Practice on Air to Water Heat Pump

MASTER'S DEGREE IN THERMAL ENGINEERING

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Introduction

Nowadays, heat pumps play a critical role in enhancing energy efficiency and reducing fossil fuel dependence by providing space heating, domestic hot water, and air conditioning. Recognized as a renewable technology by the European Directive 2009/28/EC, heat pumps serve as an essential link between electric and thermal networks. Their widespread adoption aims to lower direct electrical and fossil heating needs and integrate sustainable energy in residential applications.

The experiment proposed aims to evaluate the performance of an air-to-water heat pump developed to replace conventional direct electric heating in small domestic applications. Thermodynamic parameters and efficiencies have been analyzed thanks to numerical codes to simulate the vapor compression cycle and then validated using specific measurement devices to evaluate physical properties of the experimental model.

Brief description of the experimental unit

The CTTC experimental unit was designed to analyze an air-to-water heat pump for residential heating. This system replaces a 2 kW electric heater and prioritizes precise control and measurement, allowing for the validation of a numerical model used for further optimization studies. The setup supports flexible testing through adjustable elements, employs high-precision sensors for accurate data collection and uses a data acquisition system to facilitate real-time monitoring and calibration.

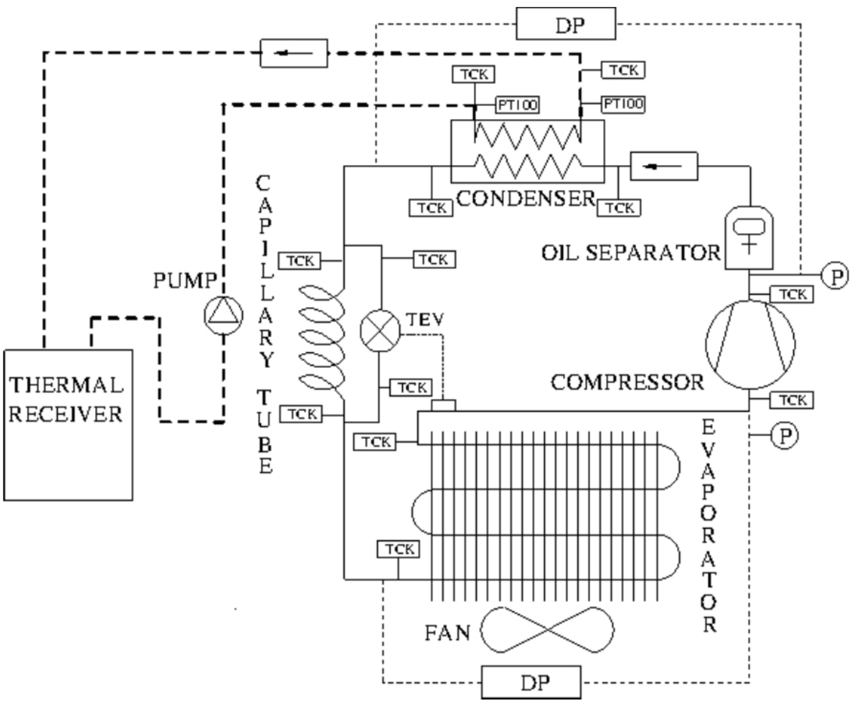

Figure 1. Schematic representation of *the CTTC's experimental unit of air-to-water heat pump.*

Specific **Sensors**, such as TCK (thermocouples), PT100, and differential pressure (DP) transducers are positioned strategically to measure temperature, pressure, and pressure drops across different components. The experimental unit is composed by:

- **Compressor**: compresses the refrigerant, increasing its pressure and temperature. It allows the refrigerant to release heat in the condenser.
- **Condenser**: here, the refrigerant releases heat to the secondary fluid (glycol-water mixture) that circulates through the condenser thanks to a **Pump** and can be further regulated in the **Thermal Receiver**. The thermal receiver would maintain control over the heat delivered to keep the outlet temperature stable.
- **Expansion Valve (TEV)**: reduces the refrigerant pressure before it enters the evaporator.
- **Evaporator**: the evaporator absorbs heat from the air, aided by the **Fan** that moves air over the evaporator. This air is pre-cooled and then heated to the test temperature, ensuring that controlled test conditions are met at the evaporator inlet.

