1 Model Theory and Assumptions

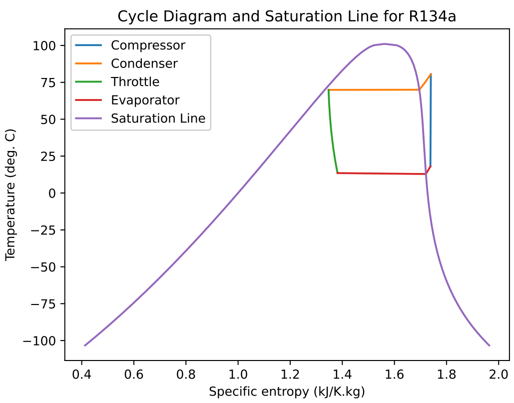
The `python` model included is able to calculate values of P, T, h, and s at all 4 points of interest of the refrigerant heat pump cycle. The specific enthalpies at these points can be used to calculate the refrigerant cycle $COP = \frac{h_3-h_2}{h_2-h_1}$. The main control inputs are the temperatures at throttle exit, T_4, and compressor exit, T_2. It can also account for variations of the compressor isentropic efficiency, pressure drop in the condenser and evaporator, enthalpy difference to saturation at condenser and evaporator exit, and different refrigerant fluids. The default refrigerant fluid used in the following report is R134a unless otherwise specified.

To calculate these salient values around the cycle, the `CoolProp` module is used. The following assumptions must be made to allow calculations and cycle plotting. First, the pressure drop in both heat exchangers is directly proportional to the enthalpy change of the refrigerant. This is a reasonable assumption based on pipe flow pressure losses and heat transfer. Second and related to this is that we must assume that the pressure at point 2 is the same as the saturation pressure for this temperature. This must be done because CoolProp cannot accept T and H as an input pair as this combination of inputs can sometimes lead to two state solutions [2], one high pressure and one lower pressure solution. It is then assumed that the rest of the condenser pressure drop occurs only when the fluid is in the two phase region which is not an ideal assumption but is not too significant so long as total condenser pressure drop is small which we found to be the case for the heat pump studied experimentally. Further improvements could be made with `RefProp` which allows T, H as an input pair. Finally, we assume that the exit of the throttle is always in the two phase region, which is very reasonable for R134a refrigerant.

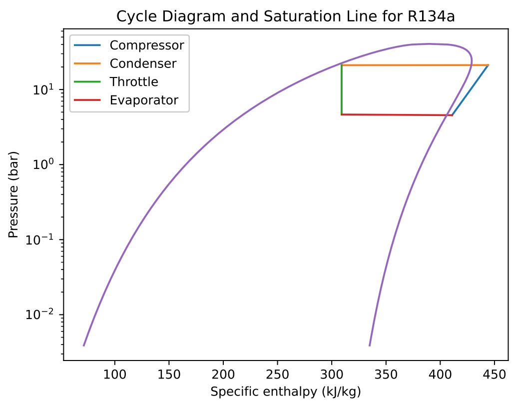
2 Example Cycle Diagrams

As can be seen from Figure 1 and Figure 2 below, the model can calculate cycle conditions even for condenser and evaporator exit conditions that are not on the saturation line. Figure 1 has a superheated gas evaporator exit and a two-phase condenser exit whereas Figure 2 has a two-phase evaporator exit and a liquid condenser exit.

The COP for these cycles is 4.07 and 3.90 respectively.

