Secondly: WET RUN

Determination of Air States:

units

$kJ := 1000J$

Ambient pressure: $p_{atm} := 102.1kPa$

At Point A:

$T_{Adb} := 22.4\,°C$ $T_{Awb} := 14.2\,°C$

Linear Interpolation for Saturation Pressure of Water:

Saturated water temperature Table A-4, p. 890 Y. A. Cengel and M. A. Boles, "Thermodynamics An Engineering Approach," 5th ed., McGraw Hill (2006).

$$p_{satH2O} := \begin{pmatrix} 0.6117 \\ 0.8725 \\ 1.2281 \\ 1.7057 \\ 2.3392 \\ 3.1698 \\ 4.2469 \\ 5.6291 \\ 7.3851 \\ 9.5953 \\ 12.352 \\ 15.763 \\ 19.947 \\ 25.043 \\ 31.202 \\ 38.597 \\ 47.416 \\ 57.868 \\ 70.183 \\ 84.609 \end{pmatrix} kPa \qquad T_{satH2O} := \begin{pmatrix} 0.01 + 273.15 \\ 5 + 273.15 \\ 10 + 273.15 \\ 15 + 273.15 \\ 20 + 273.15 \\ 25 + 273.15 \\ 30 + 273.15 \\ 35 + 273.15 \\ 40 + 273.15 \\ 45 + 273.15 \\ 50 + 273.15 \\ 55 + 273.15 \\ 60 + 273.15 \\ 65 + 273.15 \\ 70 + 273.15 \\ 75 + 273.15 \\ 80 + 273.15 \\ 85 + 273.15 \\ 90 + 273.15 \\ 95 + 273.15 \end{pmatrix} K$$

Function Definitions:

$$h_{ca}(T_{Adb}, \omega_A) := \left[T_{Adb}/°C + \omega_A \cdot (2501 + 1.805 \cdot T_{Adb}/°C)\right]\frac{kJ}{kg}$$

Solution:

1) Start by finding the saturation pressure at the wet bulb temperature:

$P_{gw} = P_{sat} @ T_{wb}$ $p_{Agw} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Awb})$ $\boxed{p_{Agw} = 1.629 \cdot kPa}$

2) Calculate the vapor pressure using Carrier's Equation:

$$p_{Av} := p_{Agw} - \frac{(p_{atm} - p_{Agw})\cdot(T_{Adb}/°C - T_{Awb}/°C)}{1532.4 - 1.3\cdot T_{Awb}/°C}$$ $\boxed{p_{Av} = 1.085\cdot kPa}$

3) calculate the specific humidity ω:

$$\omega = 0.622\frac{P_v}{P - P_v} \qquad \omega_A := 0.622\cdot\frac{p_{Av}}{p_{atm} - p_{Av}}$$ $\boxed{\omega_A = 0.0067}$

4) Determine the saturation pressure at the dry bulb temperature p_g and calculate the relative humidity ϕ:

$$p_{Ag} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Adb})$$ $\boxed{p_{Ag} = 2.738\cdot kPa}$

$$\phi = \frac{P_v}{P_g} \qquad \phi_A := \frac{p_{Av}}{p_{Ag}}$$ $\boxed{\phi_A = 39.633\cdot\%}$

5) Calculate the enthalpy h using the function above:

$$h_A := h_{ca}(T_{Adb}, \omega_A)$$ $\boxed{h_A = 39.381\cdot\frac{kJ}{kg}}$

Determining the specific volume:

$$v = R_{mix}\cdot\frac{T}{p}$$

where Rmix is the specific gas constant for the mixture, which is given by:

$$R_{mix} = y_a\cdot R_a + y_v\cdot R_v$$ $\boxed{R_a := 287\frac{J}{kg\cdot K}}$ $\boxed{R_v := 461.4\frac{J}{kg\cdot K}}$

In these equations, the subscripts "a" and "v" indicate air and water vapor, respectivly, y indicates the mass fraction, and R is the specific gas constant. The mass fractions for our binary system can be expressed in terms of the specific humidity:

$$y_a = \frac{1}{1 + \omega} \qquad y_v = \frac{\omega}{1 + \omega}$$

performing these calculations at Point A yields:

$$y_a := \frac{1}{1 + \omega_A} \qquad y_a = 0.993 \qquad y_v := \frac{\omega_A}{1 + \omega_A} \qquad y_v = 0.00664$$

$$\boxed{R_{Amix} := y_a\cdot R_a + y_v\cdot R_v = 288.158\cdot\frac{J}{kg\cdot K}}$$

$$v_A := R_{Amix}\cdot\frac{T_{Adb}}{p_{atm}}$$ $\boxed{v_A = 0.834\frac{m^3}{kg}}$

At Point B:

$T_{Bdb} := 23.0\,°C$ $\qquad$ $T_{Bwb} := 18.8\,°C$

Function Definitions:

$$h_{cb}(T_{Bdb}, \omega_B) := \left[T_{Bdb}/°C + \omega_B \cdot (2501 + 1.805 \cdot T_{Bdb}/°C)\right]\frac{kJ}{kg}$$

Solution:

1) Start by finding the saturation pressure at the wet bulb temperature:

$P_{gw} = P_{sat} @ T_{wb}$ $\qquad$ $p_{Bgw} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Bwb})$ $\qquad$ $\boxed{p_{Bgw} = 2.187 \cdot kPa}$

2) Calculate the vapor pressure using Carrier's Equation:

$$p_{Bv} := p_{Bgw} - \frac{(p_{atm} - p_{Bgw}) \cdot (T_{Bdb}/°C - T_{Bwb}/°C)}{1532.4 - 1.3 \cdot T_{Bwb}/°C} \qquad \boxed{p_{Bv} = 1.909 \cdot kPa}$$

3) calculate the specific humidity ω:

$$\omega = 0.622 \frac{P_v}{P - P_v} \qquad \omega_B := 0.622 \cdot \frac{p_{Bv}}{p_{atm} - p_{Bv}} \qquad \boxed{\omega_B = 0.0119}$$

4) Determine the saturation pressure at the dry bulb temperature p_g and calculate the relative humidity ϕ:

$p_{Bg} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Bdb})$ $\qquad$ $\boxed{p_{Bg} = 2.838 \cdot kPa}$

$$\phi = \frac{P_v}{P_g} \qquad \phi_B := \frac{p_{Bv}}{p_{Bg}} \qquad \boxed{\phi_B = 67.272 \cdot \%}$$

