

# Refrigeration

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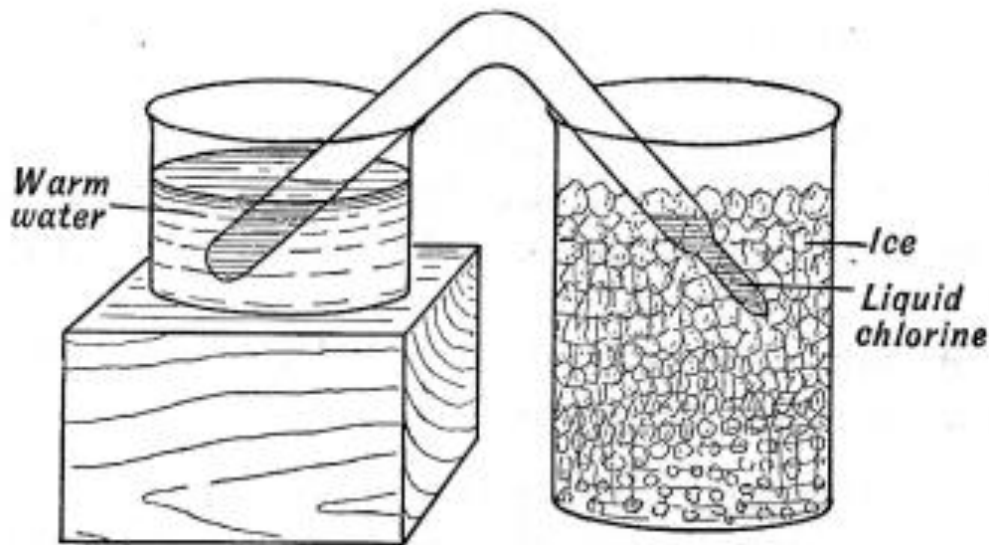
# the Science of Heat

- In ancient west, **Earth, Water, Air** and **Fire** are the generally known and often quoted ancient elements of nature.
- In China, we have **土木火水金**.
- By the end of the 18th century, French **rich aristocrat** chemist **Lavoisier** advocated a theory explaining that the phenomena involving the transfer of heat are the result of a weightless fluid substance, he called **“caloric.”**

- This “caloric,” as Lavoisier assumed, permeated the gaps between atoms of a solid causing thermal expansion and whose loss through the surface could explain Newtonian cooling.
- **Count (伯爵) Rumford** found that heat can be produced by the boring of cannons and one can generate “**unlimited amount**” of heat simply by keeping boring the cannon—**mechanical to heat**
- **Count Rumford’s** opinion eventually prevailed, but not after **Lavoisier** being severed by a **Guillotine**.

# Faraday's Experiment

- **Michael Faraday** heated the **chlorine hydrate** in a sealed bent tube. He placed the end of the tube containing the hydrate of chlorine in a hot bath (100 degrees) and the empty end in ice water (32 degrees).



- A faint yellow atmosphere filled the tube and, after some time, yellow **liquid chlorine** formed in the cool end of the tube.

- **Faraday** broke the tube, sending shards of glass flying. Then he found that the yellowish **liquid** had **vanished**.
- **Faraday** concluded that **energy** had been **transferred**, in the **form of heat**, to **change the state** of matter of the chlorine. The absorption of heat cooled the surroundings, producing **refrigeration**.
- **Amontons'** Law of Pressure-Temperature  
The pressure of a gas of fixed mass and fixed volume is directly proportional to the gas's absolute temperature.

# Siemens cycle

- **Carl Wilhelm Siemens** patented the **Siemens cycle** in 1857
  - **1. Heated - by compressing the gas - adding external energy into the gas, to give it what is needed for running through the cycle,  $PV=nRT$**
  - **2. Cooled - by immersing the gas in a cooler environment, losing some of its heat (and energy)**
  - **3. Cooled through heat exchanger with returning gas from next (and last stage)**
  - **4. Cooled further by decompression the gas, removing heat (and energy)**

**The gas which is now at its coolest in the current cycle, may be used as coolant, is then, recycled and sent back to be**

- 5. Heated - when participating as the coolant for stage 3, and then**
- 6. resent to stage one, to start the next cycle, and be slightly reheated by compression.**

# Hampson–Linde cycle

William Hampson and  
Carl von Linde

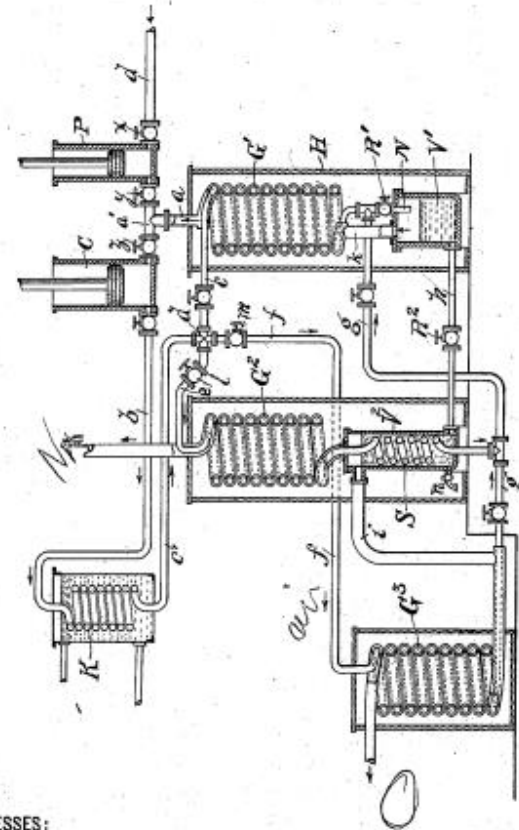
independently filed for  
patent of the cycle

in 1895.

GASES, AND THE SEPARATION OF THE CONSTITUENTS  
OF GASEOUS MIXTURES.

APPLICATION FILED JULY 9, 1895.

NO MODEL.



WITNESSES:

Chas. W. Thomas.  
Geo. W. Eschbaum.

INVENTOR:

Carl Linde,

BY

*Alfred D. ...*

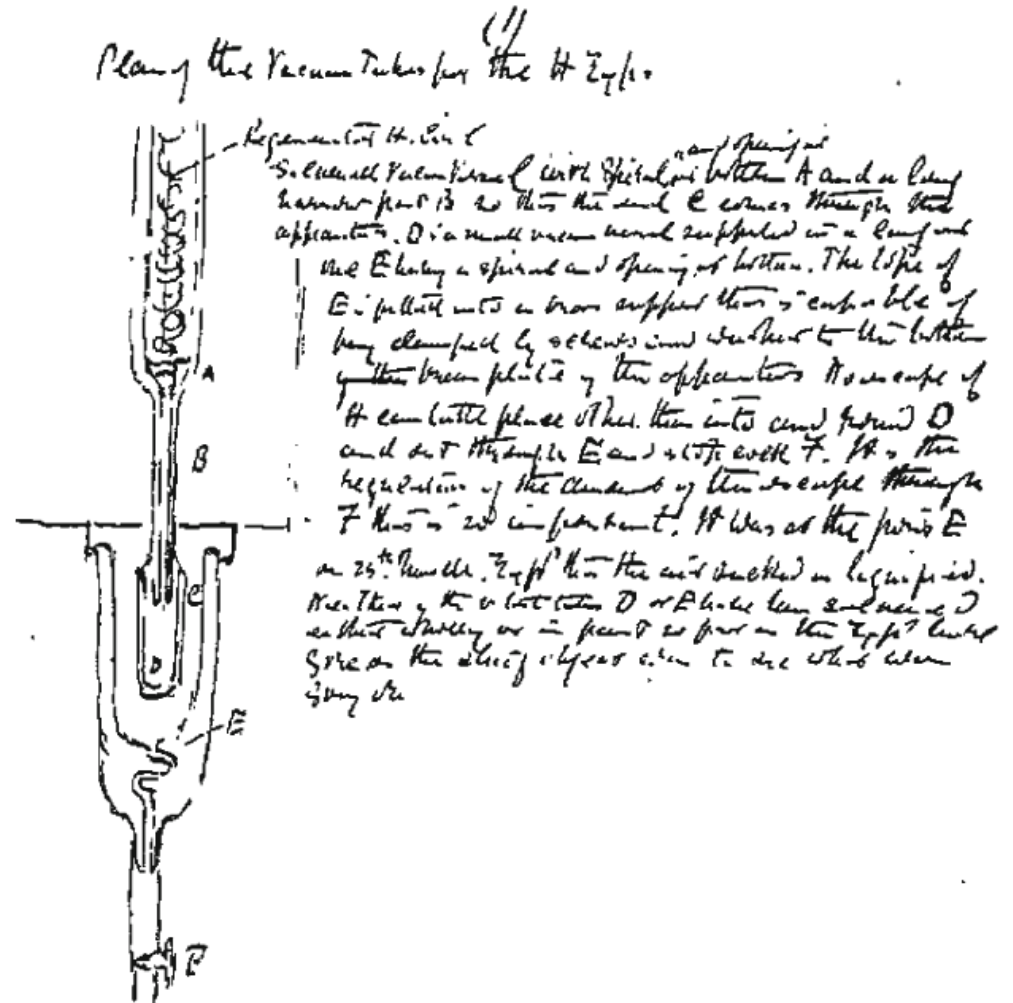


# Hampson–Linde cycle

- 1 - 3 is same as Siemens'
- 4. Cooled further by passing the gas through a **Joule-Thomson orifice**, removing heat, but conserving energy.
- The rest of the cycle is again, the same