Data analysis

To analyze the setup some previous results are given in Tables 1 and 2 (average values). They consist of five steady states, covering thermostatic bath temperatures of 20°C, 30°C, 40°C, 50°C and 60 ° C, obtained using a fixed refrigerant charge and the thermostatic valve as expansion device.

Table 1. Experimental data of the primary (refrigerant) circuit.

Variable	Test 1	Test 2	Test 3	Test 4	Test 5
T_{compress-out}[ºC]	68.57	73.74	83.83	93.87	108.10
T_{condenser-in}[ºC]	65.29	72.41	77.60	82.91	89.72
T_{condenser-out}[ºC]	26.77	36.65	45.96	54.88	65.37
T_{expvalve-in}[ºC]	25.87	38.06	47.92	56.30	65.07
T_{expvalve-out}[ºC]	2.35	4.15	6.09	7.65	8.89
T_{evaporator-in}[ºC]	2.30	3.79	5.54	6.94	8.13
T_{evaporator-out}[ºC]	8.01	10.06	12.93	15.63	17.43
T_{compressor-in}[ºC]	9.04	10.99	13.78	16.36	18.21
p_{compressor-out}[bar]	9.441	11.606	14.188	16.950	20.445
p_{compressor-in}[bar]	2.920	3.097	3.314	3.499	3.664
m_{refrigerant}[kg/h]	34.77	35.07	34.92	34.15	32.18
We_{compressor}[W]	531.8	589.0	652.4	727.5	813.0

Table 2. Experimental data from secondary (coolant and air) circuits.

Variable	Test 1	Test 2	Test 3	Test 4	Test 5
Tcoolant_{in-PT100}[ºC]	20.55	30.86	40.67	49.91	59.96
Tcoolant_{out-PT100}[ºC]	27.18	37.05	46.58	55.42	64.77
Tair_{in}[ºC]	20.10	20.05	20.10	19.99	20.00
Tair_{out}[ºC]	6.63	7.7	8.85	9.72	11.17
HRair_{in}[%]	32.60	33.64	35.33	35.43	34.92
HRair_{out}[%]	76.00	72.93	68.77	65.54	59.31
m_{coolant}[kg/s]	0.080	0.079	0.078	0.077	0.076
m_{air}[kg/s]	0.122	0.121	0.121	0.120	0.119

p-h Diagrams of the Tests

To have a first look to the data given it is common to plot the p-h diagram. During the plotting process some assumptions were made:

- pressure drop has been considered negligible in each pressure level;
- the expansion valve has been considered isenthalpic.

The second assumption has been made because of the measurement of temperature and pressure at the outlet of the expansion valve. This specific point should be located inside the dome of the refrigerant of the cycle, the zone where pressure and temperature are strictly related. Just knowing temperature and pressure it's impossible to determinate the enthalpy of the point, it could be anywhere between saturated liquid or saturated vapor. That's why the valve has been considered isenthalpic, even if in reality it's not.

