3.8.3. Vapor Compression Refrigeration and Heat Pump Cycles

The objective of a vapor compression refrigeration cycle is to remove energy from a cold reservoir and move it to the hot reservoir. For example, the cold reservoir may be the interior of a refrigerator or freezer and the hot reservoir is the kitchen, or it could be the interior of a home and the hot reservoir is the outdoors. In this latter case, the device running the refrigeration cycle is known as an air conditioner.

The objective of a heat pump cycle is to move energy from the cold reservoir into the hot reservoir. For example, the cold reservoir might be the outdoors and the hot reservoir would be the interior of a home. Another example is a heat pump water heater where the cold reservoir is the room containing the water heater and the hot reservoir is the water within the water heater tank.

The component schematics for refrigeration and heat pump cycles are identical. Similarly, the cycles traced out by the two cycles on a T-s plot have the same features. A schematic of the components and the T-s diagram are shown in Figure 3.45.

The temperature of the working fluid, typically a refrigerant, must be smaller than the temperature of the cold reservoir in order to have heat transfer into the working fluid in the evaporator (Process 4-1). Once the energy is absorbed, the compressor increases the pressure and temperature of the working fluid to raise its temperature to value larger than the hot reservoir temperature (Process 1-2). This increase in temperature is required in order for heat transfer to be from the working fluid in the condenser to the hot reservoir (Process 2-3). In order to return the working fluid to a cold temperature in the evaporator, a throttling device is used to decrease the pressure and temperature (Process 3-4).

Notes:

(1) Expansion of the working fluid from State 3 to State 4 is achieved through a throttling device rather than using a turbine. Although expansion through a throttling device is inherently non-isentropic, expansion through a turbine would also be inefficient and produce little power due to the low quality of the saturated vapor-liquid mixture and, thus, low specific enthalpies. In addition, a turbine is a more complex device and would present additional engineering and maintenance challenges. In a heat pump or air conditioner, the throttling device is in the form of an expansion valve. In a refrigerator, the throttling device is simply a long, narrow tube called a capillary tube. The pressure decreases across both devices (going from State 3 to State 4) since a higher pressure at the inlet side is required in order to push the viscous working fluid through the valve/tube.

(2) Real compressors operate best on superheated vapor rather than saturated liquid-vapor mixtures and, thus, State 1 is often in the superheated vapor region. Compression of superheated vapor is known as “dry compression” while compression of a saturated liquid-vapor mixture is called “wet compression”.

(3) For a refrigeration cycle, the rate of heat transfer from the cold reservoir is known as the refrigeration capacity. One ton of refrigeration capacity = the rate of heat transfer required to freeze one ton of water in 24 hrs with an enthalpy of fusion of 334 kJ/kg = 200 Btu/min ≈ 211 kJ/min = 3.517 kW.

(4) Recall that the “efficiency” of refrigeration and heat pump cycles are quantified using coefficients of performance,

$$COP_{\text{ref}} = \frac{\dot{Q}_C}{\dot{W}_C}, \tag{3.194}$$

$$COP_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_C}. \tag{3.195}$$

(5) A refrigeration/heat pump cycle with a 100% isentropic efficiency is called an “ideal” refrigeration/heat pump cycle. Note that the throttling device is still non-isentropic, however. In an actual heat pump cycle, the location of State 1 is in the superheated vapor region (discussed in a previous note) and the location of State 3 may be in the compressed liquid region.

