NTNU – Trondheim
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Development of an energy efficient and environmentally friendly drum dryer using a heat pump with CO2 as working fluid

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MASTER THESIS

for

Student Øyvind Lomeland Knoph

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Development of an energy efficient and environmentally friendly drum dryer using a heat pump with CO_2 as working fluid

Utvikling av energieffektiv og miljøvennlig tørketrommel med bruk av CO_2 varmepumpe

Background and objective

Drum dryers have been used for a long time in homes for drying of laundries. These dryers have been developed over years from the simple type with direct electric heaters and rejection of the humid air to the ambient. The next generation included a heat exchanger between the inlet air and the exit air from the drum (condensation units). In this case it is possible to reduce the electric consumption. More modern system have been developed with a heat pump for cooling of the air to a temperature below the dew point (condensation and removal of water) and then reheating of the inlet air to the drum dryer. In this case we have a closed loop of the air in the dryer. Typical working fluid in this type of dryer is R134a. This refrigerant has a GWP factor of 1300. The industry of these type of dryers like to reduce the environmental impact factor of their systems and have looked into using CO_2 as the working fluids in the heat pump system. Our department has over a period worked together with an industrial partner to develop a new system using CO_2 as working fluid.

In this project we will change the existing compressor with a new two stage compressor from Sanyo. This compressor will give the possibilities to have a liquid separator at an intermediate pressure.

The following tasks are to be considered:

1. Literature review for use of CO_2 in drum dryers
2. Plan the experimental setup
3. Optimize charge of the working fluid and size of the external heat exchanger
4. Develop control strategy for the drum dryer
5. Write a scientific paper with the main results from the project
6. Make proposal for further work

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents must be given to the supervisor in digital format. All relevant data collected and produced during the project shall be delivered to the supervisor on a CD at the end of the project.

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, January 14th 2014

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Preface

This report is the result from a master thesis at the Department of Energy and Process Engineering at the Norwegian University of Science and Technology. It accounts for 30 credits, which is one full semester.

The main focus of this study has been to implement modifications to a heat pump system, handle problems as they arise and perform tests with accompanying data processing.

A lot of the work has been practical work in the laboratory, which I am very grateful for. I have not been doing much laboratory work before, and relating practical incidents to theory have helped me to understand the processes better.

I want to thank my supervisor Trygve Magne Eikevik and my co-supervisor Inge Håvard Rekstad for their help and support with both laboratory and process-related issues. I also want to thank Lars Konrad Sørensen and Odin Hoff Gardå for their help regarding installation and solutions related to mechanics and instrumentation respectively.

Øyvind L. Knoph

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Trondheim 10.06.14

Summary

The drum dryer is a common appliance for drying clothes in households today. It is constantly developed in order to reduce its environmental impact and energy consumption. An industrial partner has initiated a research project with the aim to substitute R134a with CO_2 as refrigerant. R134a has a global warming potential (GWP) 1300 times the GWP of CO_2, thus CO_2 is a much more environmental friendly refrigerant.

A drum dryer driven by a heat pump was previously modified to use CO_2 as refrigerant. This system has now been further developed to include a two-stage compressor and a liquid separator at an intermediate pressure. The flash gas is separated on the intermediate pressure stage and injected into the compressor. Theoretically the result should be a higher mass flow on the high-pressure stage and reduced work required for 1st stage compression. Closing a blocking valve at the intermediate stage can disable the flash gas recycling process. This gives the opportunity to compare the results to the results from a regular cycle.

The results from the experiments in this study are compared to results from similar experiments. Results are available from experiments by former students and initial tests by the industrial partner. The industrial partner also provided an ultimate goal of 0.26 $kWh/kg_{textiles}$, which is the state of the art consumption for R134a heat pumps.

The two main performance measures are specific energy consumption based on weight of the textile load and specific energy consumption based on weight of the removed water. The best experiment in this study consumed 0.37 $kWh/kg_{textiles}$ and 0.61 $kWh/kg_{removed\ water}$. These results were achieved by disabling the flash gas injection. The corresponding results using flash gas injection were 0.41 $kWh/kg_{textiles}$ and 0.70 $kWh/kg_{removed\ water}$. In the current setup the flash gas injection has not contributed to reduced energy consumption. There has been a challenge to control the flash gas injection while keeping the superheat low out of the evaporator. Suitable modifications have been suggested to overcome this issue.

Sammendrag

Tørketrommelen er et svært vanlig hjelpemiddel for tørking av klær i husholdninger og blir stadig utviklet for å redusere energiforbruk og miljøpåvirkning. En industripartner har tatt initiativ til et forskningsprosjekt med NTNU for å bytte ut R134a med CO_2 som kuldemedium. R134a har et globalt oppvarmingspotensial (GWP-verdi) som er 1300 ganger så høyt som CO_2, CO_2 er altså et mye mer miljøvennlig kuldemedium.

En tørketrommel som drives av en varmepumpe har tidligere blitt modifisert for å kunne bruke CO_2 som kuldemedium. Dette systemet har nå blitt videreutviklet til å inkludere en to-stegs kompressor og en separator på et mellomtrykk. Arbeidsmediet kommer inn i separatoren som en blanding av gass og væske. Disse fasene separeres og gassen føres inn i kompressoren igjen på mellomtrykket. Teoretisk sett skal dette gi en høyere massestrøm på høytrykksiden og et lavere kompressorarbeid. Man kan velge å kjøre systemet uten å ta ut gassen av massestrømmen ved å stenge en stengeventil på mellomtrykket, og dermed kjøre systemet som en vanlig varmepumpekrets. På denne måten kan effekten av denne modifikasjonen finnes ved å sammenligne resultatene.

Resultatene fra eksperimentene som er gjort i denne studien sammenlignes med resultater fra lignende eksperimenter. Resultater fra eksperimenter gjort av tidligere studenter samt innledende eksperimenter fra industripartneren er tilgjengelige. Denne partneren har også satt et mål om å bruke høyst 0.26 $kWh/kg_{tekstiler}$, som er forbruket til en moderne varmepumpekrets ved bruk av R134a.

De to hovedmålene for ytelse som er brukt er spesifikt energiforbruk basert på vekt av tørre klær og spesifikt energi forbruk basert på vekt av vannet som er fjernet. Det beste eksperimentet i denne studien brukte 0.37 $kWh/kg_{tekstiler}$ og 0.61 $kWh/kg_{vann\ fjernet}$. Disse resultatene oppnådd ved å stenge stengeventilen på mellomtrykket. De beste resultatene som ble oppnådd ved å ta ut gassen på mellomtrykket var 0.41 $kWh/kg_{tekstiler}$ og 0.70 $kWh/kg_{vann\ fjernet}$. Med det nåværende oppsettet har ikke væske/gass-separasjonen bidratt til å redusere energiforbruket. Det har vært utfordrende å kontrollere uttaket av gass og samtidig begrense overhetingen ut av fordamperen. Forslag til forbedringer har blitt foreslått.

Content list

List of figures

List of tables

Nomenclature

CO_2 = Carbon dioxide
R744 = Carbon dioxide
R134a = Tetrafluoroethane
R290 = Propane
R410a = mix of difluoromethane and pentafluoroethane
HFC = Hydrofluorocarbons
COP = Coefficient of performance [-]
MER = Moisture Extraction Rate [$Kg_w\ s^{-1}$]
SMER = Specific moisture extraction rate [$Kg_w\ kJ^{-1}$]
d = diameter [m]
e = surface roughness [m]
g = acceleration of gravity [$m\ s^{-2}$]
h = specific enthalpy [$kJ\ kg^{-1}$]
L = length of the capillary tube [m]
M = mass flux [$kg\ s^{-1}\ m^{-2}$]
p = Pressure [bar]
T = Temperature [°C]
t = Temperature [°C]
U = overall heat transfer coefficient [$W\ m^{-2}\ K^{-1}$]
x = gas fraction
ϕ = two-phase multiplier
α = convection heat transfer coefficient [$W\ m^{-2}\ K^{-1}$]
η = efficiency
η_v = volumetric efficiency
ρ = density [$kg\ m^{-3}$]
v = specific volume [$m^3\ kg^{-1}$]
$\dot{m}$ = mass flow rate [$kg\ s^{-1}$]
$\dot{V}$ = Volume flow rate [$m^3\ s^{-1}$]
V = Volume [m^3]
n = Rated speed [s^{-1}]
Re = Reynolds number
We = Weber number
Fr = Froude number
f = friction factor

A = Area [m^2]
σ = surface tension [$N\ m^{-1}$]
u = velocity [$m\ s^{-1}$]
h = specific enthalpy [$kJ\ kg^{-1}$]
x = specific humidity [$kg_w\ kg_{dry\ air}^{-1}$]
Q = heat energy [kJ]
$\dot{Q}$ = Heat energy per second [kW]
C_p = Specific heat capacity [$kg\ kJ^{-1}\ K^{-1}$]

Subscripts

cap = capillary tube
evap = evaporator
gc = gas cooler
is = isentropic
liq = liquid
SP = single-phase
TP = two-phase

1. Background and Objectives

The drum dryer has been a subject of constant development since the first simple types with direct electrical heaters. Modern units include a heat pump to dehumidify and heat the air in a closed cycle. The working fluid for the heat pump cycle is typically R134a. The industry is now aiming to reduce the environmental impact from their tumble dryers by developing a new system using CO_2 as the working fluid. An industrial partner has initiated collaboration with the Norwegian University of Science and Technology (NTNU) to develop such a system. This study is a part of that collaboration.

A tumble dryer system has previously been converted from using R134a to CO_2 as refrigerant in the heat pump cycle. A few modifications to the existing rig will be made in order to investigate whether they are appropriate in terms of reduced energy consumption. A new two-stage compressor will replace the old single-stage compressor. The new compressor gives a possibility to include a liquid separator at the intermediate pressure stage. During throttling some flash gas will be produced and this can be injected into the compressor at intermediate pressure.

By introducing the intermediate pressure stage it is also necessary to throttle from the gas cooler to the evaporator in two stages. Different throttling methods and pressure ratios will be discussed and tested.

2. Literature Review

2.1. The Tumble Dryer

A tumble dryer is an appliance for drying clothes. The first cloth tumble dryer was invented around 1800 and was powered only by a hand crank. The first electrical dryer was introduced early in the 1900s. The tumble dryer is now a common appliance in households. The first dryers drew ambient air and heated it before passing it through the drum to dry the clothes. The hot humid air was released to the ambient, which represents huge losses. Modern tumble dryers, often called condenser dryers, use heat pump technology where the cold side cools and dehumidifies the circulating air and the hot side heats the air. This way the heat absorbed from the dehumidifying process is recycled and losses to the ambient are greatly reduced.

2.2. Working Fluid

In the early history of refrigeration CO_2 was used as the working fluid [1]. When the chlorofluorocarbons like R12 (also known as CFC12 or Freon12) were introduced early in the 1900s, they took over as the preferred refrigerants. The reason why CO_2 was phased out is because of its loss of capacity in high ambient temperatures and expensive, inefficient CO_2 compressors.

As a direct response to the Montreal Protocol of 1989 hydrofluorocarbons (HFCs) like R134a were introduced to replace the CFCs because of their very high ozone depletion level. R134a and other HFCs has no ozone depletion potential (ODP) because chlorine is eliminated from their chemical structure. They also offer a global warming potential (GWP) much lower than for CFCs, but it’s still much higher than for non-fluorinated alternatives[2]. R134a has a GWP of 1300. GWP is a measure of how much heat the greenhouse gas traps in the atmosphere, and the reference is CO_2 with a GWP of 1.

CO_2 is a colourless gas and is also odourless at low concentrations. A phase diagram for CO_2 is included in Figure 1. CO_2 in solid state is called “dry ice” and is used for purposes like food preservation, industrial cleaning and in fog machines for theatres, nightclubs and so on [3].

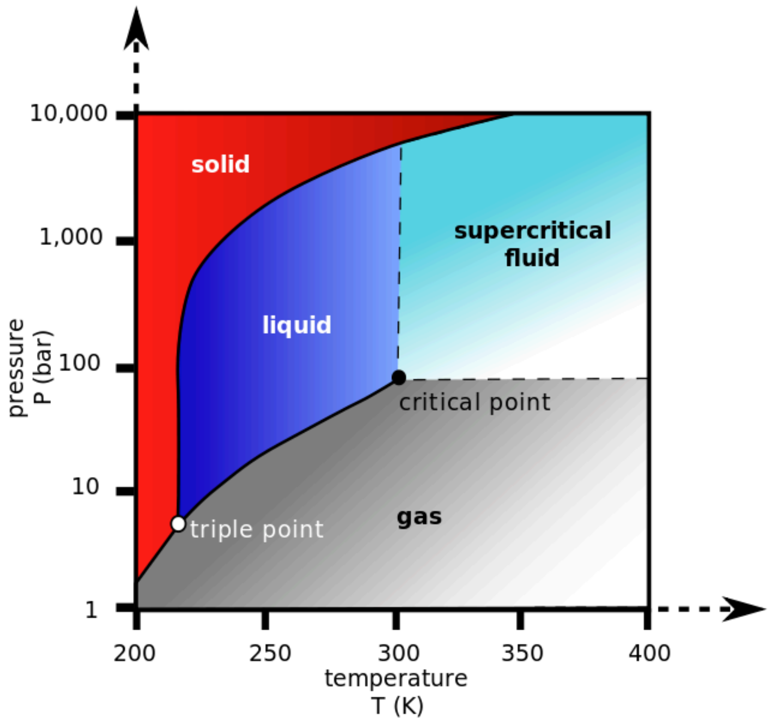

Figure 1: Phase diagram CO2

The industry that produces drum dryers aim to reduce their environmental impact by substituting their refrigerants from R134a to CO_2. There are other suitable gases in the atmosphere that can be used as refrigerants as well, an example being NH_3. CO_2 is often preferred before other atmospheric gases because it is non-toxic and non-flammable. A study has been made on a drum dryer comparing its performance running with R134a and the same dryer running with CO_2. The conclusion from this study suggests that use of CO_2 as refrigerant should not lead to higher energy consumption [4]. Properties of the working fluids are given in Table 1. The main difference during operation is that CO_2 generally operates in a transcritical process.

Table 1: Properties of R134a and R744

Property	R134a	R744 (CO_2)
Ozone depletion potential	0	0
Global warming potential	1300	1
Density at room temperature and atmospheric pressure	4.25 kg/m^3	1.98 kg/m^3
Critical pressure	40.7 bar	73.8 bar
Critical temperature	101.2 °C	31.1 °C
Pressure at 22 °C (saturated gas)	6.08 bar	59.69 bar
Refrigeration capacity at 0 °C	2.86 kJ/m^3	22.6 kJ/m^3

2.3. The Transcritical Process

CO_2 has a very low critical point, as stated in Table 1. Temperatures below 31.1 °C are rarely useful to supply heat, thus a heat pump using CO_2 as refrigerant will generally deliver heat in the supercritical area. In the supercritical area there will be no condensation, and the heat exchanger in this region is called a gas cooler rather than a condenser. As there is no condensation, the heat is released at a gliding temperature while the pressure remains constant.

2.4. Experiences

Schmidt et al. [5] compared the traditional R134a-cycle to a CO_2-cycle. They found that the exergy loss for air heating is considerably smaller for a CO_2-cycle than for the conventional R134a-process. This is because the transcritical isobaric line for the CO_2-cycle fits the air heating line much better than the isobaric line for the R134a-cycle. This is shown in the T-h diagrams in Figure 2 and Figure 3. However, the throttling losses are greater for CO_2 due to the difference between the specific heats of the fluids and the positions of the throttling curves relative to the critical points. They also found some different compression losses, which may be explained by different isentropic efficiencies and different compression ratios. Their conclusion was that a heat pump using CO_2 as a working fluid should not use more energy than using R134a as working fluid. Figure 4 compares the T-s diagrams of R134a and CO_2.

