Project ‘SaveFood’ Final Report

Elforsk project: Udvikling af CO2 baseret luft til luft varmepumpe til tørringssystem i fødevareindustrien**/** Development of a CO_2 based air-to-air heat pump for Food drying System

Prototype development at DTU

The design configuration of the prototype CO_2 heat pump unit was largely based on a Master Thesis work carried out at DTU by Jonathan Hannibal Dam with the title: Investigation of heat pump solution(s) for food drying processes [1]. This numerical work investigated different heat pump configurations with heat recovery and recirculation as well as different refrigerants. Compared to electric heat or gas burner, the study indicated huge advantages and reductions in energy consumption up to 68 %. The CO_2 configurations with heat recovery by either (system 1) exhaust directly through the evaporator or (system 2) supply-exhaust heat recovery, resulted in similar energy performance. As CO_2 cannot exist as a liquid above 31 C, it was decided to incorporate a heat recovery heat exchanger (system 2), to reduce the approach temperature difference of the evaporator, which design evaporation temperature was 20 °C. Figure 1 shows the prototype system configuration. Notice that bypassing the heat recovery heat exchanger results in system 1.

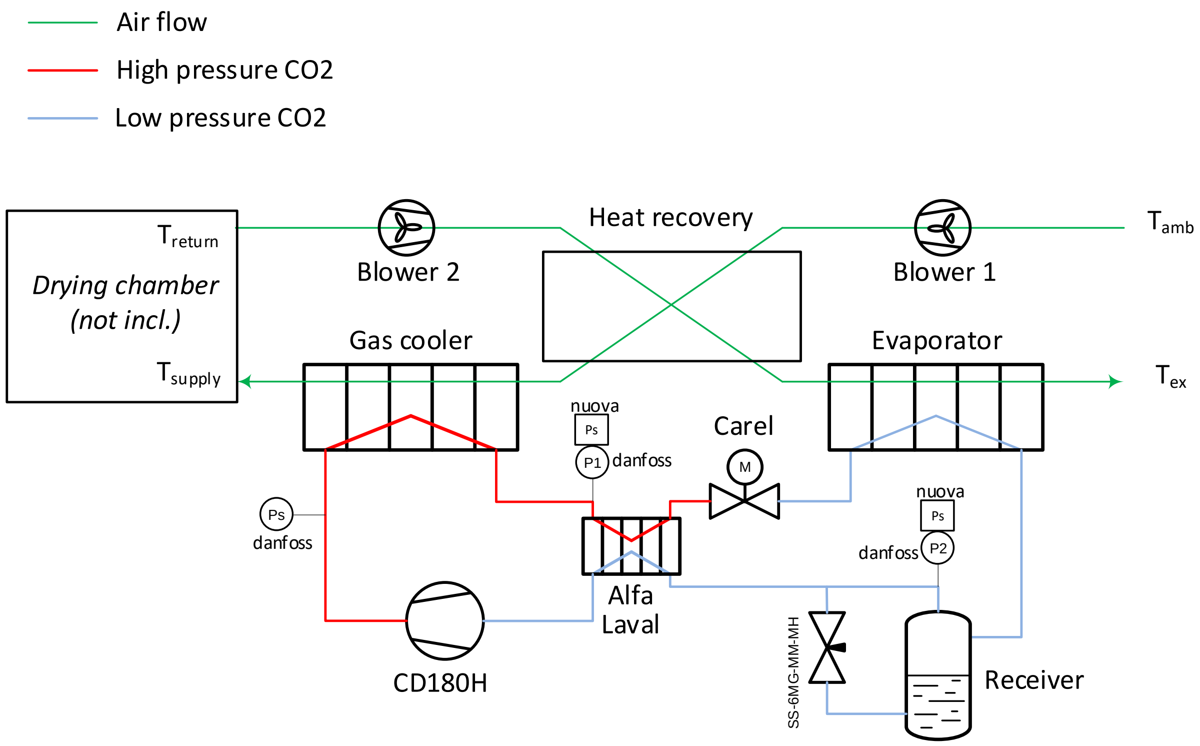

Figure 1: CO_2 prototype system configuration

The price of a commercial CO_2 condensing/gas cooler unit (7 kW) is currently 110.000 DKK through Danish refrigeration equipment resellers. CO_2 units are typically employing flash gas receiver with 3 expansion valves and controllers for high pressure, superheat and receiver pressure. Such a system is shown in Figure 2, without the evaporator. An immediate challenge with such a system is that the gas cooler only includes 3 tube rows and has a high frontal area, which is not good for utilization of the gas cooler temperature profile. On the other hand such transcritical condensing units are designed to simply cool the gas cooler without heating up the air.

Figure 2. CO_2 transcritical condensing unit by Profroid.

To minimize cost and complexity, it was decided to build the system from scratch and make it as cheap as possible. In the current design, we chose to install the simplified configuration proposed by Nekså et al. [2], which includes a suction accumulator and a suction line heat exchanger. Referring to Figure 1, the suction line heat exchanger serves to increase the temperature of the hot gas entering the gas cooler as well as evaporating partial oil-liquid mixture from the receiver. The latter is necessary for safe return of oil to the compressor. The evaporator, the gas cooler and the receiver are customized components that have been developed in collaboration with LU-VE Group and Frigomec, see Figure 3. The evaporator and gas cooler are especially designed for large air temperature differences (8 tube rows having cross-counter flow) and small refrigerant inventory (5 mm tubes).