5) Calculate the enthalpy h using the function above:

$$h_B := h_{cb}(T_{Bdb}, \omega_B) \qquad \boxed{h_B = 53.13 \cdot \frac{kJ}{kg}}$$

Determining the specific volume:

performing these calculations at Point B yields:

$$y_a := \frac{1}{1 + \omega_B} \qquad y_a = 0.988 \qquad y_v := \frac{\omega_B}{1 + \omega_B} \qquad y_v = 0.012$$

$$\boxed{R_{Bmix} := y_a \cdot R_a + y_v \cdot R_v = 289.043 \cdot \frac{J}{kg \cdot K}}$$

$$v_B := R_{Bmix} \cdot \frac{T_{Bdb}}{p_{atm}} \qquad \boxed{v_B = 0.838 \, \frac{m^3}{kg}}$$

At Point C:

$T_{Cdb} := 16.1\,°C$ $\qquad$ $T_{Cwb} := 13.5\,°C$

Function Definitions:

$$h_{cc}(T_{Cdb}, \omega_C) := \left[T_{Cdb}/°C + \omega_C \cdot (2501 + 1.805 \cdot T_{Cdb}/°C)\right]\frac{kJ}{kg}$$

Solution:

1) Start by finding the saturation pressure at the wet bulb temperature:

$$P_{gw} = P_{sat} @ T_{wb} \qquad p_{Cgw} := \mathrm{linterp}(T_{satH2O}, p_{satH2O}, T_{Cwb}) \qquad \boxed{p_{Cgw} = 1.562 \cdot kPa}$$

2) Calculate the vapor pressure using Carrier's Equation:

$$p_{Cv} := p_{Cgw} - \frac{(p_{atm} - p_{Cgw}) \cdot (T_{Cdb}/°C - T_{Cwb}/°C)}{1532.4 - 1.3 \cdot T_{Cwb}/°C} \qquad \boxed{p_{Cv} = 1.39 \cdot kPa}$$

3) calculate the specific humidity ω:

$$\omega = 0.622 \frac{P_v}{P - P_v} \qquad \omega_C := 0.622 \cdot \frac{p_{Cv}}{p_{atm} - p_{Cv}} \qquad \boxed{\omega_C = 0.0086}$$

4) Determine the saturation pressure at the dry bulb temperature p_g and calculate the relative humidity ϕ:

$$p_{Cg} := \mathrm{linterp}(T_{satH2O}, p_{satH2O}, T_{Cdb}) \qquad \boxed{p_{Cg} = 1.845 \cdot kPa}$$

$$\phi = \frac{P_v}{P_g} \qquad \phi_C := \frac{p_{Cv}}{p_{Cg}} \qquad \boxed{\phi_C = 75.328 \cdot \%}$$

5) Calculate the enthalpy h using the function above:

$$h_C := h_{cc}(T_{Cdb}, \omega_C) \qquad \boxed{h_C = 37.818 \cdot \frac{kJ}{kg}}$$

Determining the specific volume:

performing these calculations at Point C yields:

$$y_a := \frac{1}{1 + \omega_C} \qquad y_a = 0.991 \qquad y_v := \frac{\omega_C}{1 + \omega_C} \qquad y_v = 8.511 \times 10^{-3}$$

$$\boxed{R_{Cmix} := y_a \cdot R_a + y_v \cdot R_v = 288.484 \cdot \frac{J}{kg \cdot K}}$$

$$v_C := R_{Cmix} \cdot \frac{T_{Cdb}}{p_{atm}} \qquad \boxed{v_C = 0.817 \frac{m^3}{kg}}$$

At Point D:

$T_{Ddb} := 21.9\,°C$ $\qquad$ $T_{Dwb} := 15.8\,°C$

Function Definitions:

$$h_{cd}(T_{Ddb}, \omega_D) := \left[T_{Ddb}/°C + \omega_D \cdot (2501 + 1.805 \cdot T_{Ddb}/°C)\right] \frac{kJ}{kg}$$

Solution:

1) Start by finding the saturation pressure at the wet bulb temperature:

$$P_{gw} = P_{sat} @ T_{wb} \qquad p_{Dgw} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Dwb}) \qquad \boxed{p_{Dgw} = 1.807 \cdot kPa}$$

2) Calculate the vapor pressure using Carrier's Equation:

$$p_{Dv} := p_{Dgw} - \frac{(p_{atm} - p_{Dgw}) \cdot (T_{Ddb}/°C - T_{Dwb}/°C)}{1532.4 - 1.3 \cdot T_{Dwb}/°C} \qquad \boxed{p_{Dv} = 1.402 \cdot kPa}$$

3) calculate the specific humidity ω:

$$\omega = 0.622 \frac{P_v}{P - P_v} \qquad \omega_D := 0.622 \cdot \frac{p_{Dv}}{p_{atm} - p_{Dv}} \qquad \boxed{\omega_D = 0.0087}$$

4) Determine the saturation pressure at the dry bulb temperature p_g and calculate the relative humidity ϕ:

$$p_{Dg} := \text{linterp}(T_{satH2O}, p_{satH2O}, T_{Ddb}) \qquad \boxed{p_{Dg} = 2.655 \cdot kPa}$$

$$\phi = \frac{P_v}{P_g} \qquad \phi_D := \frac{p_{Dv}}{p_{Dg}} \qquad \boxed{\phi_D = 52.825 \cdot \%}$$

5) Calculate the enthalpy h using the function above:

$$h_D := h_{cd}(T_{Ddb}, \omega_D) \qquad \boxed{h_D = 43.907 \cdot \frac{kJ}{kg}}$$

Determining the specific volume:

performing these calculations at Point D yields:

$$y_a := \frac{1}{1 + \omega_D} \qquad y_a = 0.991 \qquad y_v := \frac{\omega_D}{1 + \omega_D} \qquad y_v = 8.588 \times 10^{-3}$$

$$\boxed{R_{Dmix} := y_a \cdot R_a + y_v \cdot R_v = 288.498 \cdot \frac{J}{kg \cdot K}}$$

$$v_D := R_{Dmix} \cdot \frac{T_{Ddb}}{p_{atm}} \qquad \boxed{v_D = 0.834 \frac{m^3}{kg}}$$

Mass Flow Rate of Dry Air:

The mass flow rate of dry air (m'a) is calculated using the equation for the calibrated orfice given in the lab instructions for the psychrometrics lab.