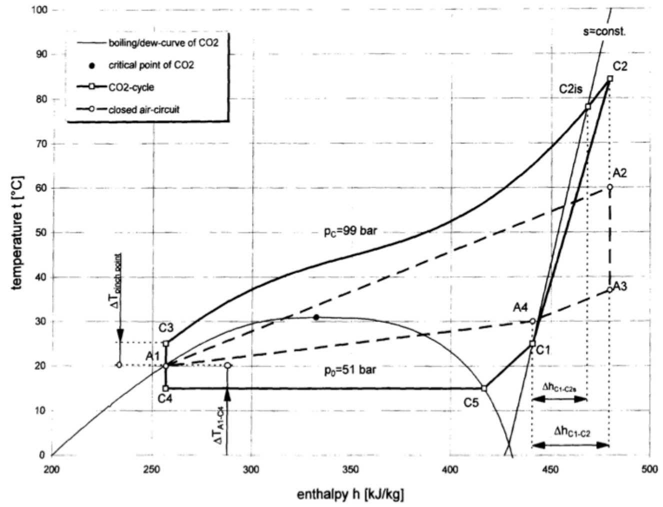

Figure 2: The transcritical process including the air cycle [5]

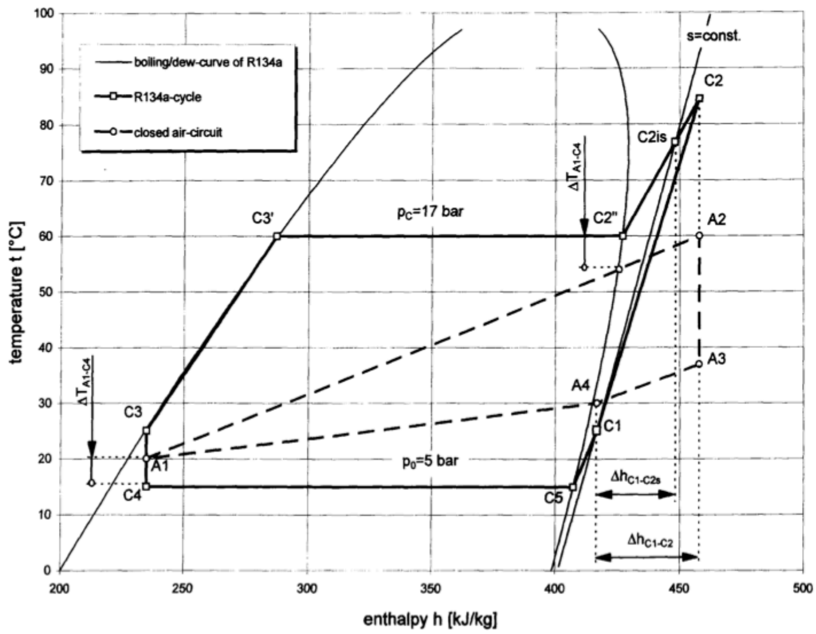

Figure 3: The R134a cycle including the air cycle [5]

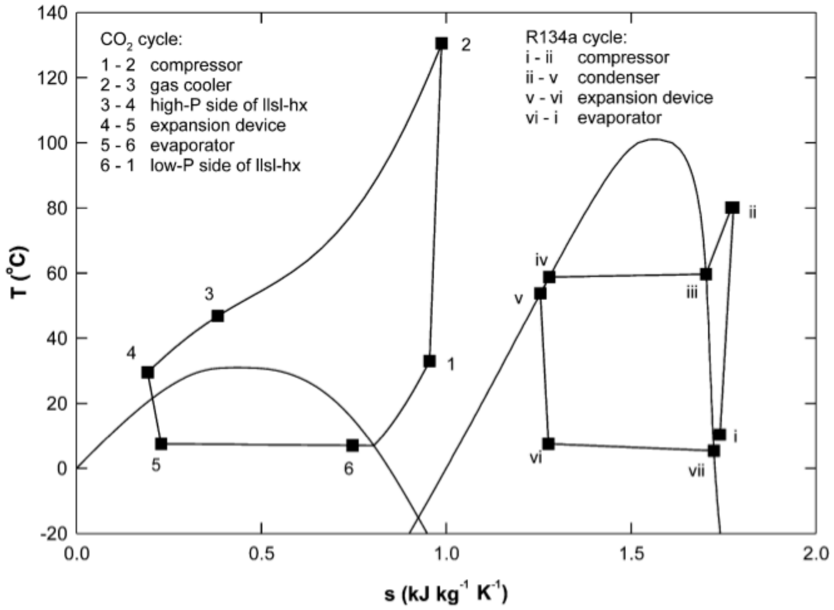

Figure 4: Comparison of a CO2 and R134a cycle [6]

Hashimoto et al. [7] designed a compact heat pump using CO_2 as refrigerant for an air heating heat pump for industrial purposes. The performance of the heat pump varied with air outlet temperature as shown in Figure 5. The prototype operated with a COP between 3 and 4, which is very good.

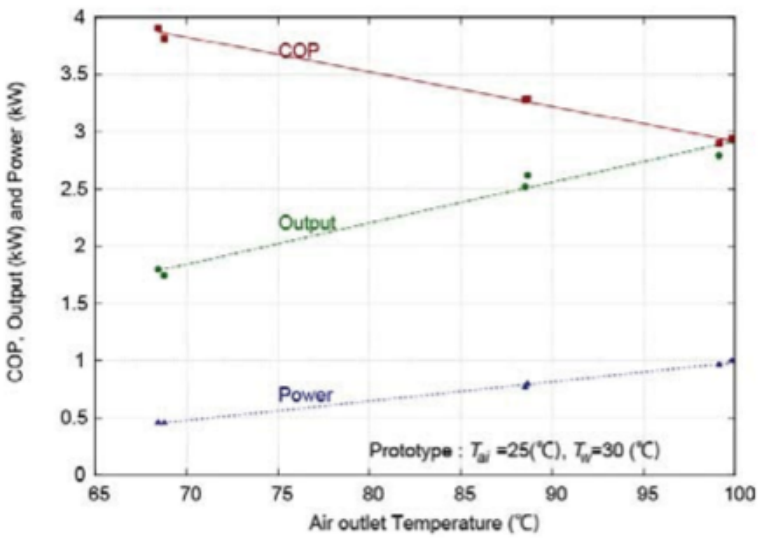

Figure 5: Power, heat output and COP related to air outlet temperature [7]

Visser built a system that can work in both a subcritical operation mode and a transcritical operation mode [8]. For transcritical operation mode he points out that the efficiency of the heat pump is reduced with higher outlet temperature from the gas cooler and higher suction temperature to the compressor. The results are given in Figure 6.

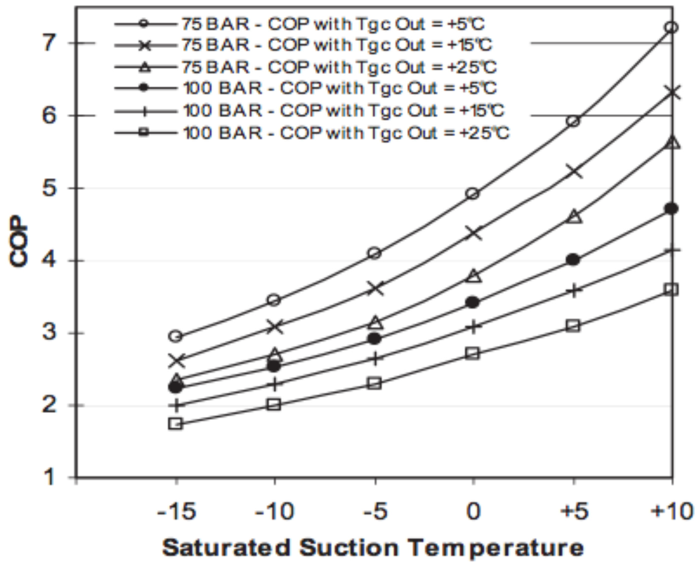

Figure 6: Performance related to GC outlet temp and suction temp. [8]

Montagner and Melo [9] investigated how a light commercial refrigeration system operates with different throttling devices. They found that the COP of the system decreases with higher temperature from the external gas cooler, regardless the kind of throttling devices. This result strengthens the result that Visser got (Figure 6) and is shown in Figure 7.

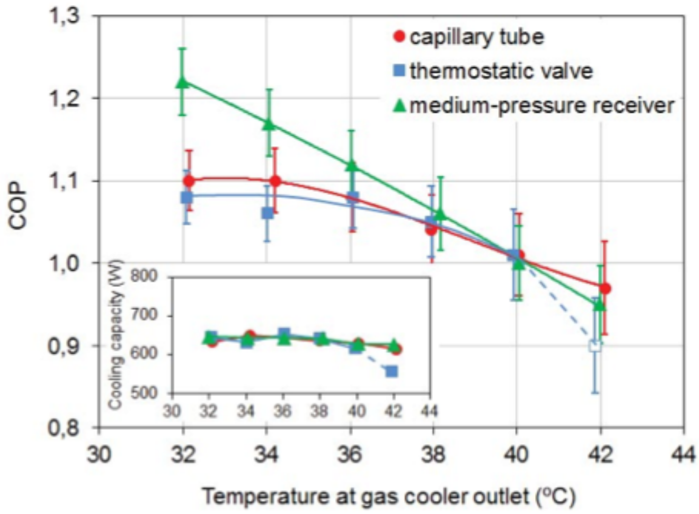

Figure 7: COP as a function of temp. from gas cooler [9]

Novak et al. [10] tested the performance of R134a, R290 (propane) and R744 (CO_2) for use in a tumble dryer. R134a performed best, but R290 performed quite similar. They also investigated the environmental impact from each of the three refrigerants. They took into account the following factors: Leakage rate, lifetime, refrigerant charge, recycling factor, energy consumption and emissions from energy generation. Using these factors they calculated the Total Equivalent Warming Impact (TEWI). The most interesting results are summarised in Table 2. Although R744 has a lower GWP, R290 got a lower TEWI. However, the application area of R290 is limited by its flammability, which is a motivation for further research to increase the efficiency of R744 tumble dryers. Novak et al. indicated that the transcritical cycle design should be further researched and developed in order to improve its efficiency to be able to compete with conventional systems.

Table 2: Energy performance comparison

	R134a	R290	R744
Specific energy consumption [kWh/kg$_{load}$]	0.29	0.30	0.34
GWP	1300	3	1
TEWI	403.6	332.5	377.3

Almeida and Barbosa [11] researched several influence factors of two-stage compression in a transcritical R744-cycle. In addition to support the theory that efficiency increases with lower gas cooler temperature, they compared two-stage compression to single-stage compression. They found that two-stage compression performed significantly better than single-stage compression. Figure 8 shows the difference in COP for different gas cooler pressures. It is worth noticing that they used an intercooler at intermediate pressure. For drying purposes it is desired to have high temperature at the compressor's 2nd outlet. An intercooler at the intermediate pressure stage would reduce the outlet temperature and is therefore not used in the experiments performed in this study.

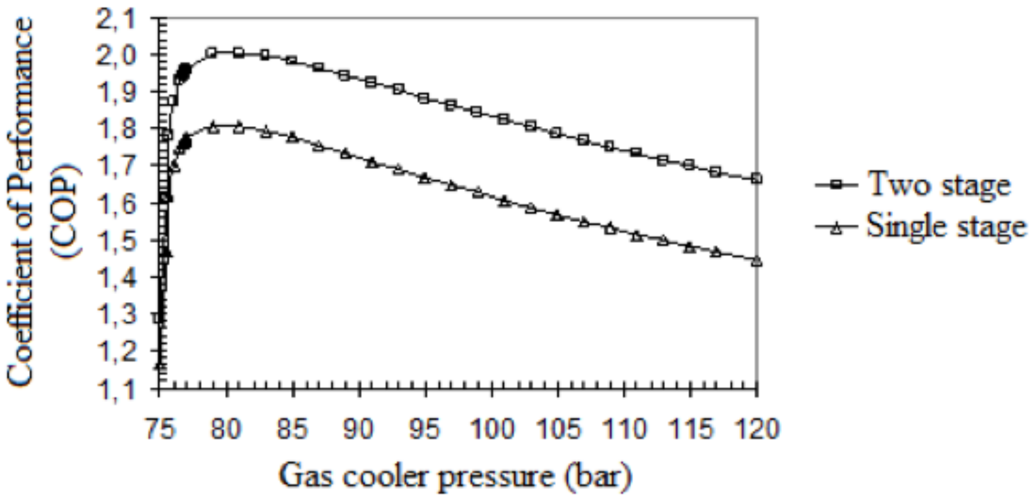

Figure 8: Two-stage vs. single stage compression

Brignoli et al. [12] tested R744 and R410A for use in a residential heat pump system for production of hot tap water and space heating/cooling. They found that the system could achieve a much higher COP for R744 for hot tap water production, but were penalised during both wintertime and summertime when space heating/cooling were required. Overall the energy consumption for the system was 20% higher when R744 was used.

Yoneda et. Al [13] investigated the performance and efficiency of the water heat source CO_2 transcritical heat pump "Eco-Cute" developed by Mayekawa/Myrcom. The heating capacity of the heat pump is 100 kW, which is the largest capacity in the world for heat pumps. The heat pump produces both hot and cold water simultaneously, which makes it very applicable for food industry and other industries with similar demands. The heat pump achieved a combined COP of over 8 with cooling water temperatures of 22°C (in)/17°C (out) and hot water temperatures of 17°C (in)/65 °C (out). The combined COP is based on both heating and cooling capacity as described in (2.1).

$$COP_{combined} = \frac{capacity_{cooling} + capacity_{heating}}{consumption_{electrical}} \tag{2.1}$$

Agrawal et al. [14] state that CO_2 transcritical vapour compression systems are more susceptible with refrigerant charge. Based on this statement they tested how sensitive the efficiency of a heat pump system using CO_2 as refrigerant is to varying refrigerant charge. A capillary tube was used as the throttling device in the system. They found that the COP of the system was reduced significantly more at undercharged conditions than at overcharged conditions.

3. Theory

3.1. The Air Cycle

The air circulates through the drum where heats and absorbs moisture from the clothes. The drying process in the drum is controlled by the temperature and relative humidity of the air. The air is able to absorb more water the higher temperature and the lower relative humidity it has when it enters the drum. The air cycle can be split into three different processes. These processes are shown in the h-x diagram in Figure 9.

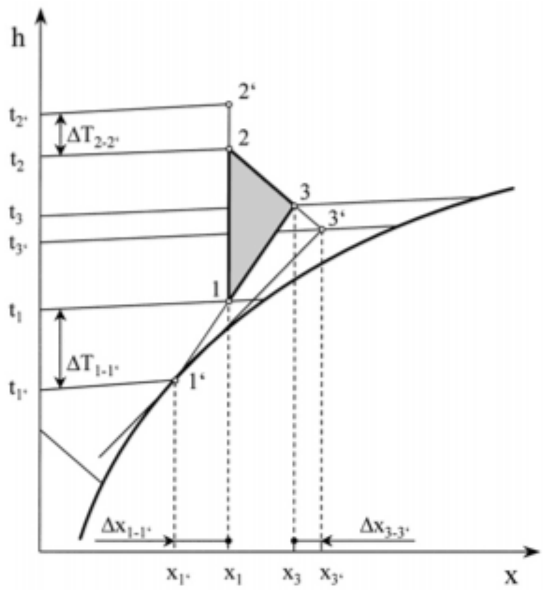

Figure 9: H-x chart of the drying process [15]

The processes are as follows:

1-2: Heating at the gas cooler

2-3: Humidification and cooling in the drum

3-1: Cooling and dehumidification at the evaporator

The points 1', 2' and 3' are included in the figure to show the Carnot process. This is the ideal process. It is not possible to achieve, but it is desirable to come as close as possible to this process.

3.2. Measurements of Performance

For comparison of experiments to be possible, some measurements of performance are essential. The most commonly used factors will be used here, which gives an opportunity to compare this facility to other systems as well.

3.2.1. Heat Pump Capacity

The performance of the system is highly dependent on the capacity of the heat exchangers. Heat is transferred from the refrigerant to the air in the main gas cooler, thus the performance of this heat exchanger is particularly important. The heat transfer rate from the heat exchanger can be calculated using (3.1).

$$\dot{Q}_{GC} = C_{p,CO_2} \cdot \Delta T_{GC,CO_2} \cdot \dot{m}_{CO_2} \tag{3.1}$$

$$\dot{Q}_{Evap} = \Delta h_{CO_2,Evap} \cdot \dot{m}_{CO_2} \tag{3.2}$$

In the gas cooler, all CO_2 is passing through as a gas. This means that the simple equation $\Delta h = C_p \cdot \Delta T$ is applicable. In the evaporator there is a phase transition, and enthalpies are found using Rnlib[1].

3.2.2. Mass Flow Air

The mass flow of air will not be measured directly. However, the heat transferred to the air will be the same as the heat absorbed from the refrigerant in the main gas cooler:

$$\dot{Q}_{air} = \dot{m}_{air} \cdot C_{p,air} \cdot \Delta T_{GC,air} \tag{3.3}$$

$$\dot{Q}_{air} = \dot{Q}_{ref} \tag{3.4}$$

$$\dot{Q}_{ref} = \dot{m}_{CO_2} \cdot C_{p,CO_2} \cdot \Delta T_{GC,CO_2} \tag{3.5}$$

$$\dot{m}_{air} = \dot{m}_{CO_2} \cdot \frac{C_{p,CO_2} \cdot \Delta T_{GC,CO_2}}{C_{p,air} \cdot \Delta T_{GC,air}} \tag{3.6}$$

[1] Rnlib is a computer program developed by NTNU/SINTEF. It is used to find properties of a refrigerant based on its properties and state.

The refrigerant mass flow and the temperature difference over the main gas cooler for both CO_2 and air will be measured. Based on these properties the mass flow of air can be calculated using equation (3.6).

3.2.3. Power Consumption

Both the heat transferred to the air as well as the air mass flow rate is known, but in order to say anything about the performance it is essential to know how much power is fed into the heat pump system. The compressor and the fan have a rated power consumption of 750W and 26W respectively. The power consumption of the motor, compressor and pump will be measured throughout the experiment. In the results the power consumption of the motor and the pump is merged and referred to as "P_motor".

$$P_{tot} = P_{comp} + P_{fan} + P_{motor} + P_{pump} \tag{3.7}$$

3.2.4. Coefficient of Performance

The coefficient of performance (COP) is a measure of the efficiency of the heat pump. It is defined as the ratio between the heat output rate from the main gas cooler and the total power consumption of the heat pump. The heat output rate from the main gas cooler can be found using equation (3.8).

$$COP = \frac{\dot{Q}_{GC}}{P_{tot}} \tag{3.8}$$

The COP is a common unit of measure for heat pumps. A typical COP-value for a conventional electrical driven heat pump for heating purposes is between 2 and 5 [16]. For purposes like the drum dryer where the cooling capacity is utilised in addition to the heating capacity, the heat absorption rate in the evaporator could be included to form a combined COP. For the purpose of comparing the COP to former students' results, COP is based onle on gas cooler performance.