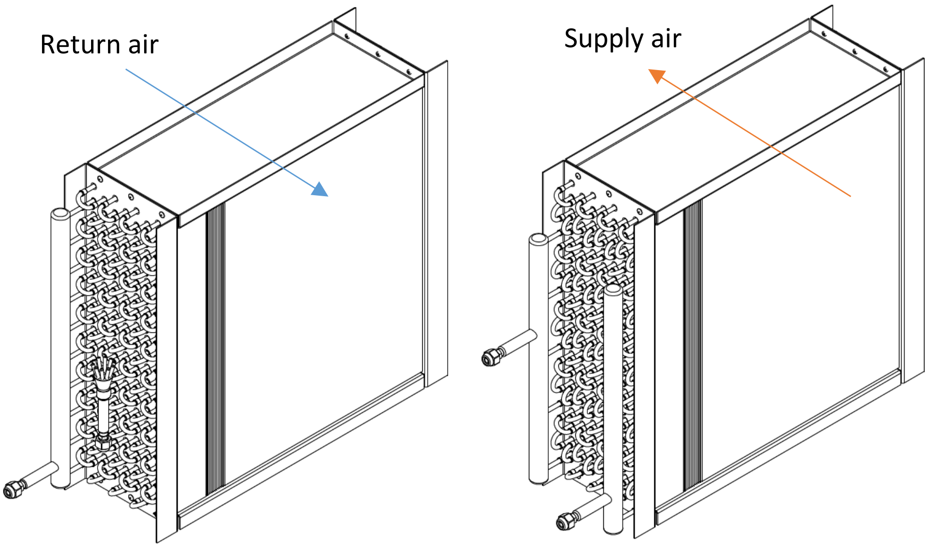

Figure 3. Evaporator (left) and gas cooler (right).

The receiver includes a sight glasses that allows to charge the refrigerant safely. The compressor is a Dorin compressor CD180H that was kindly sponsored by Advancer. The suction line heat exchanger was a plate heat exchanger that was kindly sponsored by Alfa Laval. The heat recovery heat exchanger was a Recutech counterflow heat exchanger kindly sponsored by Exhausto. Nuova General safety valves were placed on both low pressure and high pressure sides and a Danfoss high pressure switch was used to cut the compressor power in case of a high pressure excursion. A Danfoss frequency drive was used to control the speed of the compressor and the expansion valve was a Carel E2V-14 that controlled the high pressure. Systemair blowers type KV sileo 160 XL were used to move the airflow. Danfoss pressure transmitters and Wika pressure indicators were used to monitor the high and low pressures, and Danfoss temperature sensors were placed in between the various components of air and refrigerant flows. A customized controller (Danfoss MCX) was kindly sponsors by Danfoss and used to control the system and log the pressure and temperature signals. A power meter (Power Scout) is used to measure the power consumption of the compressor incl. the frequency drive. Figure 4 to 6 shows the conceptual design, cad drawings and pictures of the real unit.

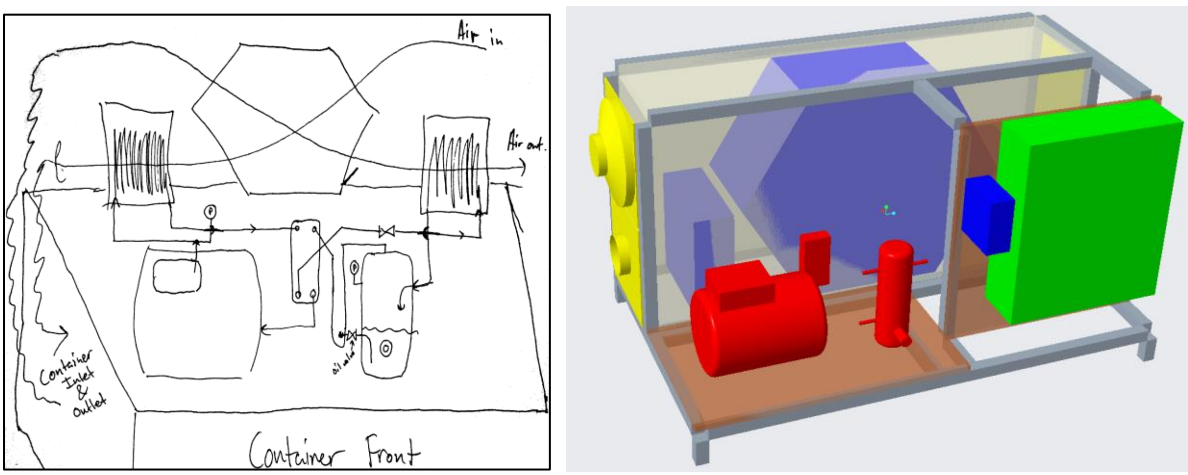

Figure 4. Conceptual drawing and cad drawing of the CO_2 unit.

