# CAE 208 / MMAE 320: Thermodynamics

Fall 2023

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## November 28, 2023

Entropy (4) and vapor compression cycles (1)

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# ANNOUNCEMENTS

# Announcements

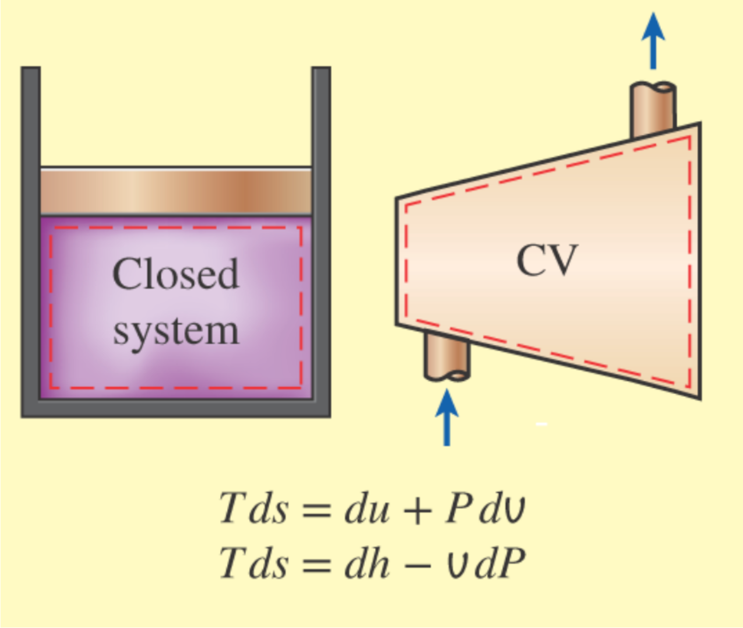
- Assignment 10 is posted (Due 12/01/23 for those who need to submit it).

- The bonus activities document is posted. Please pay attention to the deadlines and also the updates about this task (an idea submission by the 11/29/2023 is required)

# RECAP

# Recap

- Use the first T-ds (or Gibbs) equation to solve for entropy changes



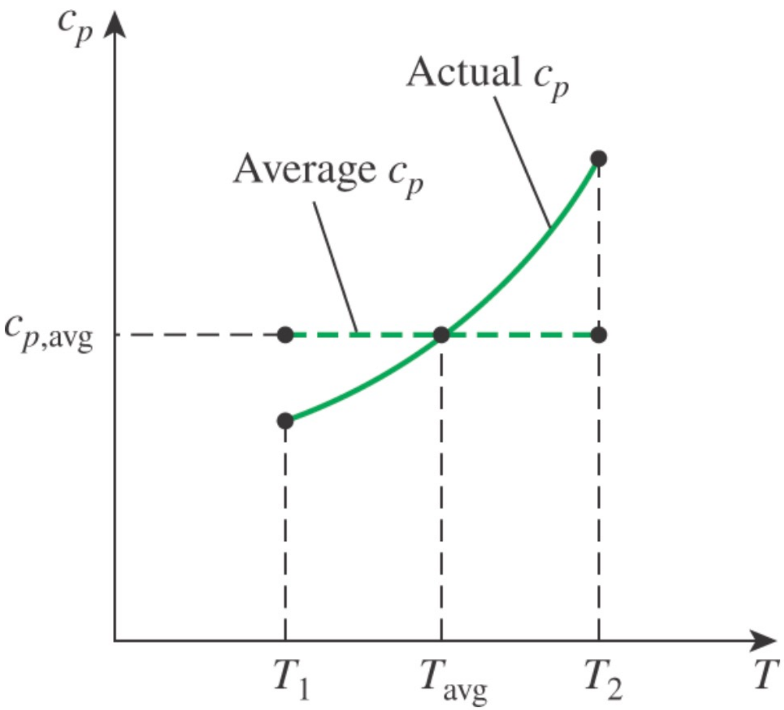
$$Tds = du + P\,dv$$
$$Tds = dh - v\,dP$$

# Recap

- Approach 1: Constant Specific Heats (Approximate Analysis):

$$s_2 - s_1 = c_{v,avg} \ln\left(\frac{T_2}{T_1}\right) + R \times ln\left(\frac{v_2}{v_1}\right)$$

$$s_2 - s_1 = c_{p,avg} \times \ln\left(\frac{T_2}{T_1}\right) - R \times \ln(\frac{P_2}{P_1})$$



# Recap

- Approach 2: Variable Specific Heats (Exact Analysis):

$$s^0 = \int_0^T c_p(T)\frac{dT}{T}$$

$$\int_0^T c_p(T)\frac{dT}{T} = s_2^0 - s_1^0$$

$$s_2 - s_1 = s_2^0 - s_1^0 - R\times\ln(\frac{P_2}{P_1})$$

$$\bar{s_2} - \bar{s_1} = \overline{s_2^0} - \overline{s_1^0} - R_u\times\ln(\frac{P_2}{P_1})$$

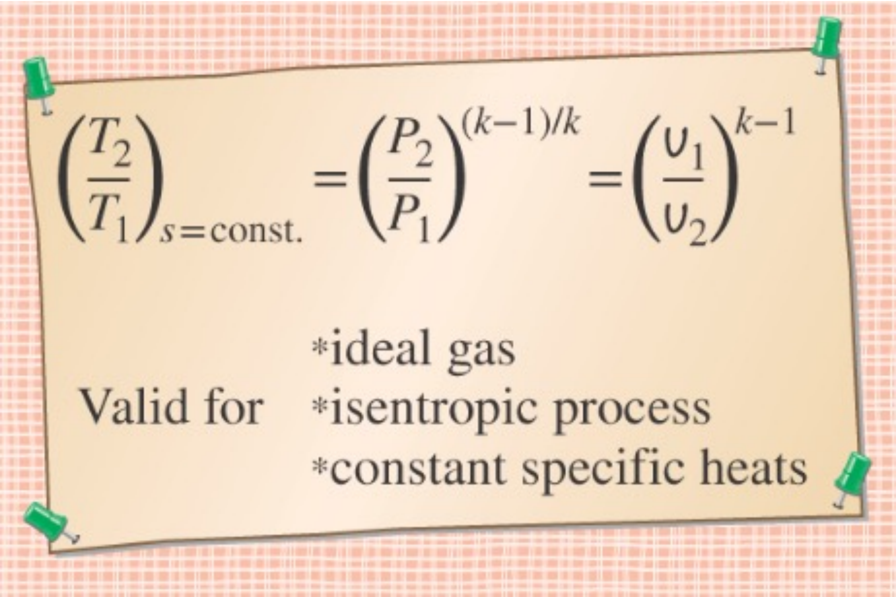
# Recap

- For Isentropic Processes of Ideal Gases, we have:

$$Tv^{k-1} = Constant$$

$$TP^{\frac{1-k}{k}} = Constant$$

$$Pv^{k} = Constant$$



$$\left(\frac{T_2}{T_1}\right)_{s=\text{const.}} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

Valid for
*ideal gas
*isentropic process
*constant specific heats

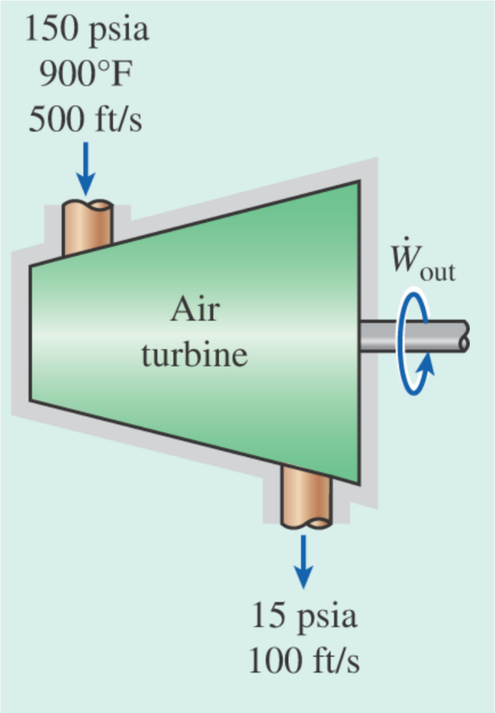
# CLASS ACTIVITY

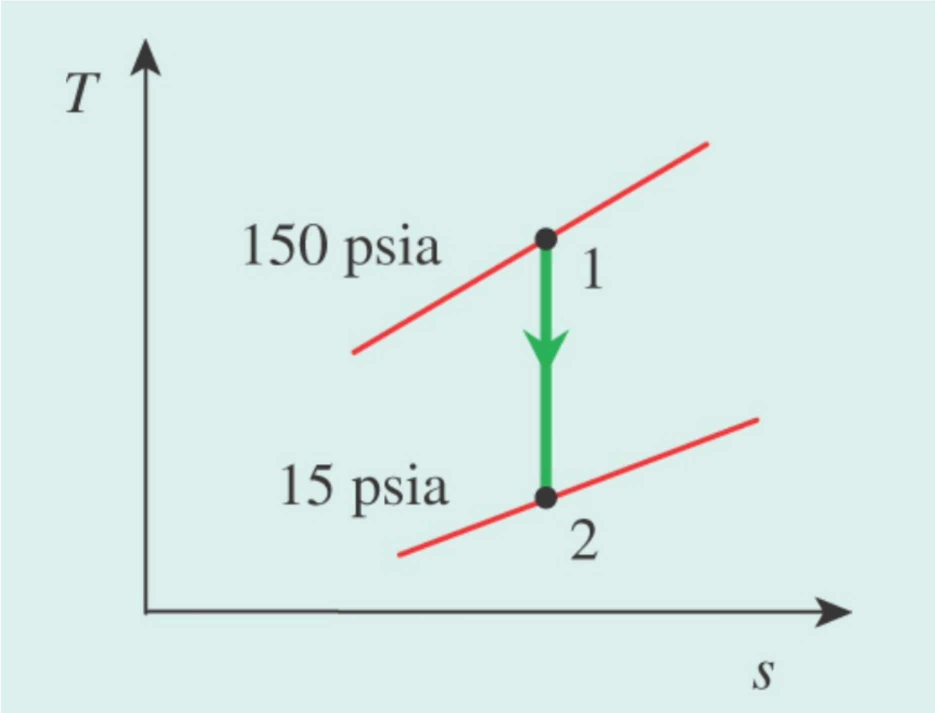
# Class Activity

- Air enters an isentropic turbine at 150 psia and 900 °F through a 0.5 ft$^2$ inlet section with a velocity of 100 ft/s. It leaves at 15 psia with a velocity of 500 ft/s. Calculate the air temperature at the turbine exit and the power produced, in hp, by this turbine.

# Class Activity

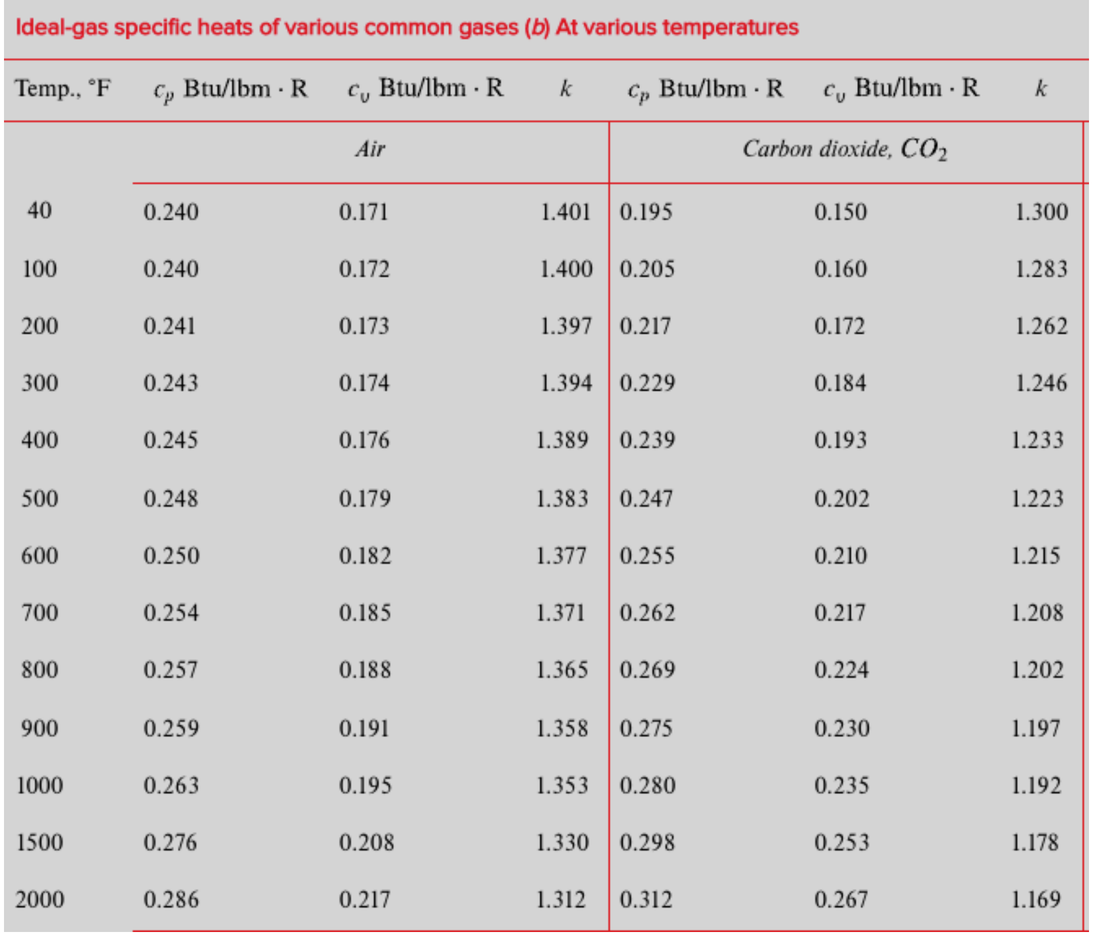
- Solution (assumptions):
  - ❑ Steady flow
  - ❑ The process is isentropic (both reversible and adiabatic)
  - ❑ Ideal gas with a constant specific heat