3.2.5. Water Extraction

The main purpose of a cloth dryer is to extract water from the clothes. It is therefore of interest to express the dryer's performance in terms of water extraction. To do this, it is essential to know how much energy is required to remove a certain amount of water from the clothes. This relation is called the dh/dx relation and is expressed in equation (3.9).

$$\frac{\Delta h}{\Delta x} = \frac{Q_{evaporator}}{\Delta x_{evaporator}} \tag{3.9}$$

The amount of condensed water will be measured during the experiment at fixed intervals. The heat transferred can be found by measuring the temperature difference of the refrigerant over the evaporator and use the following equation:

$$Q_{evaporator} = C_{p,CO2} \cdot \Delta T_{CO2} \tag{3.10}$$

3.2.5.1. Moisture Extraction Rate

The moisture extraction rate (MER) is a measure of how much moisture is extracted per time. The factor is found using the dh/dx relation and the total power consumption.

$$MER = \frac{P_{tot}}{\Delta h / \Delta x} \left[\frac{g}{s} \right] \tag{3.11}$$

3.2.5.2. Specific Moisture Extraction Rate

The specific moisture extraction rate (SMER) is a measure of how much moisture the system is able to extract for each kJ of input energy.

$$SMER = \frac{COP}{\Delta h / \Delta x} \left[\frac{kg}{kJ} \right] \tag{3.12}$$

3.2.6. Specific Energy Consumption

The specific energy consumption (SEC) is a measure of how much energy is consumed to dry a certain amount of textiles.

$$SEC_{textiles} = \frac{E_{tot}}{Weight_{dry,textiles}} \left[\frac{kWh}{kg_{dry,textiles}} \right] \tag{3.13}$$

A variant of the SEC is a measurement of how much energy is consumed per kg of removed water. This is very useful for experiment series where the moisture content varies between the experiments.

$$SEC_{water} = \frac{E_{tot}}{Weight_{removed,water}} \left[\frac{kWh}{kg_{removed,water}} \right] \tag{3.14}$$

The weight of extracted water does not equal the weight difference between textiles before and after the experiment. It is therefore reason to believe that a portion of the moisture leaves the drum as water vapour. The weight of removed water will be calculated as in equation (3.15).

$$Weight_{water,removed} = Weight_{wet,load} - Weight_{dry,after} \tag{3.15}$$

3.3. Energy Balance Calculation

In some experiments there is a strong suspicion that liquid CO_2 enters the compressor at the intermediate pressure stage. By performing an energy balance calculation for the flows that enters the 2nd suction side of the compressor this phenomenon can be proven. Point 2b in Figure 10 is in the two-phase area, but the liquid fraction and enthalpy is not known. A mass balance of the mass flows in point 2b will prove if any liquid enters the compressor or if it is saturated gas.

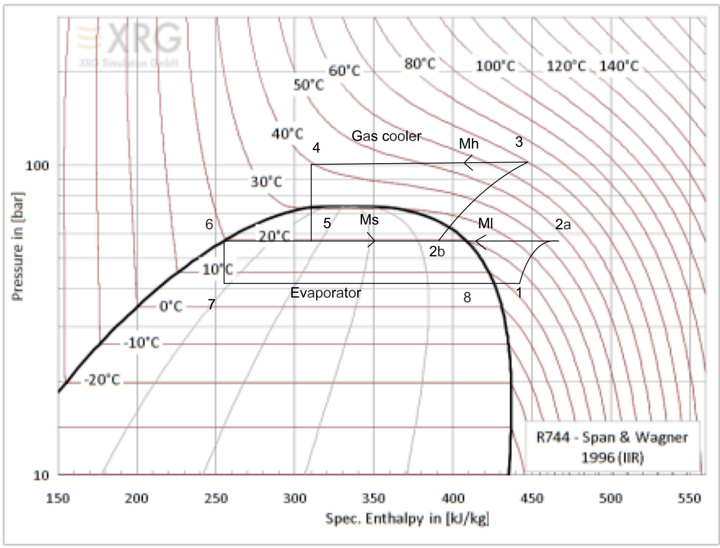

Figure 10: Process diagram: Liquid enters the compressor

The mass flow in the system is measured on the high-pressure side. Hence M_h is known. M_h is the sum of the circulating mass and the mass flow at the low-pressure side, M_s and M_l respectively. Conservation of mass gives the following equation:

$$\dot{m}_s = \dot{m}_h - \dot{m}_l \tag{3.16}$$

Energy entering point 2b equals energy leaving point 2b:

$$\dot{m}_h \cdot h_{2b} = \dot{m}_l \cdot h_{2a} + \dot{m}_s \cdot h_5 \tag{3.17}$$

This leaves us with two equations and three unknowns. Hence one more relation is required to solve the problem. The mass flow of the air is identical over both the gas cooler and the evaporator and this fact will be used to find the low-pressure side mass flow M_l. Ideally, the enthalpy of air should take into account the moisture content in the air as well as the temperature difference. A sample calculation shows that the enthalpy difference between moist air and dry air is 2.67%. Because this calculation is performed mainly to prove whether liquid enters the compressor or not, a deviation of 2.67 % is considered acceptable. Thus the following calculations will use the simple form $dh = Cp \cdot \Delta T$.

$$\dot{Q}_{GC} = \dot{m}_{air} \cdot Cp_{air} \cdot \Delta T_{air,GC} = \dot{m}_h \cdot Cp_{CO_2} \cdot \Delta T_{CO_2,GC} \tag{3.18}$$

$$\dot{m}_{air} = \dot{m}_h \cdot \frac{Cp_{CO_2} \cdot \Delta T_{CO_2,GC}}{Cp_{air} \cdot \Delta T_{air,GC}} \tag{3.19}$$

The relation for mass flow air can be used to express the performance of the evaporator.

$$\dot{Q}_{evap} = \dot{m}_l \cdot (h_1 - h_7) = \dot{m}_{air} \cdot Cp_{air} \cdot \Delta T_{air,evap} \tag{3.20}$$

$$\dot{m}_l = \frac{Cp_{CO_2} \cdot \Delta T_{CO_2,GC} \cdot \dot{m}_h \cdot \Delta T_{air,evap}}{(h_1 - h_7) \cdot \Delta T_{air,GC}} \tag{3.21}$$

With this relation for M_l and the relation (3.16) for M_s, (3.17) can be solved with respect to h_{2b}:

$$h_{2b} = \frac{\dot{m}_l \cdot h_{2a} + \dot{m}_s \cdot h_5}{\dot{m}_h} \tag{3.22}$$

4. System

The system is set up to utilise the design of the heat pump in the best possible way. Both the hot side and the cold side of the heat pump will be used to process the circulating air. This differs from common heat pumps for space heating where the evaporator extracts heat from outside air without utilising the cooling effect this process generates. Figure 11 shows how the air cycle is combined with heat pump cycle. An overview of the facility is included in Figure 12.

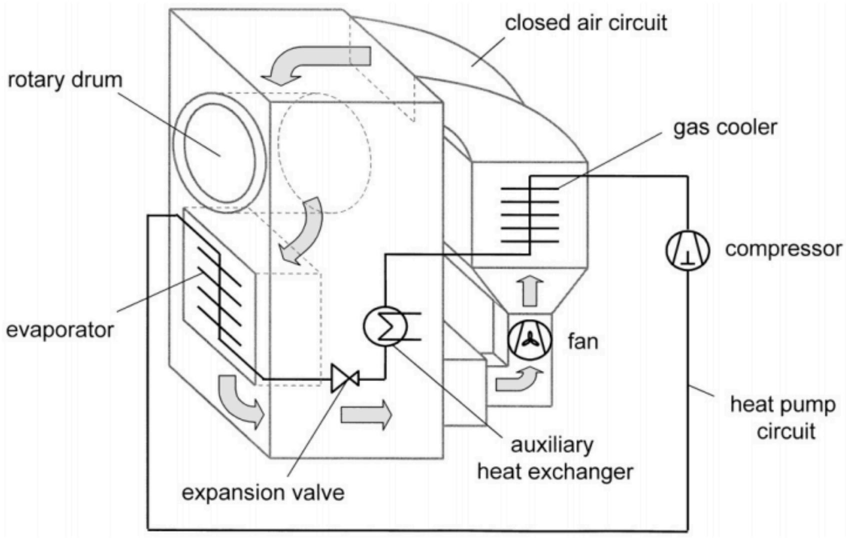

Figure 11: CO2 heat pump dryer process

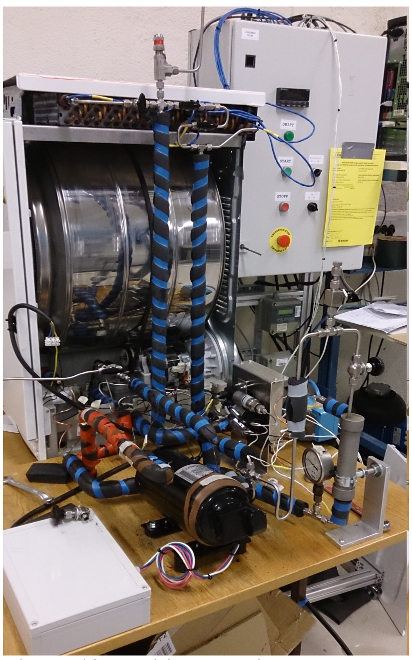

Figure 12: Facility overview

4.1. Compressor

The compressor is a hermetic Sanyo 80474035T two-stage compressor. The properties of the compressor are given in Table 3. A picture of the installed compressor is included in Figure 13.

Table 3: Compressor properties

Name	**Sanyo 80474035T**
Rated output	750 W
Speed	2850 rpm
1st stage displacement volume	2.40 cm^3
2nd stage displacement volume	1.56 cm^3
Isentropic efficiency	0.72 [17]
Volumetric efficiency	0.85 [17]

Figure 13: Two-stage compressor

4.2. Capillary Tubes

Capillary tubes are used as throttling devices in the system. This enables the possibility to extract the flash gas at the intermediate pressure stage. The calculation method developed by Madsen et al. [18] has been used to calculate the capillary tube lengths. After the first few experiments, a longer tube substituted the capillary tube between the intermediate pressure stage and the low-pressure stage. The process did not work out as planned from the start and the capillary tube between the separation tank and the evaporator was enlarged to reduce the pressure levels. The capillary tubes are made by copper and have an internal diameter of 0.9 mm.

Table 4: Length capillary tubes

Position	Length [mm]
High pressure – intermediate pressure	560.7 mm
Intermediate pressure – low pressure, version 1	327.5 mm
Intermediate pressure – low pressure, version 2	400.0 mm

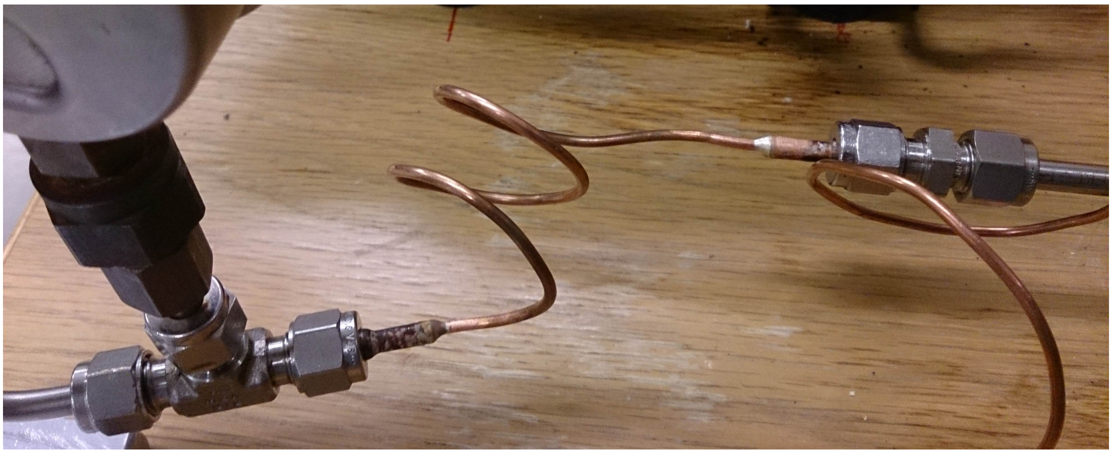

Figure 14: Capillary tube

4.3. Separation Tank

The gas from the external gas cooler will be throttled into the two-phase region where a fraction of the flow will appear as gas while the rest will be liquid. This gas is called "flash gas". The flash gas will be injected into the compressor at intermediate pressure without being throttled and sent through the evaporator. From Figure 15 it can be seen that in point 5 the mass flow is split. The flash gas is injected into the compressor in point 2 while the saturated liquid is throttled from point 6 to point 7.

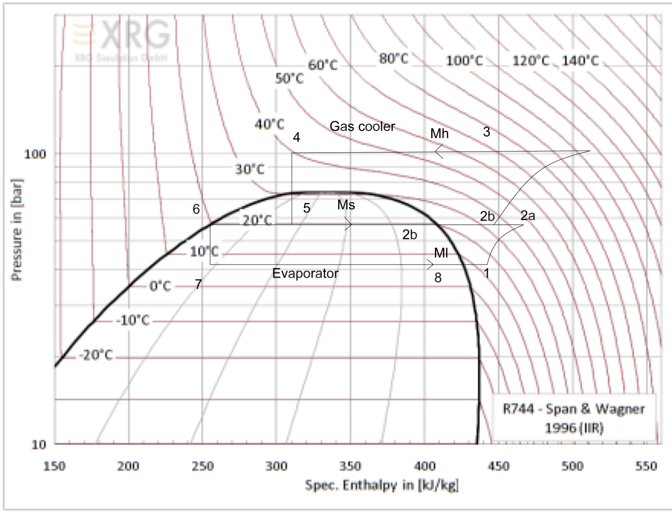

Figure 15: Theoretical p-h diagram for the process

It is beneficial to split the refrigerant flow into saturated liquid and saturated gas. The work to compress the saturated gas from the evaporating pressure to the intermediate pressure is saved, and the liquid absorbs approximately the same amount of heat because the same amount of liquid evaporates in the evaporator. Refrigerant that enters the evaporator in gas form does not contribute very much to heat absorption. It is already evaporated and will only absorb heat from temperature increase. This is called superheat. The flow enters the separation tank in an angle to the inner walls and at a distance from the separation tank outlet. The idea is that this will improve the separation process and prevent liquid from entering the compressor. Details of the tank are shown in Figure 16, Figure 17 and Figure 18.

Figure 16: Internal view of the separation tank

Figure 17: Side view of separation tank

Figure 18: Separation tank after installation

4.4. Heat Exchangers

There are three heat exchangers present in the system. On the high-pressure side there is a gas cooler and an external gas cooler. The gas cooler transfers heat from the refrigerant to the circulating air. The external gas cooler further reduces the temperature of the refrigerant, bringing it further to the left in the ph-diagram (see Figure 15). When the refrigerant is throttled, the liquid fraction will be larger due to the extra cooling. Thus the capacity of the evaporator at the low-pressure side increases.

4.5. Fan

A fan is used to blow ambient air through the external gas cooler to cool the refrigerant. The fans are of the model 8212 JH3 from the producer Ebm-papst. The fans consume 26 W each and will be switched on every time the temperature of the refrigerant exceeds 40°C at the outlet of the external gas cooler.

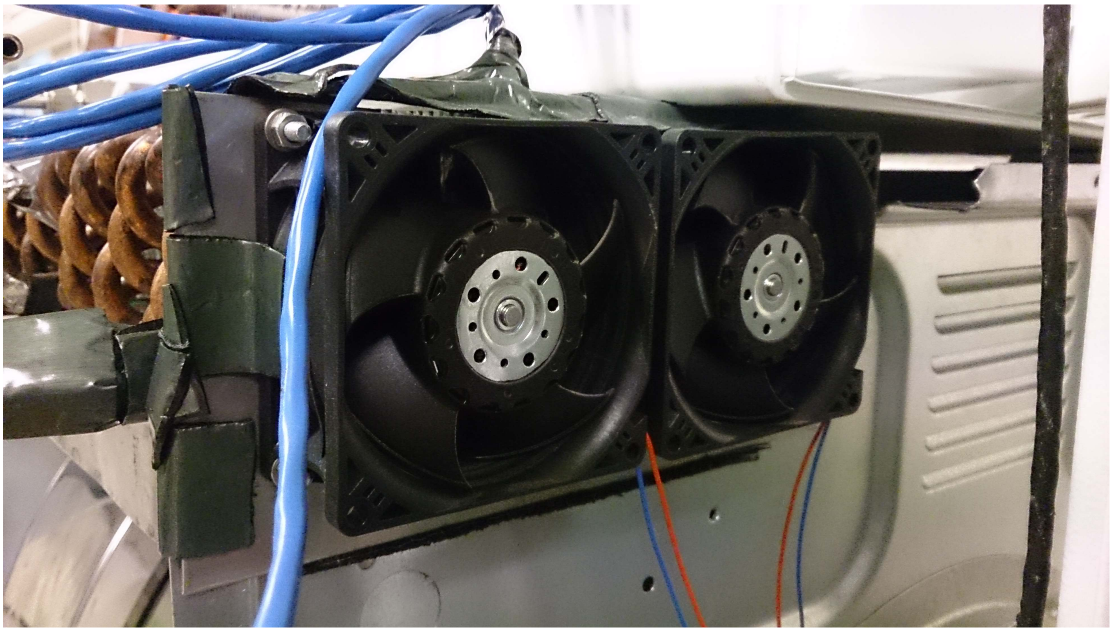

Figure 19: Ebm-papst 8212 JH3 fans

4.6. Tubes and Fittings

The system consists of a mix of 316 stainless steel and copper tubes. The heat exchangers consist of copper tubes. Copper is often preferred in heat exchangers because of its high thermal conductivity. The copper tubes that are connected to the main gas cooler and the evaporator are kept as they are. It would require very much time and resources to replace them because the internal heat exchangers are not easily accessible. The rest of the system use stainless steel tubes. The inner diameter of the stainless steel and copper tubes are calculated based on outer diameter and wall thickness as shown in (4.1).

$$d_i = d_o - 2\delta_w \tag{4.1}$$

The dimensions for each of the tube types are given in Table 5. The tubes are connected by Swagelok fittings and in some cases brazing. As few fittings and transition valves as possible are used in order to prevent leakages in the system.