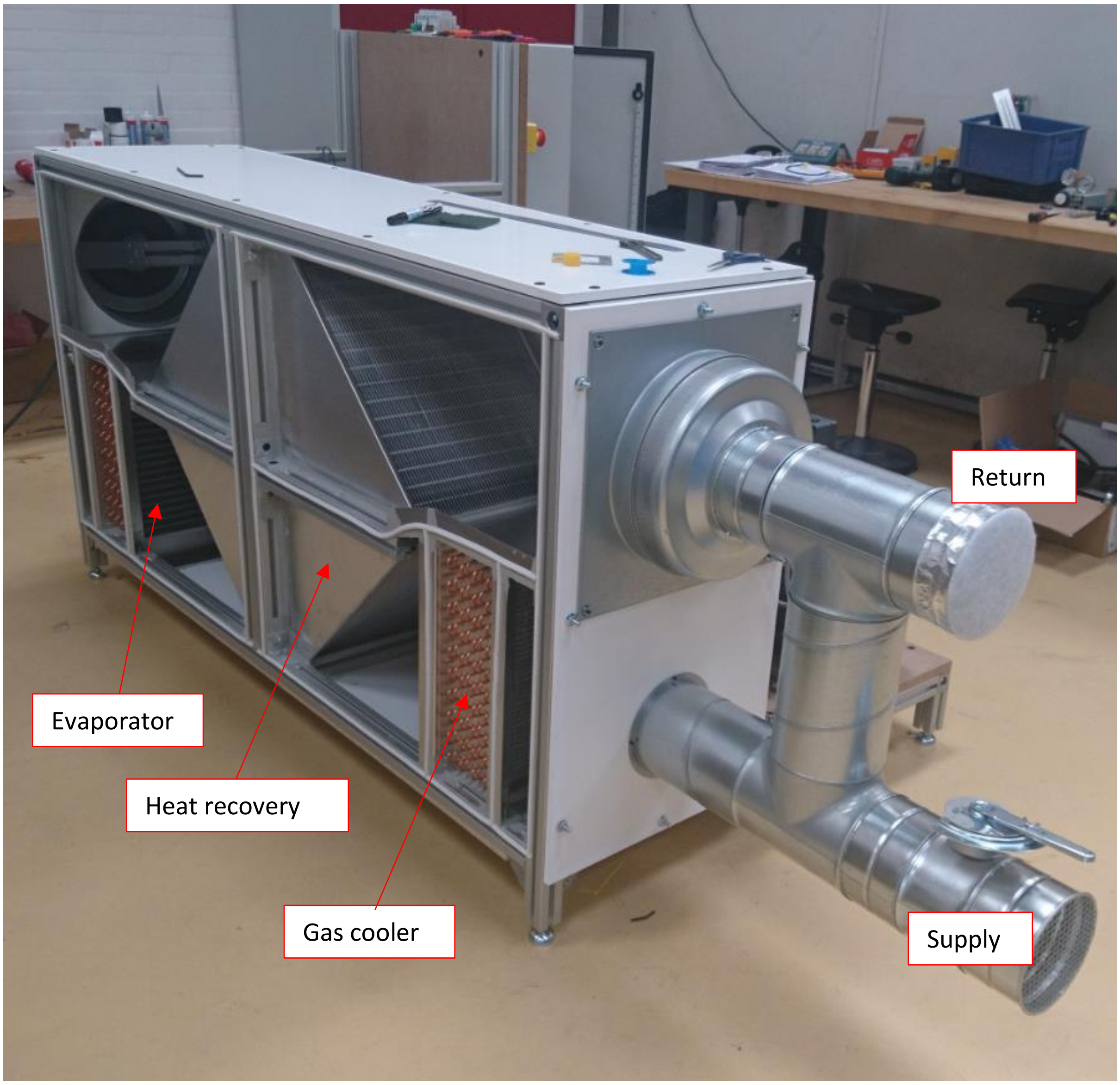

Figure 5. Air flow system

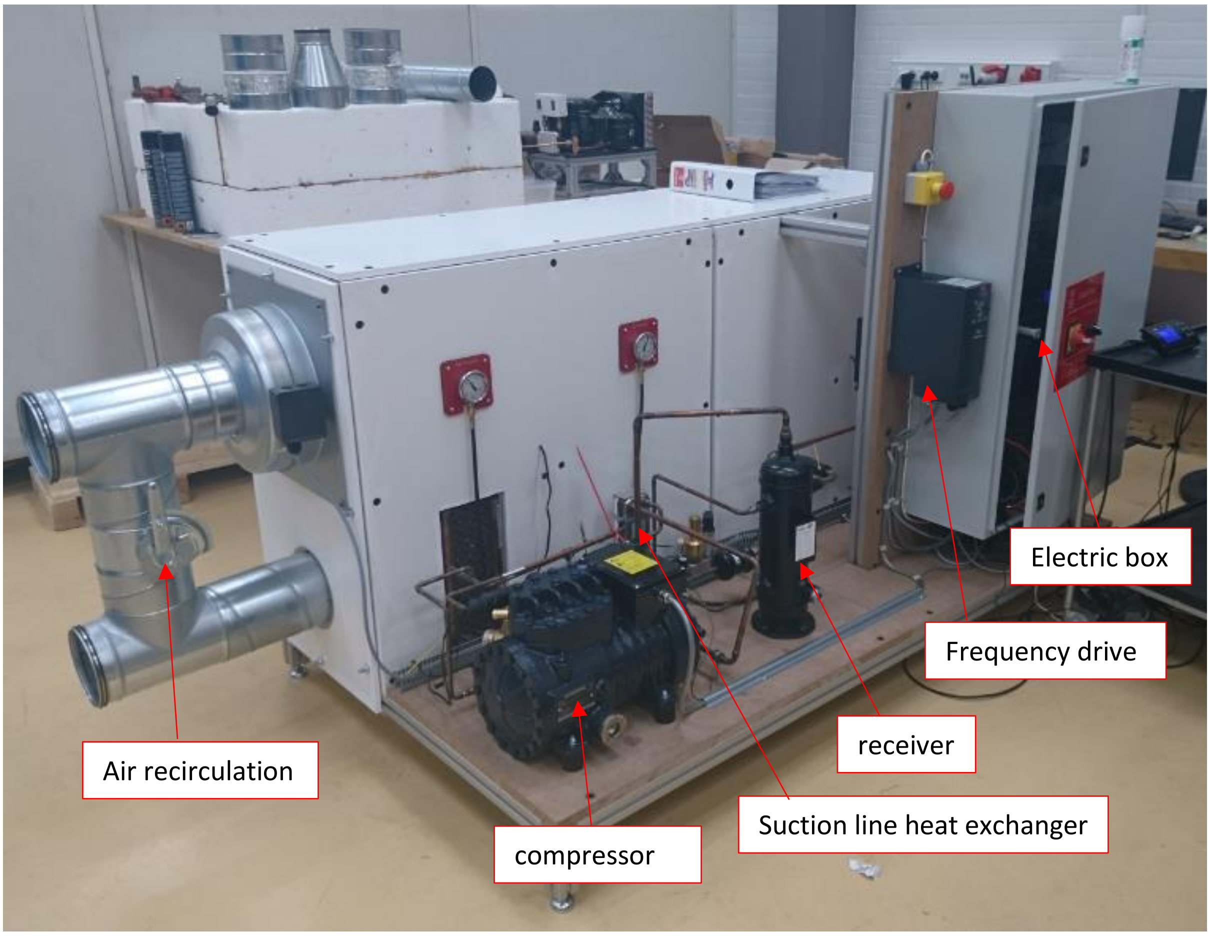

Figure 6. Heat pump system

Internet of Things (IoT)

Both Danfoss controller and power meter are interfaced by Modbus TCP/IP and a Raspberry Pi provides data to the cloud through SQL server (IoT). A customized user interface developed in python are used to monitor the data from a local pc. The local pc may also bypass the Raspberry Pi (IoT) for local operation solely. This work has been carried out by 2 students (Jonas Sønder Nielsen and Adam Mark Berman) through an ongoing special course with the title: Monitoring and control of CO_2 heat pump for drying. The special course ends during fall and many more experiments and documentation of the IoT solution are planned as well as dissemination at the 2022 Gustav Lorentzen conference on natural refrigerants. For these reasons, only few data points are provided in the results section without drying, but using air recirculation to mimic the temperature of the drying chamber exhaust.

Expenses

We had equipment expenses for about 56 kdkk and in-kind sponsorships of 33 kdkk. The budget only allowed for 30 kdkk and the rest 26 kdkk was kindly sponsored by DTU Mechanical Engineering through Weltsch's grant. These numbers are detailed in Table 1.

Table 1. Equipment expenses and sponsorships

Expenses	DKK
steel frame	4,952.93
electric box	7,541.69
Evaporator and gas cooler	12,598.15