(6) The coefficients of performance for the refrigeration/heat pump cycle shown in Figure 3.45 are less than the coefficients of performance for refrigeration/heat pump Carnot cycles operating between

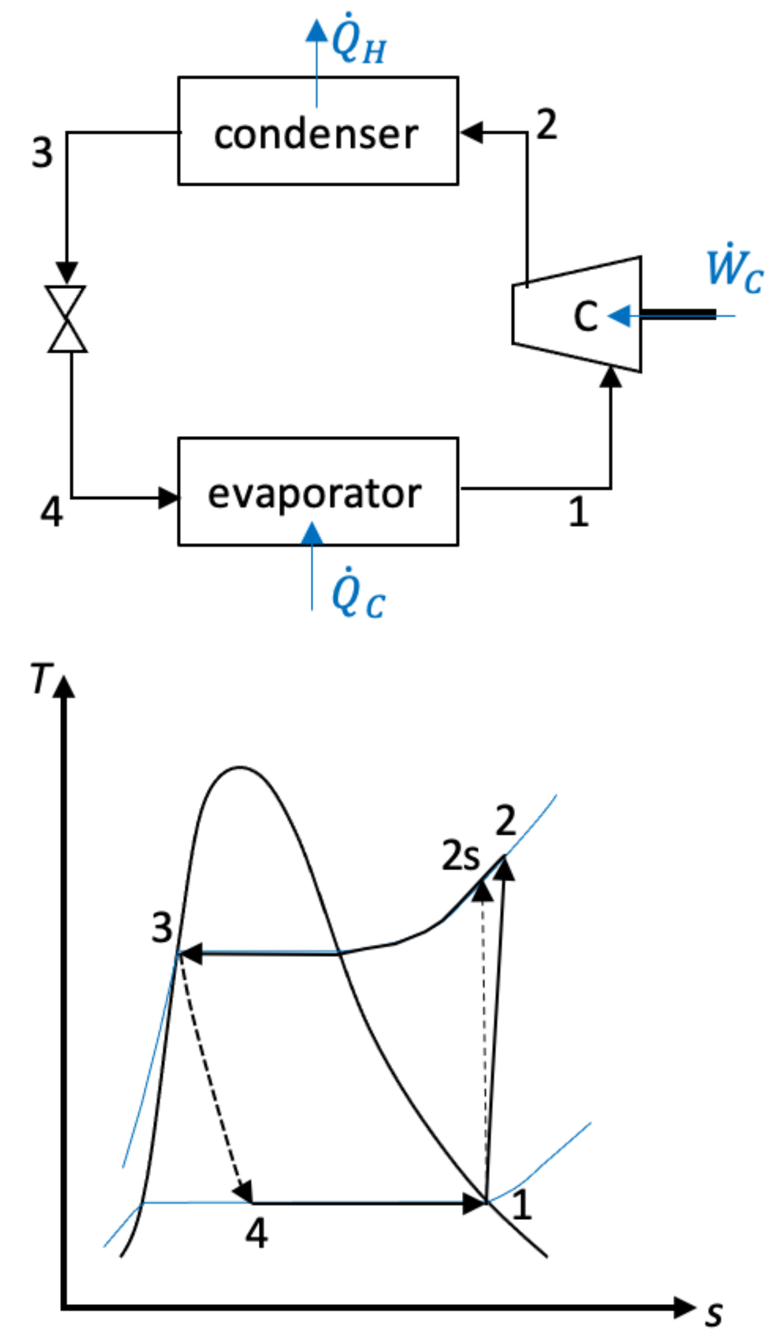


Figure 3.45. A schematic showing the components of refrigeration and heat pump cycles, and the corresponding T - s plot. The dashed line for Process 3-4 (flow through the throttling device) indicates that this process occurs abruptly and in an uncontrolled manner, i.e., the path is not well defined for this process.

identical thermal reservoirs. Recall that the Carnot cycle coefficients of performance are,

$$COP_{\text{ref,rev}} = \frac{T_C}{T_H - T_C}, \tag{3.196}$$

$$COP_{\text{HP,rev}} = \frac{T_H}{T_H - T_C}. \tag{3.197}$$

Even if the compressor has 100% isentropic efficiency, the fact that the temperature isn't constant in the condenser and evaporator (where heat transfer occurs), the non-isentropic expansion in the throttling device, and other real-world effects (viscosity, heat transfer across a finite temperature difference) results in decreased thermal efficiency.