# Class Activity

- Solution (Tables):
  - ❑ Table A-2Eb: $@900\ °F \rightarrow c_p = 0.259 \frac{Btu}{lbm{-}R}$ $and\ k = 1.358$



Ideal-gas specific heats of various common gases (*b*) At various temperatures

| Temp., °F | $c_p$ Btu/lbm · R | $c_v$ Btu/lbm · R | $k$ | $c_p$ Btu/lbm · R | $c_v$ Btu/lbm · R | $k$ |
|---|---|---|---|---|---|---|
| | *Air* | | | *Carbon dioxide, $CO_2$* | | |
| 40 | 0.240 | 0.171 | 1.401 | 0.195 | 0.150 | 1.300 |
| 100 | 0.240 | 0.172 | 1.400 | 0.205 | 0.160 | 1.283 |
| 200 | 0.241 | 0.173 | 1.397 | 0.217 | 0.172 | 1.262 |
| 300 | 0.243 | 0.174 | 1.394 | 0.229 | 0.184 | 1.246 |
| 400 | 0.245 | 0.176 | 1.389 | 0.239 | 0.193 | 1.233 |
| 500 | 0.248 | 0.179 | 1.383 | 0.247 | 0.202 | 1.223 |
| 600 | 0.250 | 0.182 | 1.377 | 0.255 | 0.210 | 1.215 |
| 700 | 0.254 | 0.185 | 1.371 | 0.262 | 0.217 | 1.208 |
| 800 | 0.257 | 0.188 | 1.365 | 0.269 | 0.224 | 1.202 |
| 900 | 0.259 | 0.191 | 1.358 | 0.275 | 0.230 | 1.197 |
| 1000 | 0.263 | 0.195 | 1.353 | 0.280 | 0.235 | 1.192 |
| 1500 | 0.276 | 0.208 | 1.330 | 0.298 | 0.253 | 1.178 |
| 2000 | 0.286 | 0.217 | 1.312 | 0.312 | 0.267 | 1.169 |

# Class Activity

- Solution (Tables):
  - Table A-1E: $R = 0.3704 \frac{psia - ft^3}{lbm - R}$

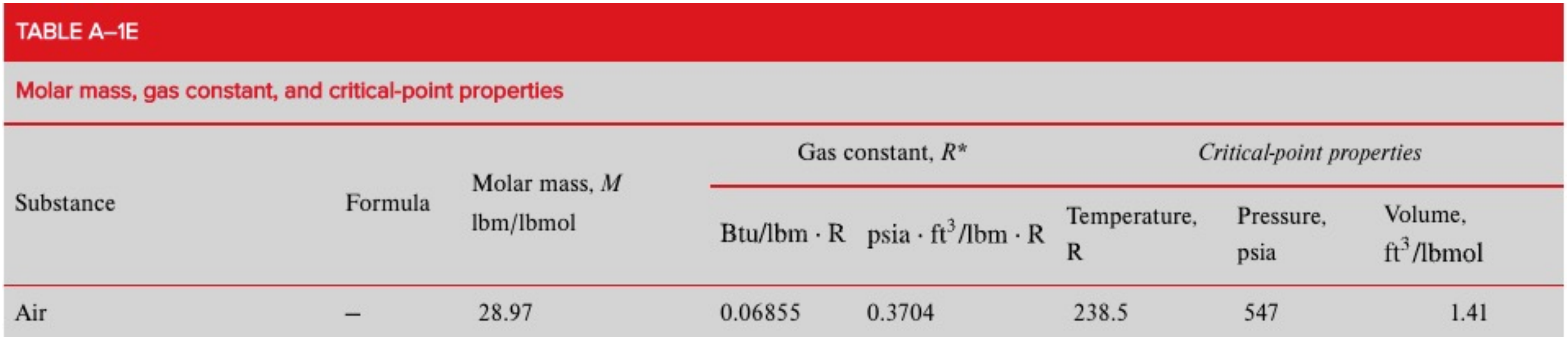


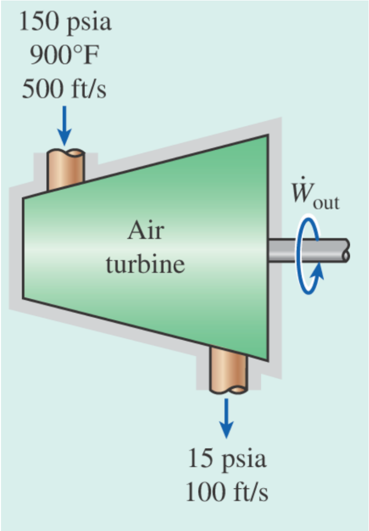
**TABLE A–1E**

Molar mass, gas constant, and critical-point properties

| Substance | Formula | Molar mass, *M* lbm/lbmol | Gas constant, *R** | | *Critical-point properties* | | |
|---|---|---|---|---|---|---|---|
| | | | Btu/lbm · R | psia · $ft^3$/lbm · R | Temperature, R | Pressure, psia | Volume, $ft^3$/lbmol |
| Air | – | 28.97 | 0.06855 | 0.3704 | 238.5 | 547 | 1.41 |

# Class Activity

- Solution (Problem solving):



$$\dot{m} = \dot{m}_1 = \dot{m}_2$$

$$\dot{E}_{in} - \dot{E}_{out} = \frac{d\dot{E}_{system}}{dt} = 0$$

$$\dot{m}(h_1 + V_1^2) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_{out}$$

$$\dot{W}_{out} = \dot{m}\left(h_1 - h_1 + \frac{V_1^2 - V_2}{2}\right)$$

# Class Activity

- Solution (Calculations):

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \rightarrow T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = (900 + 460\ R)\left(\frac{15\ psia}{150\ psia}\right)^{\frac{1.358-1}{1.358}} = 741\ R$$

$$v_1 = \frac{RT_1}{P_1} = \frac{\left(0.3704\,\frac{psia - ft^3}{lbm - R}\right)(900 + 460\ R)}{150\ psia} = 3.358\,\frac{ft^3}{lbm}$$

$$\dot{m} = \frac{A_1 V_1}{v_1} = \frac{(0.5\ ft^2)\left(500\,\frac{ft}{s}\right)}{3.358\,\frac{ft^3}{lbm}} = 74.45\,\frac{lbm}{s}$$

# Class Activity

- Solution (Calculations):

$$\dot{W}_{out} = \dot{m}\left(h_1 - h_1 + \frac{V_1^2 - V_2}{2}\right)$$

$$\dot{W}_{out} = \left(74.45\,\frac{lbm}{s}\right)[\left(0.250\,\frac{Btu}{lbm - R}\right)(1360 - 724R) + \left(\frac{\left(500\,\frac{ft}{s}\right)^2}{2} - \frac{\left(100\,\frac{ft}{s}\right)^2}{2}\right)\left(\frac{\frac{1\,\frac{Btu}{lbm}}{25.037}\,ft^2}{s^2}\right)$$

$$\dot{W}_{out} = 12{,}194\,\frac{Btu}{s}\left(\frac{1\,hp}{0.7068\,\frac{Btu}{s}}\right) = 17{,}250\,hp$$

# REVERSIBLE STEADY-FLOW WORK

# Reversible Steady-Flow Work

- Recall we had this for the closed system:

$$W_b = \int_1^2 P\, dV$$

- For steady flow:

$$\delta q_{rev} - \delta w_{rev} = dh + dke + dpe$$

$$\uparrow$$

$$\longrightarrow \quad -\delta w_{rev} = vdP + dke + dpe$$

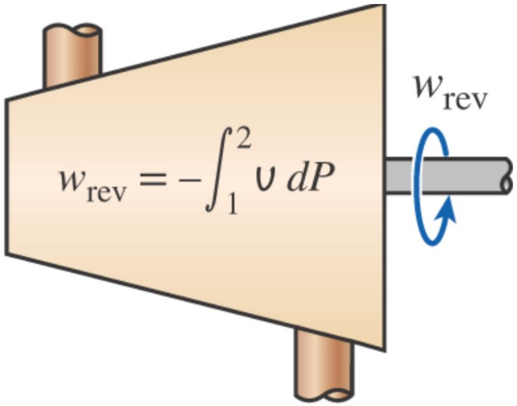
$$\left.\begin{aligned} \delta q_{rev} &= Tds \\ Tds &= dh - vdP \end{aligned}\right\} \rightarrow \delta q_{rev} = dh - vdP$$

$$w_{rev} = -\int_1^2 vdP - \Delta ke - \Delta pe$$

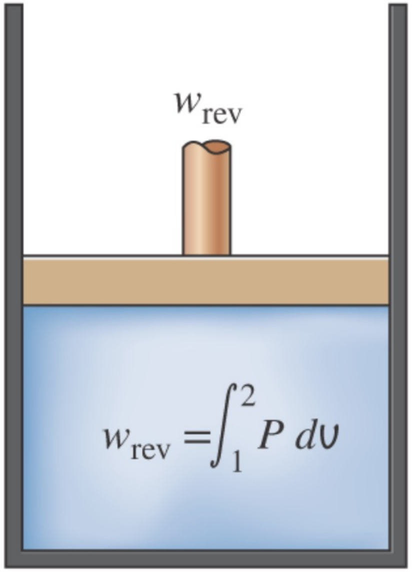
# Reversible Steady-Flow Work

- For steady flow:

$$w_{rev} = -\int_1^2 v dP - \Delta ke - \Delta pe$$



(*a*) Steady-flow system

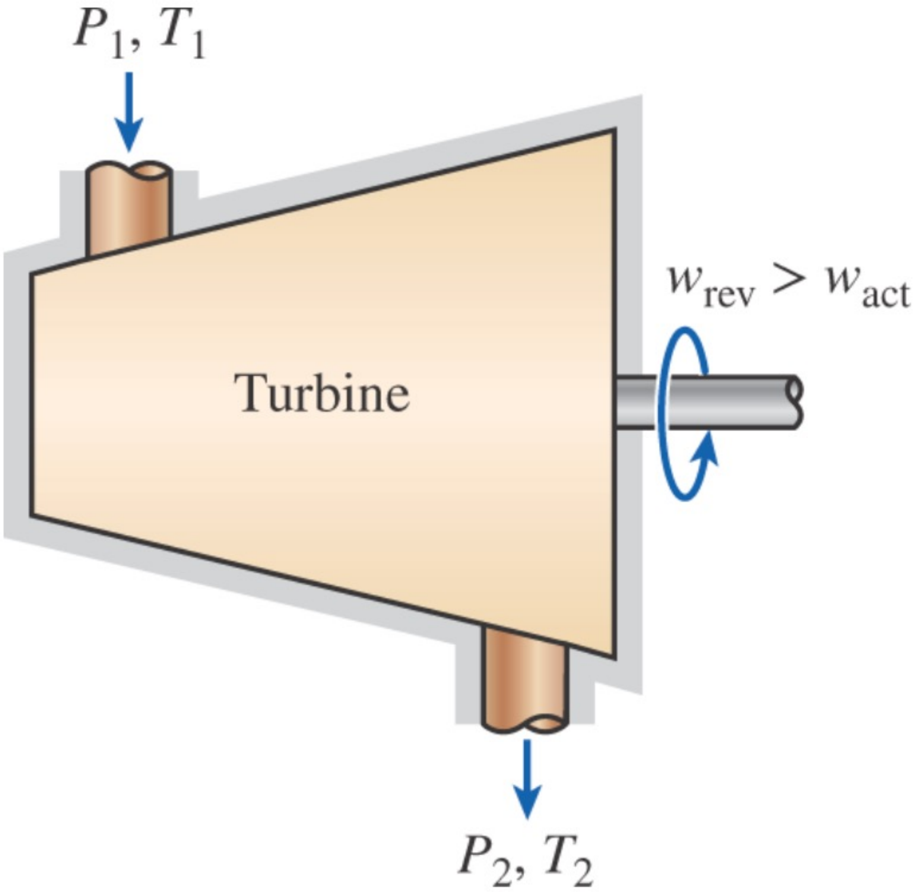


(*b*) Closed system

# ISENTROPIC EFFICIENCIES OF STEADY-FLOW DEVICES

# Isentropic Efficiencies

- Steady-flow devices deliver the most and consume the least work when the process is reversible:

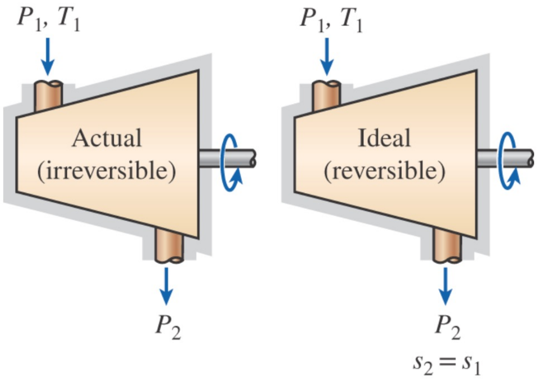


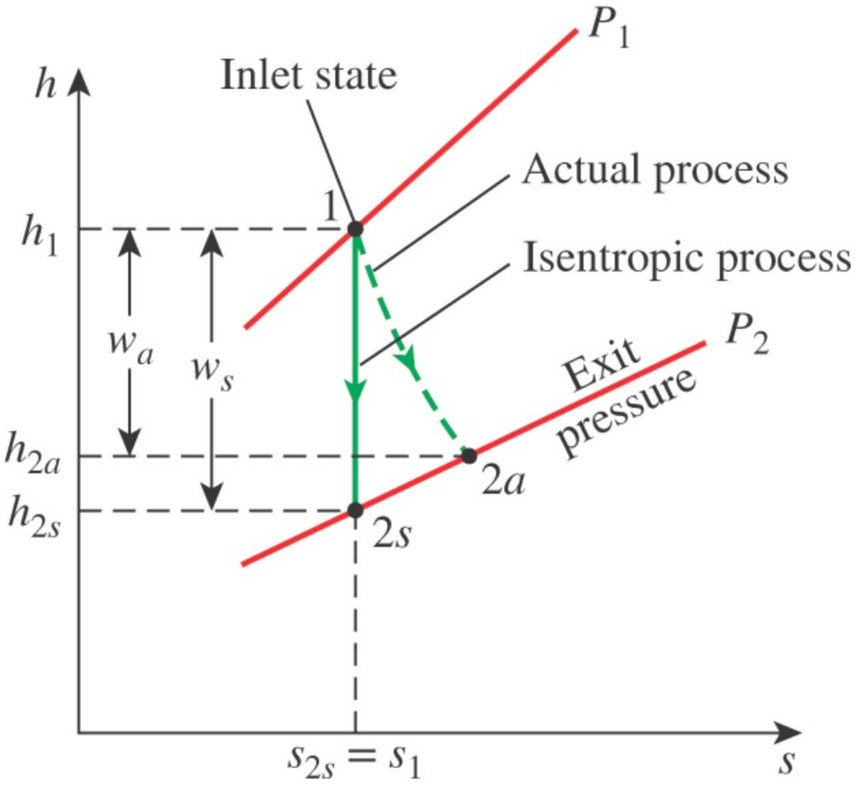
# Isentropic Efficiencies

- Isentropic efficiency of a turbine can be written as:

$$\eta_T = \frac{Actual\ turbine\ work}{Isentropic\ turbine\ work} = \frac{w_a}{w_s}$$

$$\eta_T \cong \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$



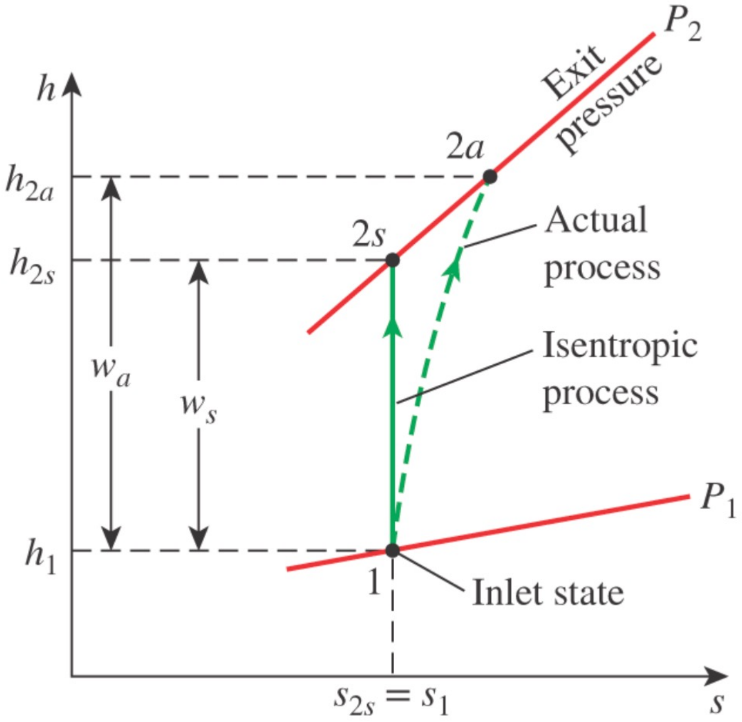


# Isentropic Efficiencies

- Isentropic efficiency of compressors and pumps

$$\eta_C = \frac{Isentropic\ compressor\ work}{Actual\ compressor\ work} = \frac{w_s}{w_a}$$

$$\eta_C \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$



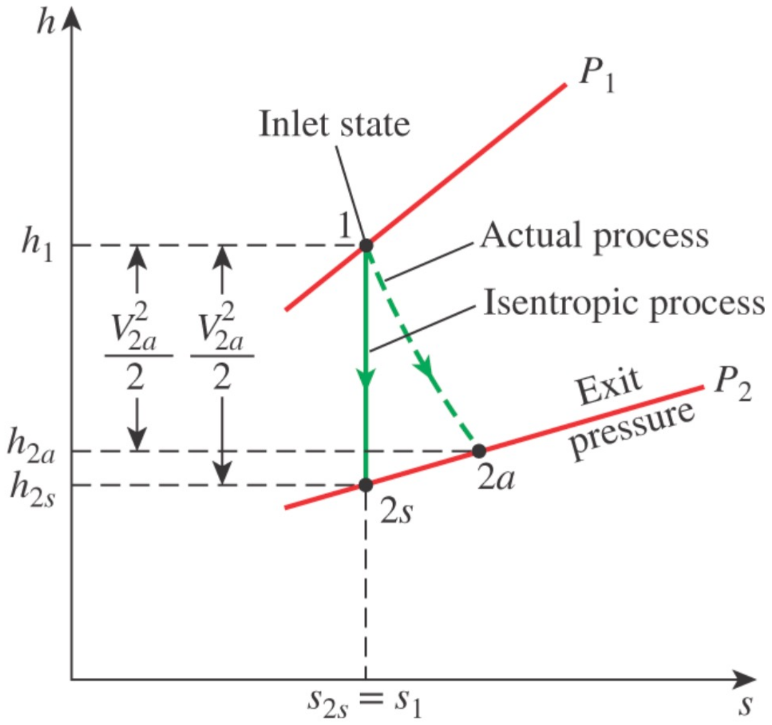
# Isentropic Efficiencies

- Isentropic efficiency of nozzles

$$\eta_N = \frac{Actual\ KE\ nozzle\ exit}{Isentropic\ KE\ at\ nozzle\ exit} = \frac{V_{2a}^2}{V_{2s}^2}$$

$$h_1 = h_{2a} + \frac{V_{2a}^2}{2}$$

$$\eta_N \cong \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$



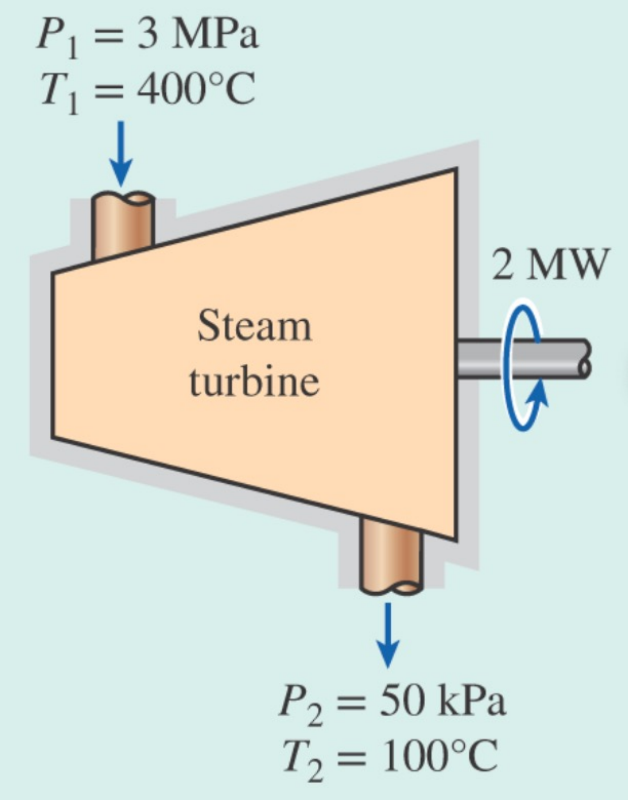
# CLASS ACTIVITY

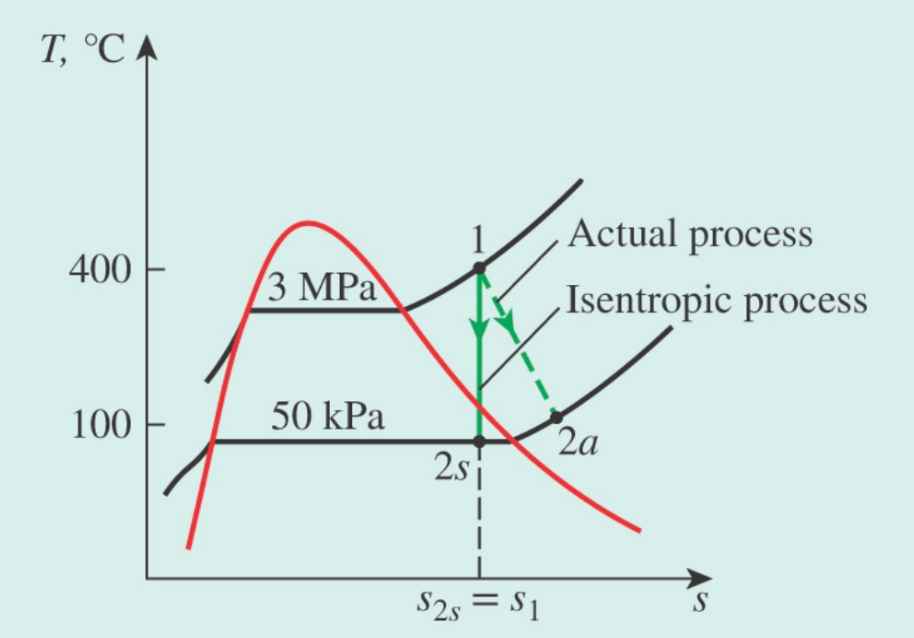
# Class Activity

- Steam enters an adiabatic turbine steadily at 3 MPa and 400 C and leaves at 50 kPa and 100 C. If the power output of the turbine is 2 MW, determine:
  a) The isentropic efficiency of the turbine
  b) The mass flow rate of the steam flowing through the turbine

# Class Activity

- Solution (assumptions):
  - ❑ Steady operating conditions exist
  - ❑ The kinetic and potential energies are negligible





# Class Activity

- Solution (Tables):

$$State\ 1: \quad \begin{cases} P_1 = 3\ MPa \\ T_1 = 400\ °C \end{cases} \rightarrow \begin{cases} h_1 = 3231.7\ kJ/kg \\ s_1 = 6.9235\ kJ/(kg-K) \end{cases}$$

$$State\ 2a: \quad \begin{cases} P_1 = 50\ kPa \\ T_{2a} = 100\ °C \end{cases} \rightarrow \quad h_{2a} = 2682.4\ kJ/kg$$

$$State\ 2s: \quad \begin{cases} P_1 = 50\ kPa \\ s_1 = s_2 \end{cases} \rightarrow \begin{cases} s_f = 1.0912\ kJ/(kg-K) \\ s_g = 7.5931\ kJ/(kg-K) \end{cases}$$

$$x_{2s} = \frac{s_{2s} - s_f}{s_{fg}} = \frac{6.9235 - 1.0912}{6.5019} = 0.897$$

$$h_{2s} = h_f + x_{2s} \times h_{fg} = 340.54 + 0.897 \times (2304.7) = 2407.9\ kJ/kg$$

# Class Activity

- Solution (a): The isentropic efficiency is:

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}} = \frac{3231.7 - 2682.4}{3231.7 - 2407.9} = 0.667\ (or\ 66.7\%)$$