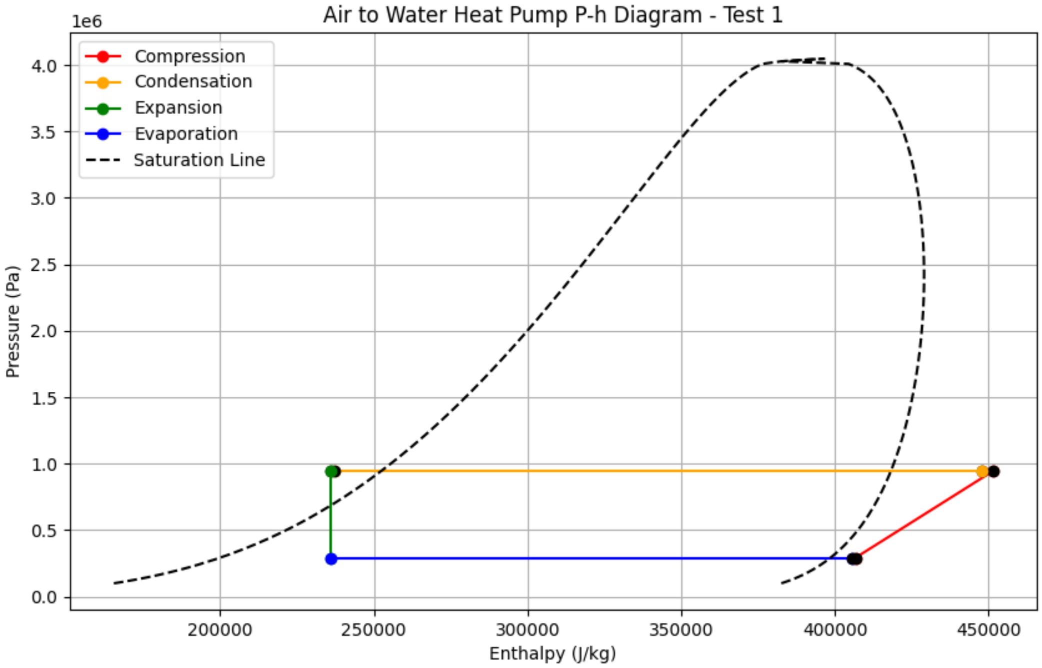

Figure 2. Air to Water Heat Pump p-h Diagram – Test 1.

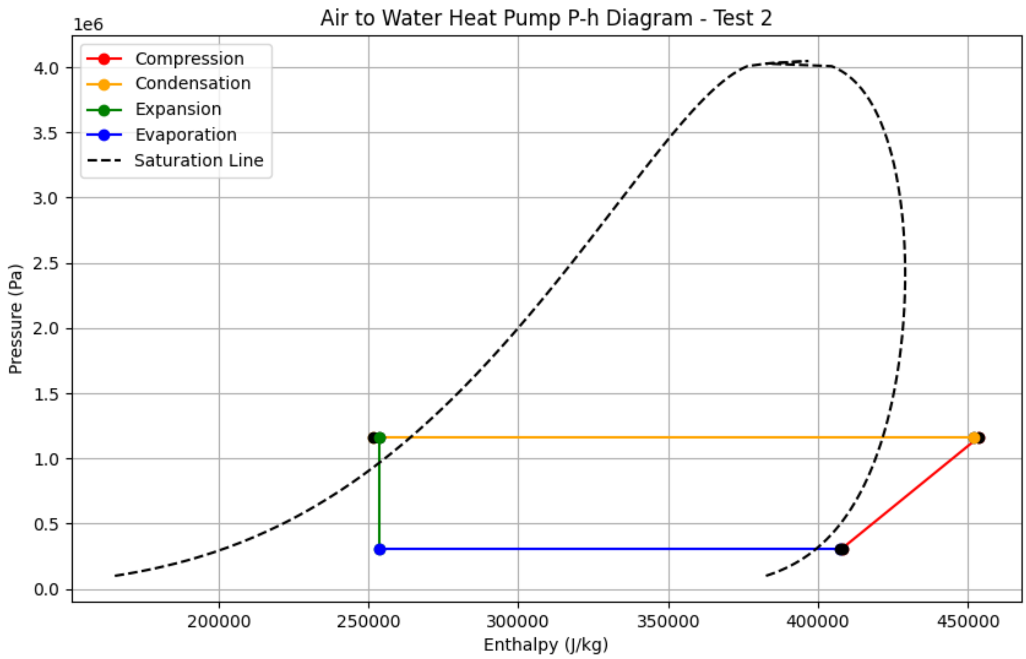

Figure 3. Air to Water Heat Pump p-h Diagram – Test 2.

