Test 3

problem 2

Purpose: * Operating pressure of evaporator & condenser.

* State the refrigeration after each elements of the vapor-compression refrigeration cycle. (provide pressure, temp., enthalpy, & quality of each state).

* P-v & T-s diagram

* COP of designed cycle.

* Required refrigerant mass flow rate

* Power required by compressor in HP.

* Waste heat rate.

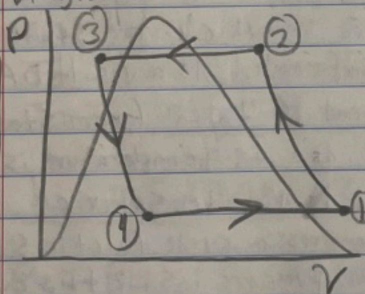
Data:

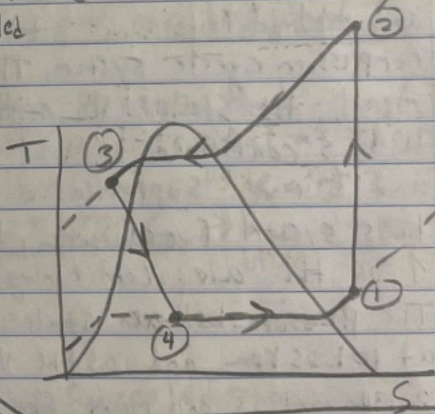
- after evaporator $T = 2.7°C$ superheated
- after condenser $T = 6.3°C$ subcooled
- $\eta^{th}_{cant} = .8$

Materials:

Refrigerant-134a

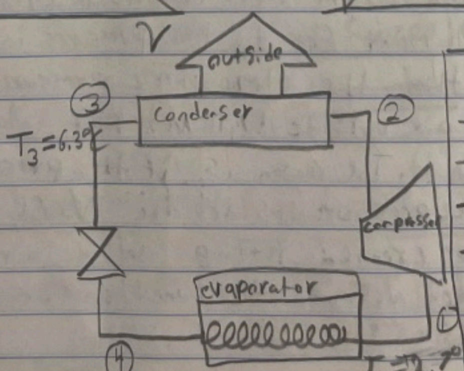
Diagrams




Sources
Thermodynamics
an engineering
approach,
Ayala,
example 11-49
Slides & notes

Diagram:




Assumptions:

- $h_3 \neq h_4$
- $T_4 \leq T_1$
- State 2 $x_2 = 1$
- heat into evaporator is equal to heat Rejected by dry air in C&D unit

procedure:

1. States: Fully define state 1 & state 3 based on given values
2. assume a design pressure 1 & pressure 2 → solve for desired values
3. fully define / calculations
4. Analysis

Summary:

This part of the system is based on a real vapor compression cycle. Refrigerant-134a is used as the working fluid in the design such that it is compressed, condensed, expanded and evaporated throughout a single full cycle. through the evaporator, the system absorbs the heat from the air in the cooling and dehumidifying unit to transfer as heat in to the vapor compression cycle system. The system being real and not ideal means the states do not lie on the saturation lines and this is confirmed in the question in which the states 1 and 3 are superheated and subcooled. As a result of this system functioning, the heat is taken from state 1 of the air conditioning cycle and its temperature is reduced. The pressure between states 1 and 1` is atmospheric at 101.25 Kpa and as the vapor compression cycle functions on R-134a and is only meant for it, the pressures inside this cycle must be higher than the atmospheric pressure so in case of a leak, the air outside will not leak into the cycle which would ruin it. The reason is if the pressure is lower than atmospheric pressure inside the V. C. cycle a vacuum will be created letting outside air in and the V. C. cycle is not designed to function on outside air and R-134a mixtures.

States:

| ① isentropic Compressor → | ② P-const condenser → | ③ expansion Valve → | ④ Pconst evaporator → |
|---|---|---|---|
| $T_1 = 2.7°C$ | $P_2 = 600 kpa$ | $T_3 = 6.3°C$ | $h_4 = 188.66 \frac{kJ}{kg}$ |
| $X_1 = 1$ | $T_2 = 37.78°C$ | $X_3 = 0$ | $X_4 = 0.71$ |
| $h_1 = 254.3236 \frac{kJ}{kg}$ | $X_2 = 1$ | $P_3 = 600 kpa$ | $T_4 = -5.38°C$ |
| $P_1 = 240 kpa$ | $h_{2s} = 274.043 \frac{kJ}{kg}$ | $h_3 = 60.36 \frac{kJ}{kg}$ | $P_4 = 240 kpa$ |
| $S_1 =$ | $h_{2a} = 278.972 \frac{kJ}{kg}$ | | |

Calculations:

③ $h_3 \rightarrow @6.3°C \rightarrow$ interpolated $\underline{h_3 = 60.36 \frac{kJ}{kg}}$ $h_3 \neq h_4$ b/c it just can't work

① Assume a design pressure → 240 kpa based on the Tsat in superheated tables → $P_1 = 240$ kpa → Tsat must be less than $T_1 = 2.7°C$

$S_1 @ T_1 \& P_1 = 0.9604 \frac{kJ}{kg \cdot K}$

② $S_2 = S_1$ → Assume a second pressure that can fit the entropy comfortably and is equal at the T@ state 2

$T_2 = 10°F$ greater than Ambient (90°F outside air)