# Refrigeration

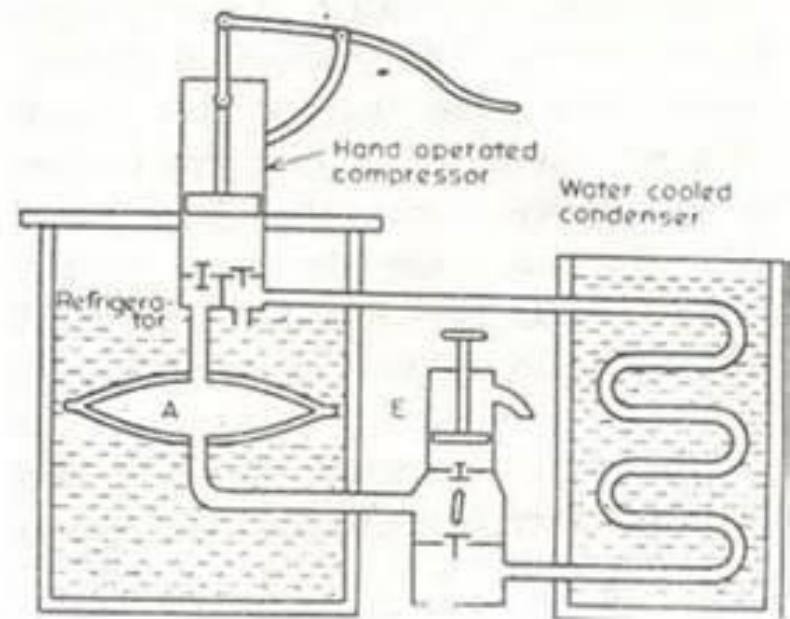
- On 10 May 1898, **James Dewar** used **regenerative cooling** to become the first to statically liquefy **hydrogen**.
- Helium** was first liquefied on July 10, 1908, by the Dutch physicist **Heike Kamerlingh Onnes** in the Netherlands, awarded **Nobel Prize in Physics (1913)**





## Portrait of Dr. John Gorrie

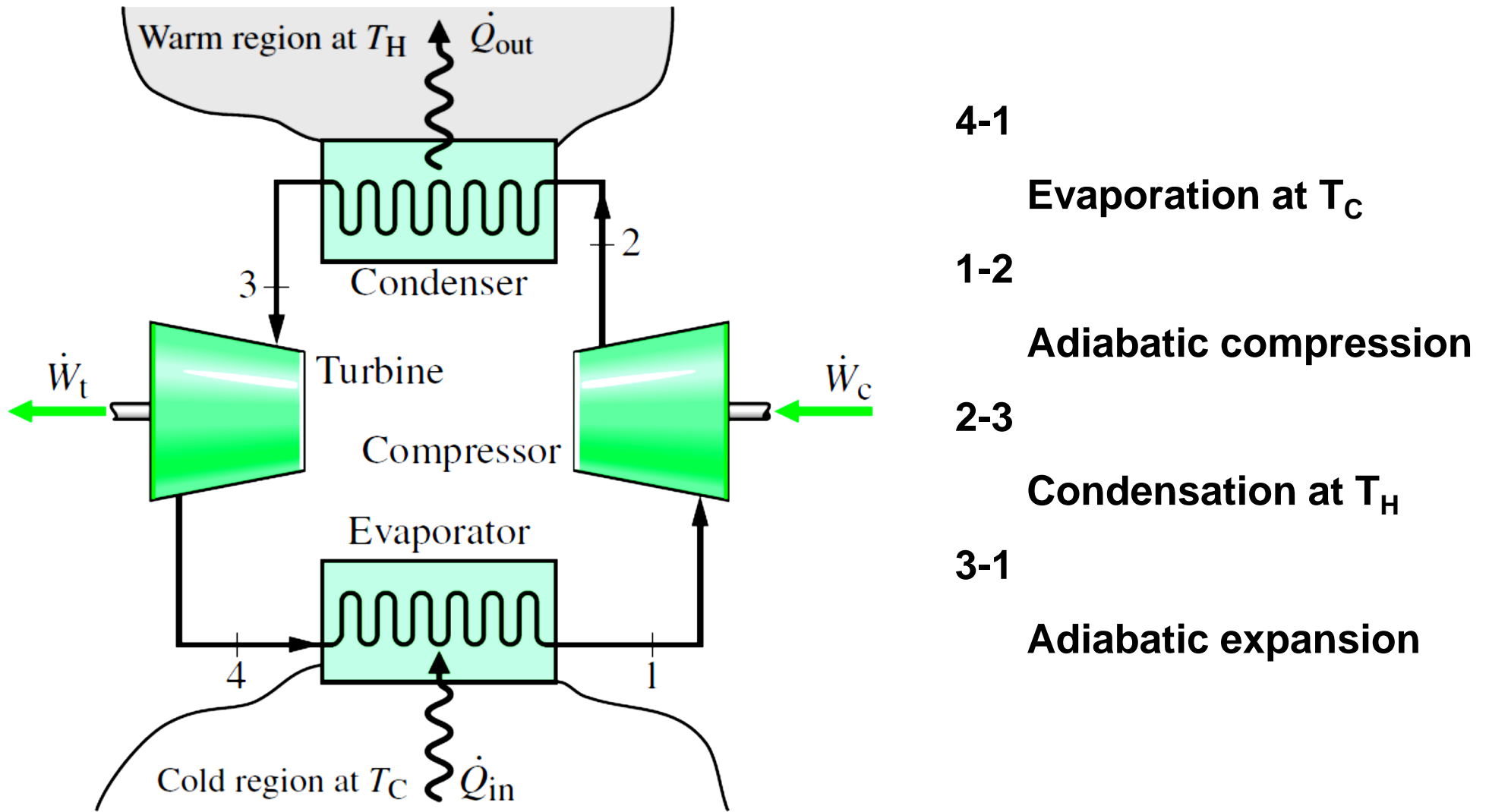
In 1851, **Dr. John Gorrie**,  
**university of Florida**,  
was granted a patent,  
No. 8080, for the machine  
to make ice.



