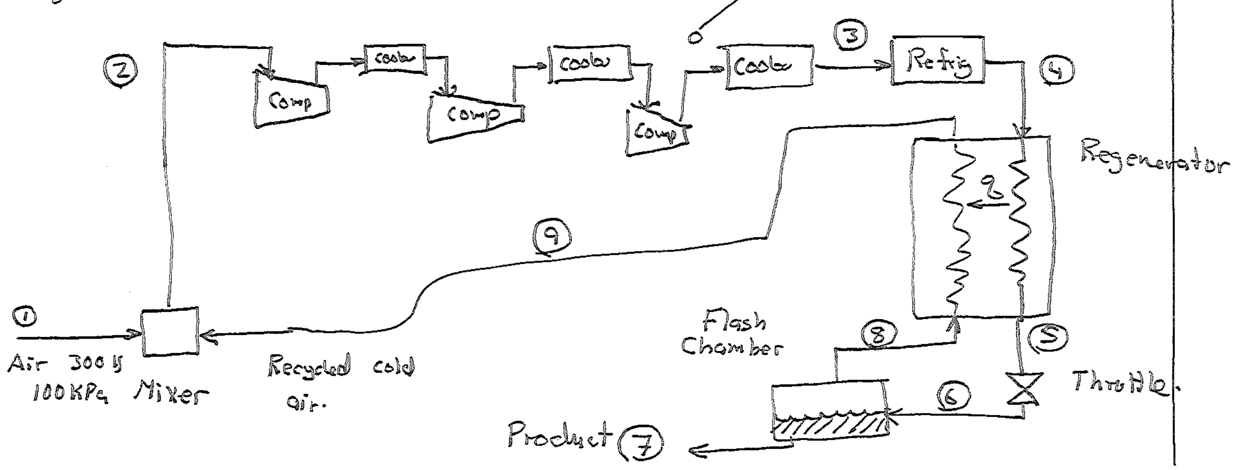
• Absorbs heat by heating up.
• Must start cold $\Rightarrow i$

