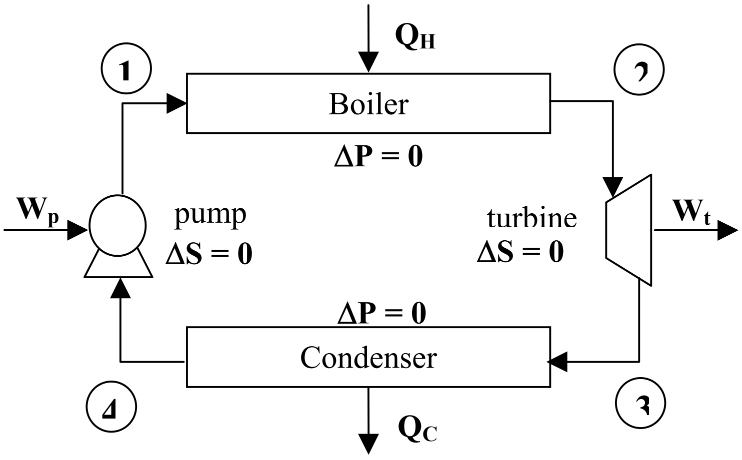
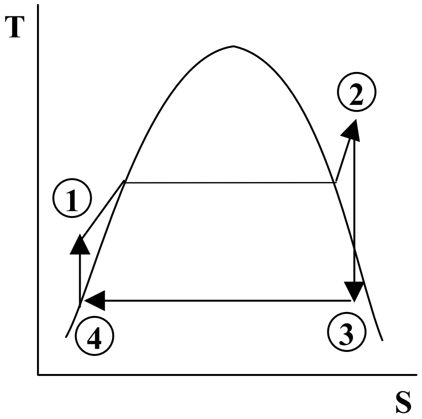
23) A schematic of the Rankine cycle and representation on a T-S diagram:





The information given to us: $P_2 = 3300$ kPa, $P_3 = 50$kPa; $T_2 = \left[\begin{array}{l} 475^\circ C \\ 500^\circ C \\ 525^\circ C \end{array} \right.$

Depends on your last name. **The solution will be given in this form (all 3 solutions).**

We are asked to find the thermal efficiency, η = (net work output) / (heat input).

Strategy: We need to step through the equipments and figure out the inlets and outlets to allow us to calculate the work and heat terms.

Turbine (2 → 3)

We want to find the quality of the outlet stream and the shaft work produced. Need to find state 3. State 2 is already defined by the temperature and pressure. We can get H_2 and S_2:

$S_2 = \left[\begin{array}{l} 7.1136 \\ 7.1873 \\ 7.2588 \end{array} \right.$ kJ/kg K and $H_2 = \left[\begin{array}{l} 3396.8 \\ 3452.8 \\ 3509.0 \end{array} \right.$ kJ/kg from steam table at 3300 kPa and $T = \left[\begin{array}{l} 475^\circ C \\ 500^\circ C \\ 525^\circ C \end{array} \right.$

$\Delta S = 0 \rightarrow S_3 = S_2$. Then we can find state 3, which is at $P_3 = 50$kPa and has the given S_2 above. When we look at the steam table for 50 kPa, we find that S_3 lies in between

$S_{\text{sat liq}} = 1.0912$ kJ/kg K and $S_{\text{sat vap}} = 7.5947$ kJ/kg K

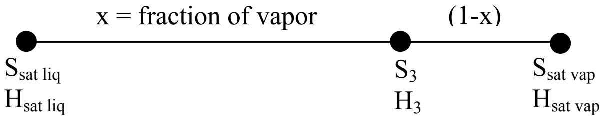
Therefore the outlet of the turbine must be a mixture of liquid and vapor. To find the quality:

$$x = \frac{S_3 - S_{\text{sat liq}}}{S_{\text{sat vap}} - S_{\text{sat liq}}} = \frac{S_3 - 1.0912\,\text{kJ/kg K}}{(7.5947 - 1.0912)\,\text{kJ/kg K}}$$

See problem 25 d) for a case where the S > $S_{\text{sat vap}}$ and how we deal with that.