# Class Activity

- Solution (b): The mass flow rate is:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{W}_{a,out} + \dot{m}h_{2a}$$

$$\dot{W}_{a,out} = \dot{m}(h_1 - h_{2a})$$

$$2\ MW\left(\frac{1000\ kJ}{1\ MW}\right) = \dot{m}(3231.7 - 2682.4)\frac{kJ}{kg}$$

$$\dot{m} = 3.64\ kg/s$$

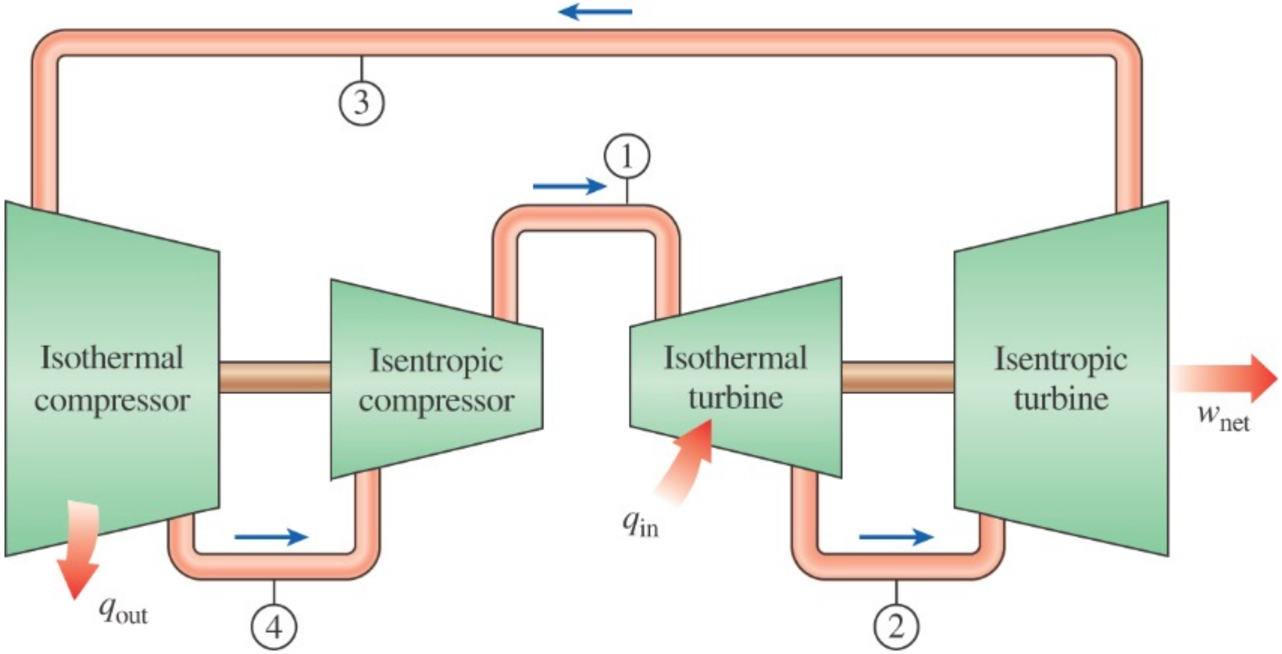
# Chapter 8 Summary

- We did not cover 8-12 (Entropy Balance)

# THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING

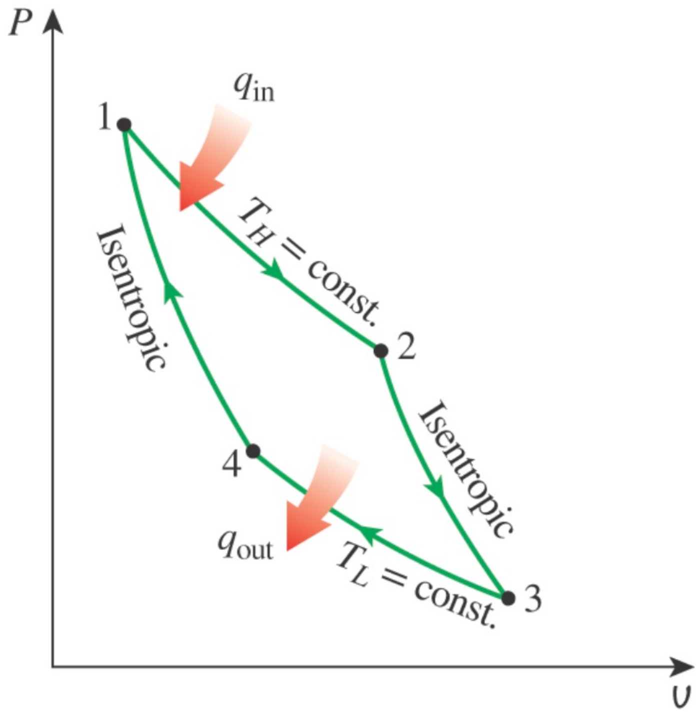
# The Carnot Cycle and Its Value in Engineering

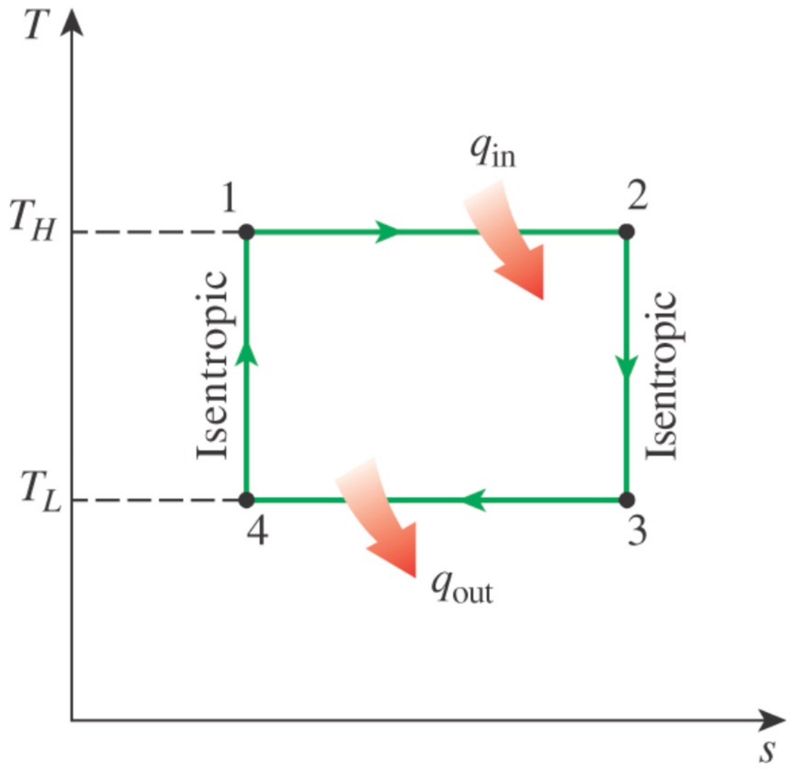
- Carnot cycle has four main processes:
  1. Isothermal heat addition
  2. Isentropic expansion
  3. Isothermal heat rejection
  4. Isentropic compression



# The Carnot Cycle and Its Value in Engineering

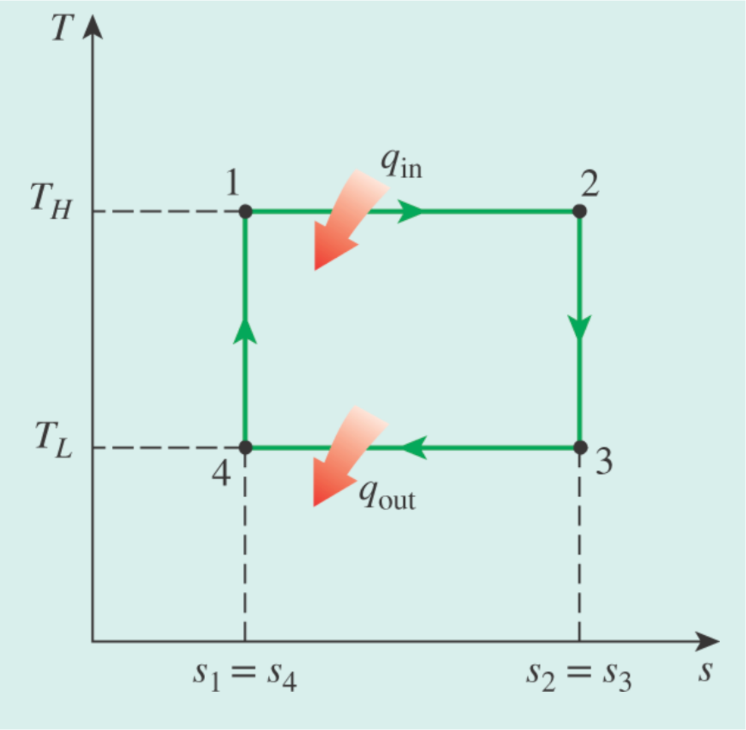
- Property diagrams such as T-s and P-V diagrams can serve as valuable aids in understanding and analysis of thermodynamics process:





# The Carnot Cycle and Its Value in Engineering

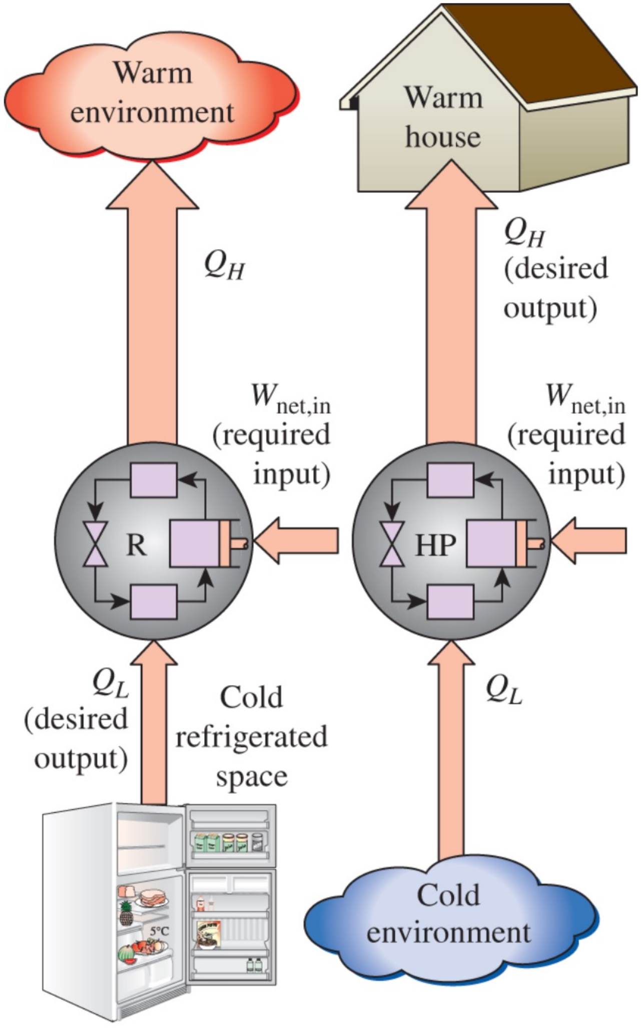
- The thermal efficiency of a Carnot cycle operating between limits of $T_H$ and $T_L$ is solely function of these two temperatures is equal to $\eta_{thermal,Carnot} = 1 - \frac{T_L}{T_H}$



# REFRIGERATORS AND HEAT PUMPS (SECTION 9-14 AND 9-15)

# Refrigerators and Heat Pumps

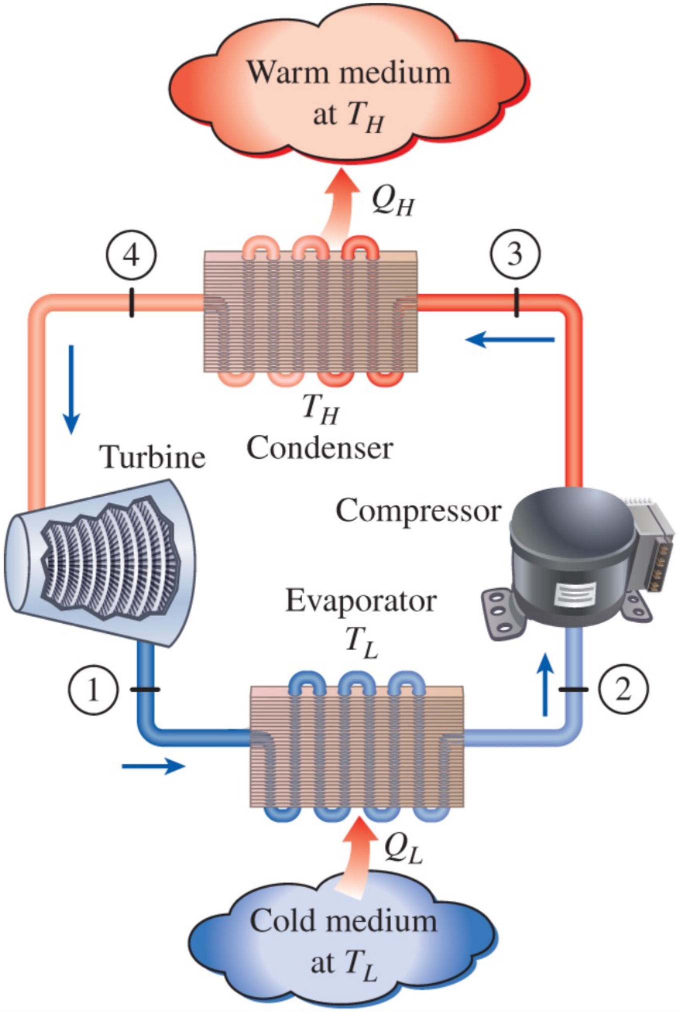
- We looked at this in Chapter 7



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

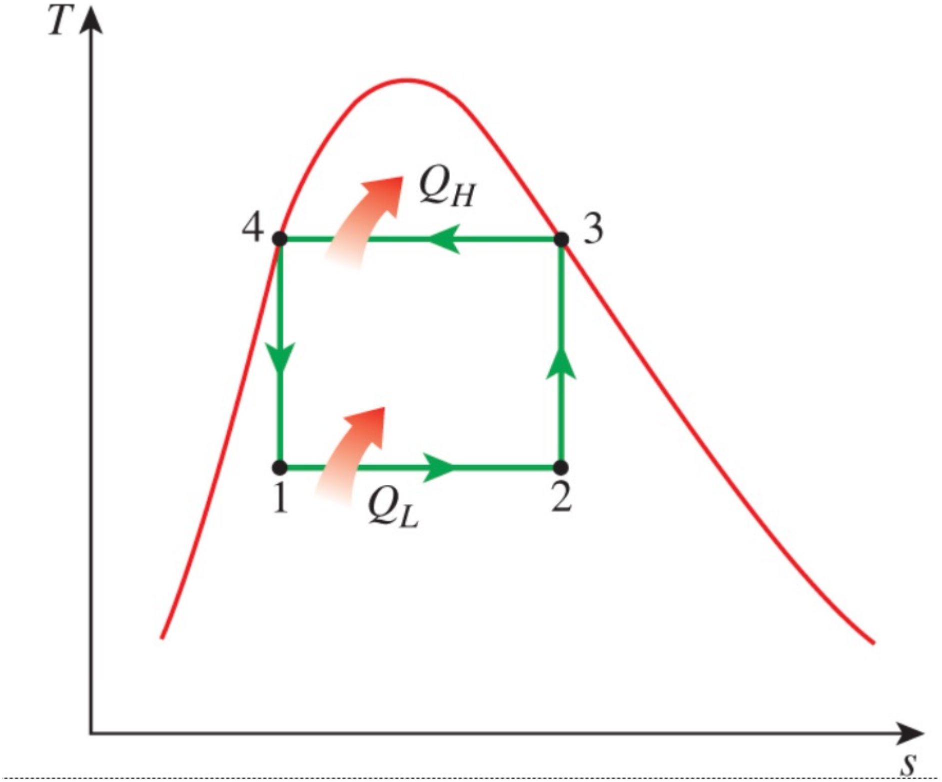
# Refrigerators and Heat Pumps

- The Carnot cycle includes:



# Refrigerators and Heat Pumps

- The T-s diagram for the Carnot cycle is:



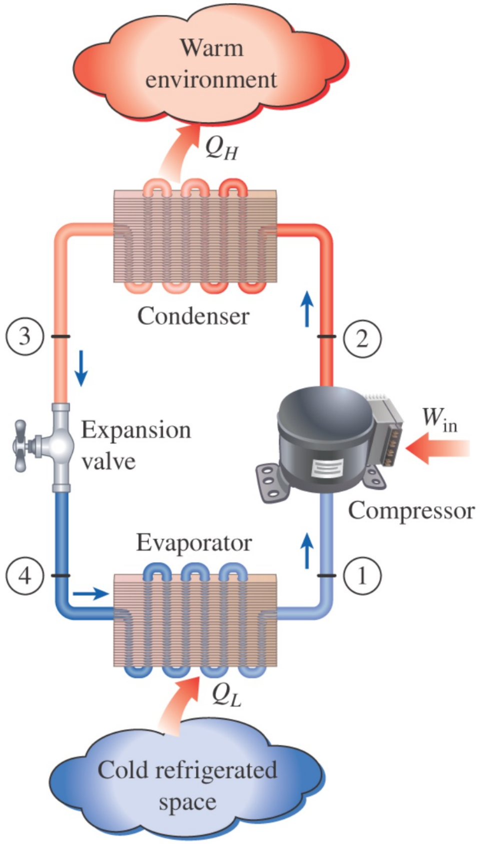
# IDEAL VAPOR COMPRESSION REFRIGERATION CYCLE (SECTION 9-16)

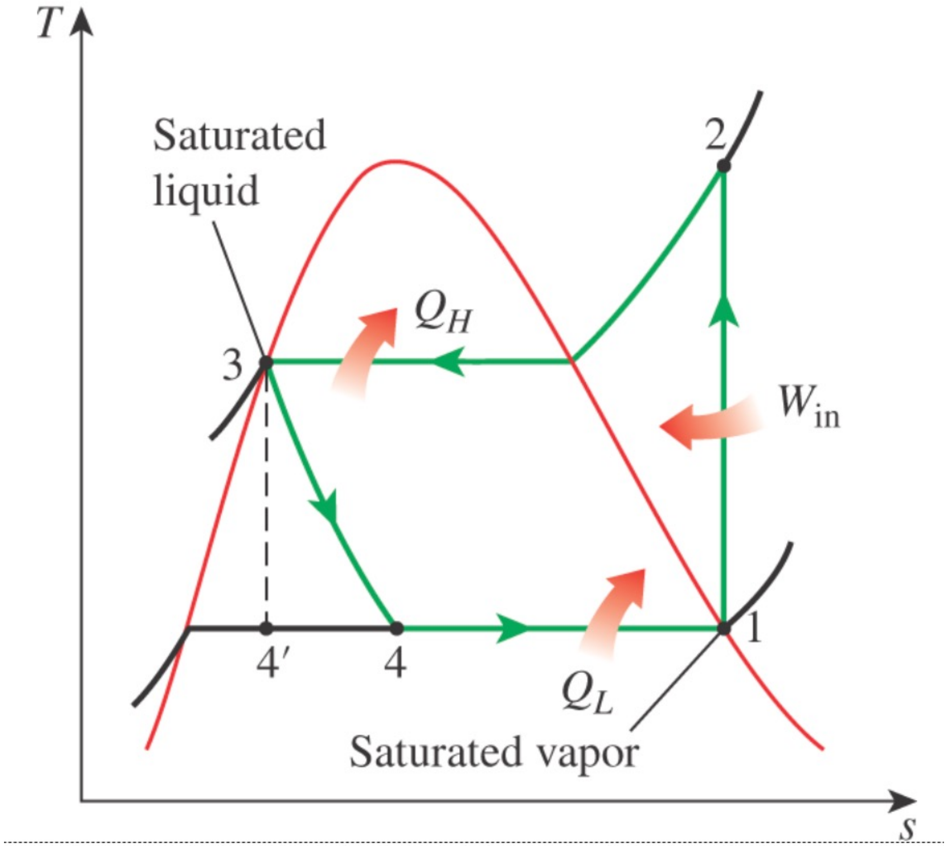
# Ideal Vapor Compression Refrigeration Cycle

- In practice, there are several issues that limit the use of Carnot vapor compression cycle:
  - ☐ 1-2: Isentropic compression in a compressor
  - ☐ 2-3: Constant pressure heat rejection in a condenser
  - ☐ 3-4: Throttling in an expansion valve
  - ☐ 4-1: Constant pressure heat absorption in an evaporator

# Ideal Vapor Compression Refrigeration Cycle

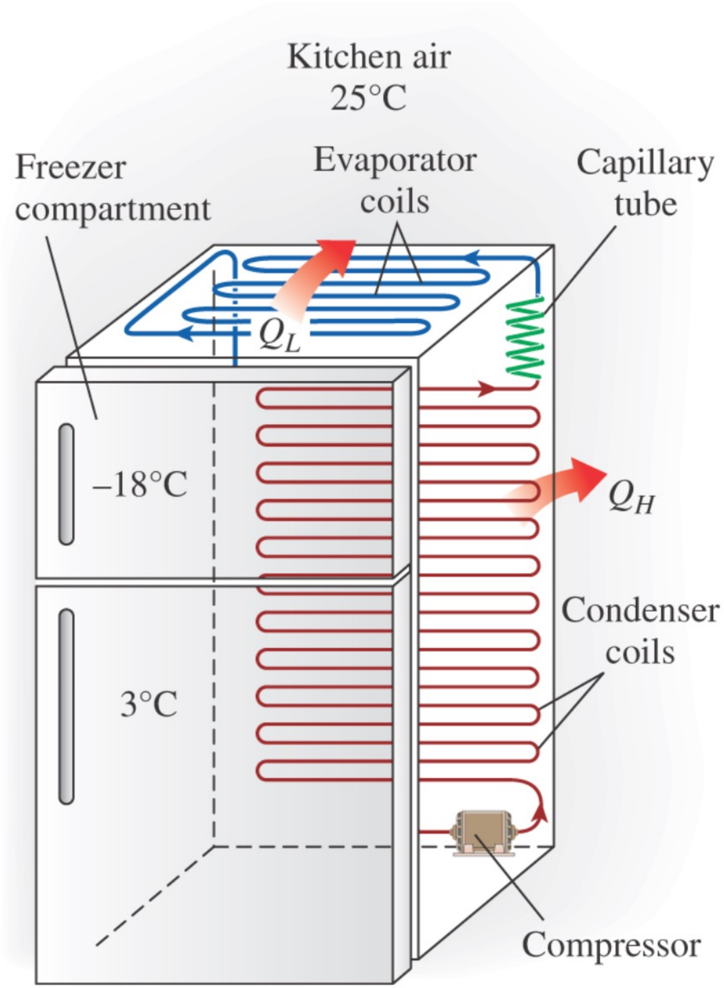
- In practice, there are several issues that limit the use of Carnot vapor compression cycle:





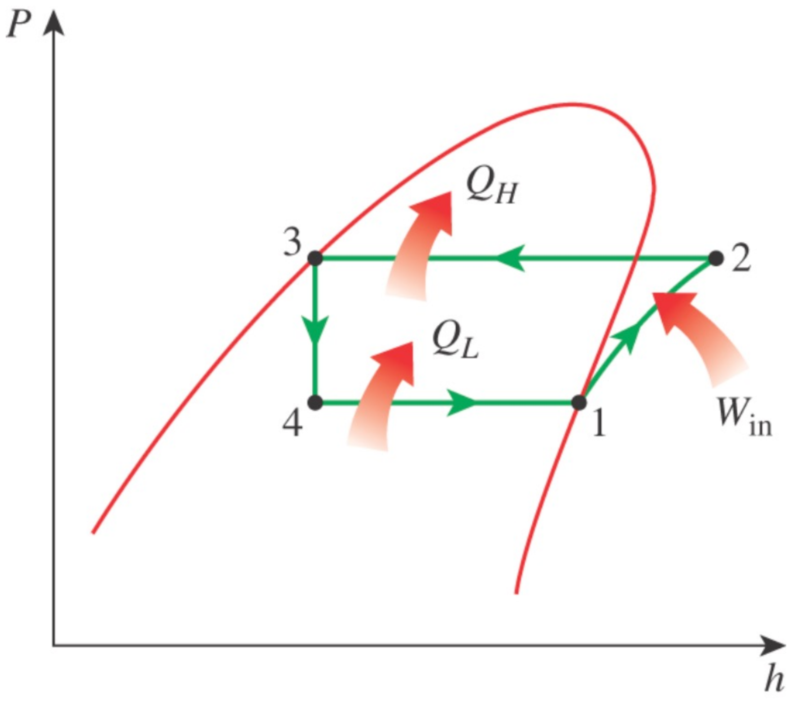
# Ideal Vapor Compression Refrigeration Cycle

- An ordinary refrigerator, has all the four main components:



# Ideal Vapor Compression Refrigeration Cycle

- P-h diagram is very helpful in analyzing the performance:



$$COP_{HP} = \frac{q_H}{w_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

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

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i$$

# CLASS ACTIVITY

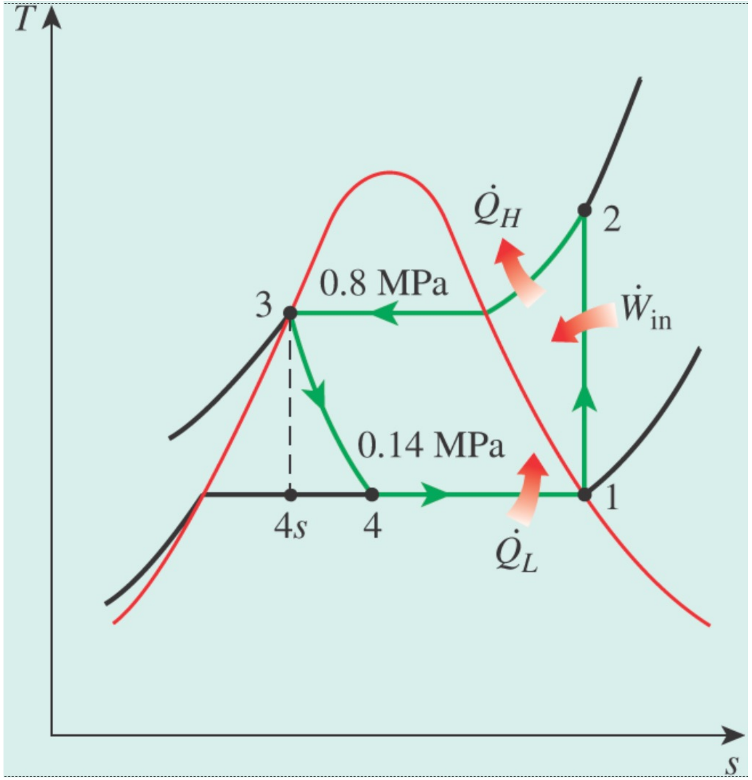
# Class Activity

- A refrigerator uses refrigerant 134-a as the working fluid and operates on an ideal vapor-compression 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
  c) The COP of the refrigerator

# Class Activity

- Solution (assumption):
  - ☐ Steady operating condition exist
  - ☐ Kinetic and potential energy are negligible

- Understanding the states:



# Class Activity

- Solution: Reading properties from the tables:

$$\begin{cases} P_1 = 0.14\ MPa \quad \rightarrow \quad h_1 = h_{g\ @\ 0.14\ MPa} = 239.19\,\dfrac{kJ}{kg} \\ s_1 = s_{g\ @\ 0.14\ MPa} = 0.94467\,\dfrac{kJ}{kg - K} \end{cases}$$