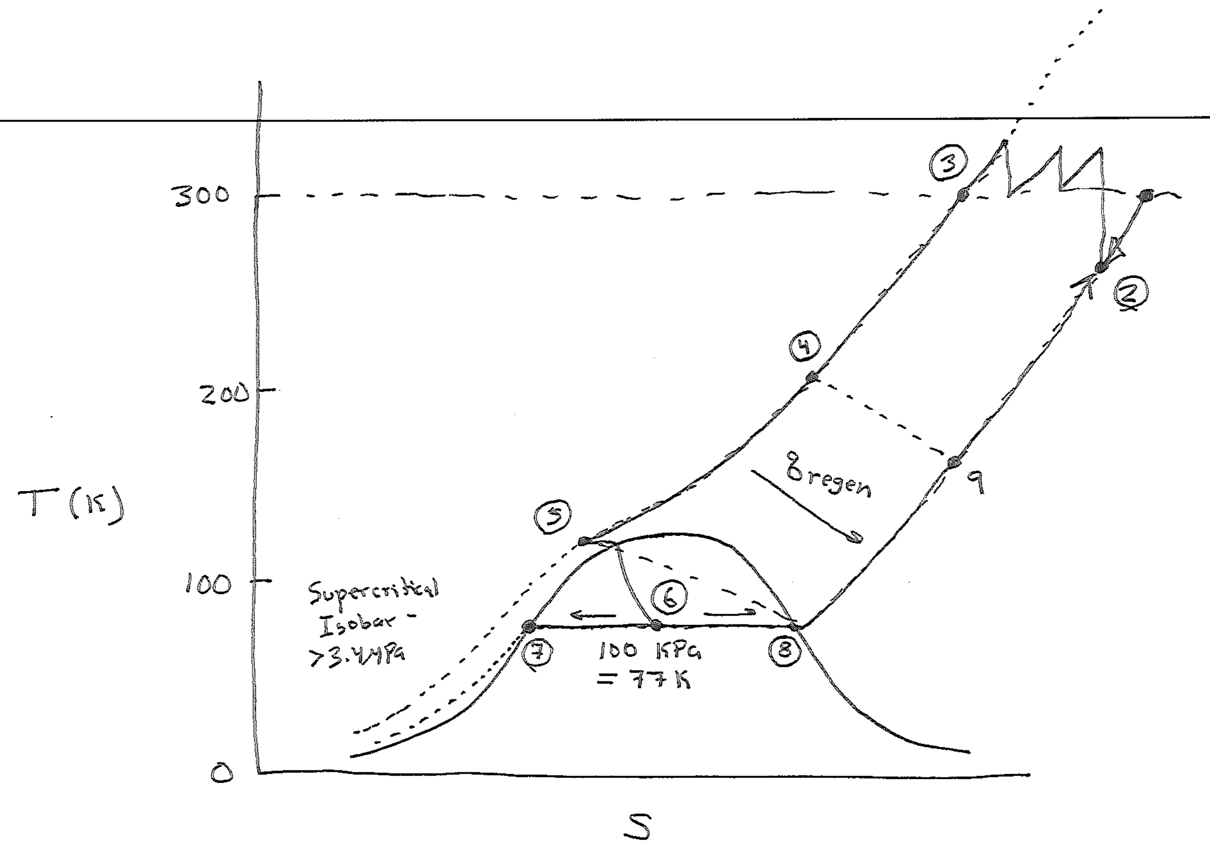
R-134a



• Absorbs heat by boiling
• Can run with small ΔT.

Gas liquifaction is an extreme example.

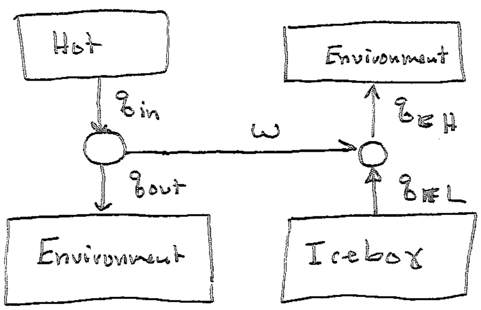




Oct. 25

Absorption Refrigeration: Colman-Type refrigerators that run off a flame.

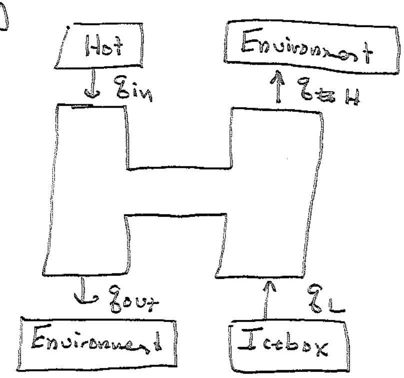
Normal refrigeration



Need w; usually get from heat engine.

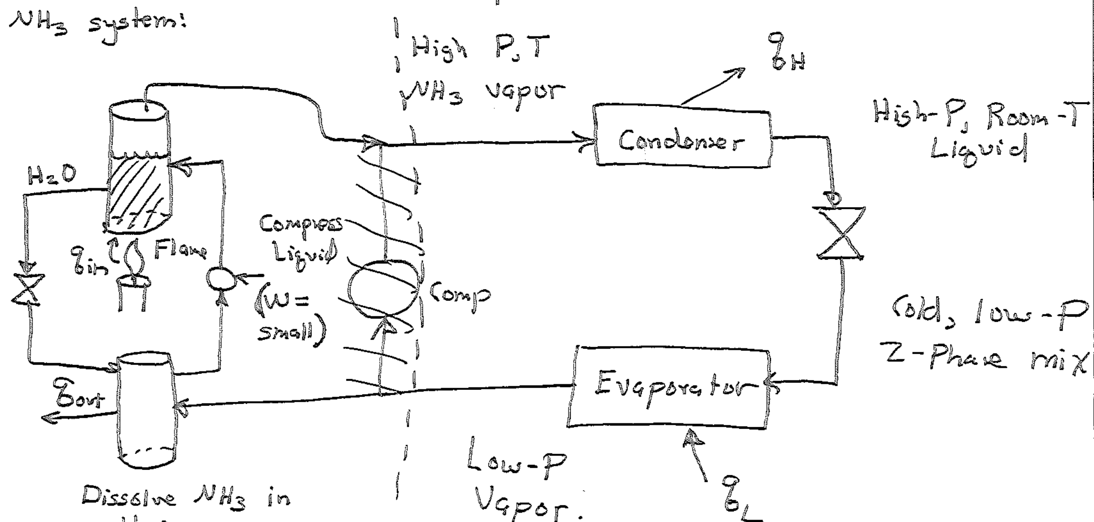
⟹

Put them together!



⟹ Eliminate discrete work step.

Look at NH_3 system:



Goal is to replace compressor. - All else stays the same.