expansion valve and drive	5,048.54
power meter	5,242.00
receiver	3,903.09
safety valve	2,730.05
fittings	6,633.91
VLT drive, pressure sensors, pressure switch	7,612.98
Total expenses	**56,263.34**
sponsorships:	
compressor	25,570.00
HRU	2,175.00
Danfoss controller	3,000.00
Alfa Laval	2,000.00
Expenses + sponsorships	**89,008.34**

Results

Two tests were carried out at a fixed nominal speed of the compressor at 50 Hz. The first test was characterized by ensuring 60 °C as the dryer supply temperature (T9), which is the design case. The second test was characterized by operating the gas cooler at 105 bar. Unfortunately, the high pressure safety valve started opening already at 110 bar instead of 120 bar, and a reimbursement is currently undertaken. This however limited the high pressure to values below 105 bar.

60 °C supply temperature and 85 bar gas cooler pressure

The time averaged measurements are shown in Figure 7, in which the bold text (temperatures T_1 to T_{12}, pressures p_{gc} and p_e, and compressor power $P_{compressor}$) are measured values and the regular text are computed values based on the measurements and the compressor polynomials. We use the compressor polynomials to calculate the actual mass flow rate of the compressor as well as the compressor shaft power, which includes the heat loss that is calculated by the measured discharge temperature (T_2). Given the measured power consumption, the loss through the frequency drive was 4.1 %.