Table 5: Tube dimensions

	316 Stainless Steel	Copper
Outer diameter	6 mm	¼ inch (6.35 mm)
Wall thickness	1 mm	1 mm
Inner diameter	4 mm	4.35 mm

5. Modifications During Test Period

As soon as the system was built, the dryer was tested. The first few tests were interrupted after a short while because of overheating of the compressor. The compressor was fully insulated from the start and removing this insulation fixed the problem. In later experiments overheating was still an issue, but shutdown was avoided by external cooling with a wet cloth.

During the following tests the system ran smoothly, but the textiles did not dry at all. It turned out that the internal heat exchangers were placed in the opposite order of what was assumed from the start. The circulating air was heated and cooled without utilising either the cooling effect for dehumidification or the heating effect for heating dry air. A great deal of work has previously been done to seal the gas cooler and the evaporator from the rest of the system and an inspection of this part would take a lot of time to execute. The tubes were redirected to utilise the internal heat exchangers properly, but still the temperatures inside the drum were to low. A picture from previous rebuilding shows that the air heater runs with parallel flow rather than counter-current flow. This picture is included as Figure 20. This setup causes the drum inlet temperatures to be too low. The inlet and outlet tubes to the air heater were switched to obtain a counter-current flow through the heat exchangers.

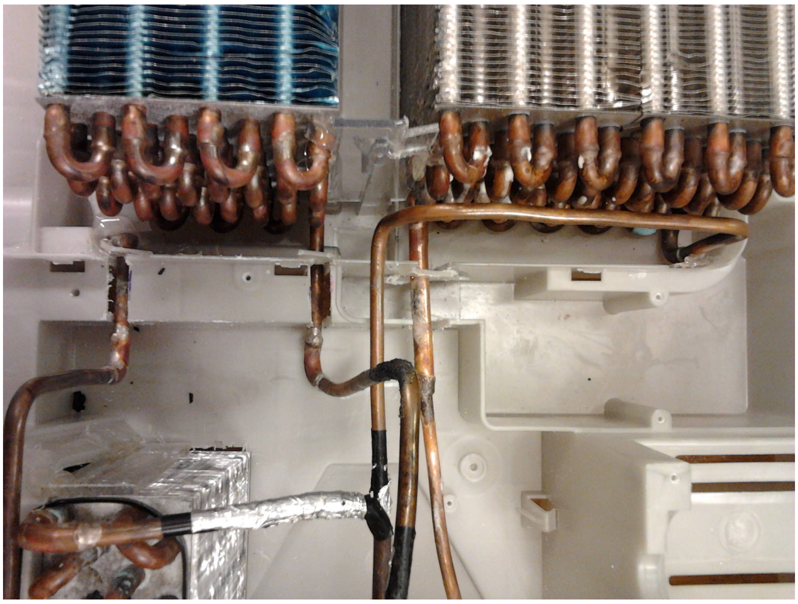

Figure 20: Internal heat exchangers setup

After doing these changes the clothes were properly dried. However, a lot of energy was consumed in the process. An analysis of the temperature levels at the intermediate stage led to the conclusion that the flash gas injection process runs in the opposite direction of what was planned. The pressure at the first stage compressor outlet turned out to be higher than the pressure in the separation tank, thus the flow is forced into the separation tank rather than the other way around. It was decided to increase the length of the capillary tube connecting the low-pressure stage to the intermediate pressure stage from 32.7 cm to 40.0 cm. The idea is that this move will reduce the pressure levels produced by the compressor and force the flow in the correct direction. This move worked, but a lower mass flow at the low-pressure stage caused a high superheat from the evaporator. An adjustable throttling valve replaced the capillary tube in order to adjust the evaporator inlet pressure and temperature.

6. Refrigerant Charge

A lot of work has previously been done to calculate the optimal amount of refrigerant. The initial test showed that this amount was not sufficient, and a more practical "trial and error"-approach proved to be easier and less time-consuming. The facility is first filled until saturation pressure for the ambient temperature is reached. This is reached when no more refrigerant flows from the gas bottle. There is equilibrium between gas and liquid in the bottle, and the facility is only filled with gas. Additional filling is executed by running the system. This produces a low pressure and more refrigerant will flow into the system. The efficiency of the system is highly dependent of the refrigerant charge, and the desired amount of refrigerant is determined by mass flow, pressure levels and evaporator superheat.

7. Lubrication

The compressor is delivered with 350 cm^3 of lubrication oil. The oil type is DAPHNE PZ68S double-end capped PAG oil. A double-end capped oil has lower hygroscopicity than a single-end capped lubrication oil [19]. Hygroscopicity is a measure of the oil's ability to absorb moisture from ambient air. Moisture dilutes the lubricant and changes its properties, which is undesired. Polyolester (POE) oils are widely used by compressor manufacturers, and were used in the system before substituting the compressor. PAG oils are less miscible with CO_2 than POE oils [20], which is a challenge concerning oil return to the compressor. To ensure that the oil returns to the compressor, the separation tank is filled with about 75 cm^3 of extra oil. This will give the oil a head start to reach the compressor before the compressor runs dry.

Table 6: Properties lubrication oil

Property	PZ68S
Viscosity @ 40°C [mm^2/s]	68.57
Viscosity @ 100°C [mm^2/s]	14.04
Water content [wt%]	214
Density @ 15°C [g/cm^3]	0.9973
Flash point [°C]	238

8. Instrumentation

8.1. Pressure Sensors

Three pressure sensors are installed in the system. One sensor is installed at the low-pressure side of the compressor, one at the high-pressure side and one at the intermediate pressure stage. They are all located close to the compressor. The sensors are of the type Druck PTX 1400. The operating temperatures for these sensors ranges from -20°C to 80°C with accuracy of ±0.15% typical and ± 0.25% maximum. The pressure sensors are included in Figure 21.

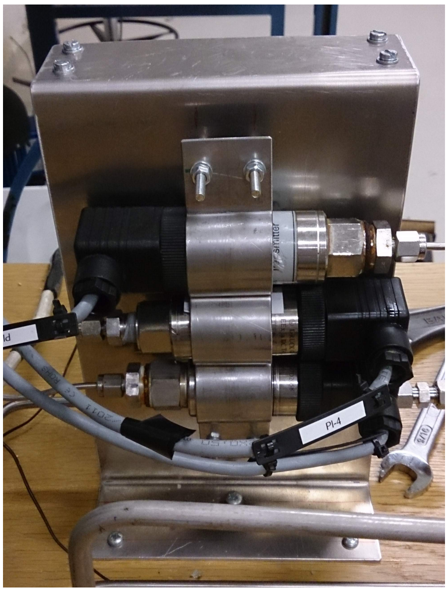

Figure 21: Pressure sensors

8.2. Mass Flow Meter

A Rheonik RHM 03 was used to measure the mass flow in the system. This meter has a typical measuring range of 0.05 kg/min to 5 kg/min, however it can measure flow rates as low as 0.0375 kg/min. The accuracy is 0.10%. It is placed after the external heat exchanger at the high-pressure stage. The mass flow meter is included in Figure 22.

Figure 22: Rheonik RHM 03 mass flow meter

8.3. Temperature Sensors

Temperature sensors are placed on the outer surface of the tubes. To achieve measurements of as high accuracy as possible, the thermocouples have been well insulated with electrical tape to avoid electrical disturbance, aluminium tape for thermal insulation and general insulation that covers all tubes in the system. The thermocouples are connected to the Thermocouple Input Module NI 9211 from National Instruments in the instrumentation locker shown in Figure 23.

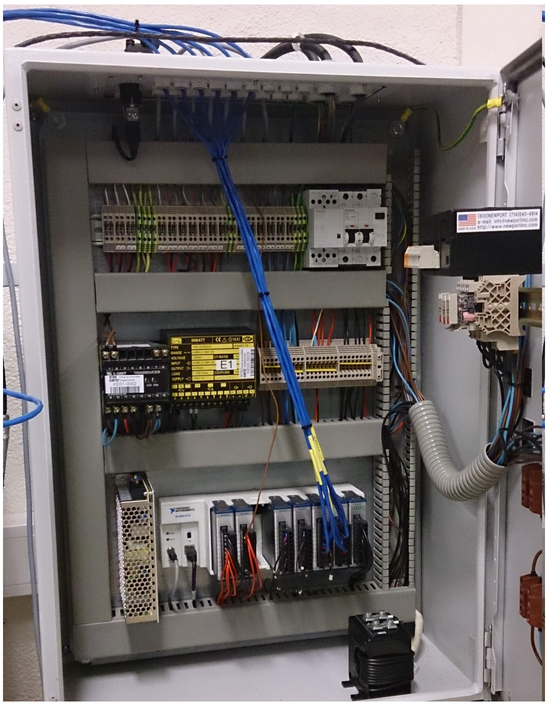

Figure 23: Instrumentation locker

8.4. Humidity Sensors

There are installed two humidity sensors in the system. One sensor is placed at the inlet of the drum and one is placed at the outlet of the drum. The sensors are of the type Vaisala HMP 235. The data from these sensors will be used to calculate the SMER and to determine when the drying process is finished.

The locations of the measuring point are shown in Figure 24. Specifications about each measuring point is summarised in Table 7.

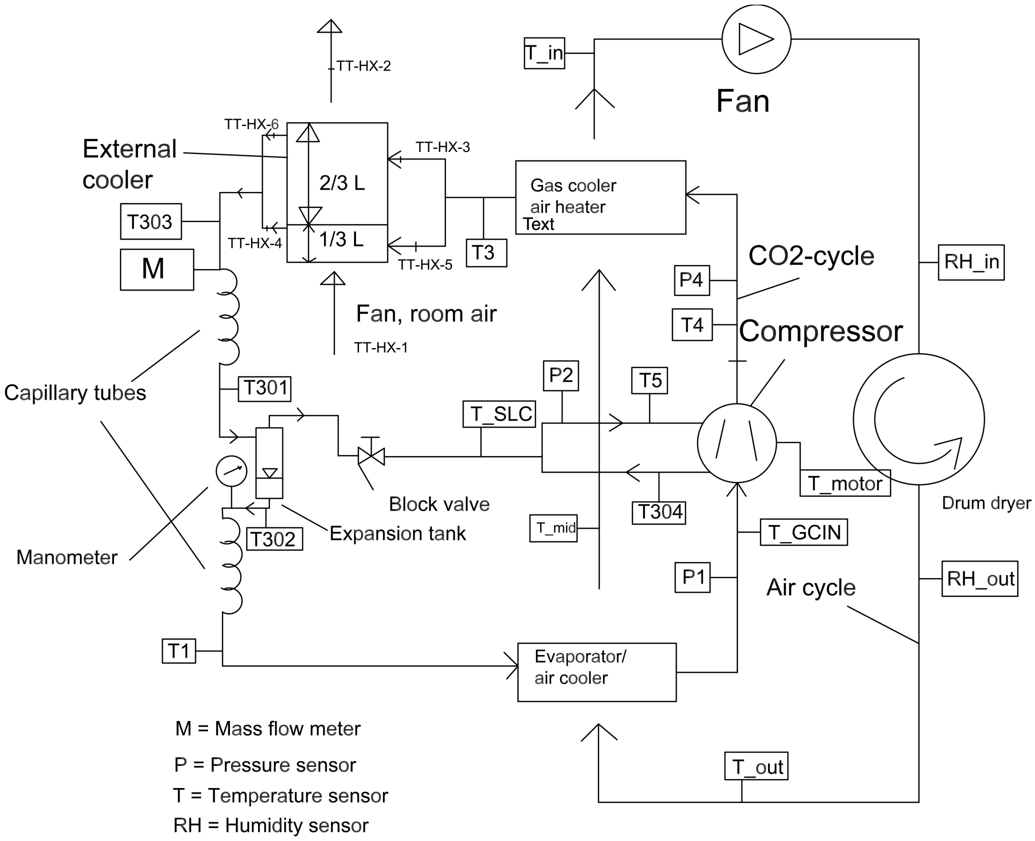

Figure 24: Process chart including measurement points

Table 7: Measuring points specifications

Displayed text	Description	Output signal	Manufacturer	Model
M	Coriolis mass flow meter	4-20 mA	Rheonik	RHM 03
RH_in	Humidity sensor	4-20 mA	Vaisala	HMP 235
RH_out	Humidity sensor	4-20 mA	Vaisala	HMP 235
P1	Pressure sensor	4-20 mA	Druck	PTX 1400
P2	Pressure sensor	4-20 mA	Druck	PTX 1400
P4	Pressure sensor	4-20 mA	Druck	PTX 1400
T1	Thermocouple	4-20 mA	National Instruments	9211
T_GCIN	Thermocouple	4-20 mA	National Instruments	9211
T3	Thermocouple	4-20 mA	National Instruments	9211
T4	Thermocouple	4-20 mA	National Instruments	9211
T5	Thermocouple	4-20 mA	National Instruments	9211
T_out	Thermocouple	4-20 mA	National Instruments	9211
T_in	Thermocouple	4-20 mA	National Instruments	9211
T_SLC	Thermocouple	4-20 mA	National Instruments	9211
T_mid	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-1	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-2	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-3	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-4	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-5	Thermocouple	4-20 mA	National Instruments	9211
TT-HX-6	Thermocouple	4-20 mA	National Instruments	9211
T-301	Thermocouple	4-20 mA	National Instruments	9211
T-302	Thermocouple	4-20 mA	National Instruments	9211
T-303	Thermocouple	4-20 mA	National Instruments	9211
T-304	Thermocouple	4-20 mA	National Instruments	9211

8.4.1. Calibration of humidity sensors

The humidity sensors provided unrealistic values during experiments, so the sensors were calibrated with an exact reference. The reference consisted of a salt solution that creates an environment with a known humidity when it is in equilibrium with air. The humidity sensor responded with a current, which was used to make a new calibration curve. Three points were measured and the curve was implemented in Labview. This calibration curve is later used to "translate" the electrical current to the correct humidity value. The salt solutions and the humidity they create are given in Table 8. The salt solutions are pictured in Figure 25 and the calibrations measurements were taken as showed in Figure 26.

Table 8: Salt solutions for calibration

Salt solution	Concentration in H_2O	Humidity [%]
LiCl	13.41 molar	25 ± 0.3
LiCl	8.57 molar	50 ± 0.3
NaCl	6.0 molar	76 ± 0.3

Figure 25: Salt solutions

Figure 26: Humidity calibration setup

The humidity sensor referred to as "RH_in" still showed a high value after calibration and had a very slow response. However, the sensor sensed correct values without the cover around it. The cover was cleaned with boiling water and rinsed with pressurised air. After this procedure the sensor's response time improved greatly. Figure 27 shows the position of "RH_out".

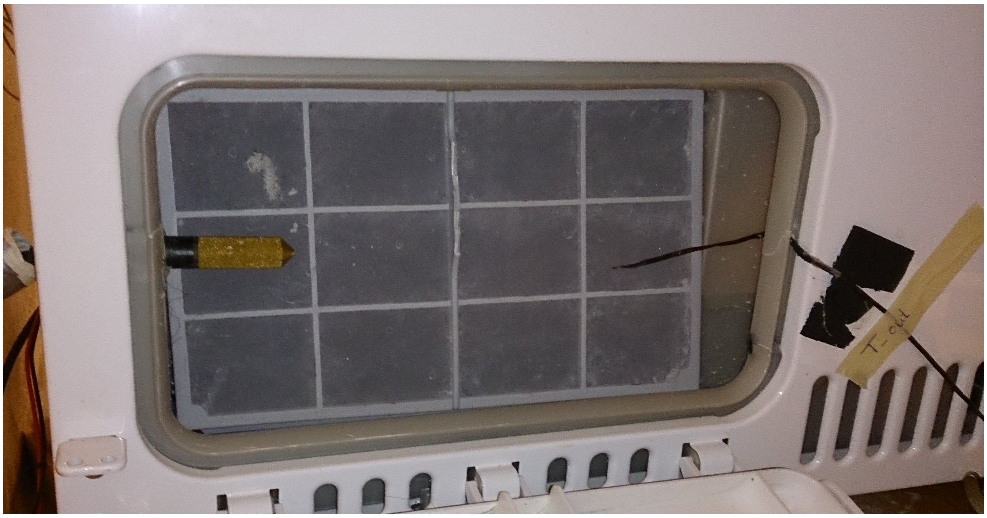

Figure 27: Humidity sensor and temperature sensor at the outlet of the drum dryer

9. Procedure for Preparation of Experiment

In order to be able to compare the results from the experiments to each other as well as to results from drum dryers currently on the market, the experiments will follow a standard procedure:

- The system will be in equilibrium with environment before start-up of the facility.
  - Temperatures in tubes and components are equal to the surrounding temperature.
  - Pressure levels are equal and stable within the system.
  - The humidity in the air cycle has stabilised.

- The textiles are in equilibrium with environment before weighed and moistened.
  - They will be kept in the same room as the drum dryer to reach the same temperature and moisture content as surrounding air.

- The filter in the tumble dryer is vacuumed to assure that the flow rate of the circulating air is equal in each experiment.

- The same amount of textiles (same initial weight) will be used for each experiment.

- Textiles are carefully weighed whilst inside a plastic box before moistened. The weight of the box is subtracted from the total weight.

- Textiles are moistened and centrifuged at 1600 rpm, water is added until desired weight is achieved.

- Textiles are weighed inside the same plastic box. The textiles will contain water corresponding to 60% of the textile's weight before being placed in the drum dryer. This is in accordance with the international standard for tumble dryers for household use [21].

- The dryer will be stopped when all the water is extracted from the textiles. Thus the weight of the textiles should be equal to the initial weight before moistened.

10. Experiment Results

10.1. Comparable results

Experiments have been executed on the rig before making the modifications described earlier in this study. The main results from these experiments will be compared to the results from the modified setup to evaluate the effect of the modifications. Results from the work by Åsmund Elnan [22] and Johannes Single [23] will be considered.