Plugging in S_3 (= S_2) from above, gives us:

$\mathbf{x} = \left[\begin{array}{l} \mathbf{0.926} \\ \mathbf{0.937} \\ \mathbf{0.948} \end{array} \right.$ (ans) for $T = \left[\begin{array}{l} 475^\circ C \\ 500^\circ C \\ 525^\circ C \end{array} \right.$



The quality (x) allows us to calculate H_3:

$$H_3 = x \cdot H_{\text{sat vap}} + (1 - x) H_{\text{sat liq}}$$

From steam table (50 kPa): $H_{\text{sat vap}} = 2646.0$, $H_{\text{sat liq}} = 340.564$ kJ/kg

$$H_3 = \left[\begin{array}{l} 2475.5 \\ 2501.6 \\ 2526.9 \end{array}\right. \text{kJ/kg} \rightarrow \quad W_t = H_3 - H_2 = \left[\begin{array}{l} -921.3 \\ -951.2 \\ -982.1 \end{array}\right. \text{kJ/kg.}$$

(since the turbine is adiabatic, $Q = 0$ and the shaft work, $W_t = \Delta H = H_3 - H_2$)

Condenser (3 → 4)

All we need to know from the condenser is that the output is saturated liquid at $P_4 = P_3 = 50$ kPa (since for condenser $\Delta P = 0$). This output goes into the pump.

Pump (4 → 1)

W of a reversible pump, $W_p \approx V_{liq}\Delta P$.

From lecture notes. We have assumed V_{liq} is constant because liquid is fairly incompressible.

Looking at the steam table at 50kPa for saturated liquid: $H_4 = 340.6$ kJ/kg. $V_4 = 1.030\ \text{cm}^3/\text{g}$.

$$W_p = V_4(P_1 - P_4) = 1.030\,\text{cm}^3/\text{g}\,(3300-50)\,\text{kPa}\,\frac{10^{-6}\,\text{m}^3}{1\,\text{cm}^3}\,\frac{1000\,\text{g}}{1\,\text{kg}}\,\frac{1\,\text{kJ}}{1\,\text{kPa m}^3} = 3.35\,\text{kJ/kg}$$

Then $H_1 = H_4 + W_p = 343.9$ kJ/kg.

Note that W_{pump} is negligible compared to $W_{turbine}$.

Boiler (1 → 2)

We want to calculate the heat required, $Q_H = H_2 - H_1$. But we already have these values:

$$Q_H = \left[\begin{array}{l} 3396.8 \\ 3452.8 \\ 3509.0 \end{array}\right. \text{kJ/kg} - 343.9\,\text{kJ/kg} = \left[\begin{array}{l} 3052.9 \\ 3108.9 \\ 3165.9 \end{array}\right. \text{kJ/kg.}$$

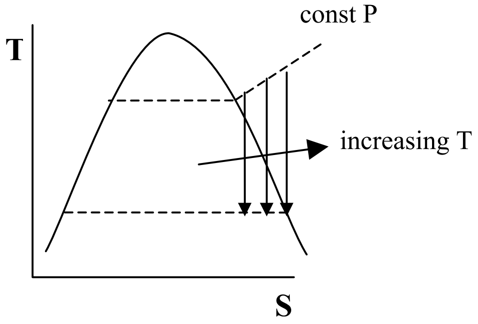
Putting it all together

Finally we can calculate the thermal efficiency:

$$\eta \equiv \frac{|W_{net}|}{|Q_H|} = \frac{|W_t| - |W_p|}{|Q_H|} = \left[\begin{array}{l} \mathbf{0.301} \\ \mathbf{0.305} \\ \mathbf{0.309} \end{array}\right. \quad \text{(ans)}$$

d)

We can see from the calculation above that increasing the turbine-inlet temperature will increase the quality of the outlet. We can also see this graphically.

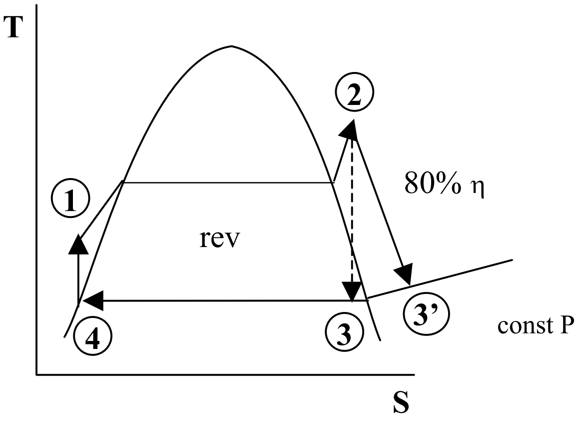


Increasing turbine inlet-temperature increases the thermal efficiency. Generally, for a power cycle, larger separation between T_{high} and T_{low} → higher thermal efficiency.

e) What if the turbine's efficiency is 80%?