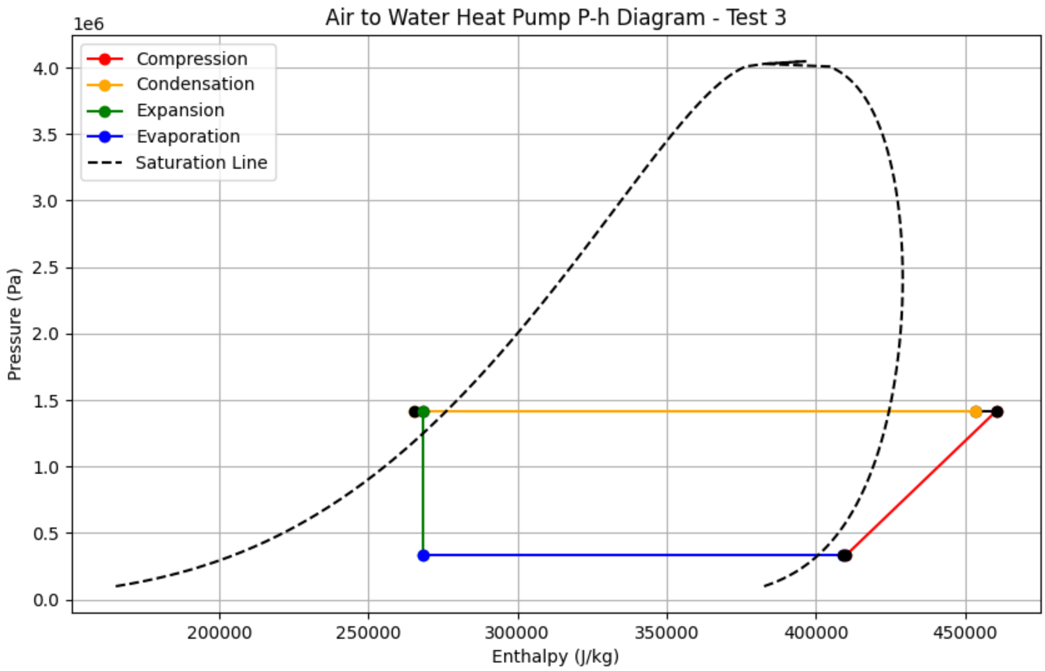

Figure 4. Air to Water Heat Pump p-h Diagram – Test 3.

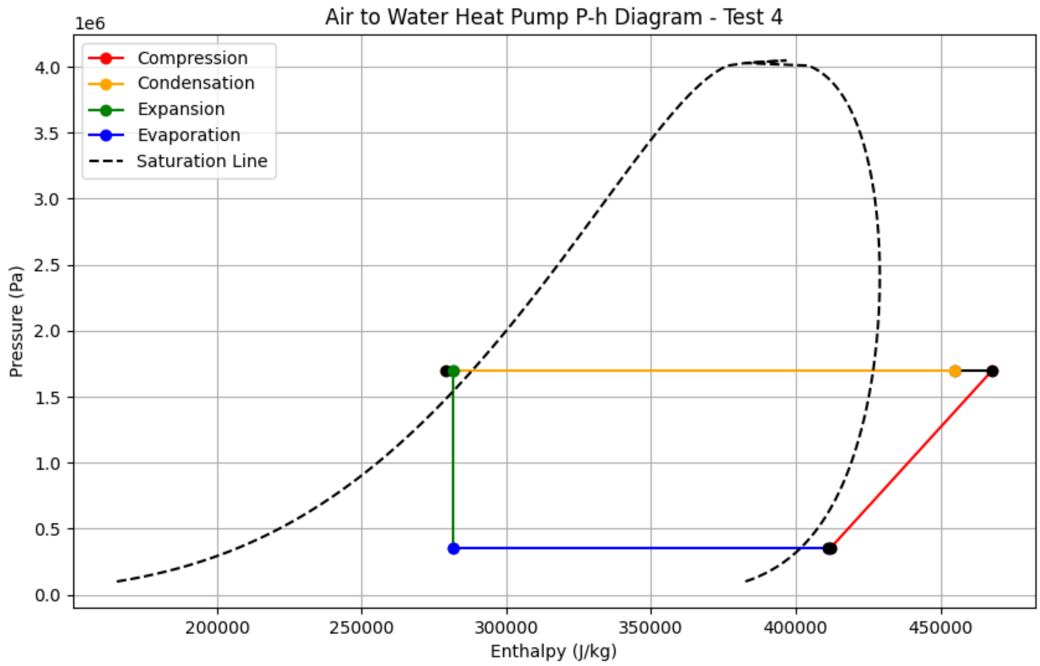

Figure 5. Air to Water Heat Pump p-h Diagram – Test 4.