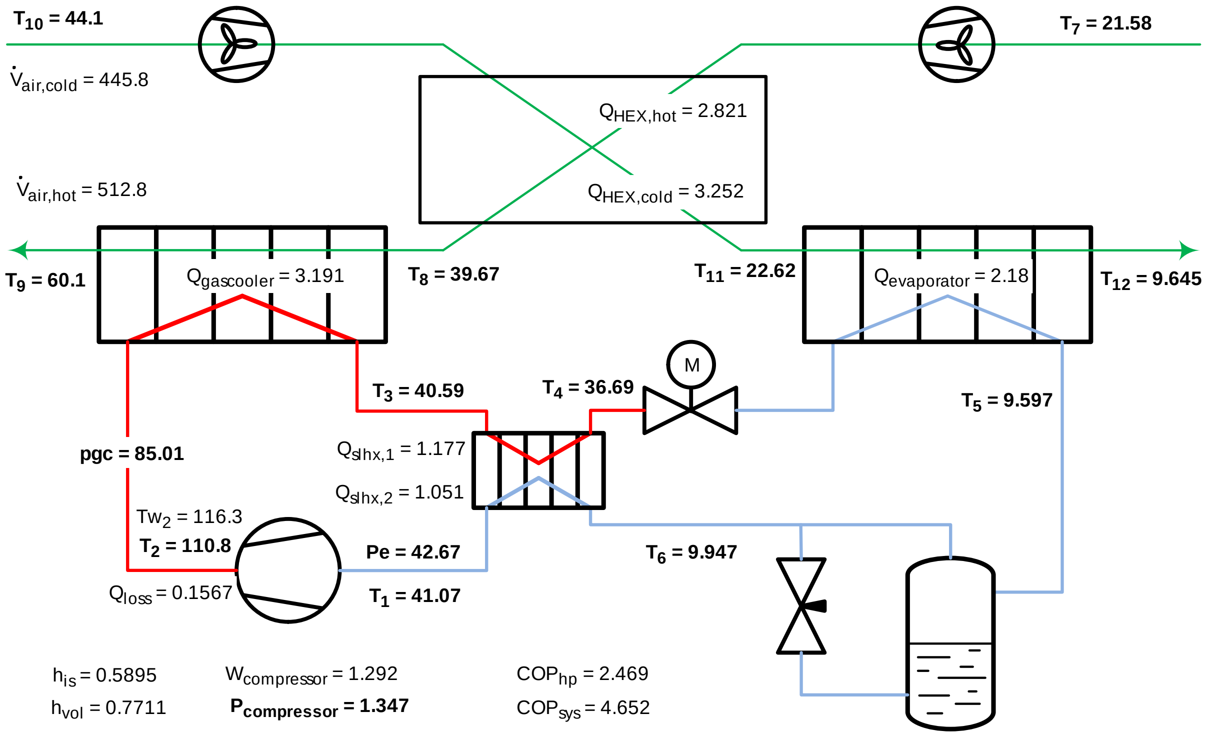

Figure 7. Tests at 60 °C supply temperature and 85 bar gas cooler pressure. Bold text are measured. Regular text are calculated.

The COP of the cycle and the system are defined by

$$COP_{hp} = \frac{Q_{gascooler}}{W_{compressor}}$$

$$COP_{sys} = \frac{Q_{gascooler} + Q_{HEX,hot}}{W_{compressor}}$$

and based on the shaft power. They resulted in 2.47 for the heat pump and 4.65 for the system, i.e. the latter heating from ambient to the supply. Ambient temperature in the laboratory was 21.58 C, and it determines through the heat recovery unit the approach temperature to the evaporator (22.62 C), by having a high efficiency. The result is an evaporation temperature of about 10 °C. This is lower than the designed value and partly caused by the lower volume flow of the cold air vs. the hot air, due to different pressure drops.

The cycle does not have superheat, which can be observed by comparing T_5 and T_6. This results in good evaporator performance without necessary superheat. The energy balances of the gas cooler and the evaporator are used to estimate the volume flow rate of the cold return and hot supply air. These volume flows are further used to calculate the heat exchange in the heat recovery unit, and the energy balance are within 13.2 %, which is good considering these repeatable use of energy balances and measurement uncertainties. The energy conservation in the suction line heat exchanger is within 10.7 %. The energy balance of the heat pump CO_2 cycle is within 3.7 %.

As mentioned, the COP of the heat pump is lower than that of the system COP, which takes into account the total heating from ambient temperature. If the heat recovery was bypassed, the COP of the heat pump would have been increased as the temperature difference across the gas cooler have been much higher. Moreover, the gas cooler exit temperature could have been cooled even further, but it all comes with a penalty of delivering lower supply temperatures. It is therefore important to take the whole heating into account, which in this case resulted in 6.0 kW total heat. On the other hand, more work is needed to understand the implications of the heat recovery heat exchanger.

The log-ph diagram and the temperature profiles in the gas cooler, suction line heat exchanger as well as the heat recovery unit is shown in Figure 8. It is interesting to notice also that the gas cooler outlet temperature is close to the return air temperature, because of the good heat recovery unit. This limits the level of gas cooling in the gas cooler.