(7) Using Eq. (3.196) *as a guide*, we see that as the hot reservoir temperature (T_H) decreases, the coefficient of performance for a refrigeration cycle increases. Thus, keeping a refrigerator/freezer in a cool basement or a garage during the winter will improve the device's thermodynamic efficiency.

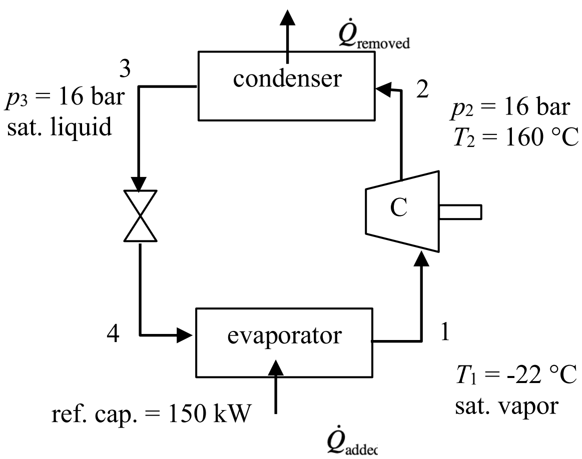
(8) Using Eq. (3.197) *as a guide*, we see that as the cold reservoir temperature (T_C) decreases, the coefficient of performance for a heat pump decreases. Home heat pumps using atmospheric air as the cold reservoir often have electric back-up heaters in order to account for the decreased efficiency when the air temperature becomes very cold. For example, if $T_H = 20\,^\circ\text{C}$ and $T_C = 10\,^\circ\text{C}$, then $COP_{\text{HP,rev}} = 29.3$. However, if $T_C = 0\,^\circ\text{C}$, then $COP_{\text{HP,rev}} = 14.7$. Of course a real heat pump won't have the same coefficients of performance as a Carnot cycle heat pump, but the principle that the coefficient of performance decreases as the cold reservoir temperature decreases still holds. Heat pumps using the Earth or a large body of water as the cold reservoir are less susceptible to this issue since the cold reservoir temperature in those cases remains nearly constant.

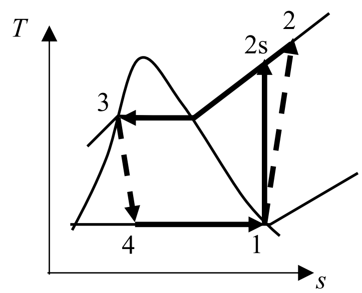
(9) A heat pump can be designed to operate as an air conditioner (i.e., a refrigeration cycle) through the use of a reversing valve. To learn more on this topic, the reader is encourage to look online for more detailed descriptions on practical heat pump design.

In a vapor-compression refrigeration cycle, ammonia exits the evaporator as saturated vapor at -22 °C. The refrigerant enters the condenser at 16 bar (abs) and 160 °C, and saturated liquid exits at 16 bar (abs). There is no significant heat transfer between the compressor and its surroundings, and the refrigerant passes through the evaporator with a negligible change in pressure. If the refrigerating capacity is 150 kW, determine:

a. the mass flow rate of refrigerant,
b. the power input to the compressor,
c. the coefficient of performance, and
d. the isentropic compressor efficiency.

SOLUTION:





First determine the properties at the various states using Tables from Moran et al., 7th ed.

State 1: T_1 = -22 °C, saturated vapor (Table A-13)
$\Rightarrow$ p_1 = 1.7390 bar, h_1 = 1415.08 kJ/kg, s_1 = 5.6457 kJ/(kg.K)

State 2: p_2 = 16 bar, T_2 = 160 °C $\Rightarrow$ superheated vapor (Table A-15)
$\Rightarrow$ h_2 = 1798.45 kJ/kg, s_2 = 5.7475 kJ/(kg.K)

State 3: p_3 = 16 bar, saturated liquid (Table A-14)
$\Rightarrow$ T_3 = 41.03 °C, h_3 = 376.46 kJ/kg, s_3 = 1.3729 kJ/(kg.K)