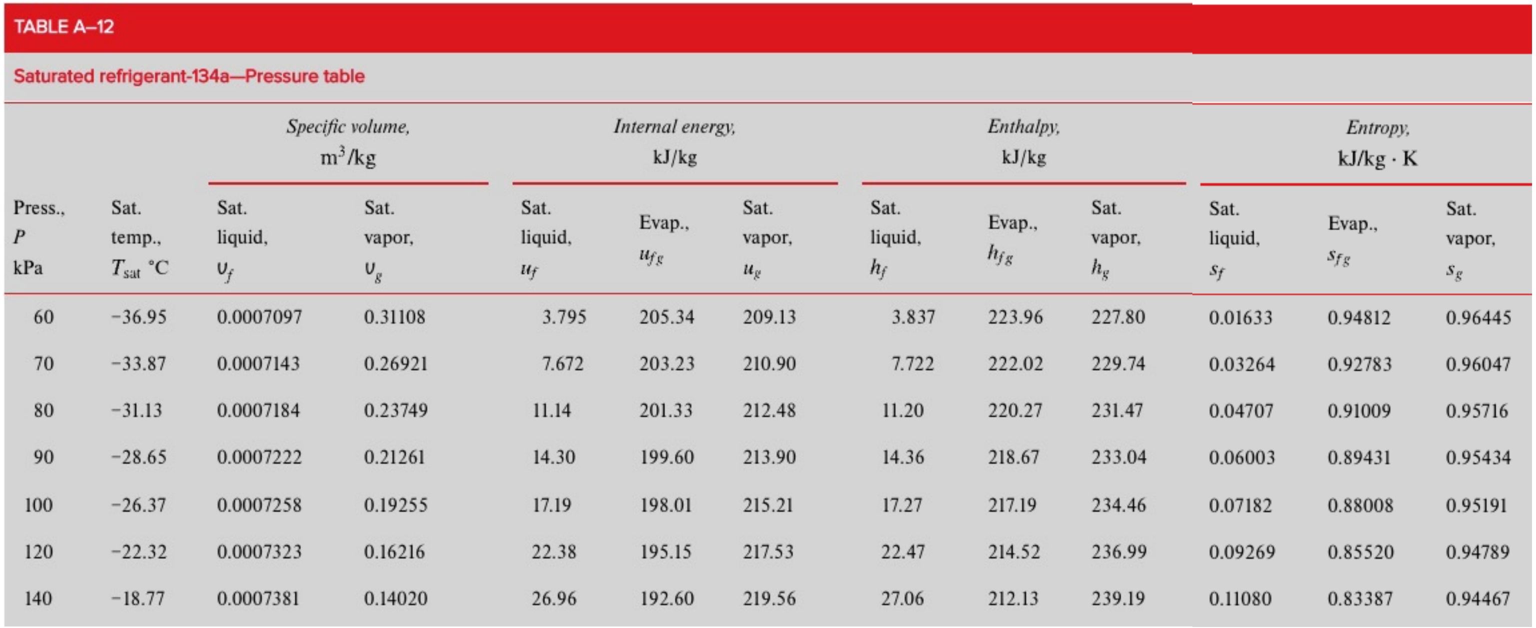


TABLE A–12

Saturated refrigerant-134a—Pressure table

| | | Specific volume, $m^3/kg$ | | Internal energy, kJ/kg | | | Enthalpy, kJ/kg | | | Entropy, kJ/kg · K | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Press., $P$ kPa | Sat. temp., $T_{sat}$ °C | Sat. liquid, $v_f$ | Sat. vapor, $v_g$ | Sat. liquid, $u_f$ | Evap., $u_{fg}$ | Sat. vapor, $u_g$ | Sat. liquid, $h_f$ | Evap., $h_{fg}$ | Sat. vapor, $h_g$ | Sat. liquid, $s_f$ | Evap., $s_{fg}$ | Sat. vapor, $s_g$ |
| 60 | −36.95 | 0.0007097 | 0.31108 | 3.795 | 205.34 | 209.13 | 3.837 | 223.96 | 227.80 | 0.01633 | 0.94812 | 0.96445 |
| 70 | −33.87 | 0.0007143 | 0.26921 | 7.672 | 203.23 | 210.90 | 7.722 | 222.02 | 229.74 | 0.03264 | 0.92783 | 0.96047 |
| 80 | −31.13 | 0.0007184 | 0.23749 | 11.14 | 201.33 | 212.48 | 11.20 | 220.27 | 231.47 | 0.04707 | 0.91009 | 0.95716 |
| 90 | −28.65 | 0.0007222 | 0.21261 | 14.30 | 199.60 | 213.90 | 14.36 | 218.67 | 233.04 | 0.06003 | 0.89431 | 0.95434 |
| 100 | −26.37 | 0.0007258 | 0.19255 | 17.19 | 198.01 | 215.21 | 17.27 | 217.19 | 234.46 | 0.07182 | 0.88008 | 0.95191 |
| 120 | −22.32 | 0.0007323 | 0.16216 | 22.38 | 195.15 | 217.53 | 22.47 | 214.52 | 236.99 | 0.09269 | 0.85520 | 0.94789 |
| 140 | −18.77 | 0.0007381 | 0.14020 | 26.96 | 192.60 | 219.56 | 27.06 | 212.13 | 239.19 | 0.11080 | 0.83387 | 0.94467 |

# Class Activity

- Solution: Reading properties from the tables:

$$\begin{cases} P_3 = 0.8\ MPa \\ s_2 = s_1 = 0.94467\,\dfrac{kJ}{kg-K} \end{cases} \rightarrow.\ \ h_2 = 275.40\,\frac{kJ}{kg}$$

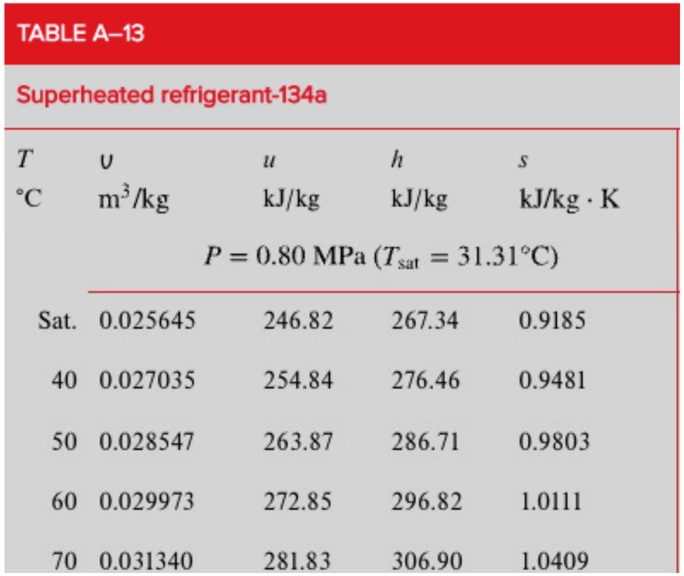


TABLE A–13

Superheated refrigerant-134a

| *T* °C | *v* m$^3$/kg | *u* kJ/kg | *h* kJ/kg | *s* kJ/kg · K |
|---|---|---|---|---|
| | *P* = 0.80 MPa ($T_{sat}$ = 31.31°C) | | | |
| Sat. | 0.025645 | 246.82 | 267.34 | 0.9185 |
| 40 | 0.027035 | 254.84 | 276.46 | 0.9481 |
| 50 | 0.028547 | 263.87 | 286.71 | 0.9803 |
| 60 | 0.029973 | 272.85 | 296.82 | 1.0111 |
| 70 | 0.031340 | 281.83 | 306.90 | 1.0409 |

# Class Activity

- Solution: Reading properties from the tables:

$$P_3 = 0.8\ MPa \quad \rightarrow \quad h_3 = h_{f\ @\ 0.8\ MPa} = 95.48\,\frac{kJ}{kg}$$

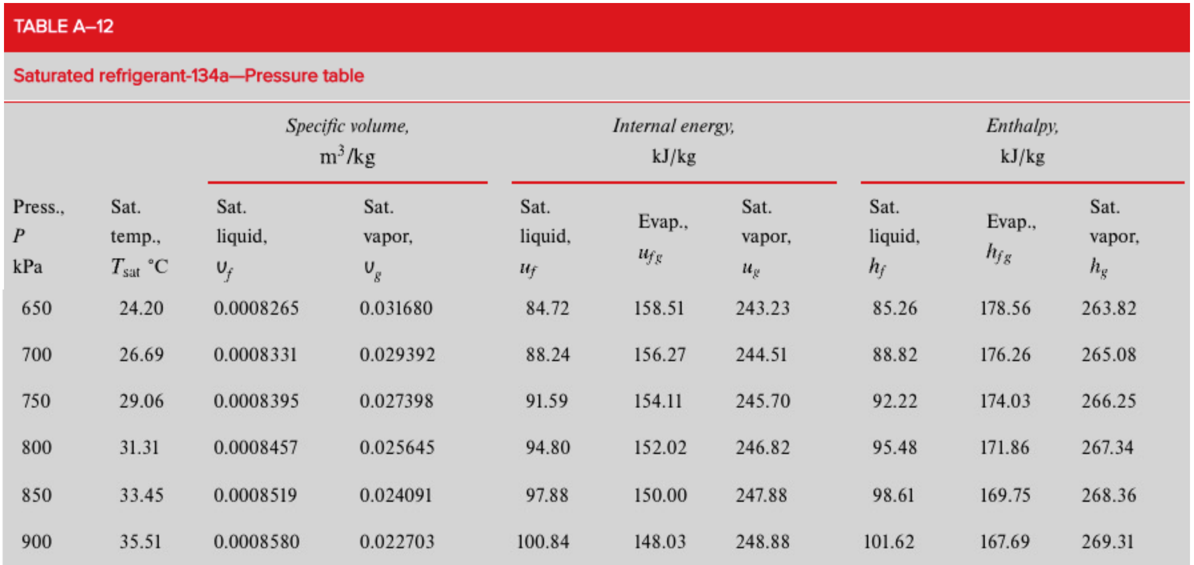


**TABLE A–12**

Saturated refrigerant-134a—Pressure table

| | | *Specific volume,* m$^3$/kg | | *Internal energy,* kJ/kg | | | *Enthalpy,* kJ/kg | | |
|---|---|---|---|---|---|---|---|---|---|
| Press., $P$ kPa | Sat. temp., $T_{sat}$ °C | Sat. liquid, $v_f$ | Sat. vapor, $v_g$ | Sat. liquid, $u_f$ | Evap., $u_{fg}$ | Sat. vapor, $u_g$ | Sat. liquid, $h_f$ | Evap., $h_{fg}$ | Sat. vapor, $h_g$ |
| 650 | 24.20 | 0.0008265 | 0.031680 | 84.72 | 158.51 | 243.23 | 85.26 | 178.56 | 263.82 |
| 700 | 26.69 | 0.0008331 | 0.029392 | 88.24 | 156.27 | 244.51 | 88.82 | 176.26 | 265.08 |
| 750 | 29.06 | 0.0008395 | 0.027398 | 91.59 | 154.11 | 245.70 | 92.22 | 174.03 | 266.25 |
| 800 | 31.31 | 0.0008457 | 0.025645 | 94.80 | 152.02 | 246.82 | 95.48 | 171.86 | 267.34 |
| 850 | 33.45 | 0.0008519 | 0.024091 | 97.88 | 150.00 | 247.88 | 98.61 | 169.75 | 268.36 |
| 900 | 35.51 | 0.0008580 | 0.022703 | 100.84 | 148.03 | 248.88 | 101.62 | 167.69 | 269.31 |

$$h_4 \cong h_3\ (throttling) \quad \rightarrow \quad h_4 = 95.48\,\frac{kJ}{kg}$$

# Class Activity

- Solution (a): The rate of heat removal from the refrigerated space and the power input to the compressor is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = \left(0.05\,\frac{kg}{s}\right)\left((239.19 - 95.48)\,\frac{kJ}{kg}\right) = 7.19\ kW$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \left(0.05\,\frac{kg}{s}\right)\left((275.40 - 239.19)\,\frac{kJ}{kg}\right) = 1.18\ kW$$

# Class Activity

- Solution (b): The rate of heat rejection from the refrigerant to the environment is:

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = \left(0.05\,\frac{kg}{s}\right)\left((275.40 - 95.48)\,\frac{kJ}{kg}\right) = 9.00\ kW$$

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{in} = 7.19 + 1.81 = 9.00\ kW$$

# Class Activity

- Solution (c): The coefficient of performance of the refrigerator is:

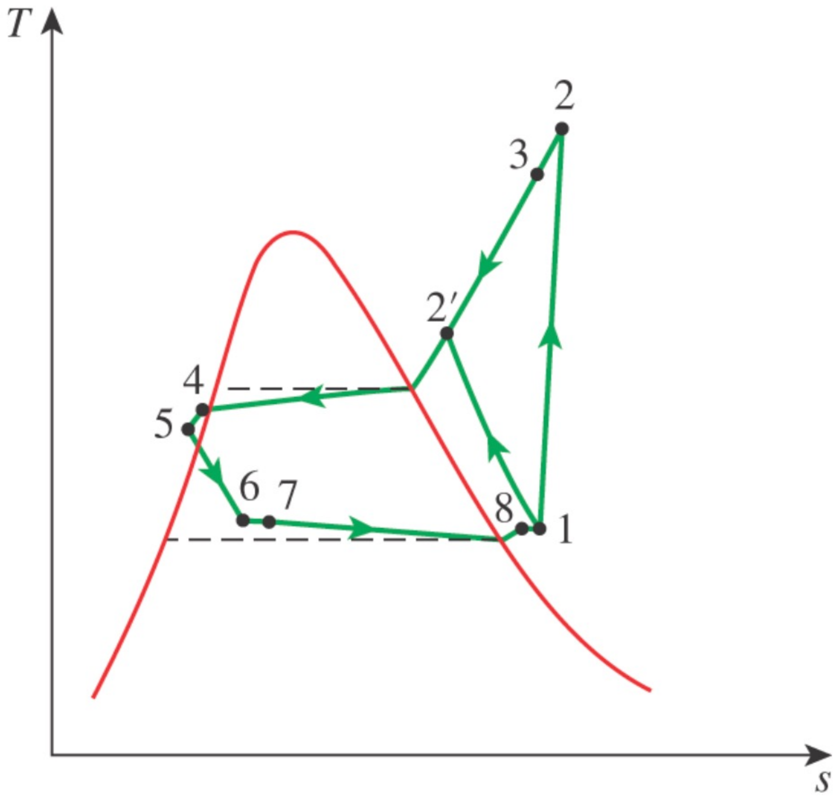
$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{7.19\ kW}{1.81\ kW} = 3.97$$

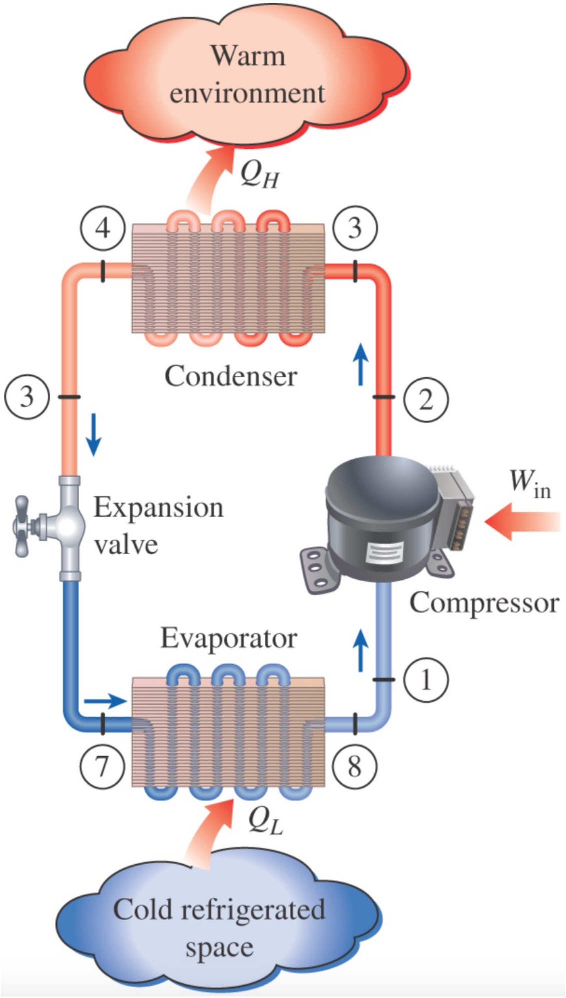
*What would be the COP is the throttling process is isentropic?*

# ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE (SECTION 9-17)

# Actual Vapor-Compression Refrigeration Cycle

- An actual vapor-compression refrigeration cycle varies from the ideal one because of two common sources of irreversibilities:





# CLASS ACTIVITY

# Class Activity

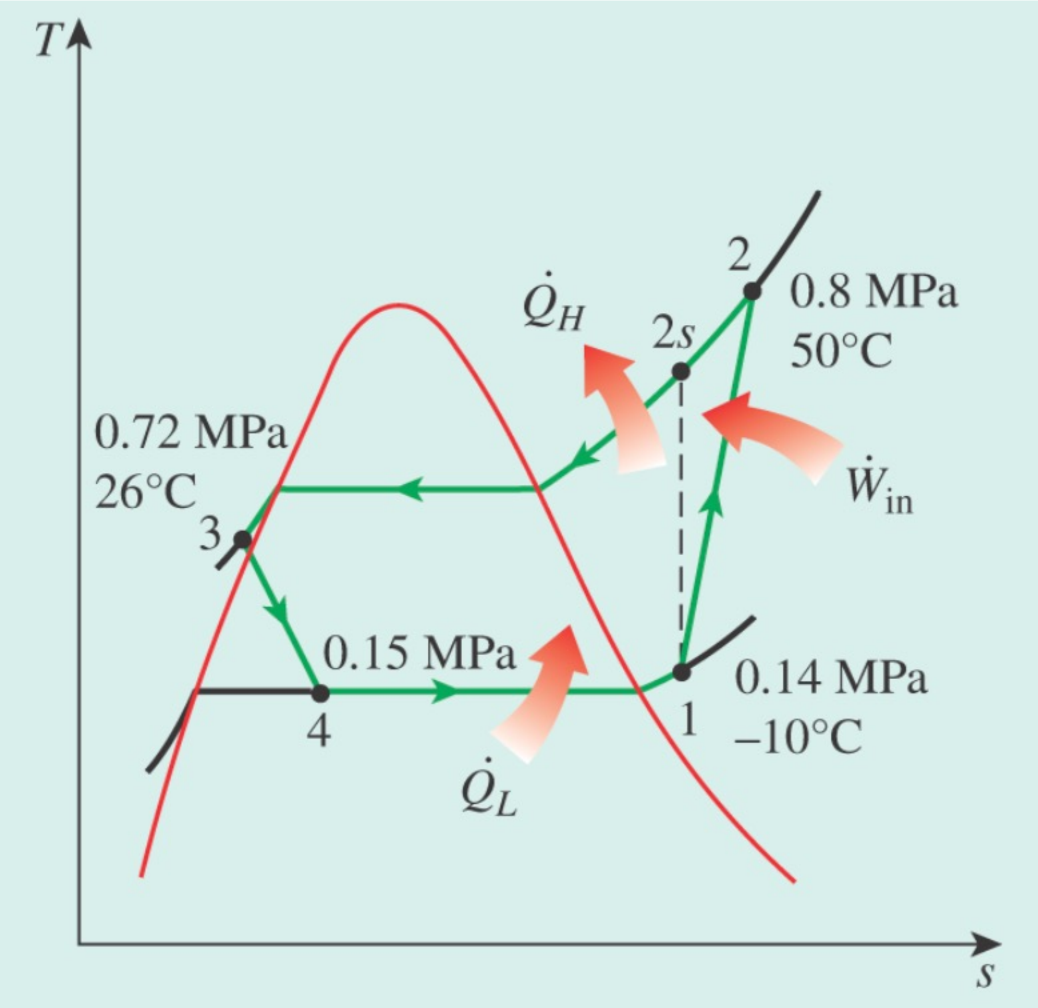
- ***(The actual vapor-compression refrigeration cycle – almost similar inputs to the previous class activity):*** Refrigerant 134-a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and -10 °C at a rate of 0.05 kg/s and leaves at 0.8 MPa and 50 °C. The refrigerant is cooled in the condenser to 26 °C and 0.72 MPa and id throttled to 0.15 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components determine
  a) The rate of heat removal from the refrigerated space and the power pressure drops in the connecting lines between the components
  b) The isentropic efficiency of the compressor
  c) The coefficient of performance of the refrigerator

# Class Activity

- Solution (assumption):
  - ☐ Steady operating condition exist
  - ☐ Kinetic and potential energy are negligible

# Class Activity

- Solution (T-s diagram)



# Class Activity

- Solution (Tables and Calculations):

$$\begin{cases} P_1 = 0.14\ MPa \\ T_1 = -10\ °C \end{cases} \rightarrow \quad h_1 = 246.37\,\frac{kJ}{kg}$$

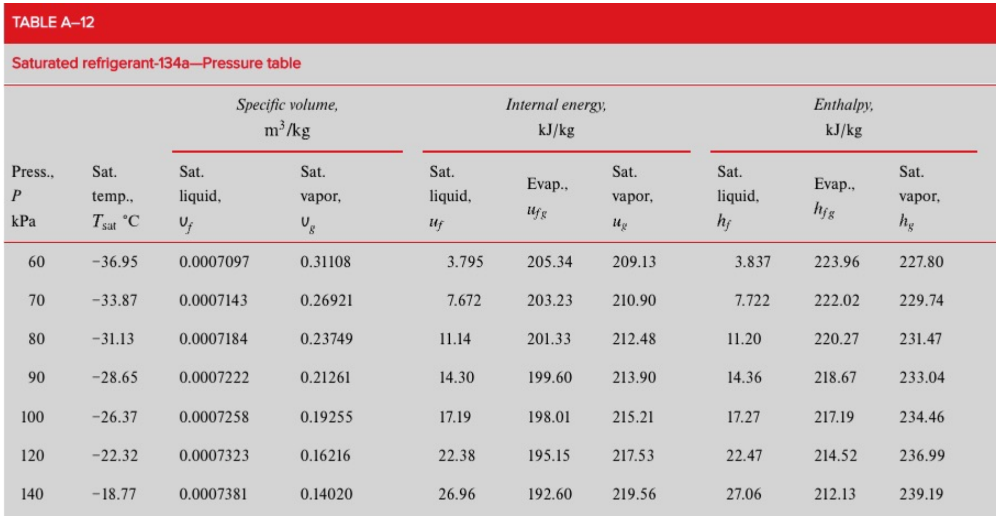


**TABLE A–12**

Saturated refrigerant-134a—Pressure table

| | | *Specific volume,* m$^3$/kg | | *Internal energy,* kJ/kg | | | *Enthalpy,* kJ/kg | | |
|---|---|---|---|---|---|---|---|---|---|
| Press., $P$ kPa | Sat. temp., $T_{sat}$ °C | Sat. liquid, $v_f$ | Sat. vapor, $v_g$ | Sat. liquid, $u_f$ | Evap., $u_{fg}$ | Sat. vapor, $u_g$ | Sat. liquid, $h_f$ | Evap., $h_{fg}$ | Sat. vapor, $h_g$ |
| 60 | −36.95 | 0.0007097 | 0.31108 | 3.795 | 205.34 | 209.13 | 3.837 | 223.96 | 227.80 |
| 70 | −33.87 | 0.0007143 | 0.26921 | 7.672 | 203.23 | 210.90 | 7.722 | 222.02 | 229.74 |
| 80 | −31.13 | 0.0007184 | 0.23749 | 11.14 | 201.33 | 212.48 | 11.20 | 220.27 | 231.47 |
| 90 | −28.65 | 0.0007222 | 0.21261 | 14.30 | 199.60 | 213.90 | 14.36 | 218.67 | 233.04 |
| 100 | −26.37 | 0.0007258 | 0.19255 | 17.19 | 198.01 | 215.21 | 17.27 | 217.19 | 234.46 |
| 120 | −22.32 | 0.0007323 | 0.16216 | 22.38 | 195.15 | 217.53 | 22.47 | 214.52 | 236.99 |
| 140 | −18.77 | 0.0007381 | 0.14020 | 26.96 | 192.60 | 219.56 | 27.06 | 212.13 | 239.19 |

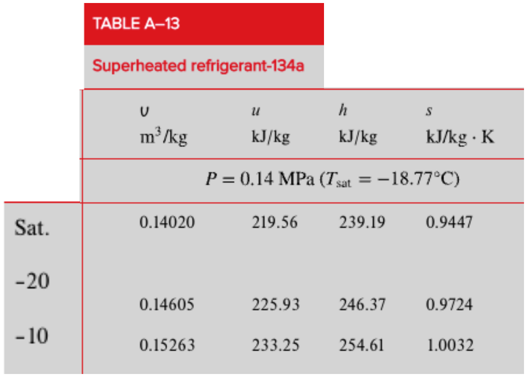


**TABLE A–13**

Superheated refrigerant-134a

| | $v$ m$^3$/kg | $u$ kJ/kg | $h$ kJ/kg | $s$ kJ/kg · K |
|---|---|---|---|---|
| | $P$ = 0.14 MPa ($T_{sat}$ = −18.77°C) | | | |
| Sat. | 0.14020 | 219.56 | 239.19 | 0.9447 |
| -20 | 0.14605 | 225.93 | 246.37 | 0.9724 |
| -10 | 0.15263 | 233.25 | 254.61 | 1.0032 |

# Class Activity

- Solution (Tables and Calculations):

$$\begin{cases} P_1 = 0.14\ MPa \\ T_1 = -10\ °C \end{cases} \rightarrow \quad h_1 = 246.37\, \frac{kJ}{kg}$$

$$\begin{cases} P_2 = 0.8\ MPa \\ T_2 = -50\ °C \end{cases} \rightarrow \quad h_2 = 286.71\, \frac{kJ}{kg}$$

$$\begin{cases} P_3 = 0.72\ MPa \\ T_3 = 26\ °C \end{cases} \rightarrow \quad h_3 \cong h_{f\ @\ 26\ °C} = 87.83\ \frac{kJ}{kg}$$

$$\begin{cases} h_4 \cong h_3 = 87.83\, \frac{kJ}{kg} \end{cases}$$

# Class Activity

- Solution (a): The rate of heat removal from the refrigerated space and the power input to the compressor are:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = \left(0.05\,\frac{kg}{s}\right)\left((246.37 - 87.83)\,\frac{kJ}{kg}\right) = 7.93\ kW$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \left(0.05\,\frac{kg}{s}\right)\left((286.71 - 246.37)\,\frac{kJ}{kg}\right) = 2.02\ kW$$