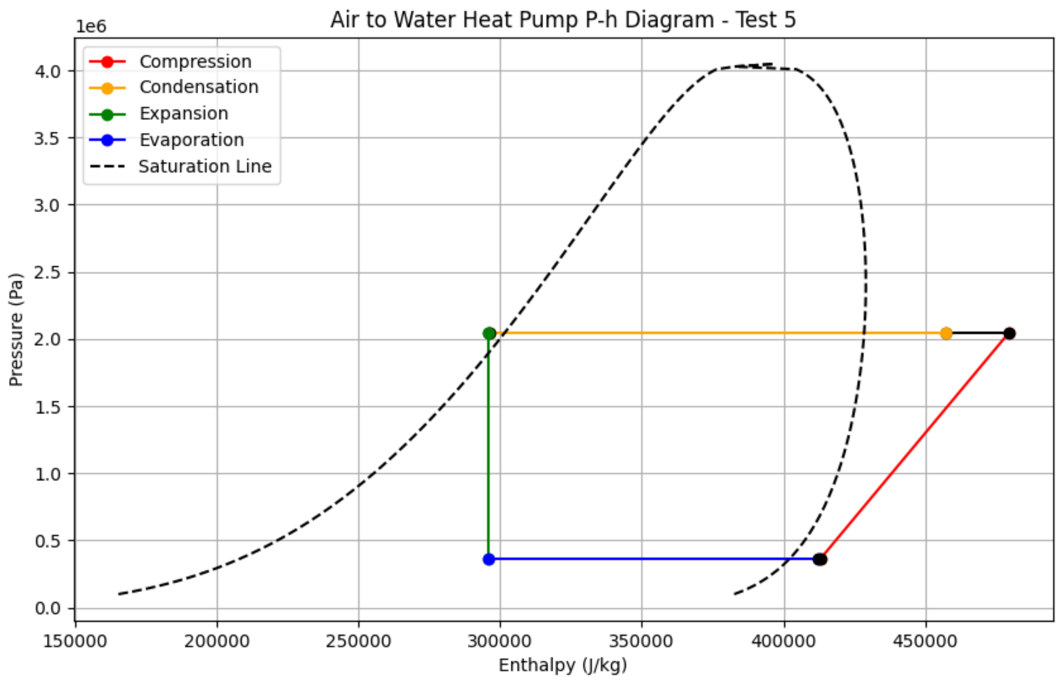

Figure 6. Air to Water Heat Pump p-h Diagram – Test 5.

When visualizing the cycle graphs, it becomes evident that as the required heating temperature rises, the temperature difference between condensation and evaporation also increases. This results in significantly higher compression work and, consequently, greater electrical consumption.

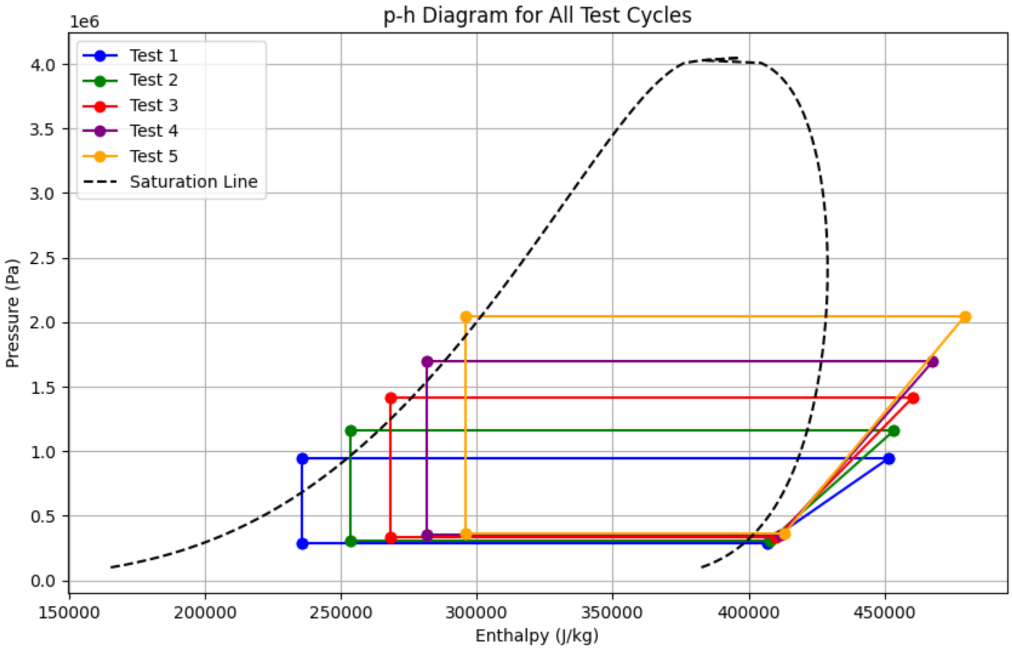

Figure 7. All Test's Cycles Comparison.

All the enthalpies used to build the graphs were obtained thanks to the physical properties library of CoolProp.