$\eta = W_t / W_{t,rev}$

The process is represented on the T-S diagram →



We see that the only point affected by the change is point 3'. (Point 3 is the reversible turbine case). The work of the pump is the same as before, and so is the heat required by the boiler.

Then:

$$\eta \equiv \frac{|W_{net}|}{|Q_H|} = \frac{|W_t| - |W_p|}{|Q_H|} = \frac{\eta|W_{t,rev}| - |W_p|}{|Q_H|} = \frac{0.80|W_{t,rev}| - |W_p|}{|Q_H|}$$

$$W_{t,rev} = \left[\begin{array}{l} -921.3 \\ -951.2 \\ -982.1 \end{array}\right. = \text{kJ/kg.} \quad W_p = 3.35 \text{ kJ/Kg.} \quad Q_H = \left[\begin{array}{l} 3052.9 \\ 3108.9 \\ 3165.9 \end{array}\right. \text{kJ/kg.}$$

Plugging in these values, we get

$$\eta = \left[\begin{array}{l} \mathbf{0.240} \\ \mathbf{0.244} \\ \mathbf{0.247} \end{array}\right.$$

In general, the coefficient of performance of the cycle has dropped to roughly 80% of the reversible case. This is because $W_{pump} << W_{turbine}$. So the performance of the cycle is mainly determined by the efficiency of the turbine.

Equipment sizing: Since we now only produce 80% of the maximum work possible, we need to use more steam than before; to be precise, by a factor of 1.25 (= 1/0.8). The equipment, then, has to accommodate this higher flow rate of steam.

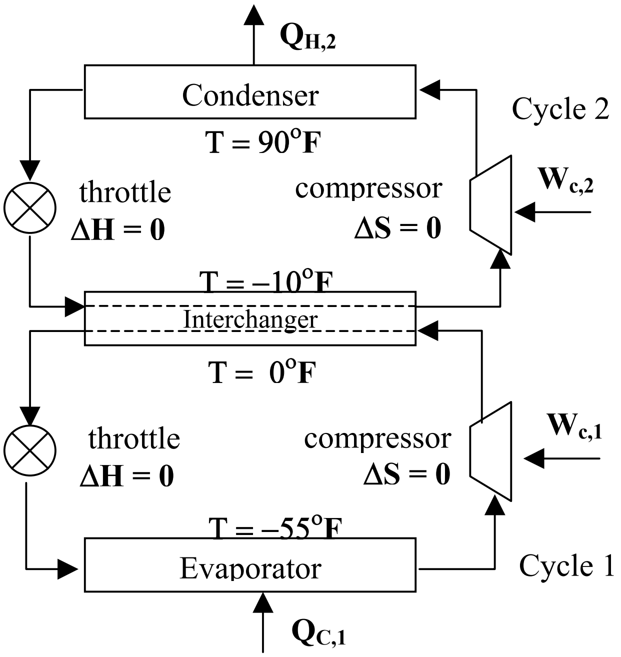
24)

Two-stage cascade refrigeration system.

Cycle 2:
Tetrafluoroethane
Between 0°F and 90°F.

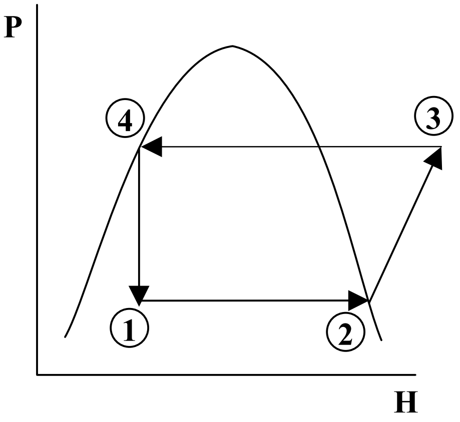
Compressors are 75% efficient.