**The model of the original ice. The ice collects in the wooden box near the top.**

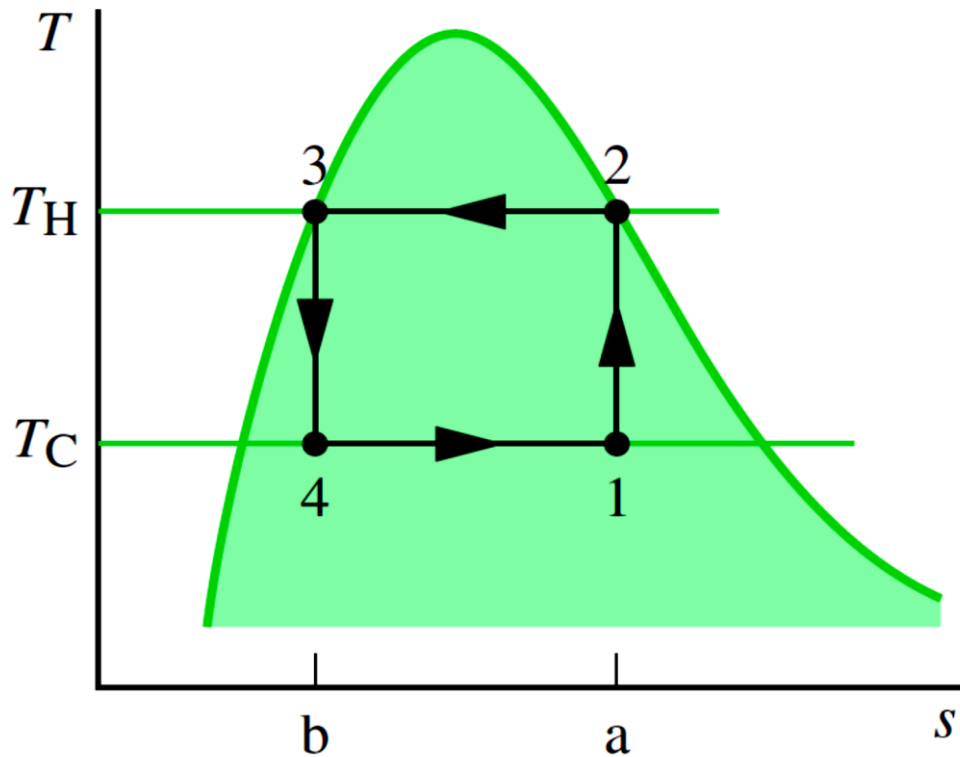
**Notice: the four components design**

# Carnot Refrigeration Cycle



# Carnot Refrigeration Cycle

- Coefficient of Performance

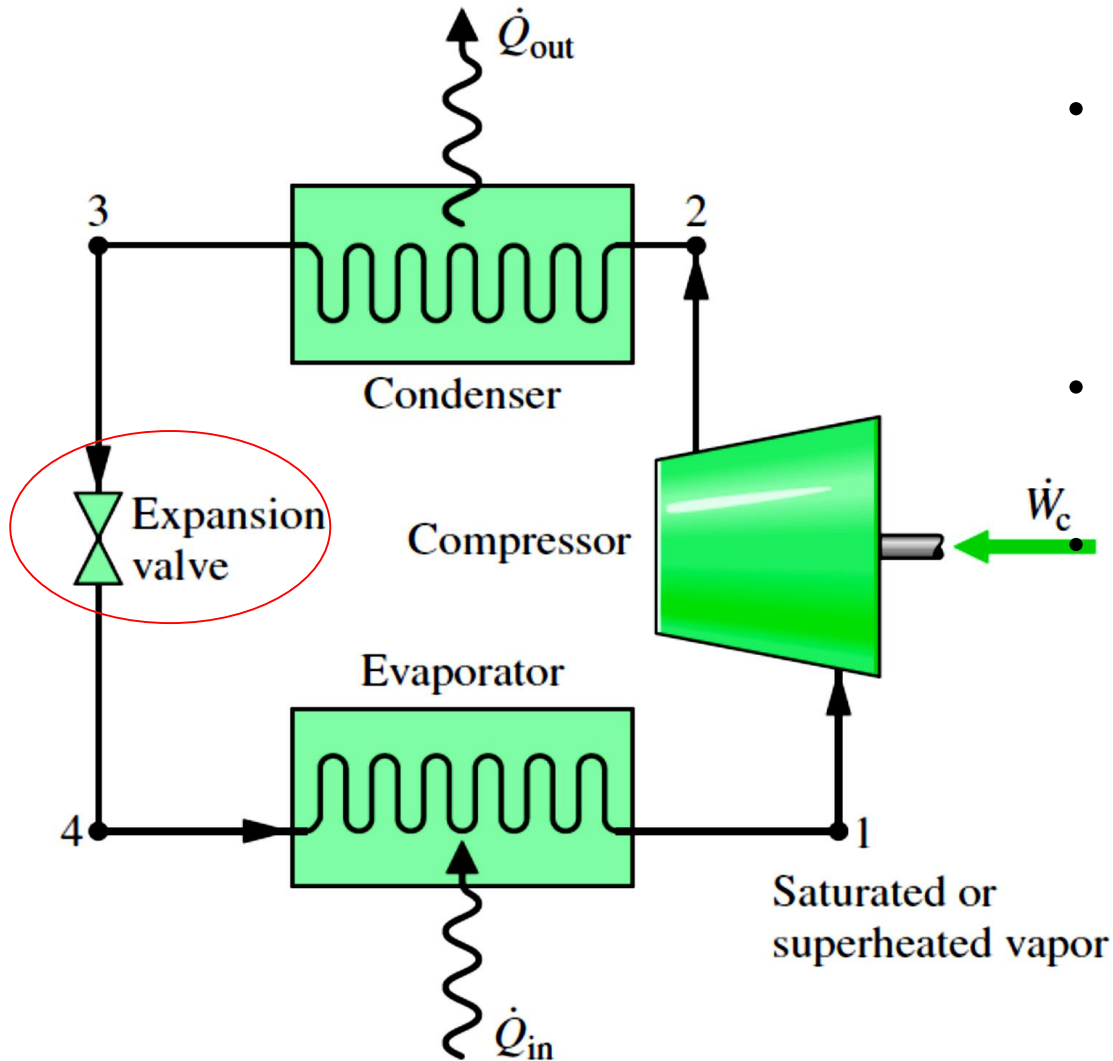


$$\begin{aligned}
 \text{cop} &= \frac{\dot{q}_{in}}{\dot{w}_C - \dot{w}_H} \\
 &= \frac{\text{Area}_{1-a-b-4-1}}{\text{Area}_{1-2-3-4-1}} \\
 &= \frac{T_C (s_a - s_b)}{(T_H - T_C)(s_a - s_b)} \\
 &= \frac{T_C}{(T_H - T_C)}
 \end{aligned}$$

$$\Delta S = \int \frac{dQ_{rev}}{T} = \int \frac{nRT}{vT} dv = nR \ln \frac{v_2}{v_1} = -nR \ln \frac{p_2}{p_1}$$

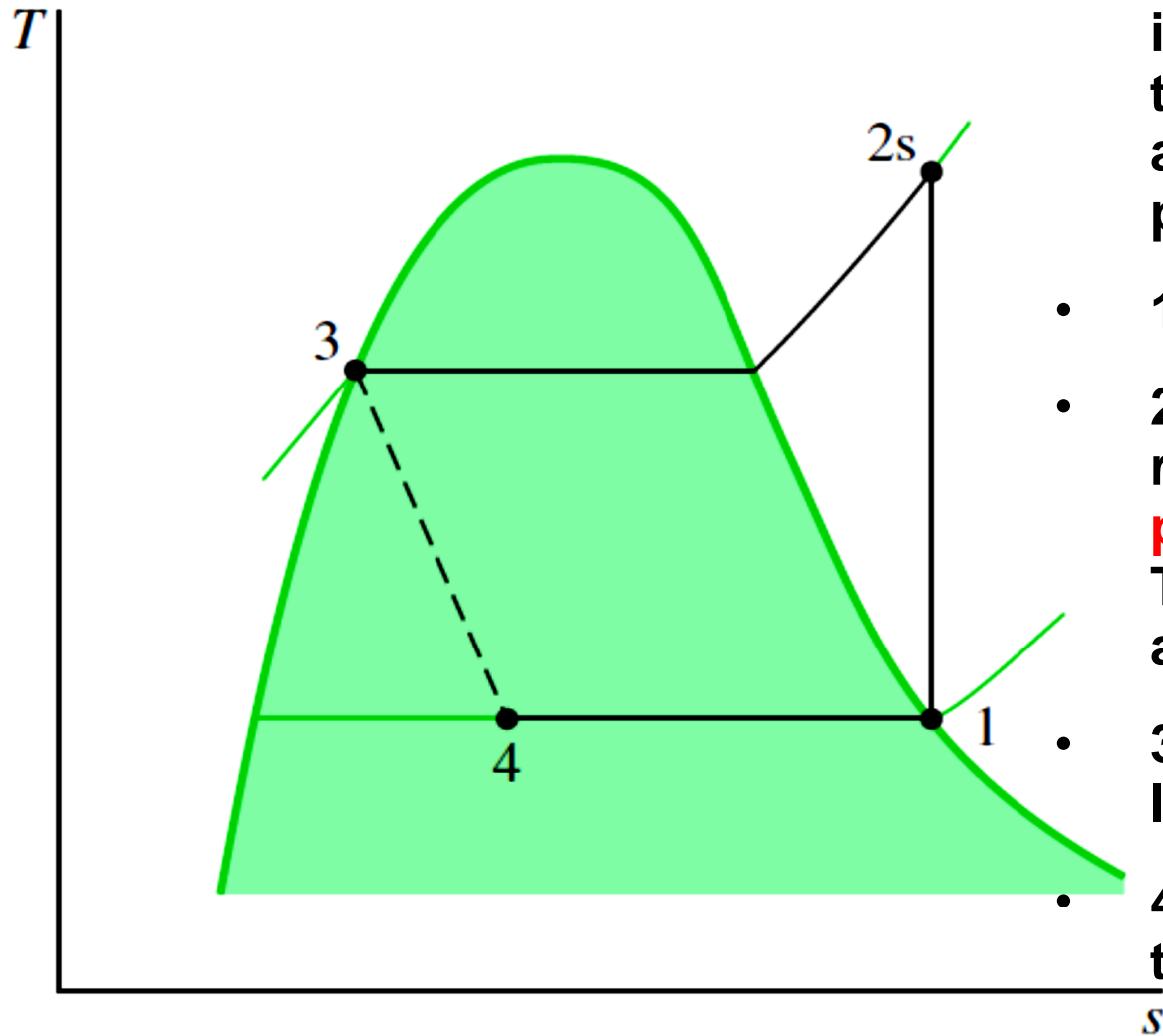
$$W_{rev} = -nR \ln \frac{p_2}{p_1}$$

# Vapor-Compression Refrigeration Systems



- The expansion produces a relatively small amount of work compared to the work input in the compression process.
  - Usually low in efficiency consider its operating condition.
- More practically, the work output of the turbine is normally sacrificed by substituting a simple throttling valve for the expansion turbine, with consequent savings in initial and maintenance costs.