10.1.1. Å. Elnan

Elnan converted the facility from using R134a to CO_2 as the working fluid. He investigated the effect of refrigerant charge and capillary tube length in order to minimise the energy consumption. He achieved some good results, the best being 0.534 $kWh/kg_{removed\ water}$ corresponding to 0.27 $kWh/kg_{textiles}$. The moisture content of the loaded textiles he used varied between 50-60%, whereas 60% is used for all experiments in this study. The results should still be comparable, if anything Elnan's specific energy consumption should be slightly higher. He did one experiment using R134a and consumed 0.27 $kWh/kg_{textiles}$. This is slightly better than the test results from ASKO of 0.31 $kWh/kg_{textiles}$. ASKO's results are given in Appendix B.

10.1.2. J. Single

Single did more experiments on the existing rig and aimed to reduce the superheat out of the evaporator and increase energy efficiency. He did this by investigating the effect of simulating different lengths of capillary tubes, varying the refrigerant charge and varying the use of a small external heat exchanger. He also designed a larger external heat exchanger, but it did not get implemented in time to be included in the experiments. His best result was 0.56 $kWh/kg_{removed\ water}$ and 0.32 $kWh/kg_{textiles}$.

10.2. Experiments overview

- Experiment 1: Original capillary tube lengths, closed valve at intermediate pressure
- Experiment 2: Original capillary tube lengths, open valve at intermediate pressure
- Experiment 3: Longer capillary tube to low pressure stage, open valve, high refrigerant charge
- Experiment 4: Longer capillary tube to low pressure stage, closed valve, high refrigerant charge
- Experiment 5: Longer capillary tube to low pressure stage, variable valve, 780 g refrigerant charge
- Experiment 6: Longer capillary tube to low pressure stage, open valve, high refrigerant charge (880 g)
- Experiment 7: Longer capillary tube to low pressure stage, throttle at intermediate pressure, 880 g refrigerant charge
- Experiment 8: Longer capillary tube, closed valve at intermediate pressure, increasing refrigerant charge
- Experiment 9: Longer capillary tube, closed valve at intermediate pressure, high refrigerant charge
- Experiment 10: Hand-adjusted throttle valve to evaporator pressure level, closed valve at intermediate pressure, 1110 g refrigerant.
- Experiment 11: Hand-adjusted throttle valve to evaporator pressure level, open valve at intermediate pressure, 1110 g refrigerant.
- Experiment 12: Hand-adjusted throttle valve to evaporator pressure level, closed valve at intermediate pressure, 1160 g refrigerant.

Experiment 1: Original capillary tube lengths, closed valve at intermediate pressure

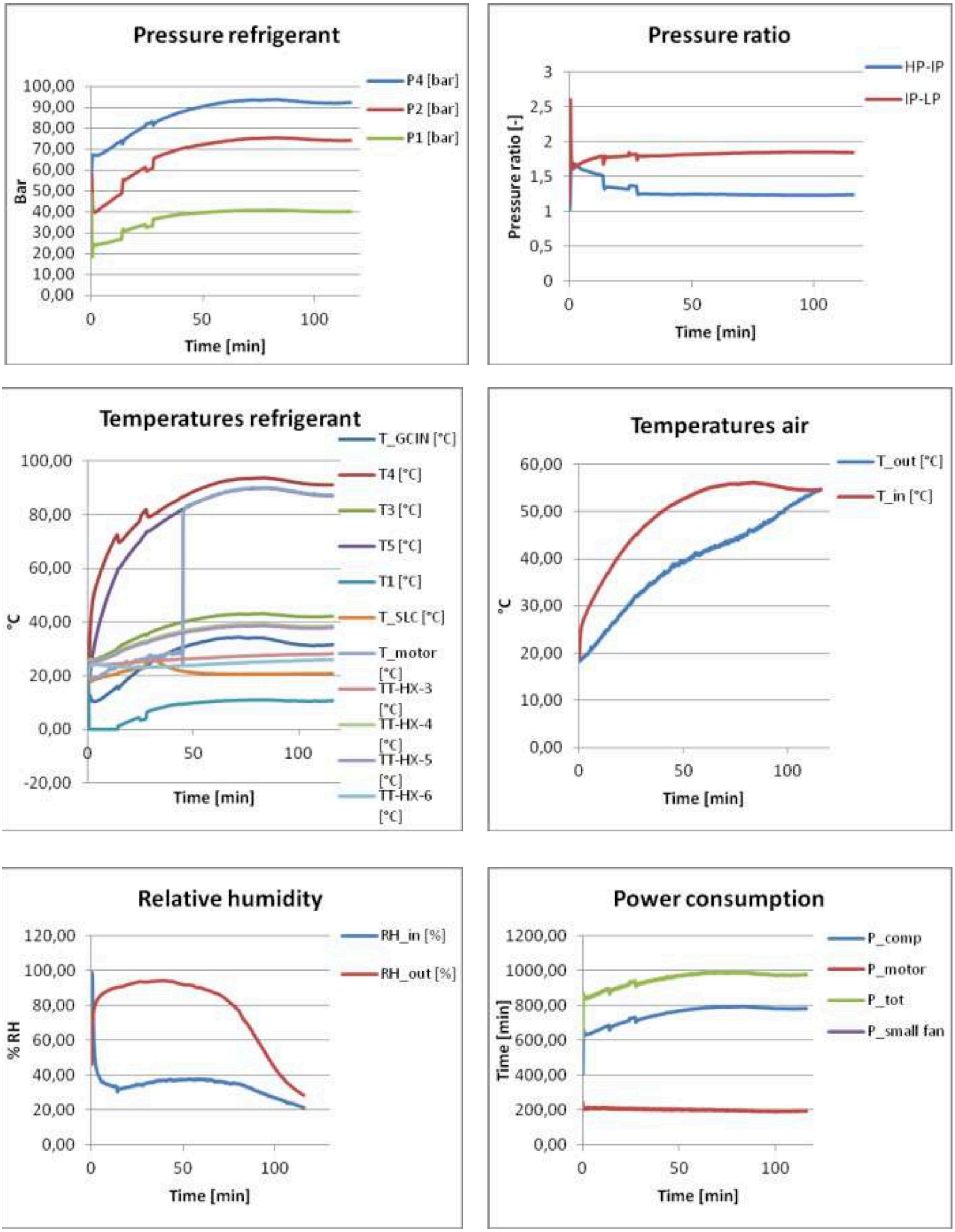

Figure 28: Results experiment 1

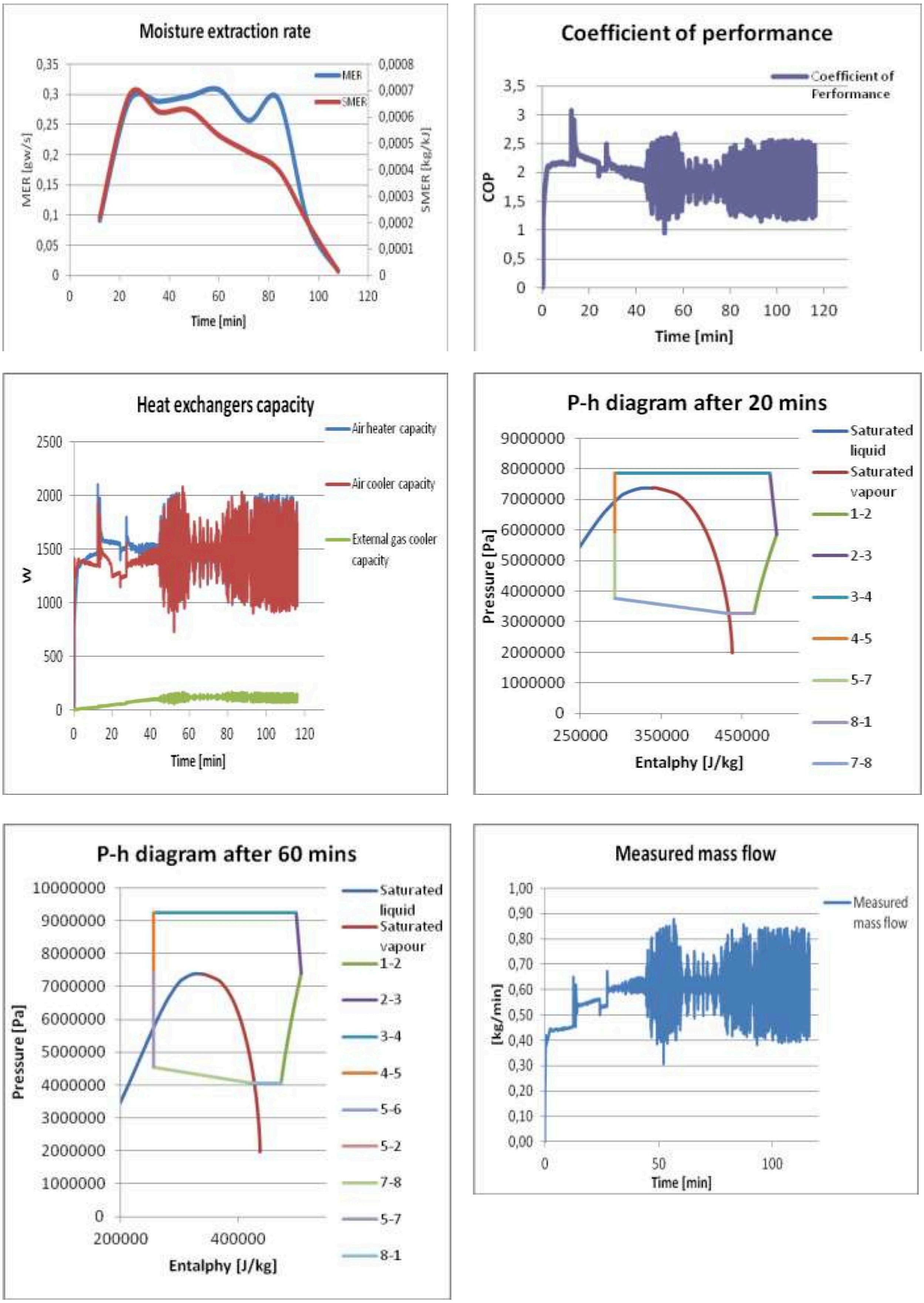

Figure 29: Results experiment 1

Table 9: Other results experiment 1

Other results		
Energy consumption:	1,84	kWh
Consumption per kg textile	0,53	$kWh/kg_{textile}$
Air leakage	6,31	%
Average COP	1,95	
Energy savings	0,49	
Average water extraction	0,018	kg/min
Moisture content end	-0,87	%
Standstill pressure before run	53	bara
Weight dry textiles	3,46	kg
Weight incl. 60% moisture	5,52	kg
Weight after experiment	3,43	kg
Mass flow high-pressure side	0,0098	kg/s average

The energy consumption is quite high for such a small load of textiles. 0.53 kWh/kg textile is way higher than any of the comparable results (see Table 23). However, the COP-value is very good. The negative moisture content in the textiles after the experiment indicates that the experiment has been running for too long. The moisture present in the clothes toward the end of the cycle is very energy consuming to remove, thus ending the experiment earlier would definitely reduce the specific energy consumption. The superheat in the evaporator is equivalent to the difference between T1 and T_GCIN. This value is constant about 15 K, which is a bit much. A superheat between 5 and 10 K is desired.

The 2nd compression stage in the ph-diagrams was expected to curve to the right rather than to the left. The reason for this is unknown, one theory is that the thermo element is located too far from the compressor and the gas may be cooled down before reaching the thermo element. The tubes are well insulated but the measuring point will still be moved in order to fix this issue. The mass flow is constantly fluctuating after the first 30 minutes. This is very strange and impacts the H-EX capacity- and the COP-curves directly.

Experiment 2: Original capillary tube lengths, open valve at intermediate pressure

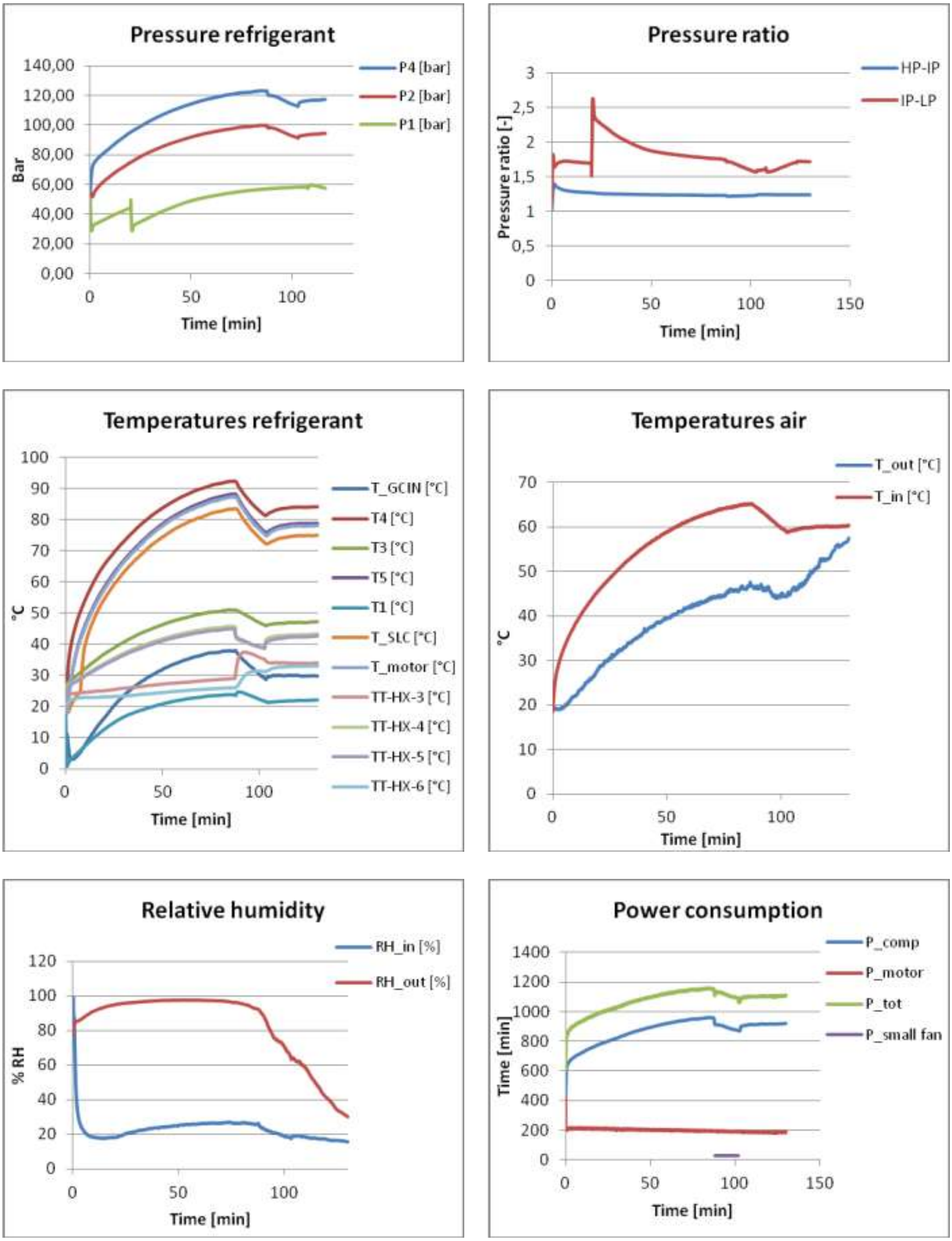

Figure 30: Results experiment 2

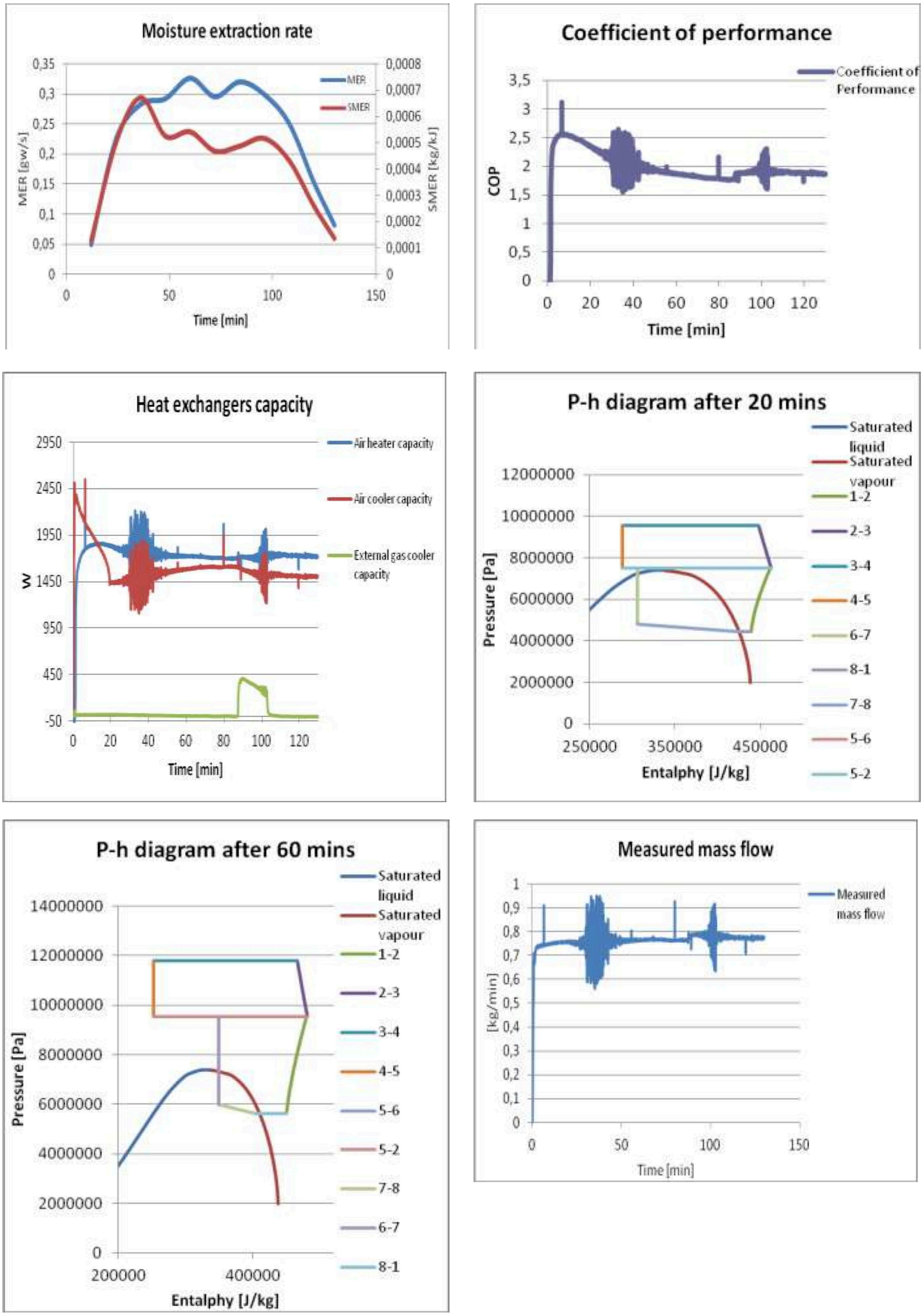

Figure 31: Results experiment 2

Table 10: Other results experiment 2

Other results		
Energy consumption:	2,32	kWh
Consumption per kg textile	0,45	$kWh/kg_{textile}$
Air leakage	11,90	%
Average COP	2,0	
Energy savings	0,50	
Average water extraction	0,0227	kg/min
Moisture content end	-2,51	%
Standstill pressure before run	54	bara
Weight dry textiles	5,17	kg
Weight incl. 60% moisture	8,272	kg
Weight after experiment	5,04	kg
Mass flow high-pressure side	0,0127	kg/s average