Cycle 1:
R32 or ammonia
Between –55°F and 0°F.



Since we are given temperatures for both cycles, we can analyze each cycle separately.

Strategy:
We want to find the heat and shaft work terms, for which we need values of enthalpy for the points 1,2,3, and 4.
Refrigeration cycle (with throttle valve) on a P-H diagram →
Point 2 and 4 are usually defined in the problem ("anchor points") because they are saturated vapor and saturated liquid, respectively.
We can start with point 4 and identify point 1.
Then we can start with point 2 and identify point 3.



Note: This solution will make a lot more sense if you follow it using Table 9.1 and Figure G.2.

Cycle 2 analysis:

Throttle valve (4 → 1)
Inlet (point 4): Saturated liquid at 90°F. (temperature specified in problem statement)
Table 9.1 (or Figure G.2) gives us for 90°F: $P_4 = P^{sat} = 119$ psia; the enthalpy = $H_4 = 41.43$ Btu/lbm.
Outlet (point 1): Mixture of vapor and liquid.
For throttle valve $\Delta H = 0$. Then $H_1 = H_4 = 41.43$ Btu/lbm. There is no heat or work produced.

Evaporator (1 → 2)
Outlet (point 2): Saturated vapor at –10°F. (temperature specified in problem statement)
Table 9.1 (or Figure G.2): $P_2 = P^{sat} = 16.626$ psia; the enthalpy = $H_2 = 101.542$ Btu/lbm.
Since there is no work done, $Q = \mathbf{Q_{C,2}} = \Delta H = H_2 - H_1 = (101.5 - 41.43)$ Btu/lbm = **60.112 Btu/lbm.**

Compressor (2 → 3)
We need to calculate the work of a reversible compressor first (to use the 75% efficiency information).

For a reversible compressor, $\Delta S = 0$, or $S_{3*} = S_2$. We want to compress from 16.626 psia to 119 psia.
On Figure G.2: We follow a constant entropy line from point 2 (sat. vap. at 16.626 psia) to 119 psia.
This gives us point 3* where H_{3*} = 120 Btu/lbm (and $T_{3*} = 116^{o}F$).
For this reversible compressor, $W_{c,2,rev} = \Delta H = H_{3*} - H_2$ = (120 – 101.5) Btu/lbm = 18.5 Btu/lbm.

For our *real* compressor, η = 75%. Then $\mathbf{W_{c,2}} = W_{c,rev} / 0.75$ **= 24.611 Btu/lbm**.

W required by real compressor always larger than W required by reversible compressor. Thus we divide, not multiply, by 0.75.

Putting it all together
For completion sake, we can calculate $Q_{H,2}$ (heat released in the condenser).
Since for a cycle ΔH must be 0, $W_c + Q_H + Q_C = 0$.
Then $Q_{H,2} = -(W_{c,2} + Q_{C,2})$ = -84.723 Btu/lbm

We are ready to answer the questions for cycle 2.

The pressure limits, as we have seen, are the P^{sat} at the specified temperatures:
16.6 psia at $-10^{o}F$ and **119 psia** at $90^{o}F$ (ans)

Coefficient of performance = COP = $|Q_c| / |W_{net}| = |Q_C| / |W_c|$ (since the valve produces no work)

For cycle 2, $\mathbf{COP_2}$ = (60.1 Btu/lbm) / (24.6 Btu/lbm) = **2.44** (ans)

Cycle 1 analysis: (this solution will show only the method **for ammonia**; method for R32 is same)

Throttle valve (4 → 1) ammonia
Inlet (point 4): Saturated liquid at $0^{o}F$. (temperature specified in problem statement)
NIST webbook gives us for $0^{o}F$: $P_4 = P^{sat}$ = 30.397 psia; the enthalpy = H_4 = 112.61 Btu/lbm.
Outlet (point 1): Mixture of vapor and liquid.
For throttle valve $\Delta H = 0$. Then $H_1 = H_4$ = 112.61 Btu/lbm. There is no heat or work produced.