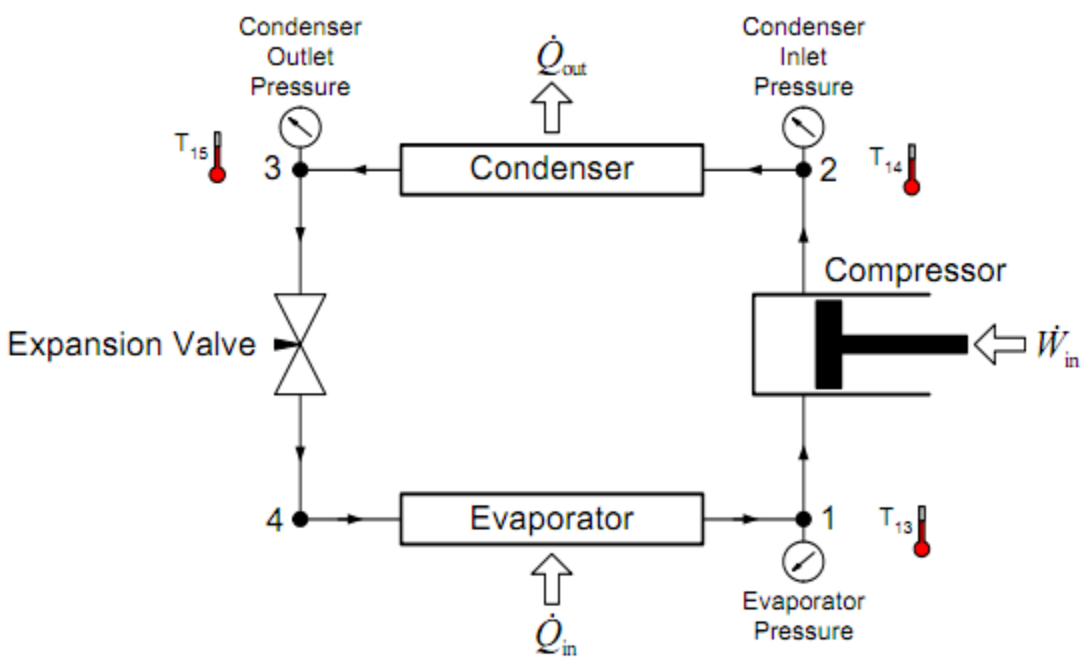
$z := 7.5$ $\qquad v_{Dtmp} := \dfrac{v_D}{\dfrac{m^3}{kg}}$ $\qquad v_{Dtmp} = 0.834$ $\qquad$ get rid of units on specific volume

$m'_a := 0.0517\sqrt{\dfrac{z}{v_{Dtmp}}}\cdot\dfrac{kg}{s}$ $\qquad \boxed{m'_a = 0.155\,\dfrac{kg}{s}}$ $\qquad$ mass flow rate of dry air

So the total mass flow rate at a point is given by: $\quad m'_{tot} = m'_a\cdot(1 + \omega)$

Determination of Refrigerant States:

The states in the closed loop refrigeration cycle will be defined as shown in the schematic below.



Refrigerant States: R-134a

Note: from the R-134a tables, this is superheated vapor as expected. We must use double linear interpolation to determine h and s.

Point 1: Evaporator Outlet/Compressor Inlet

Measurements : $p_1 := 1.9\text{bar} + p_{atm}$ $\quad \boxed{p_1 = 2.921 \cdot \text{bar}}$

$T_{13} := 9.8\,°\text{C}$ $\quad \boxed{T_1 := T_{13} = 9.8 \cdot °\text{C}}$

KLEA134a : Properties of Superheated Vapour (SI Units)

Temp C	2.9 Satn. Temp. = 272.89 K -0.26 C			3.0 Satn. Temp. = 273.83 K 0.68 C		
	D	H	S	D	H	S
Satn.	14.301	298.4	1.727	14.776	298.9	1.726
-4						
-2						
0	14.282	298.6	1.728			
2	14.135	300.4	1.734	14.674	300.1	1.731
4	13.993	302.2	1.741	14.525	301.9	1.737
6	13.855	304.0	1.747	14.380	303.7	1.744
8	13.720	305.8	1.754	14.238	305.5	1.750
10	13.590	307.6	1.760	14.101	307.3	1.757
12	13.462	309.4	1.766	13.968	309.1	1.763
14	13.338	311.2	1.773	13.838	310.9	1.769

$$h_{11} := \text{linterp}\left[(2.9 \;\; 3.0)^T \text{bar}, (305.8 \;\; 305.5)^T \frac{\text{kJ}}{\text{kg}}, p_1\right]$$

$$s_{11} := \text{linterp}\left[(2.9 \;\; 3.0)^T \text{bar}, (1.754 \;\; 1.750)^T \frac{\text{kJ}}{\text{kg} \cdot \text{K}}, p_1\right]$$

$$h_{12} := \text{linterp}\left[(2.9 \;\; 3.0)^T \text{bar}, (307.6 \;\; 307.3)^T \frac{\text{kJ}}{\text{kg}}, p_1\right]$$

$$s_{12} := \text{linterp}\left[(2.9 \;\; 3.0)^T \text{bar}, (1.760 \;\; 1.757)^T \frac{\text{kJ}}{\text{kg} \cdot \text{K}}, p_1\right]$$

$$h_1 := \text{linterp}\left[(8 \;\; 10)^T °\text{C}, \left(h_{11} \;\; h_{12}\right)^T, T_1\right]$$

$$s_1 := \text{linterp}\left[(8 \;\; 10)^T °\text{C}, \left(s_{11} \;\; s_{12}\right)^T, T_1\right]$$

$\boxed{T_1 = 9.8 \cdot °\text{C}}$ $\quad \boxed{h_1 = 307.357 \cdot \frac{\text{kJ}}{\text{kg}}}$ $\quad \boxed{s_1 = 1.759 \cdot \frac{\text{kJ}}{\text{kg} \cdot \text{K}}}$

Point 2: Compressor Outlet/Condenser Inlet

Measurements : $p_2 := 11.3\text{bar} + p_{atm}$ $\boxed{p_2 = 12.321\cdot\text{bar}}$

$T_{14} := 76.1\,°\text{C}$ $\boxed{T_2 := T_{14} = 76.1\cdot°\text{C}}$

KLEA134a : Properties of Superheated Vapour (SI Units)