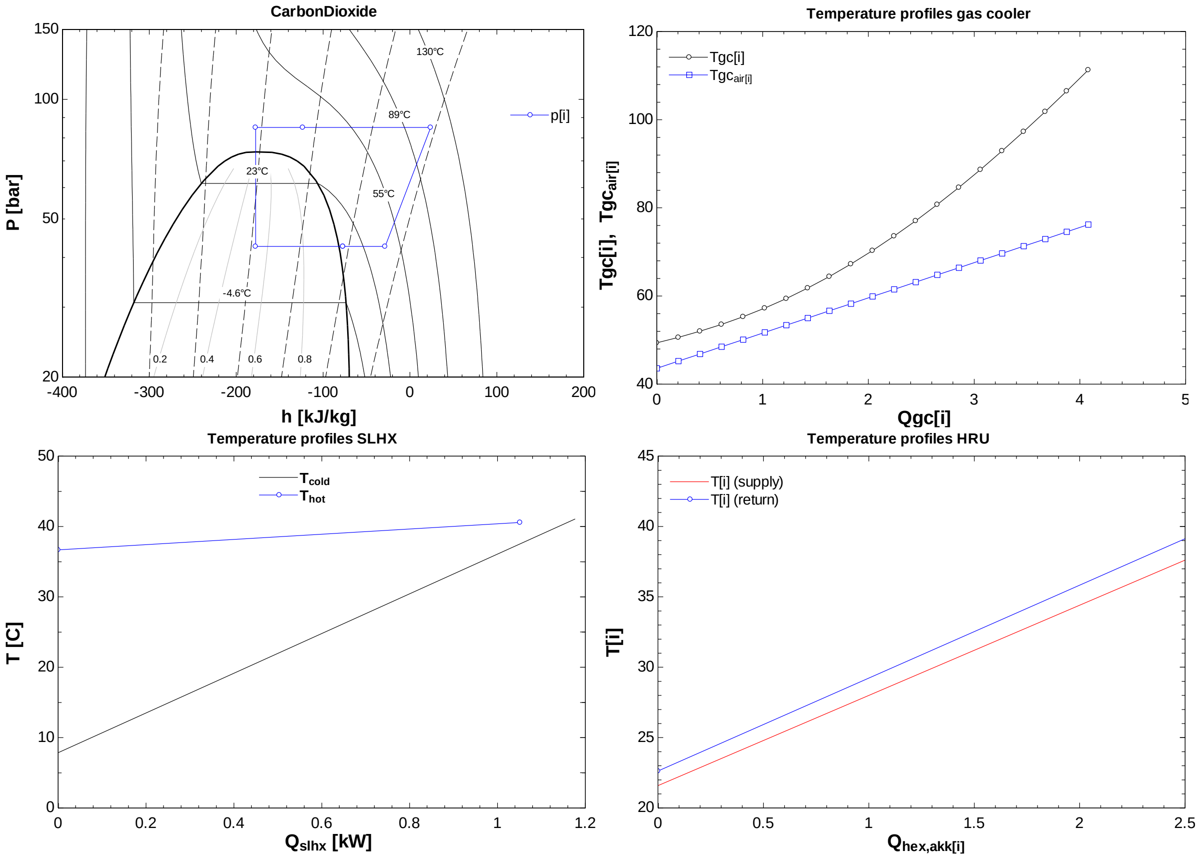

Figure 8. The log-ph diagram and the temperature profiles in the gas cooler, suction line heat exchanger as well as the heat recovery unit

105 bar gas cooler pressure

The time average measurements when operating at the highest available pressure, as controlled by the expansion valve, are shown in Figure 9. The results indicate that the COP of the heat pump as well as the COP of the system increases by 0.12 and 0.15, respectively, which is a slight increase. This means that the COP does not seem to be much dependent on the gas cooler pressure as one could expect. The benefit of increasing the gas cooler pressure is to increase the enthalpy difference across the gas cooler (see log ph diagram in Figure 10), however it comes with a penalty in terms of higher compressor power consumption. The two factors seem to more or less eliminate each other. The supply temperature are however increased to 66.4 °C.

The total heating capacity is 7.59 kW, which on the other hand is a significant increase +26.5 %. Furthermore, the evaporator temperature decreases slightly as the gas cooler pressure is increased, i.e. from 10 °C to around 7 °C. It is interesting to notice also the losses for both the frequency converter (6.7 %) as well as the heat loss from the compressor (0.31 kW) is higher when operating at the higher pressures.