Evaporator (1 → 2) ammonia
Outlet (point 2): Saturated vapor at $-55^{o}F$. (temperature specified in problem statement)
NIST webbook: $P_2 = P^{sat}$ = 6.5312 psia; the enthalpy = H_2 = 661.31 Btu/lbm.
Since there is no work done, $Q = \mathbf{Q_{C,1}} = \Delta H = H_2 - H_1$ = (661.31 – 112.61) Btu/lbm = **548.7 Btu/lbm.**

Compressor (2 → 3) ammonia

We need to calculate the work of a reversible compressor first (to use the 75% efficiency information).

For a reversible compressor, $\Delta S = 0$, or $S_{3*} = S_2$. We want to compress from 6.5312 psia to 30.397 psia.
On NIST webbook, we find $S_2 = 1.6436$ Btu/lbm R (entropy of saturated vapor at 6.5312).
Then we search at P = 30.397 psia for the point where $S_{3*} = S_2$.
This is the point where $T_{3*} = 121^{o}$F and $H_{3*} = 747.5$ Btu/lbm.

For this reversible compressor, $W_{c,1,rev} = \Delta H = H_{3*} - H_2 = (747.5 - 661.31)$ Btu/lbm = 86.19 Btu/lbm.

For our *real* compressor, $\eta = 75\%$. Then $\mathbf{W_{c,1}} = W_{c,rev} / 0.75$ **= 114.92 Btu/lbm**.

Putting it all together (ammonia)

For completion sake, we can calculate $Q_{H,1}$ (heat released in the condenser).
Since for a cycle ΔH must be 0, $W_c + Q_H + Q_C = 0$.
Then $Q_{H,1} = -(W_{c,1} + Q_{C,1}) = -663.62$ Btu/lbm
We are ready to answer the questions for cycle 2.
The pressure limits, as we have seen, are the P^{sat} at the specified temperatures:
6.5312 psia at -55^{o}F and **30.397** at 0^{o}F (ans)

Coefficient of performance = COP = $|Q_c| / |W_{net}| = |Q_C| / |W_c|$ (since the valve produces no work)
For cycle 1 (ammonia), $\mathbf{COP_1}$ = (548.7 Btu/lbm) / (114.92 Btu/lbm) = **4.775** (ans)

For R32

We can follow exactly the same method as above with the data for R32. The results are below.

Summarizing the result for the three cycles:

Cycle	P_{low} (psia)	P_{high} (psia)	W_c (Btu/lbm)	Q_C (Btu/lbm)	Q_H (Btu/lbm)	COP
1 – ammonia	6.5	30.4	114.9	548.7	-663.6	4.78
1 - R32	17.4	63.9	29.2	141.3	-170.5	4.83
2 – tetrafluoroethane	16.6	119	24.6	60.1	-84.7	2.44

c)

If cycle 1 uses ammonia:

Q_C, as calculated above, has units of Btu/lbm.

Cooling rate of 200 Btu/s at –55°F.
Flow rate of ammonia = m_1 = (cooling per second) / (cooling per pound)
= (200 Btu/s) / (Q_C) = 200 / 548.7 = **0.364 lbm ammonia/s** (ans)

How are the two cycles connected? By the heat exchanger between the two.
The heat released from the condenser in cycle 1 must equal the heat used by the evaporator in cycle 2.
Or $m_1|Q_{H,1}| = m_2|Q_{C,2}|$ (this way, the units are correct: this-many Btu/s equals this-many Btu/s)
Flow rate of tetrafluoroethane = $m_2 = m_1 (|Q_{H,1}| / |Q_{C,2}|)$ = **4.024 lbm tetrafluoroethane/s** (ans)

Power requirement of each compressor:
Cycle 1: Power = $m_1 W_{c,1}$= (0.364 lbm ammonia /s)(114.9 Btu/lbm ammonia) = **41.89 Btu/s** (ans)
Cycle 2: Power = $m_2 W_{c,2}$= (4.024 lbm tetrafluoro./s)(24.6 Btu/lbm tetrafluoro.) = **99.03 Btu/s** (ans)

Heat output from 1 = $m_1 |Q_{H,1}|$ = (0.364 lbm ammonia/s)(663.6 Btu/lbm ammonia) = **241. 9 Btu/s** (ans)

Overall COP = $(m_1 |Q_{C,1}|) / (m_1 |W_{c,1}| + m_2 |W_{c,2}|)$
= (200 Btu/s) / (41.89 Btu/s + 99.03 Btu/s)
= **1.42** (ans)

If cycle 1 uses R32:

Q_C, as calculated above, has units of Btu/lbm.