# Ideal Vapor-Compression Cycle



- If compression occurs without irreversibilities, and stray heat transfer to the surroundings is also ignored, the compression process is **isentropic**.
- 1-2s Isentropic compression
- 2s-3 Heat transfer from the refrigerant as it flows at **constant pressure** through the condenser. The refrigerant exits as a liquid at state 3
- 3-4 Throttling process to a liquid-vapor mixture at 4.
- 4-1 **constant pressure** through the evaporator

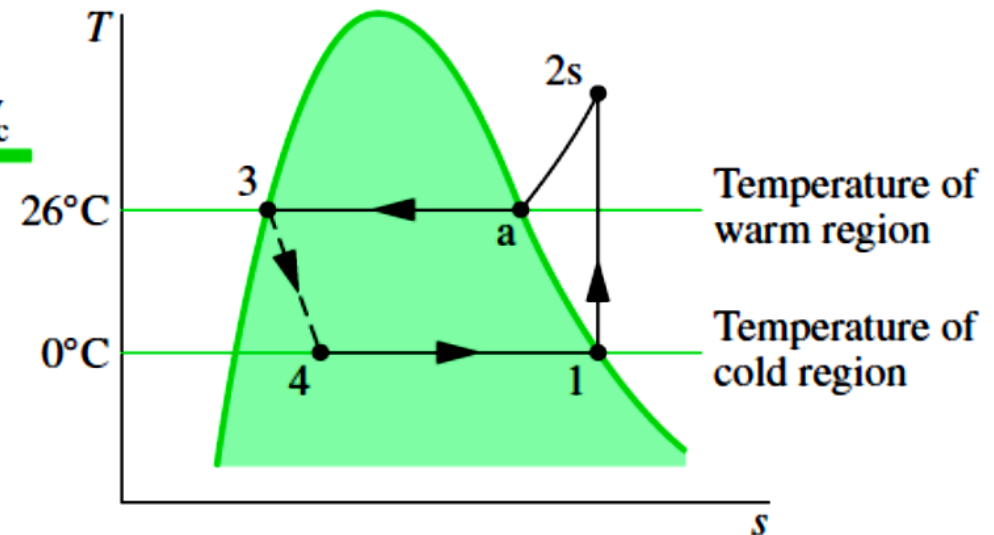
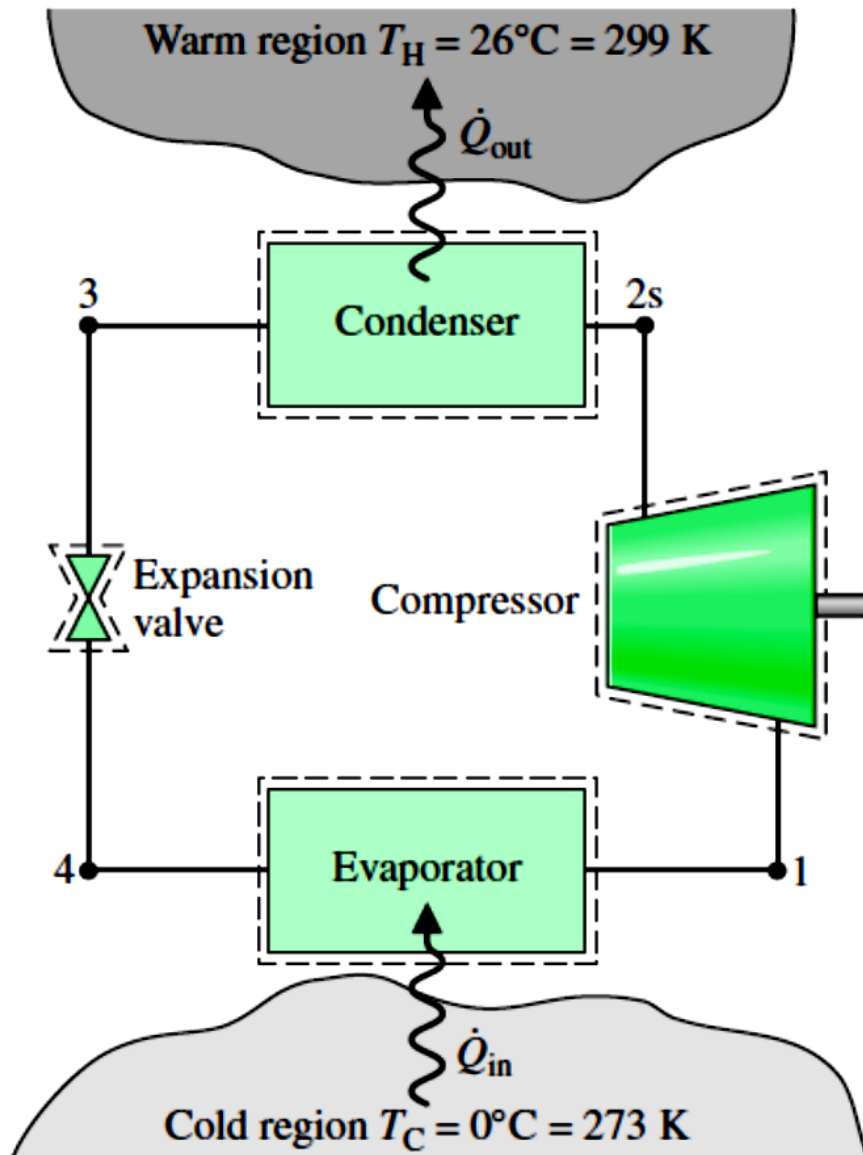


# EXAMPLE

Refrigerant 134a is the **working fluid** in an ideal vapor-compression refrigeration cycle that communicates thermally with a cold region at  $0^{\circ}\text{C}$  and a warm region at  $26^{\circ}\text{C}$ . **Saturated vapor** enters the compressor at  $0^{\circ}\text{C}$  and **saturated liquid** leaves the condenser at  $26^{\circ}\text{C}$ . The mass flow rate of the refrigerant is  $0.08\text{ kg/s}$ . Determine

- (a) the compressor power, in kW,
- (b) the refrigeration capacity, in tons,
- (c) the coefficient of performance, and
- (d) the coefficient of performance of a Carnot refrigeration cycle operating between warm and cold regions at  $26$  and  $0^{\circ}\text{C}$ , respectively.

# Schematic and Given Data



# Solution

- At the inlet to the compressor, the refrigerant is a saturated vapor at  $0^{\circ}\text{C}$ , so from the Table,  $h_1 = 247.23 \text{ kJ/kg}$  and  $s_1 = 0.9190 \text{ kJ/kg/K}$ .
- The pressure at state 2s is the saturation pressure corresponding to  $26^{\circ}\text{C}$ , or  $p_2 = 6.853 \text{ bar}$ . State 2s is fixed by  $p_2$  and the fact that the specific entropy is constant for the adiabatic, internally reversible compression process. The refrigerant at state 2s is a **superheated** vapor with  $h_{2s} = 264.7 \text{ kJ/Kg}$ .
- State 3 is saturated liquid at  $26^{\circ}\text{C}$ , so  $h_3 = 85.75 \text{ kJ/kg}$ . The expansion through the valve is a throttling process (assume reversible), so  $h_4 = h_3$ .

- **The compressor work input is**

$$\dot{W}_c = \dot{m} (h_{2s} - h_1)$$

- **where  $\dot{m}$  is the mass flow rate of refrigerant, working fluid.**

$$\begin{aligned} \dot{W}_c &= (0.08 \text{ kg} / \text{s})(264.7 - 247.23) \text{ kJ} / \text{kg} \left| \frac{1 \text{ kW}}{1 \text{ kJ} / \text{s}} \right| \\ &= 1.4 \text{ kW} \end{aligned}$$

- **The refrigeration capacity is the heat transfer rate to the refrigerant passing through the evaporator.**

$$\begin{aligned} \dot{Q}_{in} &= \dot{m} (h_1 - h_4) \\ &= (0.08 \text{ kg} / \text{s}) 60 \text{ s} / \text{min} (247.23 - 85.75) \text{ kJ} / \text{kg} \left| \frac{1 \text{ ton}}{211 \text{ kJ} / \text{min}} \right| \\ &= 3.67 \text{ ton} \end{aligned}$$