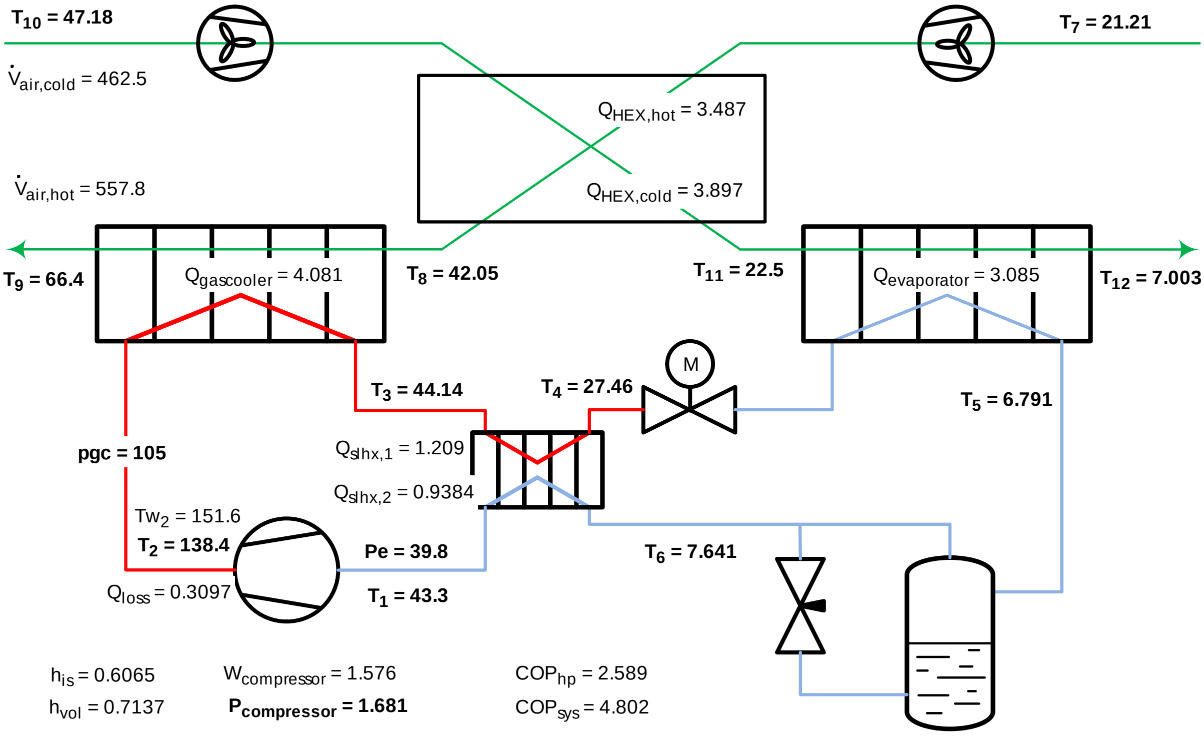

Figure 9 Tests at 105 bar gas cooler pressure. Bold text are measured. Regular text are calculated.

The log-ph diagram and the temperature profiles in the gas cooler, suction line heat exchanger as well as the heat recovery unit is shown in Figure 10. It is interesting to notice the high discharge temperature and the large temperature difference that are available for heat exchange in the gas cooler, however it only increase the supply temperature from 60 °C to 66.4 °C. Furthermore, the log-ph cycle is better by intuition, however, the calculations shows only little improvement in COPs.