Cooling rate of 200 Btu/s at –55°F.
Flow rate of R32 = m_1 = (cooling per second) / (cooling per pound)
= (200 Btu/s) / (Q_C) = 200 / 141.036= **1.415 lbm R32/s** (ans)

How are the two cycles connected? By the heat exchanger between the two.
The heat released from the condenser in cycle 1 must equal the heat used by the evaporator in cycle 2.
Or $m_1|Q_{H,1}| = m_2|Q_{C,2}|$ (this way, the units are correct: this-many Btu/s equals this-many Btu/s)
Flow rate of tetrafluoroethane = $m_2 = m_1 (|Q_{H,1}| / |Q_{C,2}|)$ = **4.015 lbm tetrafluoroethane/s** (ans)

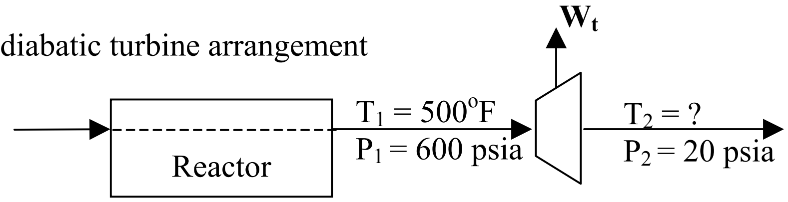
Power requirement of each compressor:
Cycle 1: Power = $m_1 W_{c,1}$= (4.015 lbm R32/s)(29.213 Btu/lbm R32) = **41.35 Btu/s** (ans)
Cycle 2: Power = $m_2 W_{c,2}$= (4.015 lbm tetrafluoro./s)(24.6 Btu/lbm tetrafluoro.) = **98.81 Btu/s** (ans)

Heat output from 1 = $m_1 |Q_{H,1}|$ = (0.364 lbm R32/s)(663.6 Btu/lbm R32) = **241.35 Btu/s** (ans)

Overall COP = $(m_1 |Q_{C,1}|) / (m_1 |W_{c,1}| + m_2 |W_{c,2}|)$
= (200 Btu/s) / (41.35 Btu/s + 98.81 Btu/s)
= **1.43** (ans)

25)

a) One adiabatic turbine arrangement

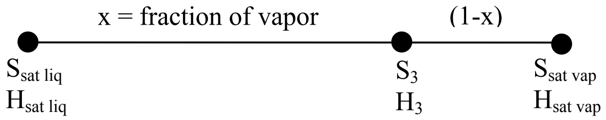


The maximum work that can be produced is when the turbine is reversible → $\Delta S = 0$
To compute the maximum work, let us look at the case of reversible turbine.
(See previous problem sets for more detailed solution to turbine/compressor problems).

$T_1 = 500^oF$, $P_1 = 600$ psia. This is supersaturated steam because $P > P^{sat}(500^oF)$.
Supersaturated steam table gives for this condition: **$H_1 = 1215.9$ Btu/lbm, $S = 1.4590$ Btu/lbm R**
(These numbers are important because we keep using them for the latter part of the problem).

$\Delta S = 0 \rightarrow S_2 = S_1 = 1.4590$ Btu/lbm R.
When we look in the steam table for $P_2 = 20$ psia we see that this entropy value lies in between $S_{sat\ liq} = 0.3358$ Btu/lbm R and $S_{sat\ vap} = 1.7320$ Btu/lbm R.
Therefore the outlet of the turbine must be a mixture of liquid and vapor. To find the quality:

$$x = \frac{S_2 - S_{sat\ liq}}{S_{sat\ vap} - S_{sat\ liq}} = \frac{(1.4590 - 0.3358)\ \text{Btu/lbm R}}{(1.7320 - 0.3358)\ \text{Btu/lbm R}} = 0.804$$



The quality (x) allows us to calculate H_2:

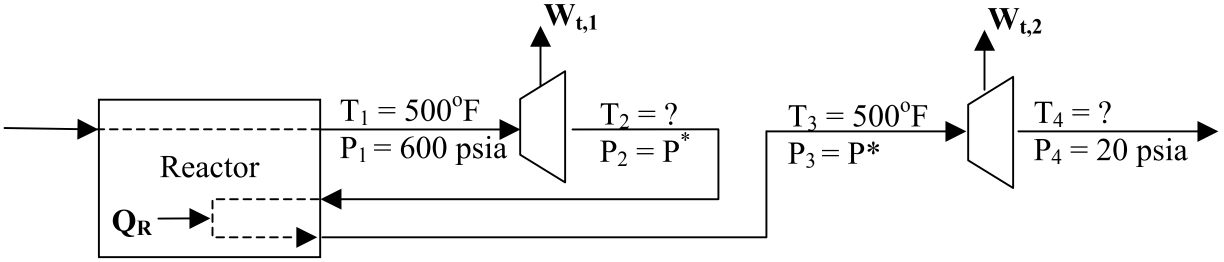
$$H_2 = x \cdot H_{sat\ vap} + (1-x)H_{sat\ liq} = (0.804)(1156.3\ \text{Btu/lbm}) + (1 - 0.804)(196.27\ \text{Btu/lbm}) = 968.6\ \text{Btu/lbm}$$

These enthalpy values from steam table saturated condition at 20 psia.

Since the turbine is adiabatic, $\Delta H = W_s + Q = W_s$
$W_s = \Delta H = H_2 - H_1 = (968.6 - 1215.9)$ Btu/lbm = **-247.3 Btu/lbm** (ans)

$W_s < 0$ for turbine because the system produces work

b) Two-turbine arrangement



There must be some heat Q_R that gets transferred from reactor to the steam to raise its temperature from T_2 (unknown for now) to 500^oF again. The process, we're told, occurs at constant $P = P^*$.

c) P* = 310 psia. Our strategy is the same as before, except with different P's and two turbines.

First turbine

We have the same inlet as before, so S_1 and H_1 are same as in part a).

$\Delta S = 0 \rightarrow S_2 = S_1 = 1.4590$ Btu/lbm R

Steam table for $P_2 = 310$ psia: $S_{\text{sat liq}} = 0.592$ Btu/lbm R and $S_{\text{sat vap}} = 1.5076$ Btu/lbm R.

Therefore the outlet of the turbine must be a mixture of liquid and vapor.

$$x = \frac{S_2 - S_{\text{sat liq}}}{S_{\text{sat vap}} - S_{\text{sat liq}}} = \frac{(1.4590 - 0.5920)\ \text{Btu/lbm R}}{(1.5076 - 0.5920)\ \text{Btu/lbm R}} = 0.947$$

This all looks kind of familiar, doesn't it?

$$H_2 = x \cdot H_{\text{sat vap}} + (1 - x)H_{\text{sat liq}} = (0.947)(1203.2\ \text{Btu/lbm}) + (1 - 0.947)(397.3\ \text{Btu/lbm}) = 1160.4\ \text{Btu/lbm}$$

$W_{t,1} = \Delta H = H_2 - H_1 = (1160.4 - 1215.9)$ Btu/lbm = -55.5 Btu/lbm

Second turbine

Inlet now is at $P_3 = 310$ psia and $T_3 = 500^{\circ}$F. Looking up in steam table for this condition:
$H_3 = 1256.5$ Btu/lbm and $S_3 = 1.5657$ Btu/lbm R.

$\Delta S = 0 \rightarrow S_4 = S_3 = 1.5657$ Btu/lbm R.

Steam table for $P_4 = 20$ psia: $S_{\text{sat liq}} = 0.3358$ Btu/lbm R and $S_{\text{sat vap}} = 1.7320$ Btu/lbm R.