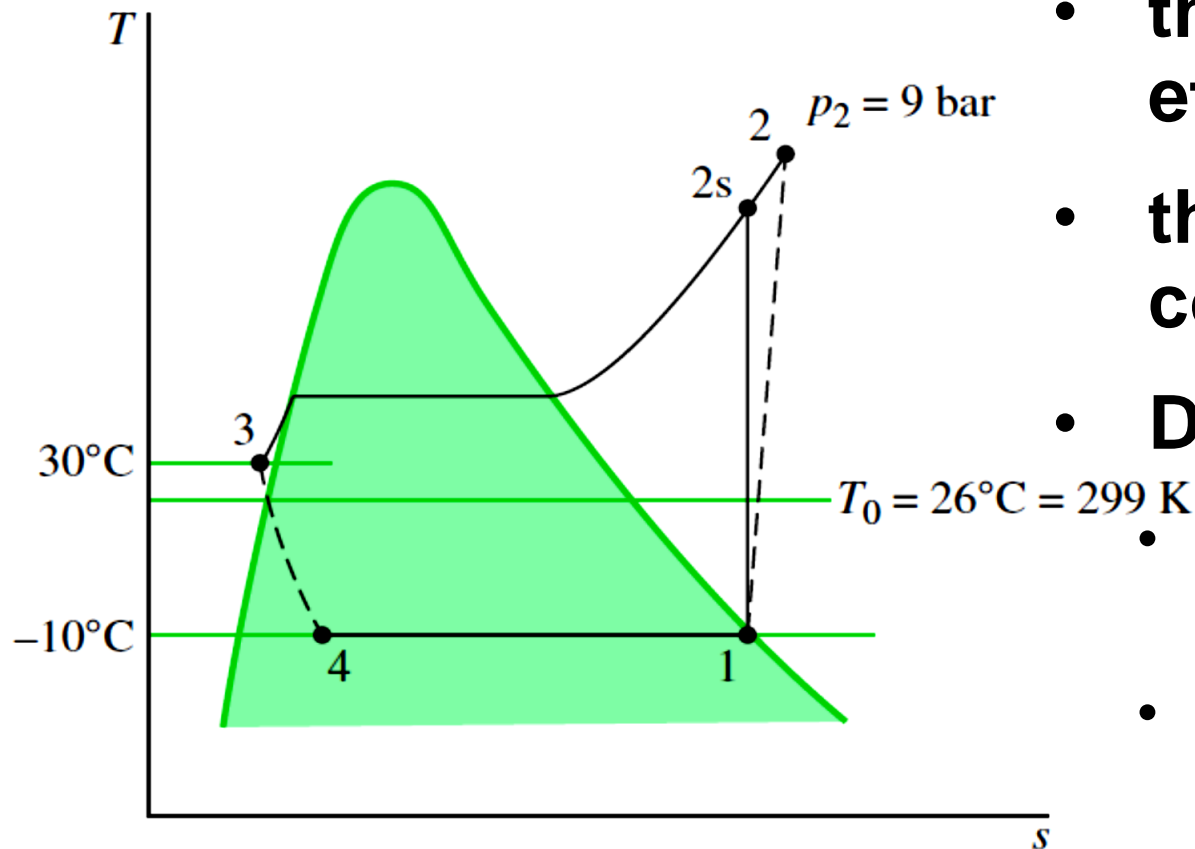
- **The coefficient of performance cop is**

$$cop = \frac{\dot{Q}_{in}}{\dot{W}_C} = \frac{h_1 - h_4}{h_{2s} - h_1} = \frac{247.23 - 85.75}{264.7 - 247.23} = 9.24$$

- **For a Carnot vapor refrigeration cycle operating at  $T_H = 299$  K and  $T_C = 273$  K, the coefficient of performance**

$$cop = \frac{T_C}{T_H - T_C} = 10.5$$

# Actual Vapor-Compression Refrigeration Cycle



- the compressor has an efficiency of 80%.
- the liquid leaving the condenser be at  $30^\circ\text{C}$ .
- **Determine**
  - the compressor power, in kW
  - the refrigeration capacity, in tons
  - the coefficient of performance

## Solution

- State 1 is the same as in previous Example, so  $h_1=241.35$  kJ/kg and  $s_1=0.9253$  kJ/kg/K.
- Owing to the presence of **irreversibilities** during the adiabatic compression process, there is an **increase in specific entropy** from compressor inlet to exit. The state at the compressor exit, state 2, can be fixed **using the compressor efficiency**

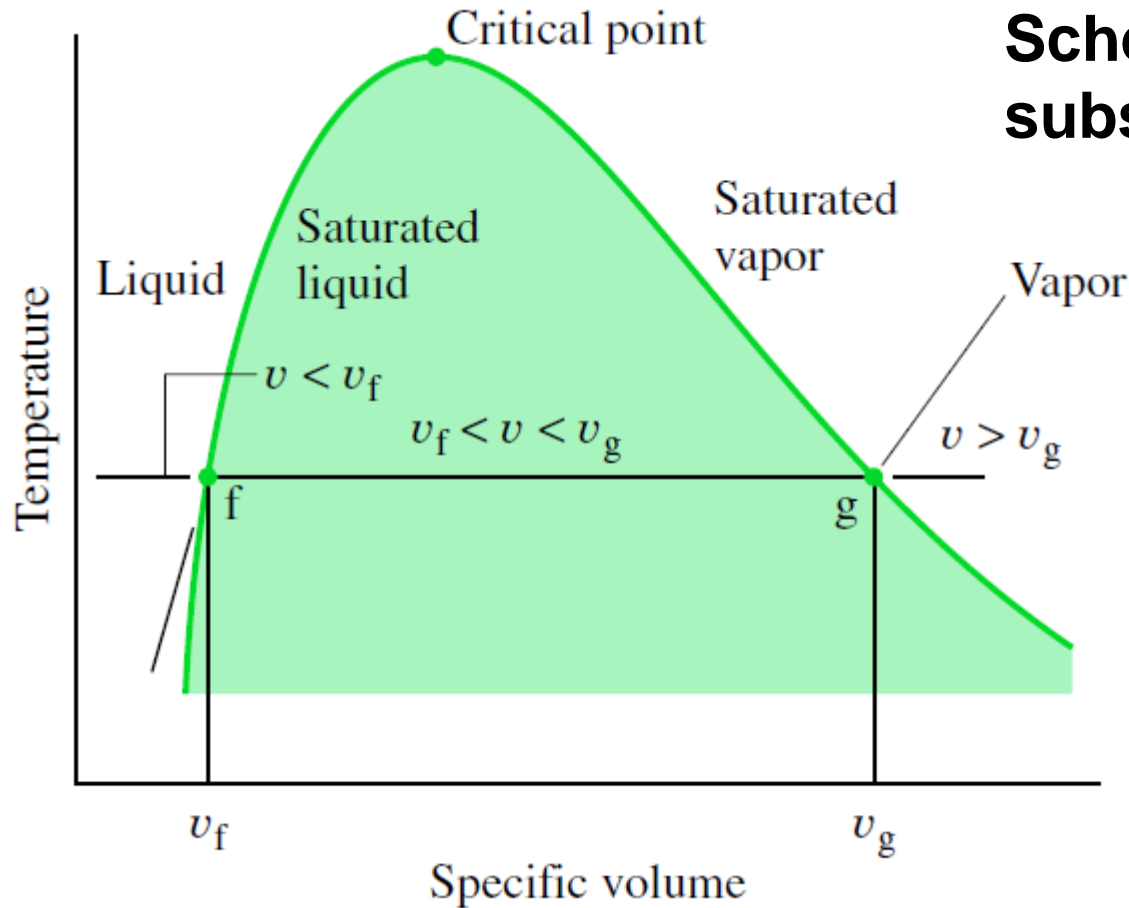
$$\eta_c = \frac{(\dot{W}_c / \dot{m})_s}{\dot{W}_c / \dot{m}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}$$

- From the previous Example,  $h_{2s}=272.39$  kJ/kg.

$$h_2 = \frac{h_{2s} - h_1}{\eta_c} + h_1 = \frac{(272.39 - 241.35)}{(0.80)} + 241.35 = 280.15 \text{ kJ / kg}$$

- **State 2 is fixed by the value of specific enthalpy  $h_2$  and the pressure,  $p_2=9$  bar. the specific entropy is  $s_2 =0.9497$  kJ/kg/K.**
- **State 3, at the condenser exit, is in the liquid region. The specific **enthalpy is approximated** by assuming that the specific volume and specific internal energy **change very little with pressure** at a fixed temperature. Together with saturated liquid data at 30°C, as follows:  $h_3 \approx h_{f3} =91.49$  kJ/kg. take the similar approach for the specific entropy  $s_3 \approx s_{f3} =0.3396$  kJ/kg/K.**
- **The expansion through the valve is a throttling process; thus,  $h_4=h_3$ .**





**Schematic diagram for subscript notation of g and f**

**state 4, the quality and specific entropy are,**

$$x_4 = \frac{h_4 - h_{f4}}{h_{g4} - h_{f4}} = \frac{91.49 - 36.97}{204.38} = 0.2667$$

$$s_4 = s_{f4} + x_4 (s_{g4} - s_{f4})$$

$$= 0.1486 + (0.2667)(0.9253 - 0.1486) = 0.3557 \text{ kJ / kg / K}$$

- **The compressor power is**

$$\begin{aligned}\dot{W}_c &= \dot{m} (h_2 - h_1) \\ &= (0.08 \text{ kg} / \text{s})(280.15 - 241.35) \text{ kJ} / \text{kg} \left| \frac{1 \text{ kW}}{1 \text{ kJ} / \text{s}} \right| = 3.1 \text{ kW}\end{aligned}$$