State 4: throttling process from 3 to 4, constant pressure from 4 to 1
$\Rightarrow$ $h_4 = h_3$ = 376.46 kJ/kg, $p_4 = p_1$ = 1.7390 bar

The mass flow rate may be determined by applying the 1st Law to the evaporator and making use of the refrigeration capacity (= $\dot{Q}_{\text{added}}$ = 150 kW),

$$\dot{Q}_{\text{added}} = \dot{m}\left(h_1 - h_4\right) \Rightarrow \dot{m} = \frac{\dot{Q}_{\text{added}}}{\left(h_1 - h_4\right)}, \tag{1}$$

$\Rightarrow$ $\boxed{\dot{m} = 0.144 \text{ kg/s}}$.

The power input into the compressor is found by applying the 1st Law to the compressor,

$$\dot{W}_{\text{on comp}} = \dot{m}\left(h_2 - h_1\right), \tag{2}$$

$\Rightarrow$ $\boxed{\dot{W}_{\text{on comp}} = 55.4 \text{ kW}}$.

The coefficient of performance for the refrigeration cycle is defined as,

$$\text{COP}_{\text{ref}} \equiv \frac{\dot{Q}_{\text{added}}}{\dot{W}_{\text{on}}}, \tag{3}$$

$\Rightarrow$ $\boxed{\text{COP}_{\text{ref}} = 2.71}$.

The isentropic efficiency of the compressor is defined as,

$$\eta_{\text{comp}} \equiv \frac{\dot{W}_{\text{on comp},s}}{\dot{W}_{\text{on comp}}} = \frac{\dot{W}_{\text{on comp},s}/\dot{m}}{\dot{W}_{\text{on comp}}/\dot{m}} = \frac{h_{2s} - h_1}{h_2 - h_1}, \tag{4}$$

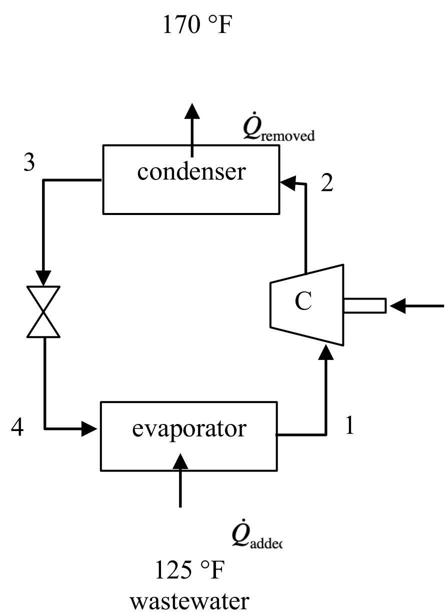
where

$p_{2s} = p_2 = 16$ bar and $s_{2s} = s_1 = 5.6457$ kJ/(kg.K) $\Rightarrow$ $h_{2s} = 1755.38$ kJ/kg, $T_{2s} = 143$ °C (interpolating from Table A-15),

$\Rightarrow$ $\boxed{\eta_{\text{comp}} = 0.888}$.

A process requires a heat transfer rate of 3*10^6 Btu/h at 170 °F. It is proposed that a Refrigerant 134a vapor-compression heat pump be used to develop the process heating using a wastewater stream at 125 °F as the lower-temperature source. The compressor isentropic efficiency is 80%. Sketch the *T-s* diagram for the cycle and determine the:

a. specific enthalpy at the compressor exit, in Btu/lb$_m$,
b. temperatures at each of the principal states in °F,
c. mass flow rate of the refrigerant in lb$_m$/h,
d. compressor power, in Btu/h, and
e. coefficient of performance and compare with the coefficient of performance for a Carnot heat pump cycle operating between reservoirs at the process temperature and the wastewater temperature, respectively.