Therefore the outlet of the turbine must be a mixture of liquid and vapor.

$$x = \frac{S_4 - S_{\text{sat liq}}}{S_{\text{sat vap}} - S_{\text{sat liq}}} = \frac{(1.5657 - 0.3358)\ \text{Btu/lbm R}}{(1.7320 - 0.3358)\ \text{Btu/lbm R}} = 0.881$$

$$H_2 = x \cdot H_{\text{sat vap}} + (1 - x)H_{\text{sat liq}} = (0.881)(1156.3\ \text{Btu/lbm}) + (1 - 0.881)(196.27\ \text{Btu/lbm}) = 1042.0\ \text{Btu/lbm}$$

$W_{t,2} = \Delta H = H_4 - H_3 = (1042.0 - 1256.5)$ Btu/lbm = -214.5 Btu/lbm

The total work = $W_{t,1} + W_{t,2}$ = (-55.5 Btu/lbm) + (-214.5 Btu/lbm) = **-270.0 Btu/lbm** (ans)

Jumping ahead to problem e) for P = 310 psia*

e) What is the heat absorbed during the reheating process?

The heat absorbed (Q_R, see diagram in part b) will be ΔH because no work is done during reheating:

$Q_R + W_s = \Delta H$

$Q_R = \Delta H = H_3 - H_2 = 1256.5$ Btu/lbm – 1160.4 Btu/lbm = **-173.9 Btu/lbm** (ans)

d) P* = 35 psia. Our strategy is the same as before, except with different P's and we have two turbines.

First turbine

We have the same inlet as before, so S_1 and H_1 are same as in part a).

$\Delta S = 0 \rightarrow S_2 = S_1 = 1.4590$ Btu/lbm R

Steam table for $P_2 = 35$ psia: $S_{\text{sat liq}} = 0.3809$ Btu/lbm R and $S_{\text{sat vap}} = 1.6872$ Btu/lbm R.

Therefore the outlet of the turbine must be a mixture of liquid and vapor.

$$x = \frac{S_2 - S_{\text{sat liq}}}{S_{\text{sat vap}} - S_{\text{sat liq}}} = \frac{(1.4590 - 0.3809)\ \text{Btu/lbm R}}{(1.6872 - 0.3809)\ \text{Btu/lbm R}} = 0.825$$

$$H_2 = x \cdot H_{\text{sat vap}} + (1-x)H_{\text{sat liq}} = (0.825)(1167.1\ \text{Btu/lbm}) + (1 - 0.825)(228.03\ \text{Btu/lbm}) = 1003.1\ \text{Btu/lbm}$$

$W_{t,1} = \Delta H = H_2 - H_1 = (1003.1 - 1215.9)$ Btu/lbm = -212.8 Btu/lbm

Second turbine

Inlet now is at $P_3 = 35$ psia and $T_3 = 500^{\circ}$F. Looking up in steam table for this condition:
$H_3 = 1285.5$ Btu/lbm and $S_3 = 1.8294$ Btu/lbm R.

$\Delta S = 0 \rightarrow S_4 = S_3 = 1.8294$ Btu/lbm R.

Steam table for $P_4 = 20$ psia: $S_{\text{sat liq}} = 0.3358$ Btu/lbm R and $S_{\text{sat vap}} = 1.7320$ Btu/lbm R.

$S_4 > S_{\text{sat vap}}$ (20 psia) → The outlet of the turbine must be vapor.

Looking up in steam table at P = 20 psia and for S = 1.7320 Btu/lbm R, we find it lies between 350°F and 400°F. Using linear interpolation (very similar to the way we find quality), we find that the temperature T_4 is approximately 382°F. And at this condition, $H_4 = 1230.7$ Btu/lbm

$W_{t,2} = \Delta H = H_4 - H_3 = (1230.7 - 1285.5)$ Btu/lbm = -54.8 Btu/lbm

The total work = $W_{t,1} + W_{t,2}$ = (-212.8 Btu/lbm) + (-54.8 Btu/lbm) = **-267.6 Btu/lbm** (ans)

e) What is the heat absorbed during the reheating process?
We calculated for the case of P* = 310 psia, that Q_R = **-173.9 Btu/lbm** (see previous page).
Similarly for the case of P* = 35 psia,

The heat absorbed (Q_R, see diagram in part b) will be ΔH because no work is done during reheating:

$Q_R + W_s = \Delta H$

$Q_R = \Delta H = H_3 - H_2$ = 1285.5 Btu/lbm – 1003.1 Btu/lbm = **-282.4 Btu/lbm** (ans)

f) and g)

The work produced by the turbine can be calculated by measuring the change in enthalpy for the vertical lines. Naturally, we should get the same answer even if we use the Mollier diagram.