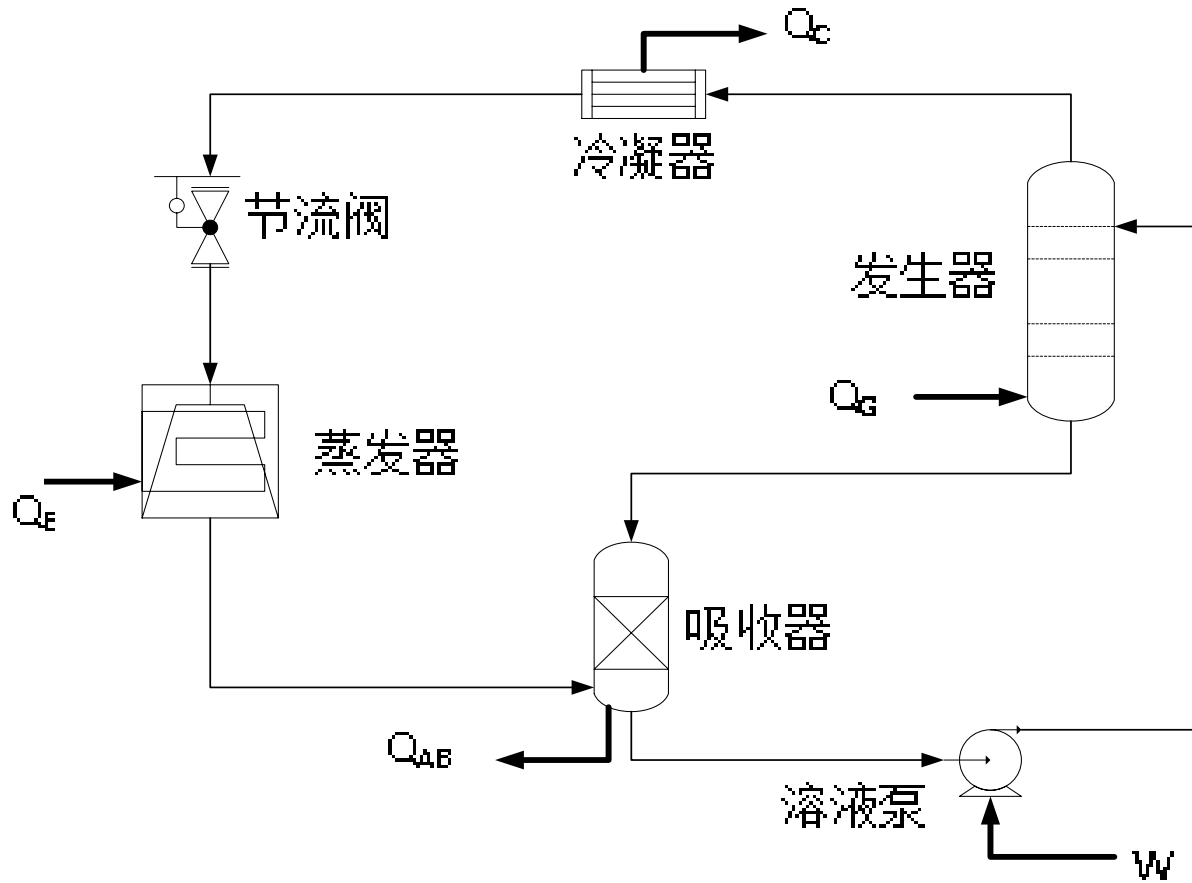
- **The refrigeration capacity is**

$$\begin{aligned}\dot{Q}_{in} &= \dot{m} (h_1 - h_4) \\ &= (0.08 \text{ kg} / \text{s})(60 \text{ s} / \text{min})(241.35 - 91.49) \text{ kJ} / \text{kg} \left| \frac{1 \text{ ton}}{211 \text{ kJ} / \text{min}} \right| \\ &= 3.41 \text{ ton}\end{aligned}$$

- **The coefficient of performance is**

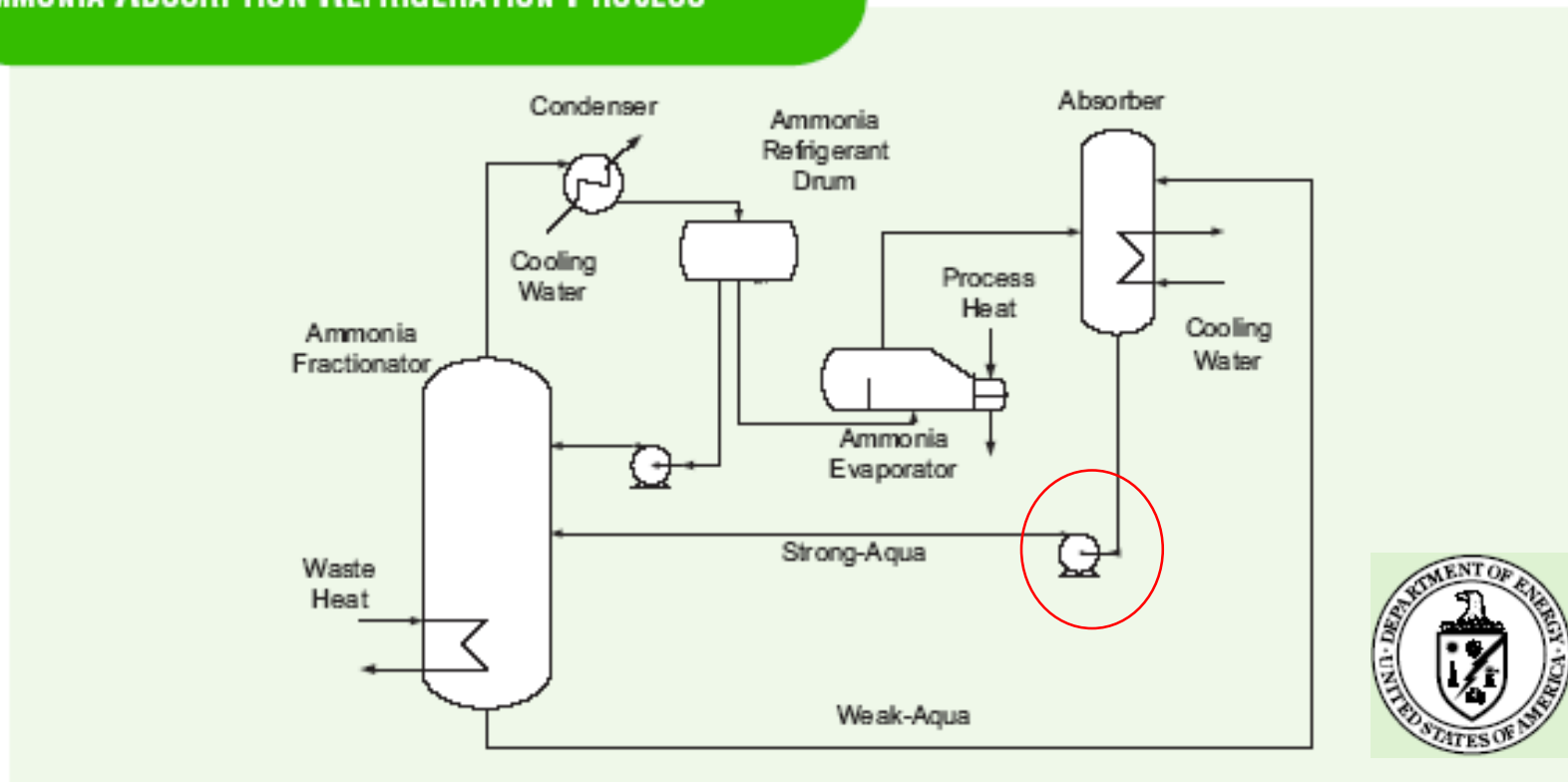
$$cop = \frac{(h_1 - h_4)}{(h_2 - h_1)} = \frac{(241.35 - 91.49)}{(280.15 - 241.35)} = 3.86$$