The energy consumption of the drum dryer is still too high but it is lower than the previous experiment. Thus the trend is positive. The COP-value is good but the experiment is again run for too long. The intermediate pressure level is supercritical after about 20 minutes. The temperature that is called “T_SLC” is located at the intermediate pressure level between the separation tank and the compressor. This temperature remains very high during the experiment, and this indicates that the refrigerant flows in the wrong direction at the intermediate pressure stage. This theory is confirmed by the ph-diagram after 60 minutes. Rather than supplying the compressor with flash gas, the compressor feeds the separation tank hot refrigerant. This is unfavourable because the mass flow through the air heater is decreased and the dehumidifying ability of the refrigerant is reduced because it absorbs heat before it is throttled and enters the evaporator with a higher gas fraction than desired. The temperature of the refrigerant will unquestionably decrease from 1st discharge to 2nd inlet, even though this is not clear from the ph-diagrams. An extra thermoelement will be installed to measure this difference in future experiments.

Experiment 3: Longer capillary tube to low pressure stage, open valve, high refrigerant charge

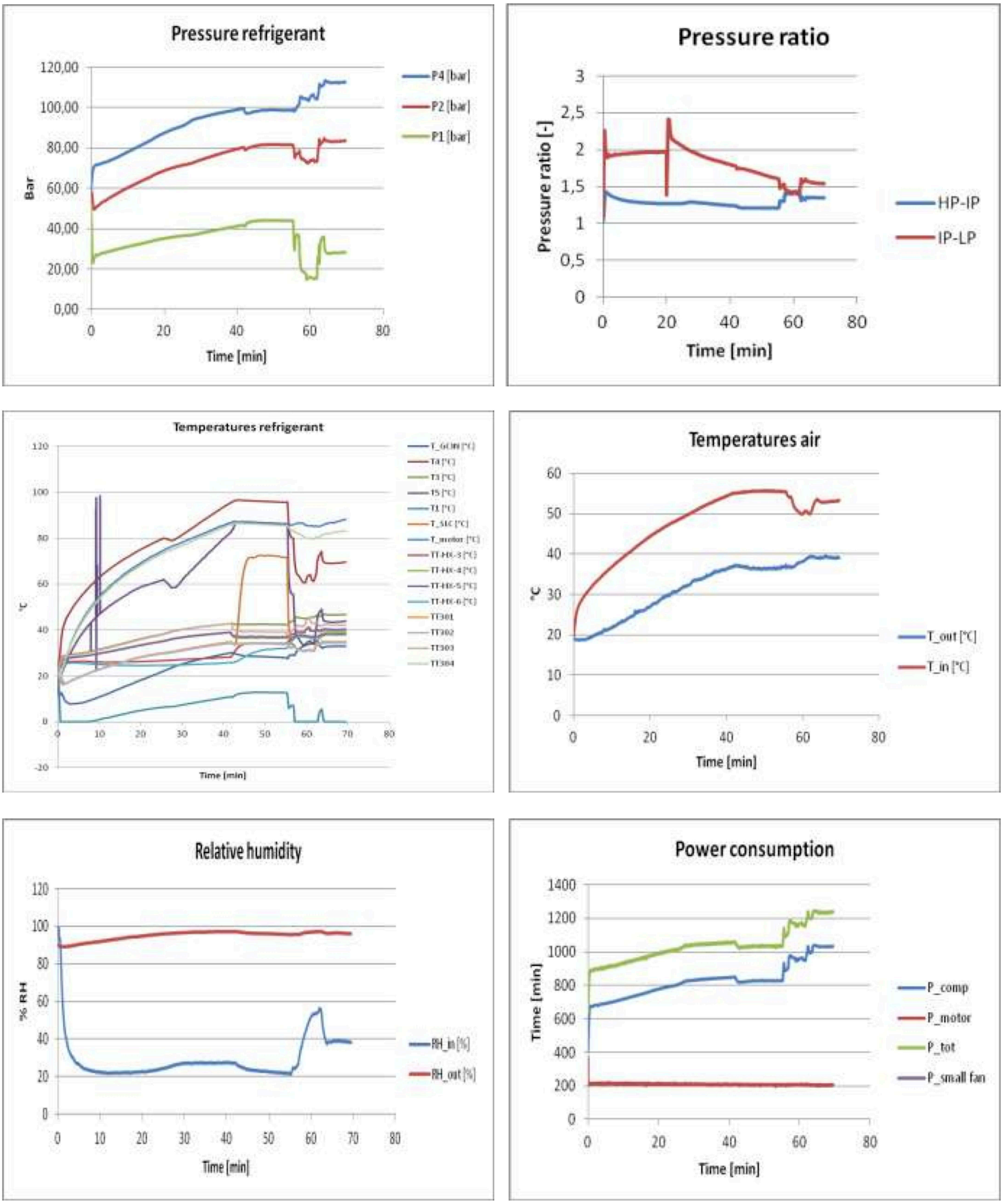

Figure 32: Results experiment 3

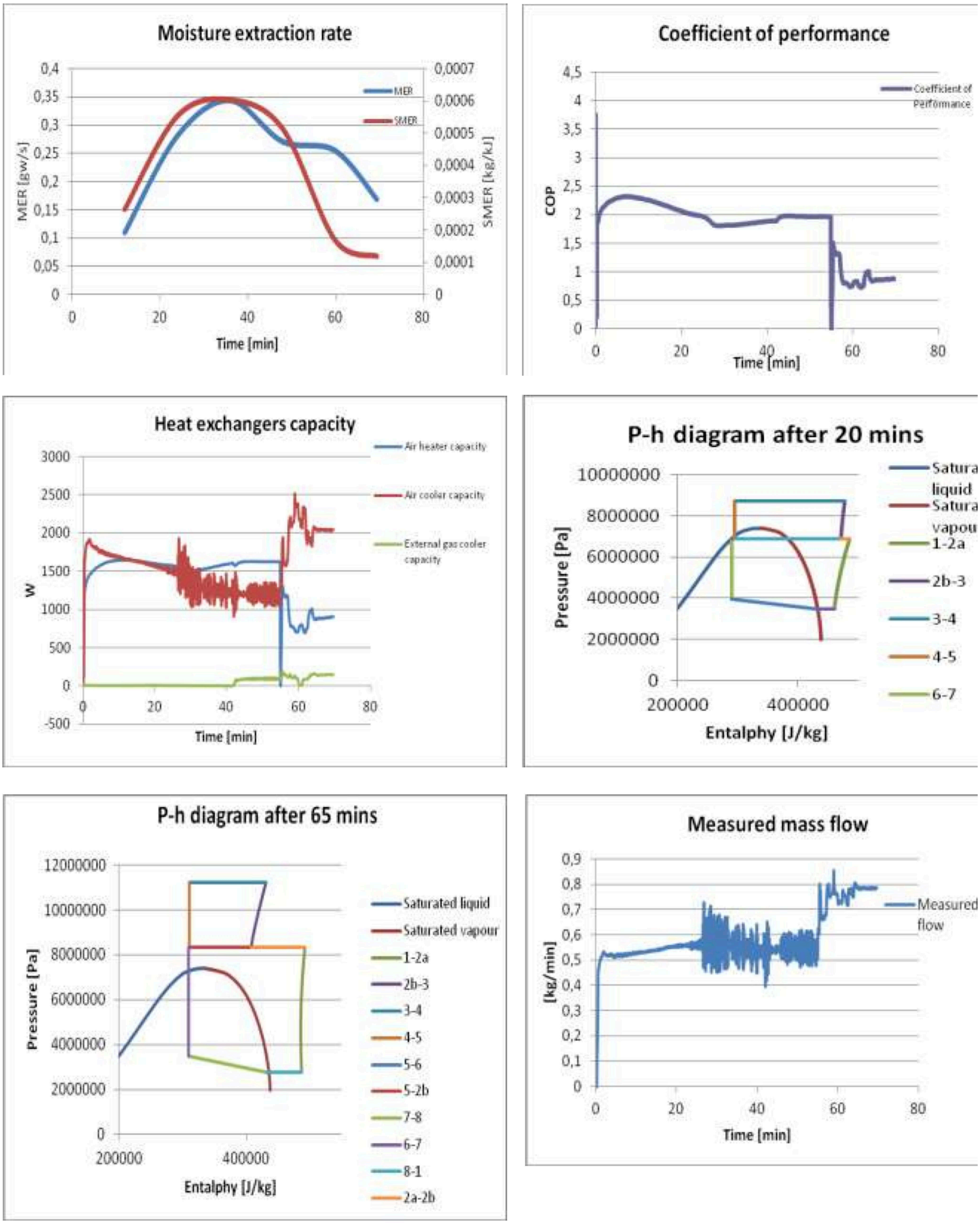

Figure 33: Results experiment 3

Table 11: Other results experiment 3

Other results		
Energy consumption:	1,23	kWh
Consumption per kg textile	0,24	$kWh/kg_{textile}$
Air leakage	9,48	%
Average COP	1,81	
Energy savings	0,45	
Average water extraction	0,014	kg/min
Moisture content end	24,31	%
Standstill pressure before run	54	bara
Weight dry textiles	5,10	kg
Weight incl. 60% moisture	8,16	kg
Weight after experiment	6,34	kg
Mass flow high-pressure side	0,0098	kg/s average

This experiment is performed with an open valve at the intermediate pressure stage. This allows the refrigerant to flow from the separation tank to the compressor. The process seems to work fine for about 40 minutes before the flow direction at intermediate pressure suddenly turns. This can be seen from the orange T_SLC – graph in the refrigerant temperature diagram Figure 32. After about 15 minutes with this flow it switches back, causing multiple reactions that decreases the efficiency. The air heater capacity drops, intermediate and low-pressure stages drops, high-pressure stage increases, 2nd stage compressor discharge temperature decreases, COP decreases and MER/SMER decrease. Compressor work increases gradually before shutting off after about 70 minutes total. The ph-diagram after 65 minutes shows a great cooling of the gas before entering the 2nd stage compression. This is a sign of liquid mixing with the compressed gas and possibly entering the compressor, causing it to shut down. The pressure and temperature suggest that the process is marginally supercritical, thus liquid should not be present. When the temperature and pressure are close to the critical point for CO_2, the phases are rather mixed and some liquid might occur even though properties are supercritical.

A short time after this experiment, a blockage in one of the capillary tubes was discovered. The blockage was removed and the tubes connected to the capillary tube were cleaned. This blockage might have contributed to the strange behaviour of the system in this experiment.

Experiment 4: Longer capillary tube to low pressure stage, closed valve, high refrigerant charge