State	p [psia]	h [Btu/lb$_m$]
1	180	116.74
2	400	?
3	400	76.11
4	180	76.11

SOLUTION:

The specific enthalpy at state 2 may be found since the compressor efficiency is known ($\eta_{\text{comp}} = 0.80$),

$$\eta_{\text{comp}} \equiv \frac{\dot{W}_{\text{on comp},s}}{\dot{W}_{\text{on comp}}} = \frac{\dot{W}_{\text{on comp},s}/\dot{m}}{\dot{W}_{\text{on comp}}/\dot{m}} = \frac{h_{2s}-h_1}{h_2-h_1} \Rightarrow h_2 = h_1 + \frac{h_{2s}-h_1}{\eta_{\text{comp}}}, \tag{1}$$

where,

$s_{2s} = s_1 = 0.2154$ Btu/(lb$_m$.°R) using Table A-11E from Moran et al., 7$^{\text{th}}$ ed. with $p_1 = 180$ psia and knowing state 1 is in a saturated vapor state ($\boxed{T_1 = 117.74\ \text{°F}}$)

$\Rightarrow h_{2s} = 123.32$ Btu/lb$_m$ knowing $p_{2s} = p_2 = 400$ psia (Table A-12E and interpolation; $T_{2s} = 186$ °F, SHV)

$\Rightarrow$ $\boxed{h_2 = 124.97\ \text{Btu/lb}_\text{m}}$ (using Table A-12E and interpolation $\boxed{T_2 = 191.63\ \text{°F}}$, SHV)

Knowing $p_3 = p_2 = 400$ psia and $h_3 = 76.11$ Btu/lb$_m$ (saturated liquid state, Table A-11E) $\Rightarrow$ $\boxed{T_3 = 179.95\ \text{°F}}$.
Knowing $p_4 = p_1 = 180$ psia and $h_4 = 76.11$ Btu/lb$_m$ (SLVM Table A-11E) $\Rightarrow$ $\boxed{T_4 = 117.74\ \text{°F}}$.

Apply the 1$^{\text{st}}$ Law to the condenser to determine the mass flow rate,

$$\dot{Q}_{\text{removed}} = \dot{m}\left(h_2 - h_3\right) \Rightarrow \dot{m} = \frac{\dot{Q}_{\text{removed}}}{\left(h_2 - h_3\right)}, \tag{2}$$

$\Rightarrow$ $\boxed{\dot{m} = 6.13 \times 10^4\ \text{lb}_\text{m}/\text{h}}$.

The compressor power is found by applying the 1st Law to the compressor,

$$\dot{W}_{\text{on comp}} = \dot{m}\left(h_2 - h_1\right), \tag{3}$$

$$\Rightarrow \boxed{\dot{W}_{\text{on comp}} = 5.05*10^5 \text{ Btu/h}}.$$

The coefficient of performance for the heat pump cycle is,

$$\text{COP}_{\text{hp}} \equiv \frac{\dot{Q}_{\text{removed}}}{\dot{W}_{\text{on comp}}}, \tag{4}$$

$$\Rightarrow \boxed{\text{COP}_{\text{hp}} = 5.95}.$$

The COP_{hp} for the corresponding Carnot cycle operating between T_C = 125 °F (= 585 °R) and T_H = 170 °F (= 630 °R) is,

$$\text{COP}_{\text{hp,rev}} = \frac{T_H}{T_H - T_C}, \tag{5}$$

$$\Rightarrow \boxed{\text{COP}_{\text{hp,rev}} = 14}.$$

The Carnot cycle COP is larger than the actual COP, as expected. Much of the cause for irreversibility in the actual system is due to the fact that the system temperatures from 2-3 and from 4-1 are substantially different than the hot and cold reservoir temperatures of T_H = 170 °F and T_C = 125 °F leading to large, finite temperature differences. Such large differences are needed for practical heat transfer rates between the condenser, evaporator, and the surroundings.

A sketch of the states and processes are shown on the following *T*-*s* diagram.