Temp C	12.5 Satn. Temp. = 321.06 K 47.91 C			13.0 Satn. Temp. = 322.61 K 49.46 C		
	D	H	S	D	H	S
Satn.	62.505	322.9	1.709	65.253	323.5	1.708
48	62.461	323.0	1.709			
50	61.472	325.4	1.717	64.961	324.1	1.710
55	59.232	331.3	1.735	62.464	330.1	1.729
60	57.262	337.0	1.752	60.288	336.0	1.746
65	55.504	342.6	1.769	58.361	341.7	1.763
70	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
75	52.470	353.7	1.801	55.061	352.8	1.796
80	51.141	359.1	1.816	53.624	358.3	1.812
85	49.911	364.5	1.831	52.300	363.8	1.827

Note: since the temps for P=12.0 and P=12.5 are different, so we have to interpolate by using 12.5 and 13.0

$$h_{21} := \text{linterp}\left[(12.5\ \ 13)^T\text{bar}, (353.7\ \ 352.8)^T\frac{\text{kJ}}{\text{kg}}, p_2\right] = 354.022\cdot\frac{\text{kJ}}{\text{kg}}$$

$$s_{21} := \text{linterp}\left[(12.5\ \ 13)^T\text{bar}, (1.801\ \ 1.796)^T\frac{\text{kJ}}{\text{kg}\cdot\text{K}}, p_2\right]$$

$$h_{22} := \text{linterp}\left[(12.5\ \ 13)^T\text{bar}, (359.1\ \ 358.3)^T\frac{\text{kJ}}{\text{kg}}, p_2\right]$$

$$s_{22} := \text{linterp}\left[(12.5\ \ 13)^T\text{bar}, (1.816\ \ 1.812)^T\frac{\text{kJ}}{\text{kg}\cdot\text{K}}, p_2\right]$$

$$h_2 := \text{linterp}\left[(75\ \ 80)^T °\text{C}, \left(h_{21}\ \ h_{22}\right)^T, T_2\right]$$

$$s_2 := \text{linterp}\left[(75\ \ 80)^T °\text{C}, \left(s_{21}\ \ s_{22}\right)^T, T_2\right]$$

$\boxed{T_2 = 76.1\cdot°\text{C}}$ $\boxed{h_2 = 355.202\cdot\frac{\text{kJ}}{\text{kg}}}$ $\boxed{s_2 = 1.806\cdot\frac{\text{kJ}}{\text{kg}\cdot\text{K}}}$

Point 3: Condenser Outlet/Expansion Valve Inlet

Measurements : $p_3 := 11\text{bar} + p_{atm}$ $p_3 = 12.021 \cdot \text{bar}$

$T_{15} := 40.9\,°C$ $T_3 := T_{15} = 40.9 \cdot °C$

From the R134a Tables, in the satruated tables, the saturation pressure at the measured temperature of 41.0 deg C is 10.4401 bar which is below the measured pressure. Therefore, the refrigerant is subcooled as expected and we use the subcooled liquid approximation to determine h and s at this point.

KLEA134a SATURATION PROPERTIES (SI Units)

Temp C	Absolute Pressure bar	Density		Enthalpy			Entropy	
		kg/m³	kg/m³	kJ/kg			kJ/(kg K)	
		Liquid	Vapour	Liquid	Latent	Vapour	Liquid	Vapour
30	7.70132	1187.5	37.517	141.736	173.156	314.892	1.14354	1.71473
31	7.92501	1183.5	38.634	143.190	172.193	315.383	1.14826	1.71441
32	8.15355	1179.6	39.779	144.649	171.220	315.869	1.15299	1.71409
33	8.38701	1175.6	40.953	146.112	170.239	316.351	1.15771	1.71377
34	8.62545	1171.5	42.157	147.580	169.247	316.827	1.16243	1.71346
35	8.86896	1167.5	43.391	149.053	168.246	317.299	1.16715	1.71314
36	9.11759	1163.4	44.658	150.530	167.235	317.765	1.17187	1.71282
37	9.37142	1159.2	45.956	152.013	166.213	318.226	1.17659	1.71250
38	9.63052	1155.1	47.288	153.500	165.181	318.681	1.18130	1.71217
39	9.89496	1150.9	48.654	154.993	164.138	319.131	1.18602	1.71185
40	10.1648	1146.7	50.055	156.491	163.084	319.575	1.19073	1.71152
41	10.4401	1142.4	51.492	157.994	162.019	320.013	1.19545	1.71119
42	10.7210	1138.1	52.967	159.503	160.942	320.445	1.20017	1.71085
43	11.0076	1133.7	54.479	161.017	159.853	320.870	1.20488	1.71051
44	11.2998	1129.4	56.031	162.537	158.752	321.289	1.20960	1.71016

Approximate to the Saturated liquid

$$h_3 := \text{linterp}\left[(40 \;\; 41)^T\,°C, (156.491 \;\; 157.994)^T \frac{kJ}{kg}, T_3\right]$$

$$s_3 := \text{linterp}\left[(40 \;\; 41)^T\,°C, (1.19073 \;\; 1.19545)^T \cdot \frac{kJ}{kg \cdot K}, T_3\right]$$

$\boxed{T_3 = 40.9 \cdot °C}$ $\boxed{h_3 = 157.844 \cdot \frac{kJ}{kg}}$ $\boxed{s_3 = 1.195 \cdot \frac{kJ}{kg \cdot K}}$

Point 4: Expansion Valve Outlet/Evaporator Inlet

Measurements : We don't have any, so we make assumptions

we know that we are under the vapor dome. Assuming that there is no pressure drop across the evaporator, then

$p_4 := p_1$ $\quad\quad$ $\boxed{p_4 = 2.921 \cdot bar}$

the adiabatic throttling process from 3 to 4 is a constant enthalpy process, therfore:

$h_4 := h_3$ $\quad\quad$ $\boxed{h_4 = 157.844 \cdot \frac{kJ}{kg}}$

since we are underneath the vapor dome, we are at the same saturation temperature for the given pressure. From the saturation tables:

-1	2.82307
0	2.92769

$$T_4 := (-1)\,°C + 1 \cdot \Delta°C \cdot \frac{(2.921 - 2.82307)}{(2.92769 - 2.82307)} = -0.064 \cdot °C \quad\quad \boxed{T_4 = -0.064 \cdot °C}$$

determination of quality in order to determine entropy:

KLEA134a SATURATION PROPERTIES (SI Units)