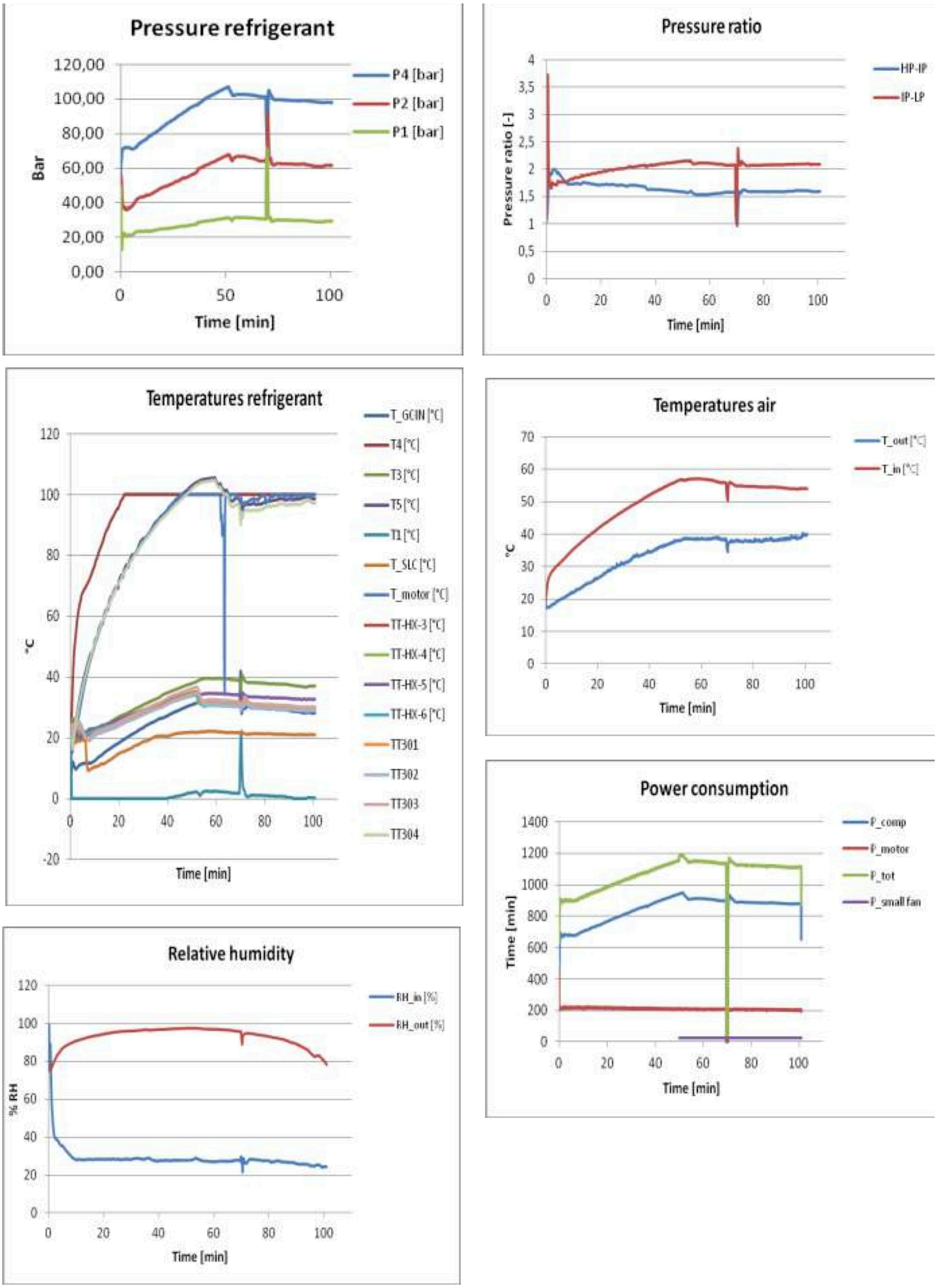

Figure 34: Results experiment 4

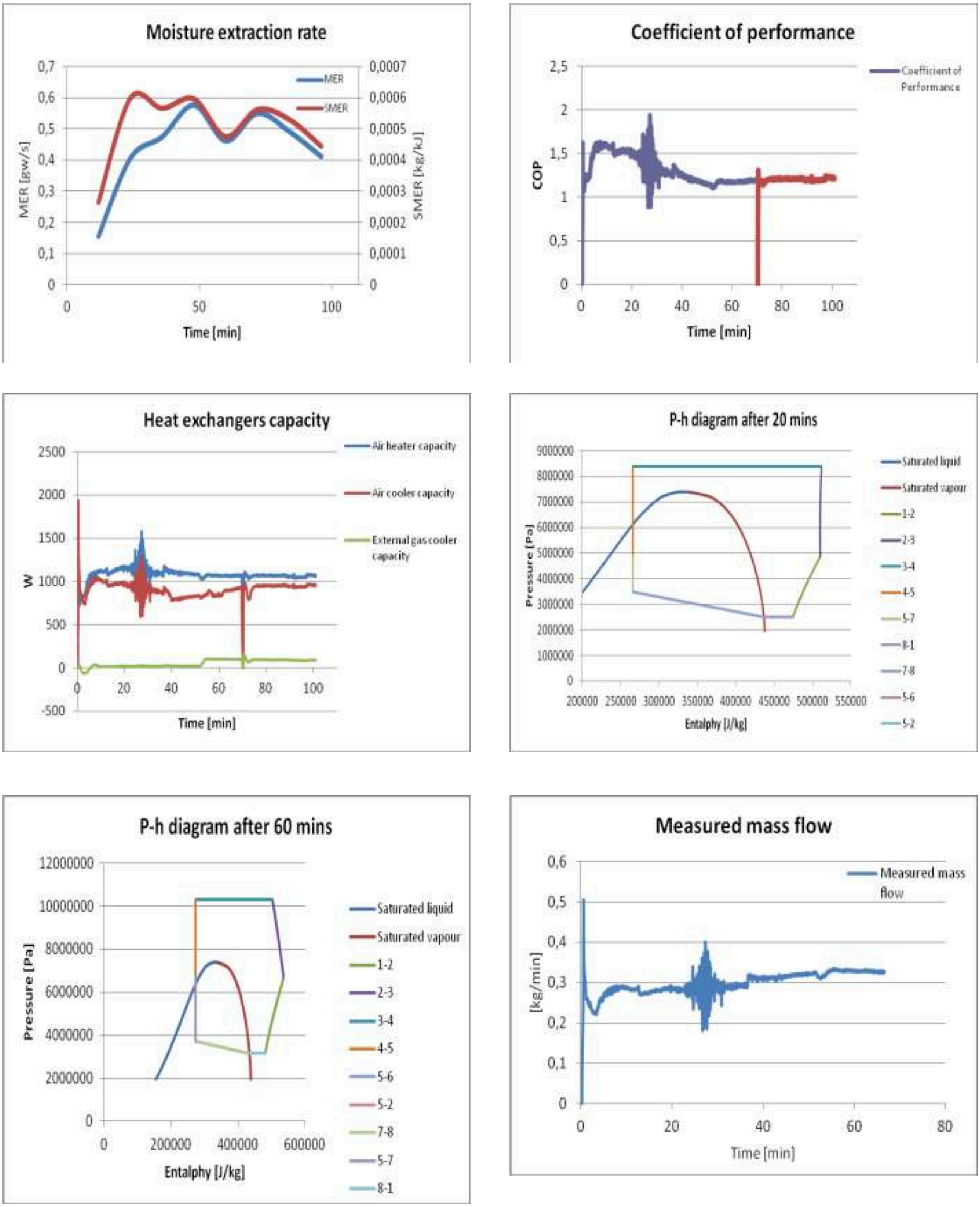

Figure 35: Results experiment 4

Table 12: Other results experiment 4

Other results		
Energy consumption:	1,79	kWh
Consumption per kg textile	0,35	$kWh/kg_{textile}$
Air leakage	20,71	%
Average COP	1,26	
Energy savings	0,20	
Average water extraction	0,021	kg/min
Moisture content	9,57	%
Standstill pressure before run	54	bara
Weight dry textiles	5,12	kg
Weight incl. 60% moisture	8,192	kg
Weight after experiment	5,61	kg
Mass flow high-pressure side	0,0051	kg/s average

This experiment is executed with a closed valve at the intermediate pressure stage, preventing refrigerant to flow between the separation tank and the compressor. After about 70 minutes the system was shut down a few seconds to look for any blockages in the air filters, resulting in the spikes seen in the graphs in Figure 34 and Figure 35. The energy consumption is still too high, even though the clothes weren't completely dry at the end of the cycle. The COP is poor, 1.25 on average. The compressor became very hot during this experiment, and needed external cooling after a short while. Both the temperature after 2nd discharge and the temperature of the compressor couldn't measure temperatures over 100 °C. This means that the air heater capacity and the COP may have been higher than they appear in the results. The 2nd compression curve in the ph-diagrams would also be more realistically curved towards the right rather than to the left. However, the energy consumption is calculated from the work of the compressor and should be accurate. Hence the performance of the system is not good enough. The superheat out of the evaporator reaches 30 K even though all the refrigerant is forced through the evaporator. However the mass flow is extremely low, 0.05 kg/s on average. This may be related to a blocking found in one of the capillary tubes a few days later. The mass flow has a large impact on the results, and a rerun of this particular experiment will be considered.

Experiment 5: Longer capillary tube to low pressure stage, variable valve, 780 g refrigerant charge

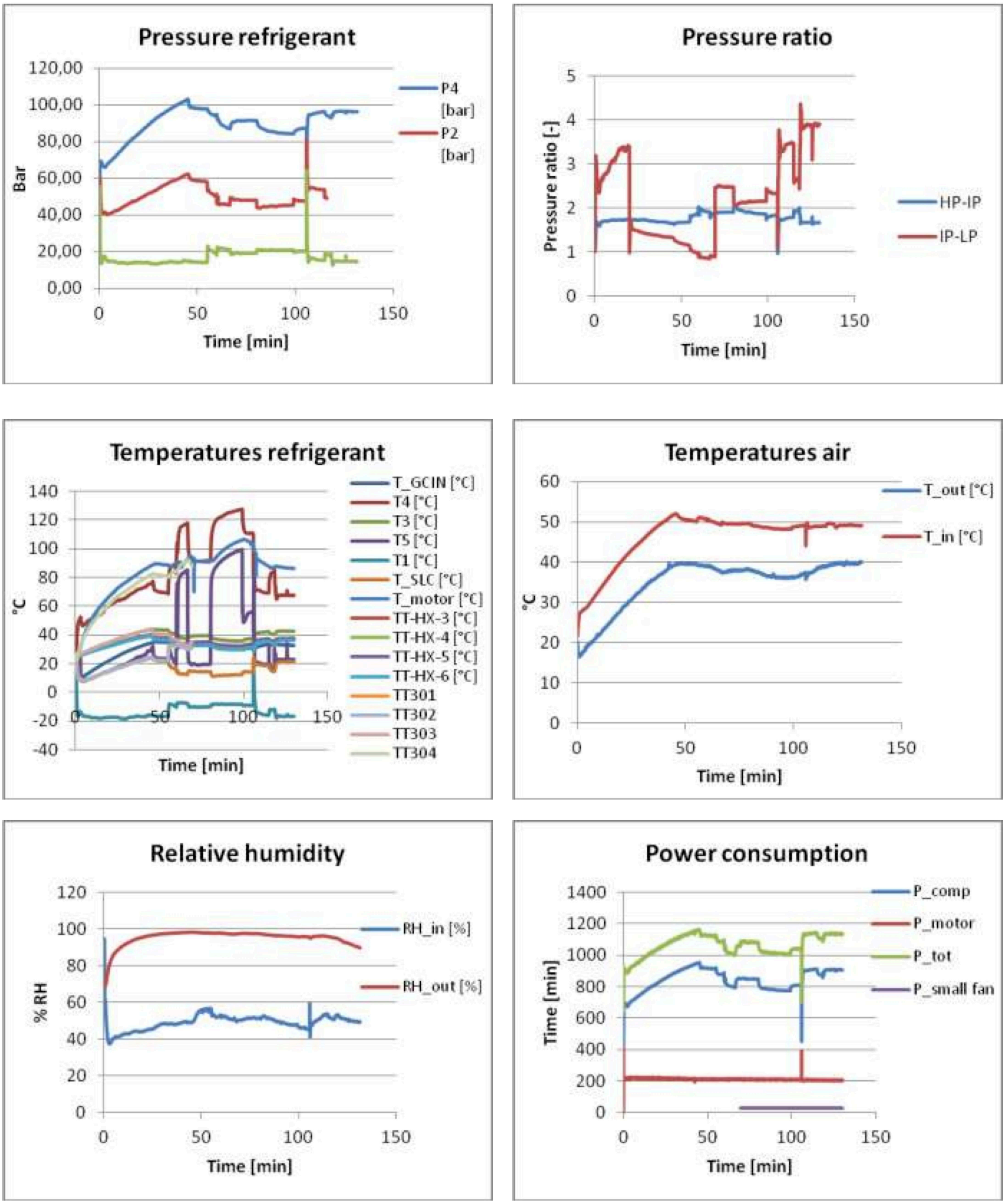

Figure 36: Results experiment 5

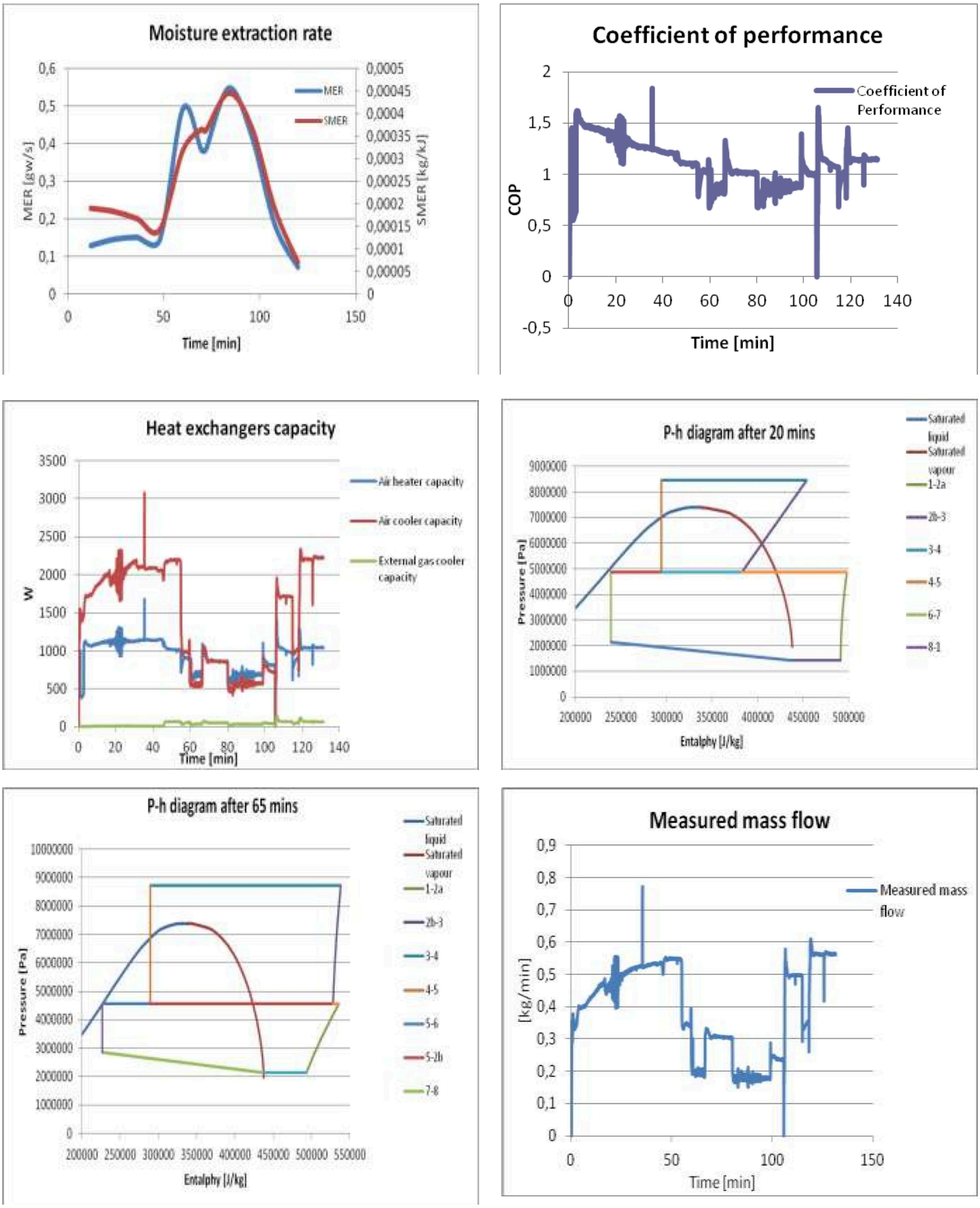

Figure 37: Results experiment 5

Table 13: Other results experiment 5

Other results		
Energy consumption:	2,31	kWh
Consumption per kg textile	0,45	$kWh/kg_{textile}$
Air leakage	23,35	%
Average COP	1,11	
Energy savings	0,10	
Average water extraction	0,016	kg/min
Moisture content end	9,39	%
Standstill pressure before run	55,5	bara
Weight dry textiles	5,11	kg
Weight incl. 60% moisture	8,176	kg
Weight after experiment	5,59	kg
Mass flow high-pressure side	0,0063	kg/s average

During this experiment, the valve at intermediate-pressure stage is varied to monitor how it impacts the process. The valve is closed just before 50 minutes, resulting in lower power consumption, the high-pressure side experience higher temperatures, lower pressure and a dramatic reduction of the mass flow. Even though the temperature of the refrigerant entering the gas cooler is higher, the gas cooler effect is drastically reduced due to the reduced mass flow. The COP-curve experiences a drop directly related to the reduced gas cooler effect. The energy consumption is 0.45 kWh/kg textile, which is way too high. The ph-diagram after 20 minutes shows that some liquid enters the compressor. The compressor can handle some liquid so it manages to run with this small quantity. At 60 minutes the process looks all right from the ph-diagram, but it is way too inefficient and the experiment was shut down after about 130 minutes because of no signs of progress.

Experiment 6: Open valve at intermediate pressure, long capillary tube, 880g refrigerant charge