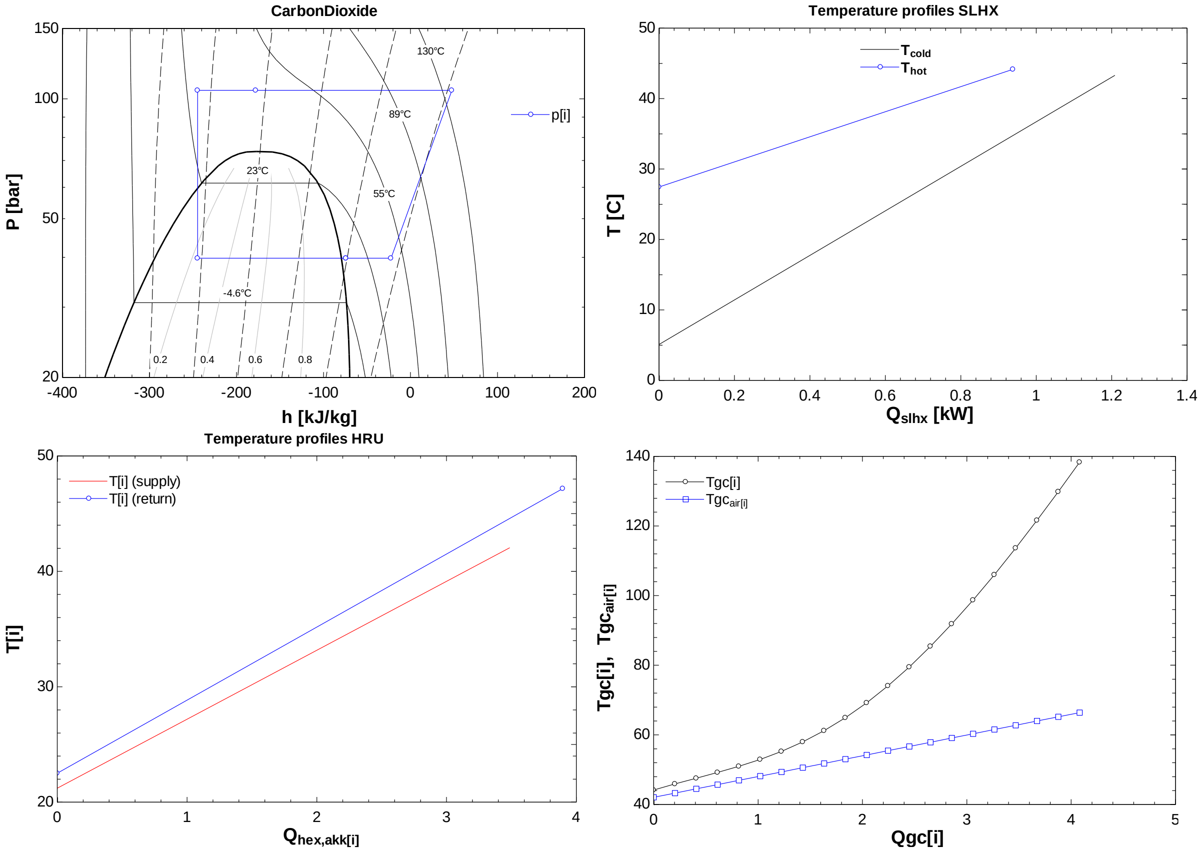

Figure 10. The log-ph diagram and the temperature profiles in the gas cooler, suction line heat exchanger as well as the heat recovery unit

Pilot Drying processing Unit Development (UPFOOD)

UpFood aim was to develop a mobile/portable drying system that can be transported to any surplus food or by- product source like a farm or a food industry etc. and save the food by drying right there. Imagine a drying system that can be easily transported and parked at a farm, catering or food production company and outside a supermarket.

Based on field experiences and requirement of the dried fruit value chain, it was decided to design a complete miniature portable drying factory instead of just a drying system. The design was based on utilizing the shipping containers which will provide easy deployment on a global scale considering food waste/loss is a global challenge. The container based modular was supposed to have both pre and post processing facility providing a complete solution where raw fruits/vegetables come and go out packed and ready for the market.

Due to COVID19 lockdown situation, the progress on development of CO_2 heat pump was halted at DTU. Considering the uncertainty at that time, UpFood then decided to develop and utilize another conventional R134a based heat pump operating a lower temperature than then CO_2 heat pump. The idea was to continue with other main goal of the project to develop and demonstrate a portable drying unit. The conventional heat pump would then be replaced by the CO_2 heat pump.

UpFood designed and developed a complete food drying facility with as shown in picture below.

Figure 11: Container based modular drying facility

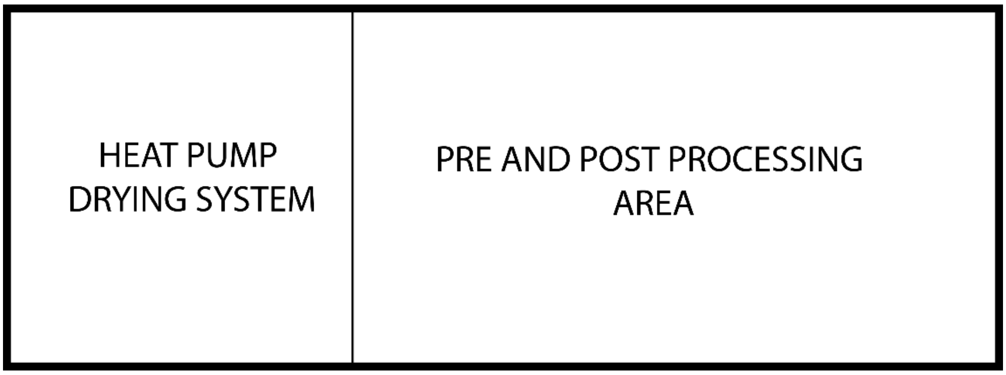

Figure 12: Inside Layout

The pilot production unit was installed at UpFood’s location at Alfalaval A/S facility in Søborg Denmark. The pilot system is now approved for processing both conventional and organic food products by Fødevarestyrelsen.

Figure 13: Pre-processing surplus tomato from a Danish producer

Surplus fruits from Danish farms (having lots of unsold fresh fruits due to COVID19 lockdown) are being received, processed, dried at this pilot unit and then also packed. Today this pilot facility is saving multiple surplus produce from Danish farms and supply to Danish supermarkets.

Figure 14: Retails packs for Dried Apple, Pear and Tomato

Discussion and future work

The results herein have demonstrated the feasibility of CO_2 as a heat pump for drying. More investigations are planned targeting the effect of the speed of the compressor, as well as the effect of the heat recovery heat exchanger. Furthermore, real dryer operation including significant dehumidification is pending. The development of the prototype and experimental results has also outlined that further investigations and evaluations are necessary to understand in more detail the losses and implications in the system especially as the gas cooler pressure as well as the return air is varied. The critical challenge here is to exploit the return air, but at the same time cool the gas cooler outlet gas as much as possible.

The modular drying unit, has demonstrated its applicability to impact the challenge of food loss at farms. The integration of CO_2 heat pump will not only improve its capability to efficiently process the surplus food but will also make it a more sustainable to the environment in terms of technology.

References

[1] Jonathan Hannibal Dam, Investigation of heat pump solution(s) for food drying processes, Technical University of Denmark, 2020.

[2] P. Nekså, H. Rekstad, G.R. Zakeri, P.A. Schiefloe, CO_2-heat pump water heater: characteristics, system design and experimental results, International Journal of Refrigeration. 21 (1998) 172–179.