Temp C	Absolute Pressure bar	Density kg/m³ Liquid	Density kg/m³ Vapour	Enthalpy kJ/kg Liquid	Enthalpy kJ/kg Latent	Enthalpy kJ/kg Vapour	Entropy kJ/(kg K) Liquid	Entropy kJ/(kg K) Vapour
-4	2.52643	1308.2	12.526	94.675	201.506	296.181	0.98047	1.72915
-3	2.62250	1305.0	12.983	96.002	200.771	296.772	0.98537	1.72855
-2	2.72136	1301.7	13.453	97.331	200.031	297.362	0.99025	1.72796
-1	2.82307	1298.4	13.936	98.664	199.286	297.950	0.99513	1.72739
0	2.92769	1295.1	14.433	100.000	198.536	298.536	1.00000	1.72684
1	3.03526	1291.8	14.944	101.339	197.780	299.120	1.00486	1.72629

$$h_{f4} := 98.664 \frac{kJ}{kg} + \left(100.000 \frac{kJ}{kg} - 98.664 \frac{kJ}{kg}\right) \cdot \frac{(2.921 - 2.82307)}{(2.92769 - 2.82307)} = 99.915 \cdot \frac{kJ}{kg}$$

$$h_{g4} := 297.950 \frac{kJ}{kg} + \left(298.536 \frac{kJ}{kg} - 297.950 \frac{kJ}{kg}\right) \cdot \frac{(2.921 - 2.82307)}{(2.92769 - 2.82307)} = 298.499 \cdot \frac{kJ}{kg}$$

$$s_{f4} := 0.99513 \frac{kJ}{kg \cdot K} + \left(1.00000 \frac{kJ}{kg \cdot K} - 0.99513 \frac{kJ}{kg \cdot K}\right) \cdot \frac{(2.921 - 2.82307)}{(2.92769 - 2.82307)} = 0.99969 \cdot \frac{kJ}{kg \cdot K}$$

$$s_{g4} := 1.72739 \frac{kJ}{kg \cdot K} + \left(1.72684 \frac{kJ}{kg \cdot K} - 1.72739 \frac{kJ}{kg \cdot K}\right) \cdot \frac{(2.921 - 2.82307)}{(2.92769 - 2.82307)} = 1.72688 \cdot \frac{kJ}{kg \cdot K}$$

$$x_4 := \frac{(h_4 - h_{f4})}{(h_{g4} - h_{f4})} \quad\quad \boxed{x_4 = 0.292}$$

$$s_4 := s_{f4} + x_4 \cdot (s_{g4} - s_{f4}) \quad\quad \boxed{s_4 = 1.212 \cdot \frac{kJ}{kg \cdot K}}$$

T-s Diagram for Refrigerant:

1. Compressor Inlet stays the same

2. Compressor outlet/Condenser Inlet - stays the same

2.a Condensor condenses to saturated vapor at condensor pressure

$p_{2a} := p_2$ $\qquad p_{2a} = 12.321 \cdot bar$

KLEA134a SATURATION PROPERTIES (SI Units)

Temp C	Absolute Pressure bar	Density kg/m³ Liquid	Density kg/m³ Vapour	Enthalpy kJ/kg Liquid	Enthalpy kJ/kg Latent	Enthalpy kJ/kg Vapour	Entropy kJ/(kg K) Liquid	Entropy kJ/(kg K) Vapour
45	11.5978	1124.9	57.623	164.062	157.639	321.701	1.21432	1.70981
46	11.9017	1120.5	59.256	165.593	156.513	322.106	1.21904	1.70945
47	12.2115	1116.0	60.933	167.130	155.374	322.504	1.22376	1.70908
48	12.5273	1111.4	62.654	168.673	154.221	322.894	1.22848	1.70870
49	12.8492	1106.8	64.421	170.222	153.055	323.277	1.23321	1.70832

$T_{2a} := 47\,°C + (48\,°C - 47\,°C) \cdot \left[\frac{(12.321 - 12.2115)}{(12.5273 - 12.2115)}\right]$ $\quad \boxed{T_{2a} = 47.347 \cdot °C}$ saturation temp

$s_{2a} := 1.70908 \frac{kJ}{kg \cdot K} + \left(1.70870 \frac{kJ}{kg \cdot K} - 1.70908 \frac{kJ}{kg \cdot K}\right) \cdot \frac{(12.321 - 12.2115}{(12.5273 - 12.211}$ $\boxed{s_{2a} = 1.70895 \cdot \frac{kJ}{kg \cdot K}}$ saturated vapor

2.b Condenser sondenses from saturated vapor to saturated liquid at constant temperature and pressure

$p_{2b} := p_{2a}$ $\qquad T_{2b} := T_{2a}$ $\qquad \boxed{T_{2b} = 47.347 \cdot °C}$

$s_{2b} := 1.22376 \frac{kJ}{kg \cdot K} + \left(1.22848 \frac{kJ}{kg \cdot K} - 1.22376 \frac{kJ}{kg \cdot K}\right) \cdot \frac{(12.321 - 12.2115}{(12.5273 - 12.211}$ $\boxed{s_{2b} = 1.225 \cdot \frac{kJ}{kg \cdot K}}$ saturated liquid

3. Condenser Outlet/Expansion Valve Inlet - stays the same

4. Expansion valve outlet/Evaporator inlet - stays the same

4.a Refrigerant is heated at constant pressure and temperature to a saturated vapor in the evaporator

$p_{4a} := p_4$ $\quad \boxed{p_{4a} = 2.921 \cdot bar}$ $\qquad T_{4a} := T_4$ $\quad \boxed{T_{4a} = -0.06395 \cdot °C}$