$T_2 = \frac{100°F - 32°F}{1.8} = \underline{T_2 = 37.78°C}$ → $h_{2s}$ interpolated btwn 30°C & 40°C

$\underline{h_{2s} = 274.043 \frac{kJ}{kg}}$

③ $P_3 = P_2$ → $\underline{600 kpa,\ X_3 = \text{subcooled} \rightarrow \text{liquid},\ 0}$

② $\eta_{th} = 80\% = 0.8$ compressor $\quad \eta_{th} = \frac{h_{2s} - h_1}{h_{2a} - h_1} = 0.8 = \frac{274.043 \frac{kJ}{kg} - 254.3236 \frac{kJ}{kg}}{h_{2a} - 254.3236 \frac{kJ}{kg}}$

$\underline{h_{2a} = 278.972 \frac{kJ}{kg}}$

1st law around evaporator / cooling & dehumidifying unit

$q_{in} = h_i - h_1 + h_w \rightarrow Cp_{air}(T_i - T_1) + h_w$ $\quad Cp_{air} = 1.005 \frac{kJ}{kg \cdot °C}$

$1.005 \frac{kJ}{kg \cdot °C}(24.72°C - 12.77°C) + 53.661 \frac{kJ}{kg}$ $\quad T_{in}/T_i = 76.5°F = 24.72°C$

$q_{in} = 65.665 \frac{kJ}{kg}$ $\quad T_{out}/T_1 = 55°F = 12.77°C$

$h_w = 23.07 \frac{btu}{lbm} = 53.661 \frac{kJ}{kg}$

$1^{st}$ law around evaporator

$q_{in} = h_1 - h_4 = 65.665\,\frac{kJ}{kg} = 254.3236\,\frac{kJ}{kg} - h_4$

$\underline{h_4 = 188.66\,\frac{kJ}{kg}}$

$T_4 \leq 2.7°C$

b/c it is just after cooling from expansion valve and goes to a cooler point than $T_1$ @ 2.7°C

$T_4$ in saturation tables

$T_4 @ P_4$ tables saturated $= -5.38°C$

$X_4 = \frac{h_4 - h_{f@P_4}}{h_{fg@P_4}} = \frac{188.66\,\frac{kJ}{kg} - 44.64\,\frac{kJ}{kg}}{202.68\,\frac{kJ}{kg}}$

$\underline{X_4 = 0.71}$

$Cp_{air} = 1.005\,\frac{kJ}{kg \cdot C}$

Previous problem 1 values

$\dot{m}_a = 4.42\,\frac{lbm}{s} = 2.005\,\frac{kg_a}{s}$

$\dot{m}_w = 0.00515\,\frac{lbv}{s} = 0.002336\,\frac{kg_v}{s}$

$h_w = 23.07\,\frac{btu}{lbm} = 53.661\,\frac{kJ}{kg}$

$\dot{Q}_{in} = \dot{m}_a \cdot Cp_a (T_1 - T_1) + \dot{m}_w \cdot h_w$

$\dot{Q}_{in} = 2.005\,\frac{kg_a}{s} \cdot 1.005\,\frac{kJ}{kg \cdot C} (24.72°C - 12.77°C) + 0.002336\,\frac{kg_v}{s} \cdot 53.661\,\frac{kJ}{kg}$

$\boxed{\dot{Q}_{in} = 24.205\ KW}$

$\dot{m}_R = \frac{\dot{Q}_{in}}{q_{in}} = \frac{24.205\ KW}{65.665\,\frac{kJ}{kg}} = \boxed{0.3686\,\frac{kg}{s} = \dot{m}_R}$

$1^{st}$ law compressor

$\dot{W}_{in} = \dot{m}_R (h_{2a} - h_1) = 0.36855\,\frac{kg}{s} (278.972\,\frac{kJ}{kg} - 254.3236\,\frac{kJ}{kg})$

$\boxed{\dot{W}_{in} = 9.085\ KW}$

$1^{st}$ law condenser → waste heat rate

$\dot{Q}_{out} = (h_{2a} - h_3)\dot{m}_R = (278.972\,\frac{kJ}{kg} - 60.36\,\frac{kJ}{kg})\,0.3686\,\frac{kg}{s}$

$\boxed{\dot{Q}_{out} = 80.58\ KW}$

$COP_{HP} = \frac{\dot{Q}_{out}}{\dot{W}_{in}} = \frac{80.58\ KW}{9.085\ KW} = \boxed{8.87 = COP_{HP}}$

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Technical analysis:

The vapor compressim cycle runs on refrigent R-134a and based on the design pressures 240kPa & 600kPa the $\dot{Q}_{out}$ / waste heat leaving the system is $80.58 \frac{kJ}{s}$, while it is taking heat in from the cooling and dehumidifying unit $\dot{Q}_{in}$ at $24.205 \frac{kJ}{s}$. based off of this, the heat rejected is nearly 4 times the heat collected from the cooling and dehumidifying unit its connected to. The system's coefficient of performance of 8.87 reflects the heat ejected from the system per the amount of work into the system. if the system were altered that is the main AC system such that the heat leaving and entering the cooling and dehumidifyunit and the evaporator was altered would affect the condition / states of the refrigerent. This will alter the enthalpies of each state to the extent of possibly changing the quality. inthe end, if the outside temp. becomes too cold the vaper compression cycle could cease to function properly. To fix this issue making both the condenser and evaporater capable of functioning as the other would make it adaptable to outside temperature.