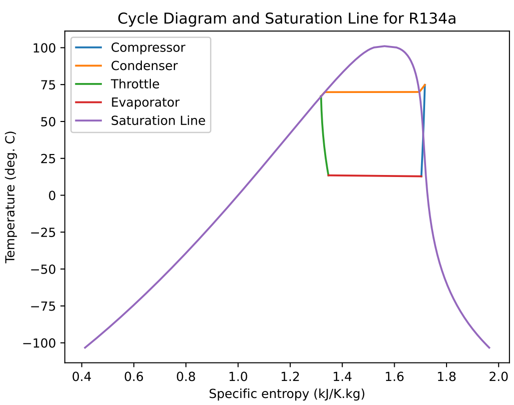


(a) T-s diagram

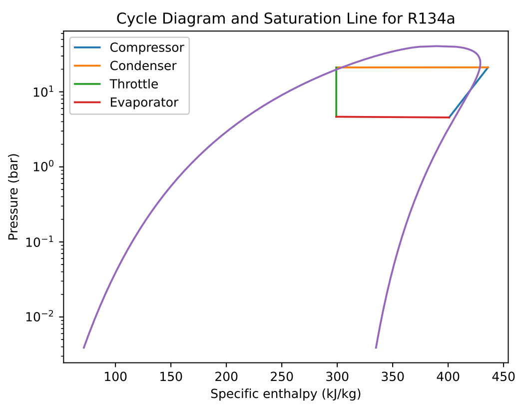


(b) P-h diagram

Figure 1: Cycle diagrams for $T_2 = 13.5^O$C, $T_4 = 70^O$C, Compressor $\eta = 0.85$, $h_1 - h_1sat$ = 5kJ/kg, $h_3 - h_3sat$ = 5kJ/kg, condenser P drop = 0.05bar, evaporator P drop = 0.1bar



(a) T-s diagram



(b) P-h diagram

Figure 2: Cycle diagrams for $T_2 = 13.5^O$C, $T_4 = 70^O$C, Compressor $\eta = 0.85$, $h_1 - h_1sat$ = -5kJ/kg, $h_3 - h_3sat$ = -5kJ/kg, condenser P drop = 0.05bar, evaporator P drop = 0.1bar

3 Input Investigations

3.1 Refrigerant fluid and Condenser and Evaporator Entry Temperatures

It can be seen from Figure 3 below that R134a is actually not a very efficient refrigerant in the range of T_2 and T_4 values tested. It was found from experiments that T_4 was approximately 5K below the ambient air temperature and T_2 was approximately 5K above the temperature of the heated water which would need to be approximately 65^oC for a domestic heating application using radiators or approx. 35^oC if using underfloor heating. Clearly we can see that a lower top temperature required by a house running underfloor heating will result in higher COPs than one using radiators. It is worth noting that R134a only just outperforms R290 (propane) even though the Global Warming Power (GWP) [[1]] of R134a is 1430 and of R290 is only 3. R32 which was the most efficient refrigerant tested over the temperature ranges has a GWP of 675, asnd therefore is a significantly better refrigerant than R134a in both GWP and COP for domestic heat pumps in the UK.

(a) COP vs T_2 for different refrigerants

(b) COP vs T_4 for different refrigerants

Figure 3: Variation of COP with T_2, T_4, and refrigerant: for Compressor $\eta = 0.85$, $h_1 - h_1sat = 5$kJ/kg, $h_3 - h_3sat = 5$kJ/kg, condenser P drop = 0.05bar, evaporator P drop = 0.1bar

3.2 Condenser and Evaporator Entry Temperatures

As would be expected for an ideal heat pump, the calculated COP for the non-ideal cycle in Figure 4 is largest when T_2 and T_4 are closer together and smallest when the difference between them is largest as the surface plot shows.

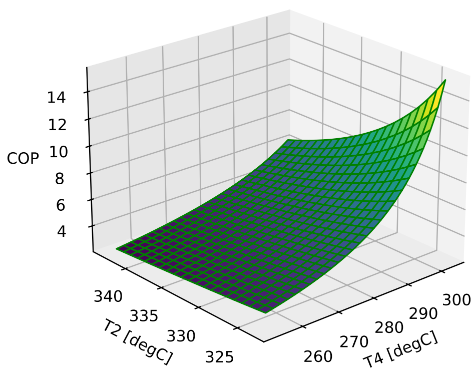


Figure 4: COP against T_2 and T_4: for Compressor $\eta = 0.85$, $h_1 - h_1sat = 5$kJ/kg, $h_3 - h_3sat = 5$kJ/kg, condenser P drop = 0.05bar, evaporator P drop = 0.1bar

References

[1] LearnMetrics. *Refrigerant GWP Chart For 61 Freons (R410A, R134A, R22, etc.) - LearnMetrics — learnmetrics.com*. `https://learnmetrics.com/refrigerant-gwp-chart/`. [Accessed 30-05-2024].

[2] *NIST Pages — pages.nist.gov*. `https://pages.nist.gov/REFPROP-docs/#multiple-and-metastable-states`. [Accessed 30-05-2024].