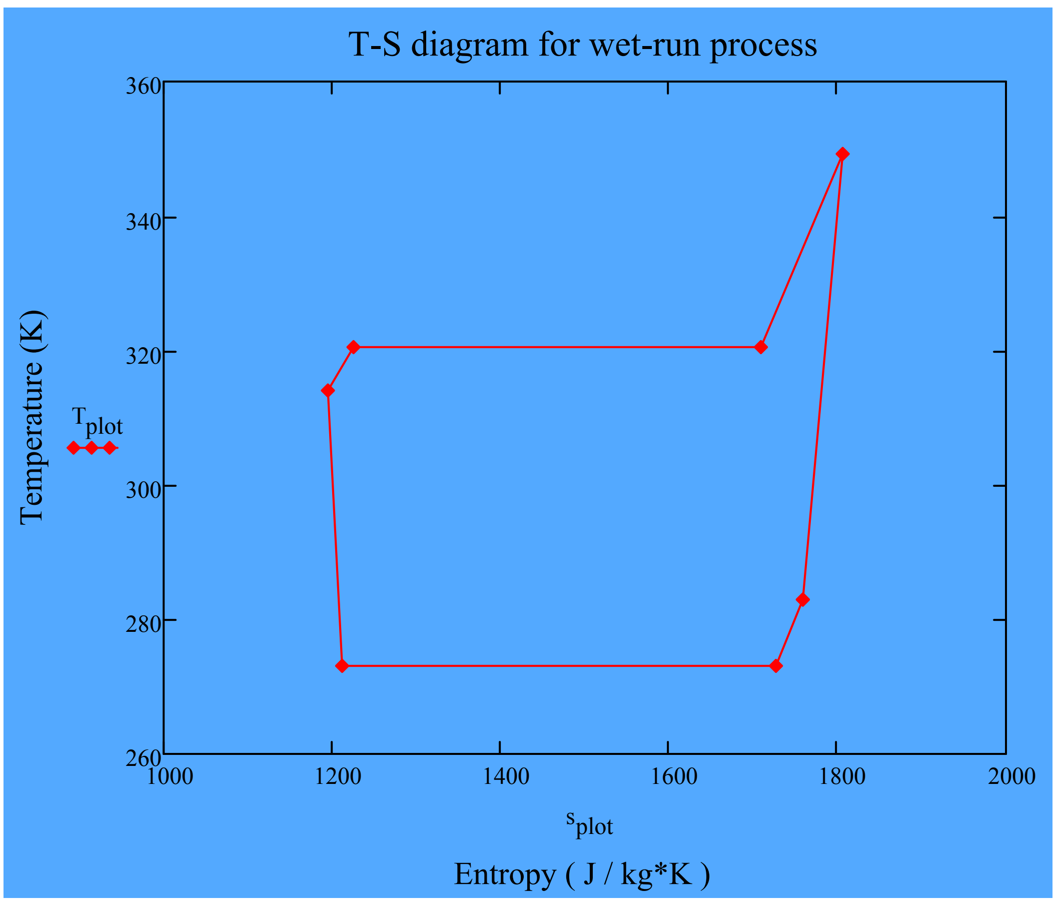
KLEA134a SATURATION PROPERTIES (SI Units)

Temp C	Absolute Pressure bar	Density kg/m³ Liquid	Density kg/m³ Vapour	Enthalpy kJ/kg Liquid	Enthalpy kJ/kg Latent	Enthalpy kJ/kg Vapour	Entropy kJ/(kg K) Liquid	Entropy kJ/(kg K) Vapour
-4	2.52643	1308.2	12.526	94.675	201.506	296.181	0.98047	1.72915
-3	2.62250	1305.0	12.983	96.002	200.771	296.772	0.98537	1.72855
2	2.72136	1301.7	13.453	97.331	200.031	297.362	0.99025	1.72796
-1	2.82307	1298.4	13.936	98.664	199.286	297.950	0.99513	1.72739
0	2.92769	1295.1	14.433	100.000	198.536	298.536	1.00000	1.72684
1	3.03526	1291.8	14.944	101.339	197.780	299.120	1.00486	1.72629

$$s_{4a} := 1.72739 \frac{kJ}{kg \cdot K} + (1.72684 - 1.72739) \cdot \frac{(1-0)}{(1-0.06395)} \frac{kJ}{kg \cdot K} \qquad \boxed{s_{4a} = 1.726802 \cdot \frac{kJ}{kg \cdot K}}$$ saturated vapor

T-s Diagram

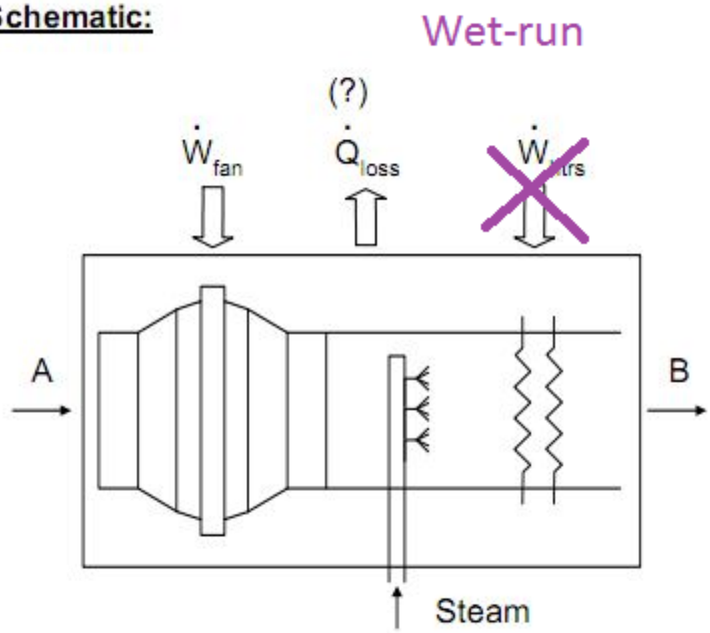
$$T_{plot} := \left(T_1 \;\; T_2 \;\; T_{2a} \;\; T_{2b} \;\; T_3 \;\; T_4 \;\; T_{4a} \;\; T_1\right)^T \qquad s_{plot} := \left(s_1 \;\; s_2 \;\; s_{2a} \;\; s_{2b} \;\; s_3 \;\; s_4 \;\; s_{4a} \;\; s_1\right)^T$$



Control-Volume Analysis:

Mass and energy balance A to B:

Schematic:



conservation of mass at steady state: mass flow rate in = mass flow rate out

$m'_A + m'_{stm} = m'_B$ $\qquad\qquad$ $m'_{stm} = m'_B - m'_A = 0$

$$m'_A := m'_a \cdot (1 + \omega_A) = 0.1561 \frac{kg}{s}$$

$$m'_B := m'_a \cdot (1 + \omega_B) = 0.1569 \frac{kg}{s}$$

$$m'_{stm} := m'_a \cdot (\omega_B - \omega_A) = 0.00080154 \frac{kg}{s}$$

conservation of energy at steady state: rate of energy in = rate of energy out

$$m'_a \cdot h_A + m'_{stm} \cdot h_{stm} + W'_{fan} + W'_{htrs} = m'_a \cdot h_B + Q'_{loss}$$

the boiler produces saturated steam at atmospheric pressure. Using the measured atmospheric pressure and the saturation tables for water gives:

$p_{atm} = 102.1 \cdot kPa$

P (kPa)	hg (kJ/kg)
101.325	2675.6
125	2684.9

From Table A-5 Thermodynaimcs

$$h_{stm} := 2675.6\frac{kJ}{kg} + \left(2684.9\frac{kJ}{kg} - 2675.6\frac{kJ}{kg}\right)\cdot\frac{(102.1 - 101.325)}{(125 - 101.325)}$$ $$h_{stm} = 2675.9044\cdot\frac{kJ}{kg}$$

the fan power can be obtained by using the fan power curve (Figure 3 in psychrometrics lab) and the measured fan supply voltage:

$v_F := 165V$ $\quad$ $W'_{fan} := 162W$ $\quad$ ***Figure 3 in psychrometrics lab shown below

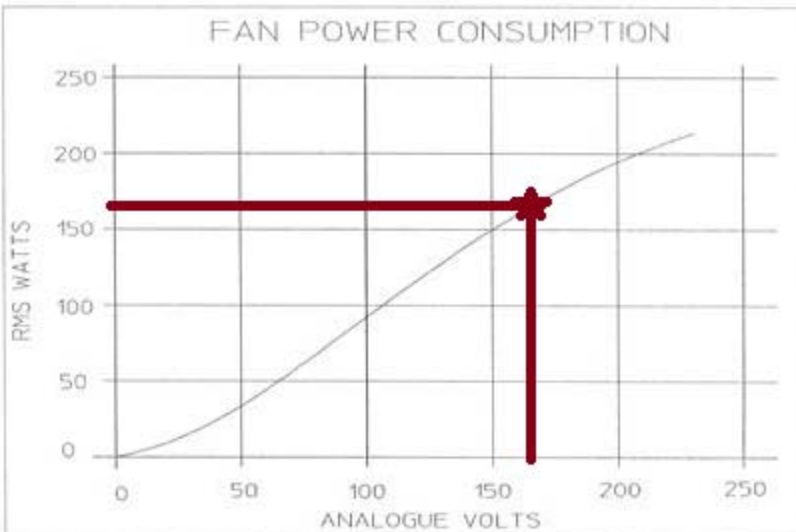


Figure 3: Fan power versus fan supply voltage [1]

No pre-heater is energized.

$V_L := 0V$ $\quad$ $R_p := 47.2\Omega$

$$W'_{htrs} := \frac{V_L^{\;2}}{R_p}$$ $$W'_{htrs} = 0$$

We can now estimate the heat loss between A and B using the energy balance:

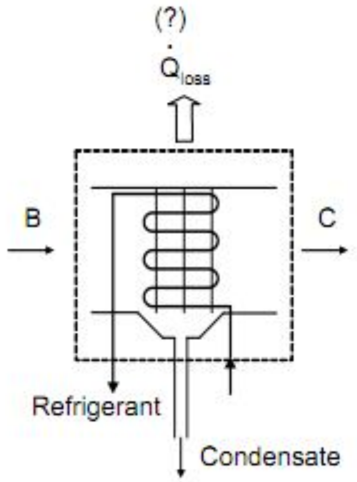
$$Q'_{lossAB} := m'_a\cdot h_A + m'_{stm}\cdot h_{stm} + W'_{fan} + W'_{htrs} - m'_a\cdot h_B$$

$$Q'_{lossAB} = 174.756\ W$$

the calculated value for "heat loss" results from several different effects including heat loss through the duct walls, inaccurate measurements, inaccuracies in calculated values for the enthalpy of air, ect.

Mass and energy balance B to C:

Schematic:



Between B and C - we are removing energy from the air and adding energy to the refrigerant in the evaporator. The data also indicated that condensate is formed and drains out of the control volume.

Conservation of Mass at Steady State: mass flow rate in = mass flow rate out

Note: the refrigerant doesn't interact with the condensate

$$m'_B = m'_C + m'_{cond} \qquad m'_{cond} = m'_B - m'_C$$

$$m'_B = 0.157\,\frac{kg}{s}$$

$$m'_C := m'_a \cdot \left(1 + \omega_C\right) = 0.156\,\frac{kg}{s}$$

$$m'_{cond} := m'_a \cdot \left(\omega_B - \omega_C\right) = 0.0005065\,\frac{kg}{s}$$ **Estimated condensate flow rate**

Observed condensate flow rate:

Volume flow rate:

$$V'_{condObs} := \frac{137mL}{10min} = 2.283 \times 10^{-7}\,\frac{m^3}{s}$$

Approximate density: $\rho_{cond} := 1000\,\frac{kg}{m^3}$

Mass flow rate: $m'_{condObs} := V'_{condObs} \cdot \rho_{cond} = 0.0002283\,\frac{kg}{s}$

Note the difference between the estimated condensate flow rate and the observed rate are caused by many different effects including measurement errors, water retention on the evaporation fins, re-entrainment of water into the air stream, ect.

Conservation of energy at steady state: rate of energy in = rate of energy out

for our energy balance from B to C, we are concerned with the refrigerant entering (state 4) and exiting (state 1), The complete energy balance is:

$$m'_a \cdot h_B + m'_{ref} \cdot h_4 = m'_a \cdot h_C + m'_{cond} \cdot h_{cond} + m'_{ref} \cdot h_1 + Q'_{loss}$$

the condensate term in the energy balance is often neglected, but we will include it in our calculations. We will assume that the condensate is a saturated liquid at the dry-bulb temperature given at C on the air-side:

$T_{cond} := T_{Cdb}$ saturated liquid at: $T_{cond} = 16.1 \cdot °C$

From the steam tables:

T (°C)	hf (kJ/kg)
15	62.982
20	83.915

From Table A-4 Thermodynamics

$$h_{cond} := 62.982 \frac{kJ}{kg} + \left(83.915 \frac{kJ}{kg} - 62.982 \frac{kJ}{kg}\right) \cdot \frac{(16.1 - 15)}{(20 - 15)} \qquad h_{cond} = 67.587 \cdot \frac{kJ}{kg}$$

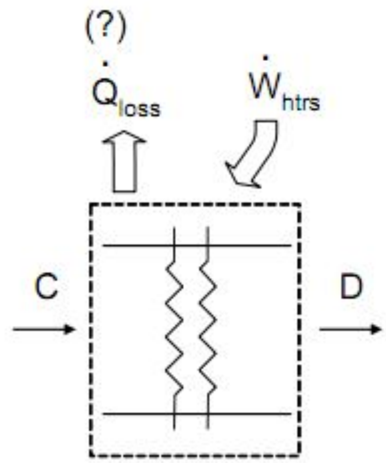
Measured Refrigerant Flow Rate: $m'_{ref} := 15.3 \frac{gm}{s}$

$$Q'_{lossBC} := m'_a \cdot h_B + m'_{ref} \cdot h_4 - m'_a \cdot h_C - m'_{cond} \cdot h_{cond} - m'_{ref} \cdot h_1$$