# Absorption Refrigeration



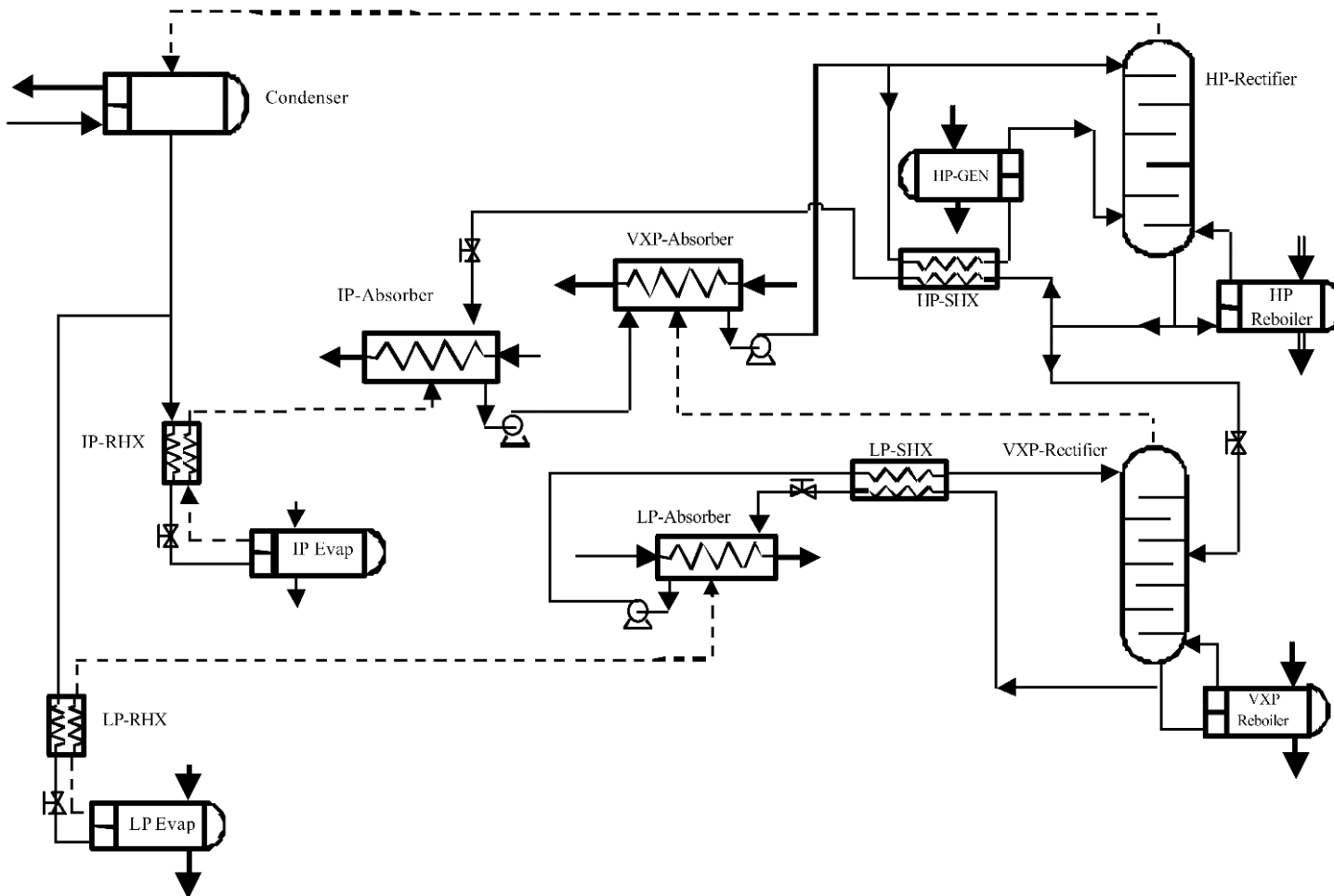
# DOE Conceptual Design (2003)

## AMMONIA ABSORPTION REFRIGERATION PROCESS



Ammonia absorption refrigeration conceptual flow diagram.

# BP Amoco Advanced Multi-Effect AAR Cycle



# **Integrated Direct Cooling Suspension Crystallization and Absorption Refrigeration Process for pX-Purification**

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ZHU Ying1, GUO Yan-zi1,2, WANG  
Yan1, GUO Si-si1, WU Jun1**

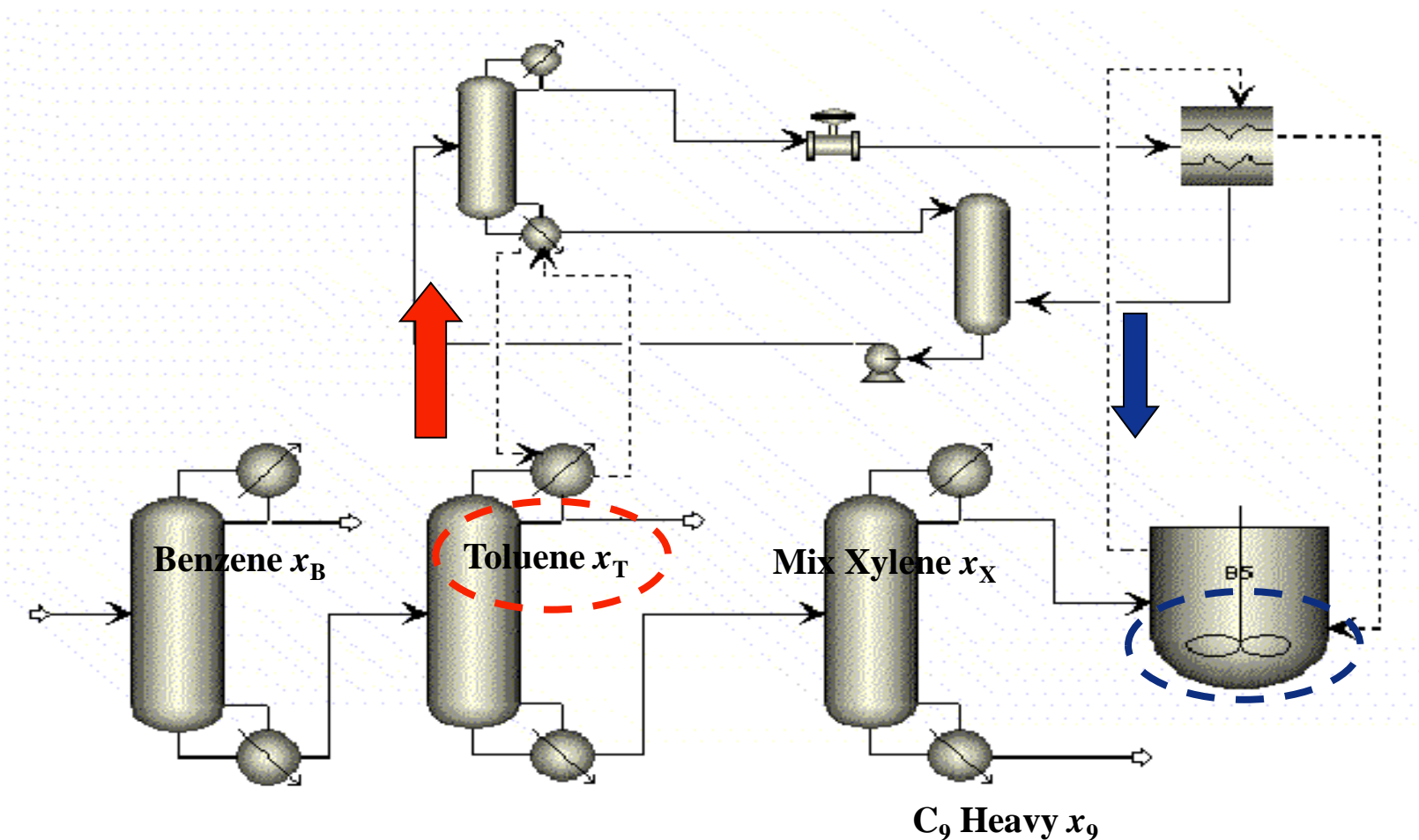
**1. Chemistry Department, Tongji  
University**

**2. Shanghai Research Institute of  
Petrochemical Technology, SINOPEC**

# **Petrochemical production consumes a great amount of energy**

- **Statistics from the China Petroleum and Chemical Industry Association (CPCIA) show that by 2006,**
  - **energy consumption in the petrochemical sector accounts for about 40 percent of the country's total, and**
  - **the sector's energy consumption per unit produced is 4.1 times higher than that in the North America.**
- **High energy consumption causes**
  - **company's competitiveness on the global stage**
  - **greater CO<sub>2</sub> green house emission, posing a major threat to the environment and sustainability of growth.**