Compression Work Analysis

To continue with the data analysis, the compression work has been evaluated through the following correlation:

$$W_C = \dot{m}_{ref}(h_2 - h_1) \quad [W]$$

Where $\dot{m}_{ref}$ is the mass flow rate of the refrigerant R-134a (kg/s).

The results obtained have been compared with the values of electrical consumption given in Table 1, then both values have been plotted in function of the secondary fluid inlet temperature of the condenser.

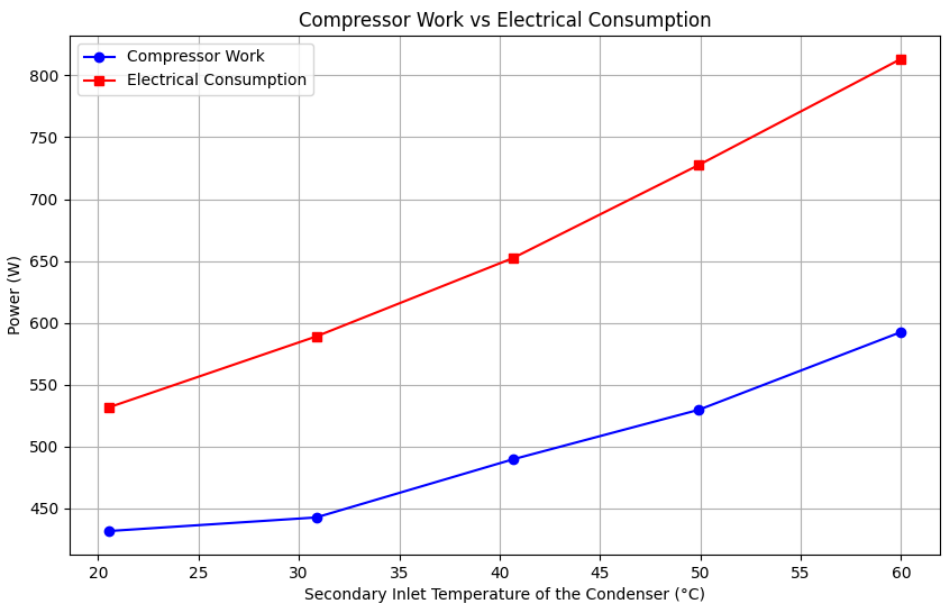

Figure 8. Compression Work and Electrical Consumption in function of the secondary inlet temperature of the condenser

Table 3. Values of Compression Work calculated compared to values of Electrical Consumption given.

	Test 1	Test 2	Test 3	Test 4	Test 5
W_C $[W]$	431.8	442.8	489.8	529.7	592.4
W_e $[W]$	531.8	589.0	652.4	727.5	813.0

The plot and the values of the table show how, increasing the temperature of the secondary flow, the work requested to the compressor raises and the same does the electrical consumption.

Heating Power and COP Evaluation

To complete the analysis, the coefficient of performance (COP) has been calculated, considering as total work the one made by the compressor plus the power of the fan. To accomplish this, it was necessary to evaluate the heating power of the condenser. This has been done for both refrigerant and the secondary fluid streams and then the COP of the heat pump has been evaluated considering the heating power of the refrigerant stream.

$$Q_{C_{refrigerant}} = \dot{m}_{ref}(h_3 - h_4)$$

$$Q_{C_{coolant}} = \dot{m}_{cool} c_{p_{cool}} \left(T_{cool_{out}} - T_{cool_{in}}\right)$$

	Test 1	Test 2	Test 3	Test 4	Test 5
$Q_{C_{refrigerant}}$ $[W]$	2037	1954	1825	1668	1440
$Q_{C_{coolant}}$ $[W]$	1963	1826	1736	1613	1402