$Q'_{lossBC} = 52.617\,W$

Mass and Energy Balance C to D:

Schematic:



Between C and D - we are adding energy to the air via the first re-heater.

$$m'_C = m'_D$$

$$m'_C = m'_a \cdot (1 + \omega_C) \qquad m'_C = 0.1564 \frac{kg}{s}$$

$$m'_D := m'_a \cdot (1 + \omega_D) = 0.1564 \frac{kg}{s}$$

Therefore conservation of mass tells us that the specific humidity should not change between C and D, which makes sense because we are not adding water to or removing water from the air. Any difference in the specific humidities would be probably caused by measurement error.

$$\omega_C = 0.008584 \qquad \omega_D = 0.0086625$$

Conservation of energy at Steady-State: rate of energy in = rate of energy out

$$m_a \cdot h_C + W_{rhtrs} = m_a \cdot h_D + Q_{loss}$$

The first re-heater is energized. The heater power can be determined using the measured line voltage and the known resistance of the 1st re-heater.

$V_L := 205V$ and $R_r := 46.8\Omega$

1st Re-heater, 1kW	R_r	46.8 Ω

$$W'_{rhtrs} := \frac{V_L^2}{R_r} \qquad W'_{rhtrs} = 897.97\,W$$

$$Q'_{lossCD} := m'_a \cdot h_C + W'_{rhtrs} - m'_a \cdot h_D$$

$$Q'_{lossCD} = -46.276\,W$$

Refrigeration Effect/Capacity:

Q'_L : rate of heat transfer to the refrigerant in the evaporator

$$Q'_L := m'_{ref} \cdot (h_1 - h_4) \qquad \boxed{Q'_L = 2287.553\ W}$$

$$Ton := 12000 \frac{BTU}{hr} \qquad \boxed{Q'_L = 0.6505 \cdot Ton}$$

COP (Coefficient of Performance):

$$COP = \frac{\text{Desired Output}}{\text{Required Input}} = \frac{\text{Refrigeration Effect}}{\text{Net Power Input}} = \frac{Q'_L}{W'_{netin}}$$

Note that the net power input is generally based on the total electric power required to run the refrigeration cycle, which for our case would include the compressor power and the power for the condenser fan. At this time, we have no way to determine the condenser fan power, so we will neglect it. We could try to use the power imparted to the fluid in the compressor as the net power input. In that case,

$$W'_{netinFluid} := m'_{ref} \cdot (h_2 - h_1) \qquad \boxed{W'_{netinFluid} = 732.033\ W}$$

$$COP_{netinFluid} := \frac{Q'_L}{W'_{netinFluid}} \qquad \boxed{COP_{netinFluid} = 3.125}$$

note that the value obtained for Cop is very hight when using the fluid power. In order to use the electric power to the compressor, we need the line voltage (we have it), the compressor current (Ic, which we don't have), and the power factor (pf, which is the cosine of the phase angle) for the compressor motor under our operating conditions (which we don't have). The resulting real electric power that is consumed by the compressor is given by:

$$W'_{comp} = V_L \cdot I_C \cdot pf$$

The manufacturer states that "typical" values during tests are:

$\boxed{I_C := 7A}$ and $\boxed{0.5 \le pf \le 0.8}$

The resulting maximum and mininum compressor powers and their associated COP's are:

$$pf_{max} := 0.8 \qquad W'_{compMax} := V_L \cdot I_C \cdot pf_{max} \qquad \boxed{W'_{compMax} = 1.148 \cdot kW}$$

$$COP_{low} := \frac{Q'_L}{W'_{compMax}} \qquad \boxed{COP_{low} = 1.993}$$

$$pf_{min} := 0.5 \qquad W'_{compMin} := V_L \cdot I_C \cdot pf_{min} \qquad \boxed{W'_{compMin} = 0.718 \cdot kW}$$

$$COP_{high} := \frac{Q'_L}{W'_{compMin}} \qquad \boxed{COP_{high} = 3.188}$$

EER:

$$EER = \frac{\text{Refrigeration Effect in Btu/hr}}{\text{Net Power Input in Watts}} = 3.413 \cdot COP$$

$$\boxed{EER_{high} := 3.413 \cdot COP_{high} \cdot \frac{BTU}{W \cdot hr} = 10.881 \cdot \frac{BTU}{W \cdot hr}}$$

$$\boxed{EER_{low} := 3.413 \cdot COP_{low} \cdot \frac{BTU}{W \cdot hr} = 6.801 \cdot \frac{BTU}{W \cdot hr}}$$

Overall Compressor Efficiency:

The overall compressor efficiency is just the ratio of the power imparted to the fluid by the compressor to the real electric power consumed by the compressor. Since we made some assumptions to get the electric power to the compressor, we could ge different values for the compressor efficiency.

$$\eta_{compMax} := \frac{W'_{netinFluid}}{W'_{compMin}} \qquad \boxed{\eta_{compMax} = 102.026 \cdot \%}$$

$$\eta_{compMin} := \frac{W'_{netinFluid}}{W'_{compMax}} \qquad \boxed{\eta_{compMin} = 63.766 \cdot \%}$$

Note: We can tell the efficiency is bigger than 100%, which is not correct. It happened maybe because of measurement errors(as Dr. Hoxie said, the evaporator outlet temperature T13 in the wet run is much lower than the normal value, which will reduce the h1 value and increase the Wnetinfluid value and finally increase the efficiency value to over 100% for this calculation) , water retention on the evaporation fins, re-entrainment of water into the air stream as listed above for other values' offset. Besides, there is one more important factor we cannot neglect is that the power factor (pf), which provided by the manufacturer. The pf ranges from 0.5 to 0.8, may not appliable to this wet-run process. In order to fit our answer into the reasonable one, it can be seen that we need to increase the lower limit for the pf.

Thus we assume:

$$pf_{min} := 0.55 \qquad W'_{compMin} := V_L \cdot I_C \cdot pf_{min} \qquad \boxed{W'_{compMin} = 0.789 \cdot kW}$$

$$\eta_{compMax} := \frac{W'_{netinFluid}}{W'_{compMin}} \qquad \boxed{\eta_{compMax} = 92.751 \cdot \%}$$