# Class Activity

- Solution (b): The isentropic efficiency of the compressor is determined from:

$$\eta_C \cong \frac{h_{2s} - h_1}{h_2 - h_1}$$

- Where the enthalpy at state $2s$ ($P_{2s} = 0.8\ MPa$ and $s_{2s} = s_1 = 0.9724 \frac{kJ}{kg-K}$) is $284.20 \frac{kJ}{kg}$. Thus:

$$\eta_C \cong \frac{284.20 - 246.37}{286.71 - 246.37} = 0.938\ or\ 93.8\%$$

# Class Activity

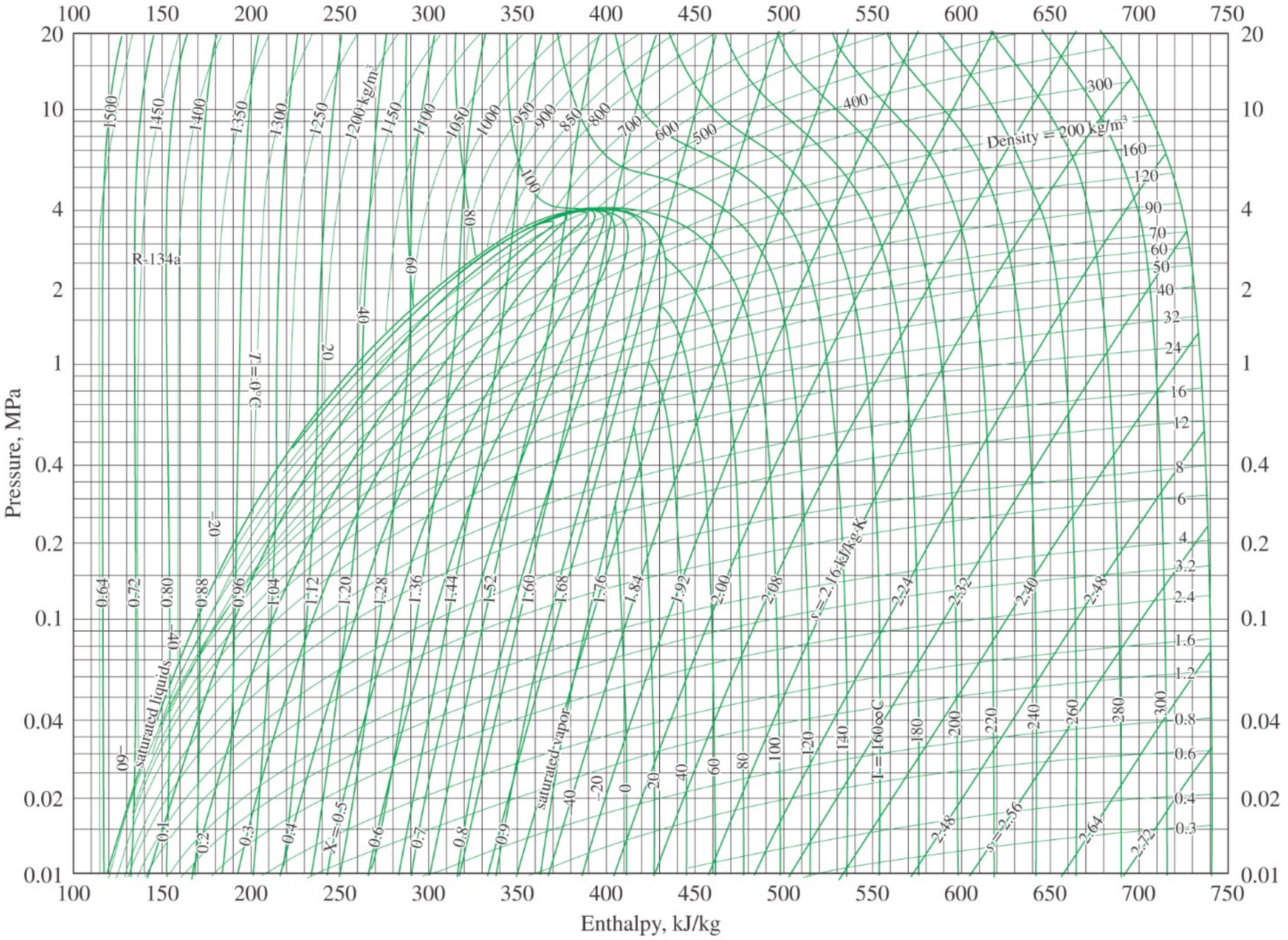
- Solution (c): The coefficient of performance of the refrigerator is:

$$COP_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{7.93\ kW}{2.02\ kW} = 3.93$$

# CLASS ACTIVITY

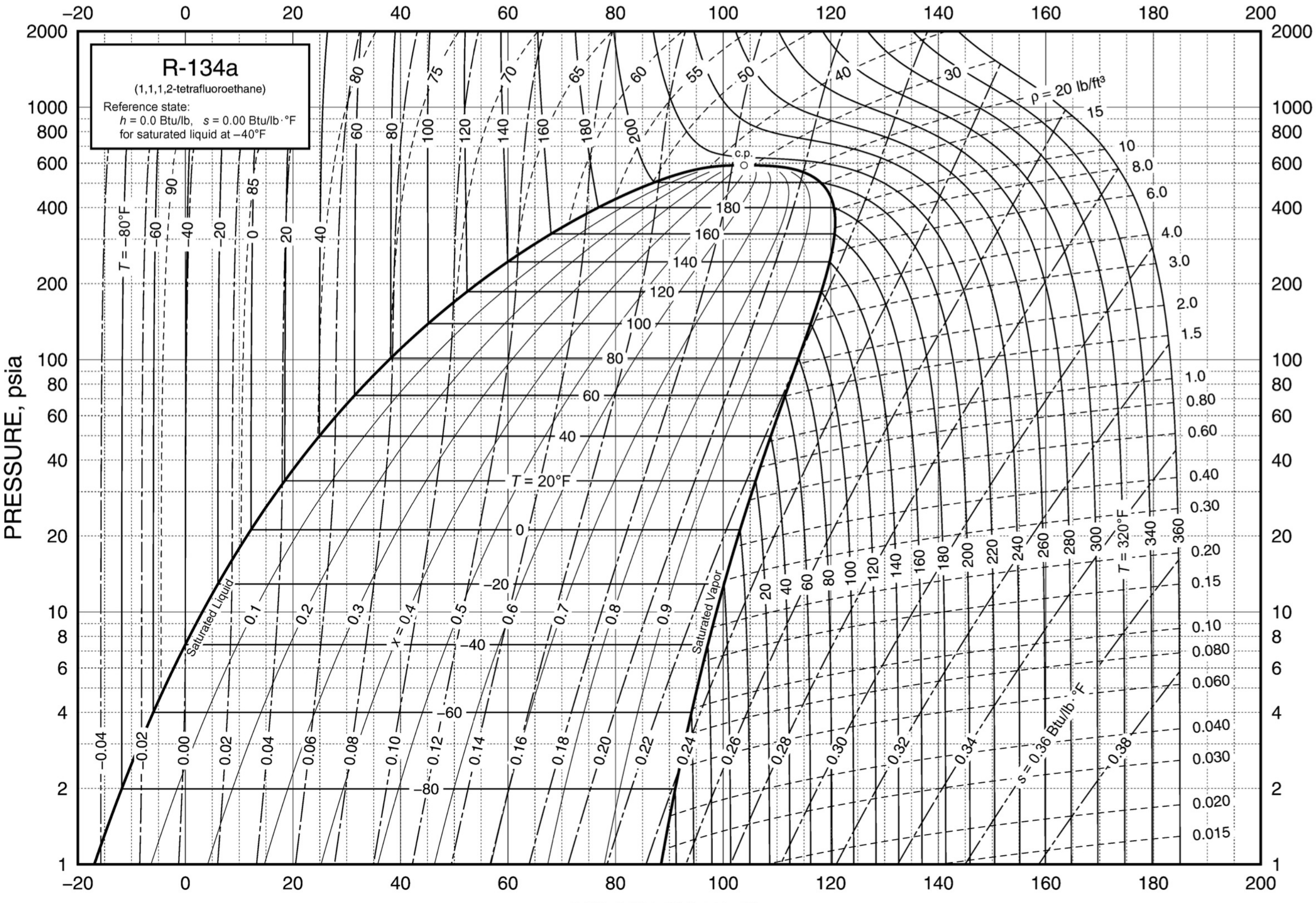
# Class Activity

- Solve the previous example using P-h diagram (Figure A-14)



# Class Activity

- Solve the previous example using P-h diagram (ASHRAE)



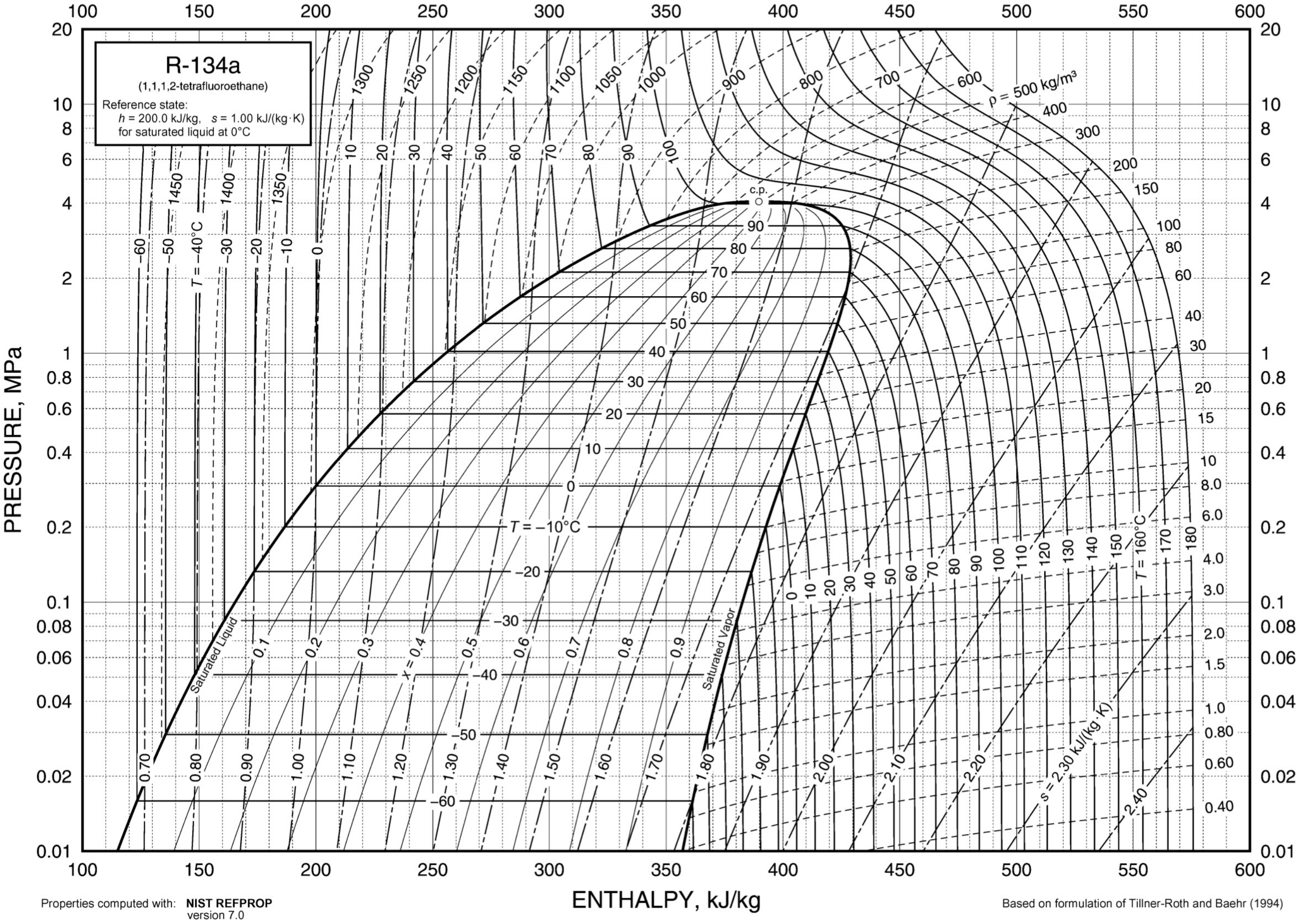
Properties computed with: **NIST REFPROP** version 7.0

Based on formulation of Tillner-Roth and Baehr (1994)

**Fig. 8 Pressure-Enthalpy Diagram for Refrigerant 134a**

# Class Activity

- Solve the previous example using P-h diagram (ASHRAE)



Properties computed with: **NIST REFPROP** version 7.0

Based on formulation of Tillner-Roth and Baehr (1994)

**Fig. 8 Pressure-Enthalpy Diagram for Refrigerant 134a**