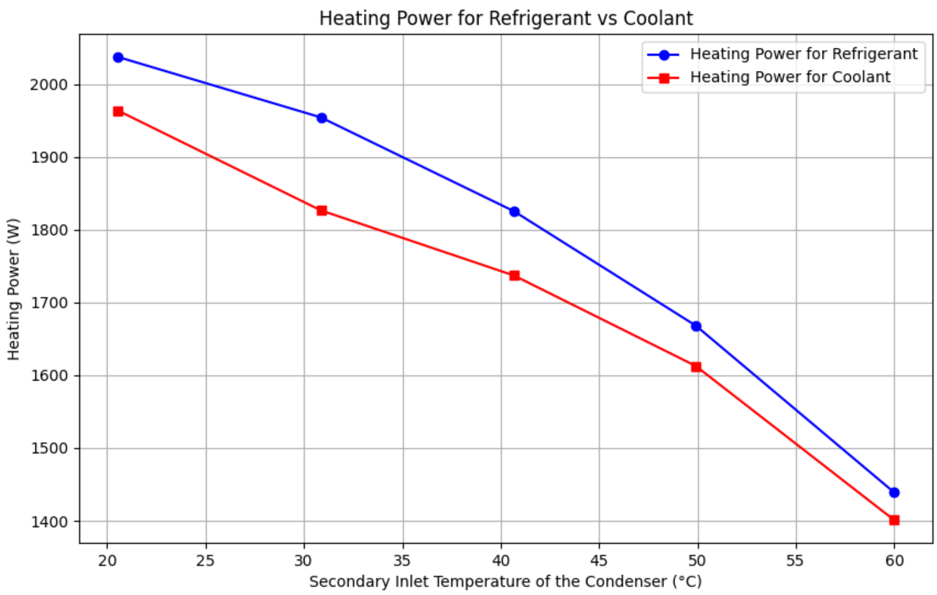

Figure 9. Heating power released by the refrigerant vs Heating power absorbed by the coolant.

The picture above compares the heating power released by the refrigerant and the heating power absorbed by the coolant, showing that there could be some heat losses through the condenser. Additionally, this loss seems to be less if working with higher temperatures, probably due to the less heating power that has to be exchanged.

Another important plot is the one that relates the COP of the cycles with the secondary inlet temperature of the condenser.

Table 4. Coefficient of performance of each test.

	Test 1	Test 2	Test 3	Test 4	Test 5
COP	3.90	3.66	3.14	2.69	2.11

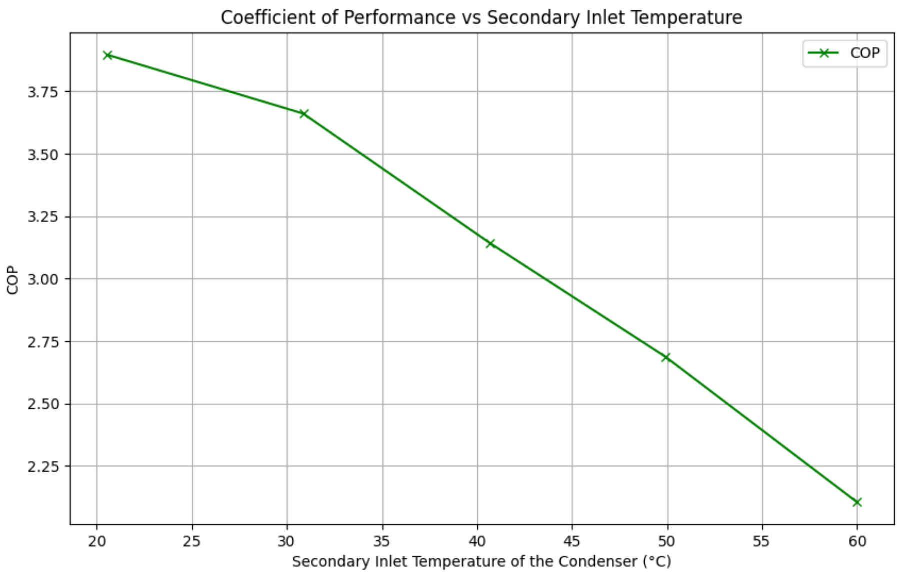

Figure 10. COP as a function of the secondary inlet temperature of the condenser.

From the data obtained and the figure that has been plotted it’s clear that the performance of the heat pump critically decreases with the increase of the temperature requested by the secondary flow.

Conclusions

The results align with the theoretical model, confirming both the accuracy of the measurement devices and the proper dimensioning of the experimental setup. The increase in requested heating temperature led to a higher temperature differential between condensation and evaporation, resulting in greater compression work and electrical consumption. Additionally, as the secondary inlet temperature rises, the coefficient of performance (COP) decreases, highlighting the balance needed between heating demands and efficiency.

References

- Practice: Air-to-water heat pump. Material prepared by professors and researchers from CTTC
- ASHRAE. (2021). *ASHRAE Handbook: Fundamentals.*
- CoolProp (v6.4.1) [Computer software]. (2024). Retrieved from http://www.coolprop.org

Annex – 1

R134a p-h Diagram