# Assess if the waste heat from the toluene tower is enough

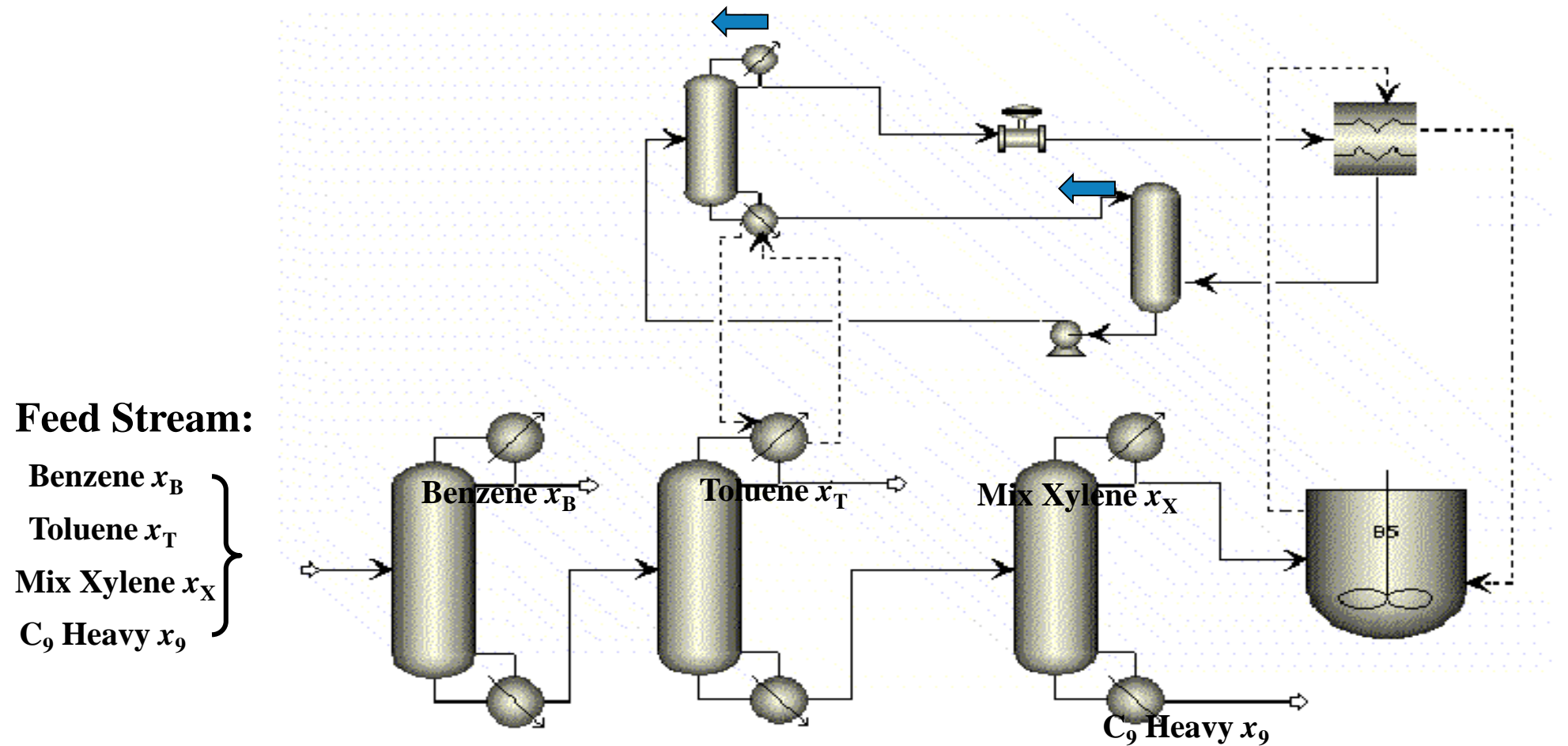


**Feed Stream:**  
 Benzene  $x_B$   
 Toluene  $x_T$   
 Mix Xylene  $x_X$   
 C<sub>9</sub> Heavy  $x_9$

$$\frac{Q_{\text{Crystallization}}}{Q_{\text{Toluene}}} = \frac{[144\text{kJ} / (\text{kg mixed xylenes})] \times [0.15 (\text{kg mixed xylenes}) / (\text{kg CSTDP product})]}{[319\text{kJ} / (\text{kg toluene})] \times [0.70 (\text{kg toluene}) / (\text{kg CSTDP product})]} \approx 0.097$$



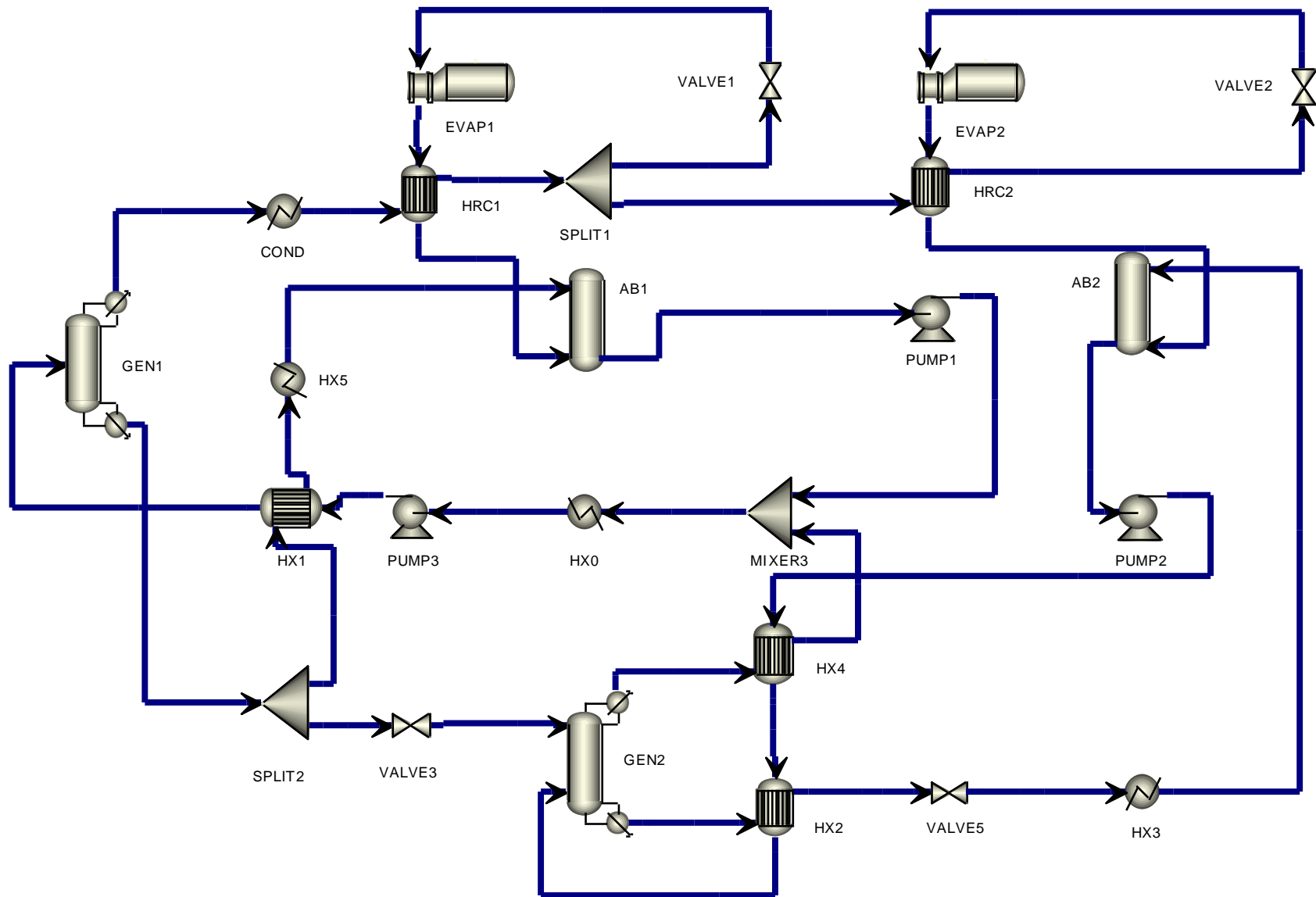
# Assess if the waste heat from the toluene tower is enough



$$\frac{Q_{\text{Crystallization}}}{Q_{\text{Toluene}}} = \frac{[144\text{kJ} / (\text{kg mixed xylenes})] \times [0.15 (\text{kg mixed xylenes}) / (\text{kg CSTDP product})]}{[319\text{kJ} / (\text{kg toluene})] \times [0.70 (\text{kg toluene}) / (\text{kg CSTDP product})]} \approx 0.097$$

# Multi effect propane-hydrocarbon absorption refrigeration system

Chinese Patent (Granted) 200910056897.9B



# Summary

- Xylene isomer ternary solid-liquid phase equilibrium was investigated experimentally, mathematical equations were derived.
- The optimized first stage temperature was found to be at around 248K~265K, which agrees well with the industrial operation conditions.
- **Multi effect propane-hexane absorption refrigeration process can deliver COP  $\geq 0.1$  at 245K and COP  $\geq 0.1$  at 271K.**
- Use refrigeration process to replace a compression refrigeration system would save
  - 47.1KWHr electricity per MT pX,
  - For every new increase pX capacity of 1000KMTA, annual saving of electricity will be **at 5.7 MW,**
  - equivalent to **indirect reduce CO2 emission 46K metric ton.**

# Exergy

- In distillation columns, this work is supplied by heat being injected at the reboiler  $q_{\text{reb}}$  and rejected at the condenser  $q_{\text{cond}}$ . The net work available from the heat energy (or the net exergy from the heat transferred) is:

$$Ex_{\text{heat}} = q_{\text{reb}} \left( 1 - \frac{T_0}{T_{\text{reb}}} \right) - q_{\text{cond}} \left( 1 - \frac{T_0}{T_{\text{cond}}} \right)$$