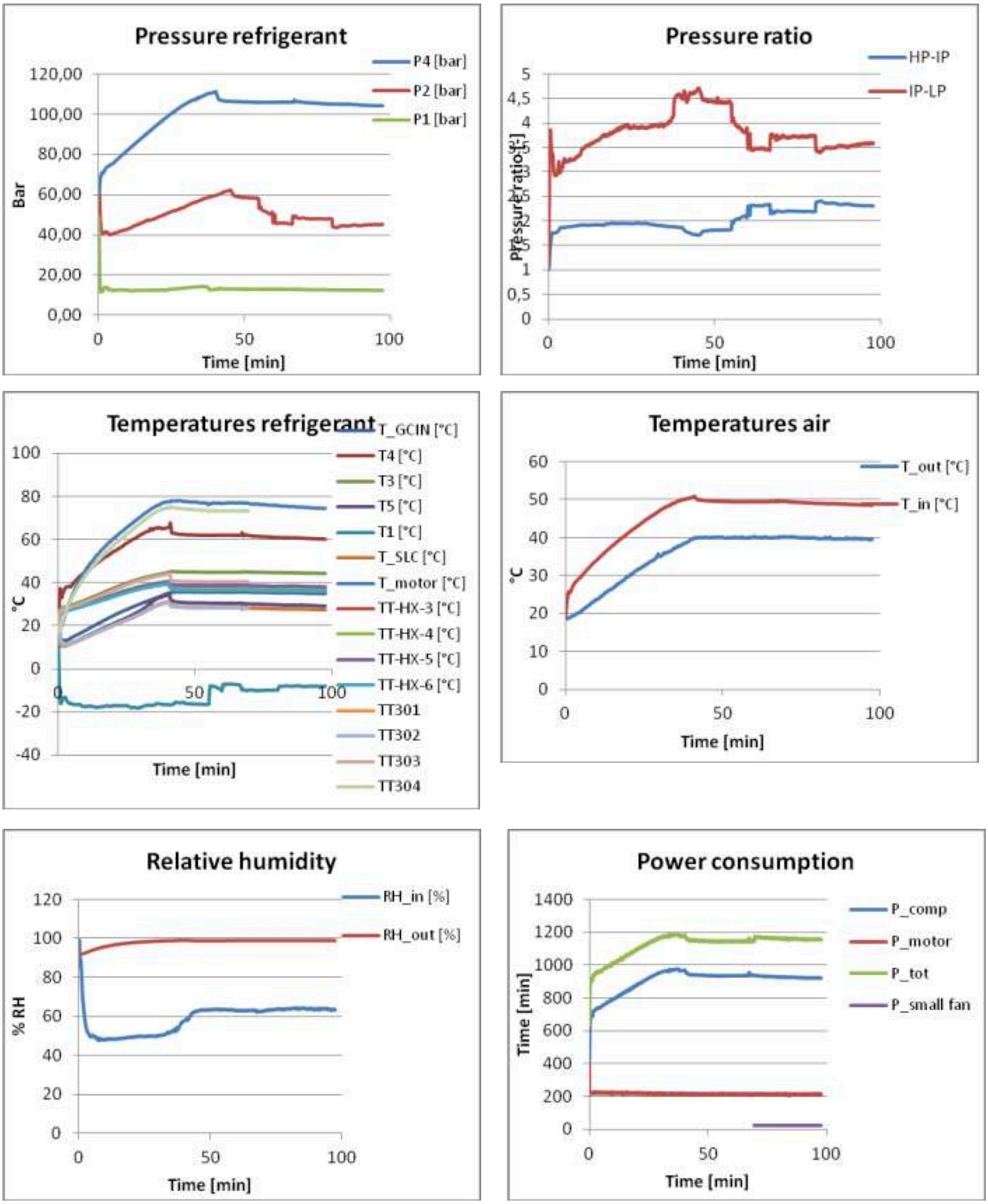

Figure 38: Results experiment 6

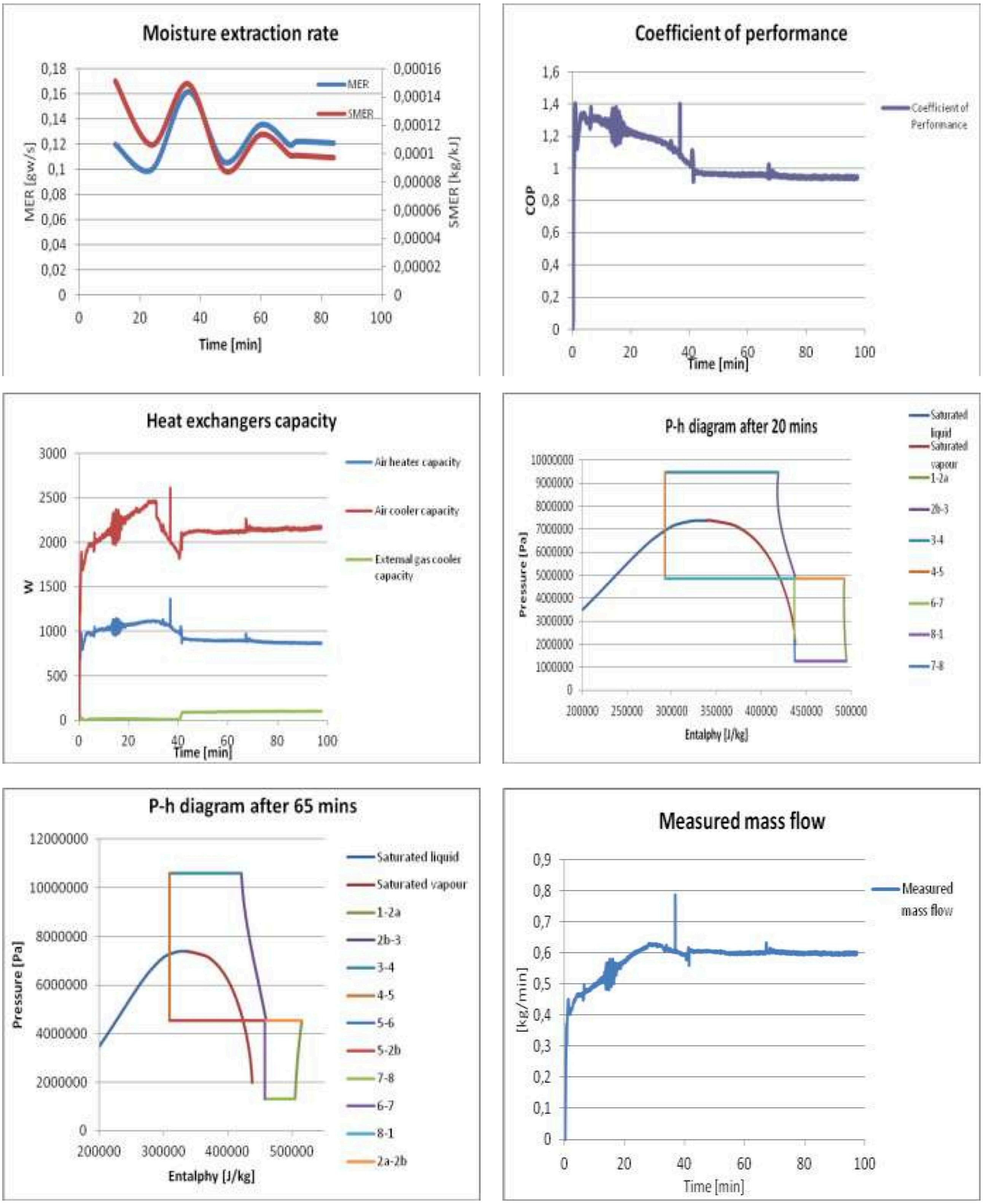

Figure 39: Results experiment 6

Table 14: Other results experiment 6

Other results		
Energy consumption:	1,83	kWh
Consumption per kg textile	0,36	$kWh/kg_{textile}$
Air leakage	16,34	%
Average COP	1,06	
Energy savings	0,05	
Average water extraction	0,01	kg/min
Moisture content end	27,43	%
Standstill pressure before run	54,7	bara
Weight dry textiles	5,14	kg
Weight incl. 60% moisture	8,224	kg
Weight after experiment	6,55	kg
Mass flow high-pressure side	0,0097	kg/s average

This experiment is executed with a 40 cm capillary tube between the intermediate-pressure stage and the low-pressure stage. The purpose of this experiment is to see if the flow turns multiple times throughout the experiment like it did during experiment 3. The valve at the intermediate-pressure stage is kept open through the entire experiment. The temperature curve for "T_SLC" in Figure 38 is stable, this indicates that the flow direction is constant during the experiment. The behaviour in experiment 3 was arguably a special case, probably caused by a capillary tube blockage that was found soon after the experiment.

The ph-diagrams in Figure 39 indicate that the flash gas injection process runs in the wrong direction and the liquid entering the flash tank from the capillary tube is evaporated immediately. This represents a huge energy loss. The average COP is 1.06, and most of the time the COP is below 1. The superheat out of the evaporator is still way too high, which may be a sign of too little refrigerant running through the evaporator. The experimented ended before the textiles were dry because of poor performance and no signs of improvement.

Experiment 7: Longer capillary tube to low pressure stage, throttle at intermediate pressure, 880 g refrigerant charge

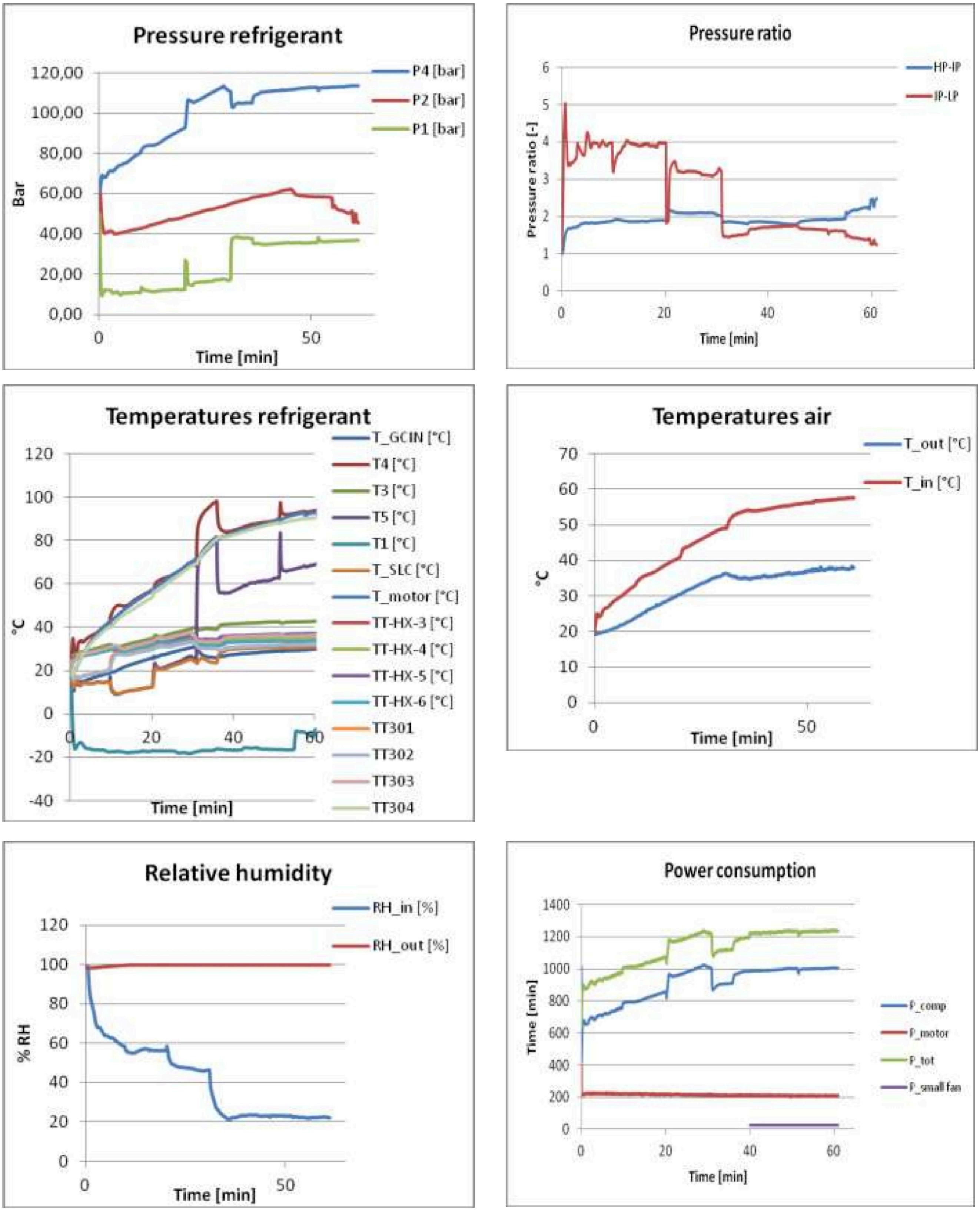

Figure 40: Results experiment 7

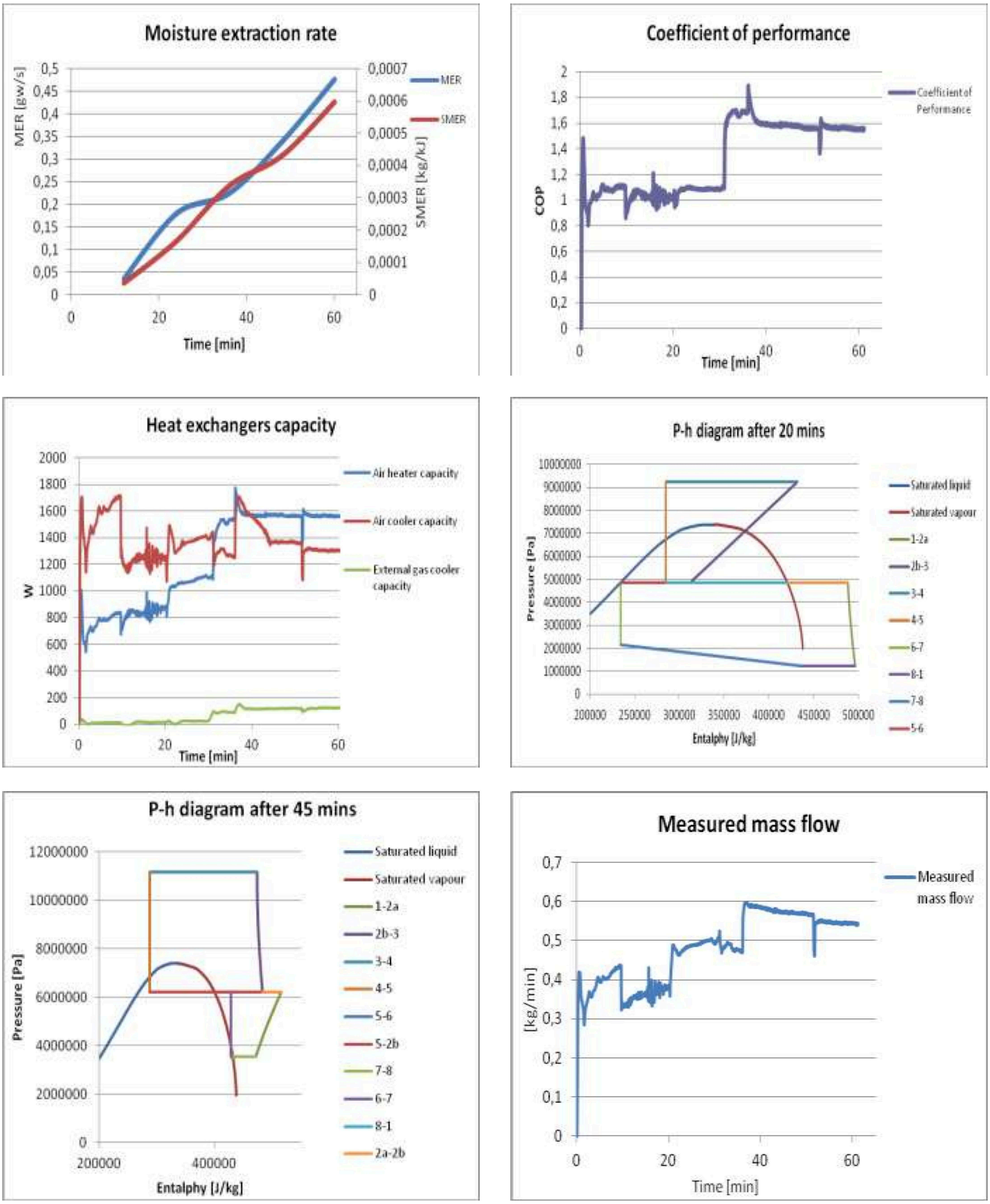

Figure 41: Results experiment 7

Table 15: Other results experiment 7

Other results		
Energy consumption:	1,15	kWh
Consumption per kg textile	0,23	$kWh/kg_{textile}$
Air leakage	14,71	%
Average COP	1,32	
Energy savings	0,24	
Average water extraction	0,0092	kg/min
Moisture content end	31,76	%
Standstill pressure before run	55	bara
Weight dry textiles	5,10	kg
Weight incl. 60% moisture	8,16	kg
Weight after experiment	6,72	kg
Mass flow high-pressure side	0,0080	kg/s average

In this experiment the valve at the intermediate pressure was barely open, allowing a small quantity of gas to enter the compressor for recompression. Previous experiments have shown that too much refrigerant enters the compressor when the valve is fully open, leading to a shortage of refrigerant running through the evaporator. In the start of this experiment the evaporator pressure and temperature was very low, which combined with very little condensed water led to a suspicion of icing on the evaporator. In order to fix this issue, 100 g of CO_2 was fed into the low-pressure side of the system after about 20 minutes. The evaporator pressure rose immediately, and more water were drained from the evaporator. The compressor consumes a lot of energy, but the high-pressure side temperature is quite low. The valve is shut for a while after about 30 minutes, causing the compressor work to drop and the temperature to increase. Slightly opening the valve again causes the temperature to drop drastically again. The mixing temperature T5 is the temperature before the refrigerant enters the compressor for the 2nd stage compression. This temperature is remarkable lower than the 1st discharge temperature T304, even though only a small quantity of refrigerant should mix with the discharged gas. The ph-diagram after 20 minutes in Figure 41 shows that the flow is cooled down and transformed to liquid before entering the compressor for the 2nd stage compression. This is not healthy for the compressor and it is making the process very inefficient. Whenever the valve is open, even just slightly open, the compressor work is much larger than when it’s closed. This strengthens the theory that the flow is throttled into the two-phase region over the adjustable valve at intermediate pressure. The compressor shuts down suddenly after 60 minutes, very likely caused by liquid CO_2 in the compressor. The average COP-value is not too bad, but the experiment finished way too early.

Experiment 8: Effects of increased mass flow, closed valve at intermediate pressure stage

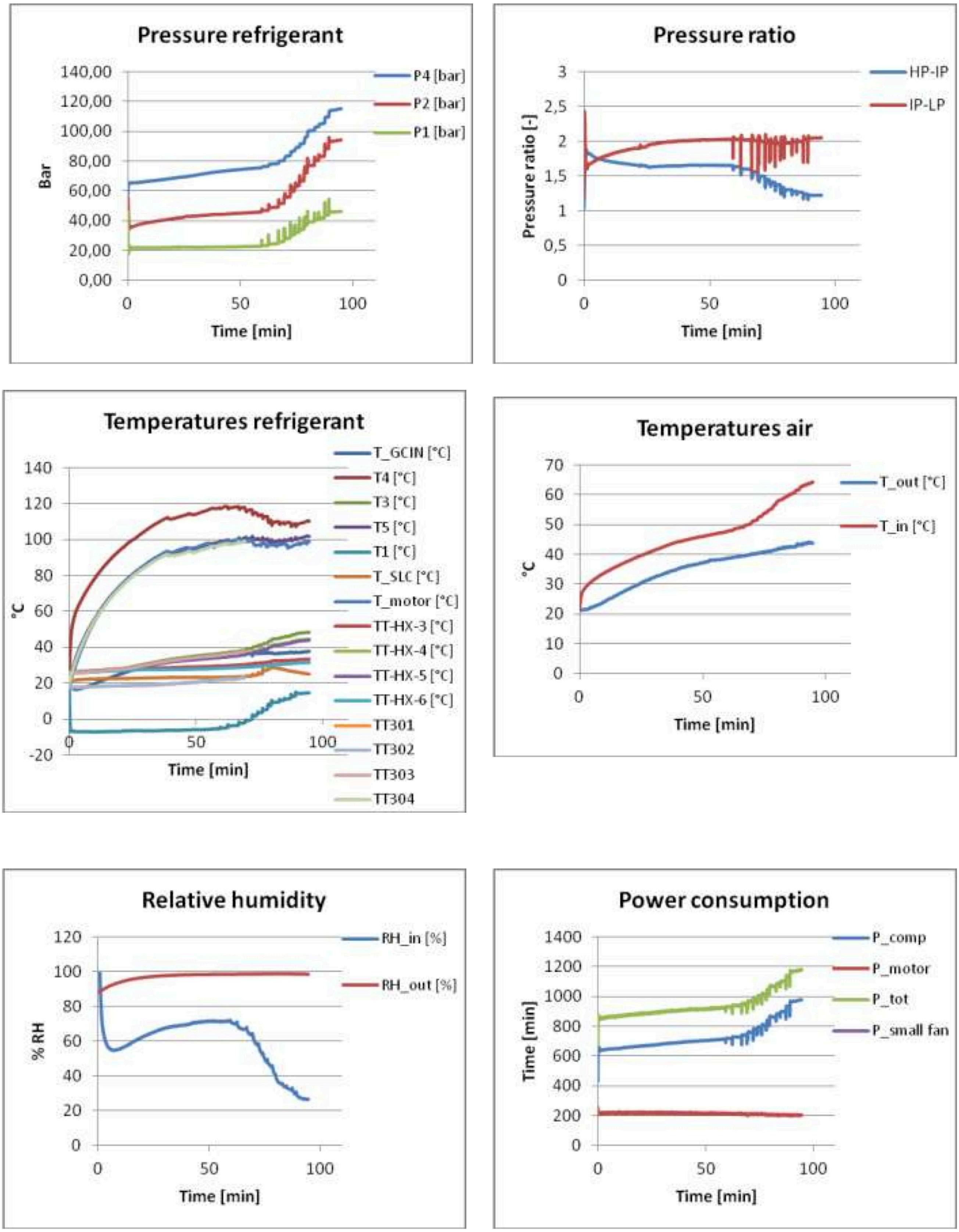

Figure 42: Results experiment 8

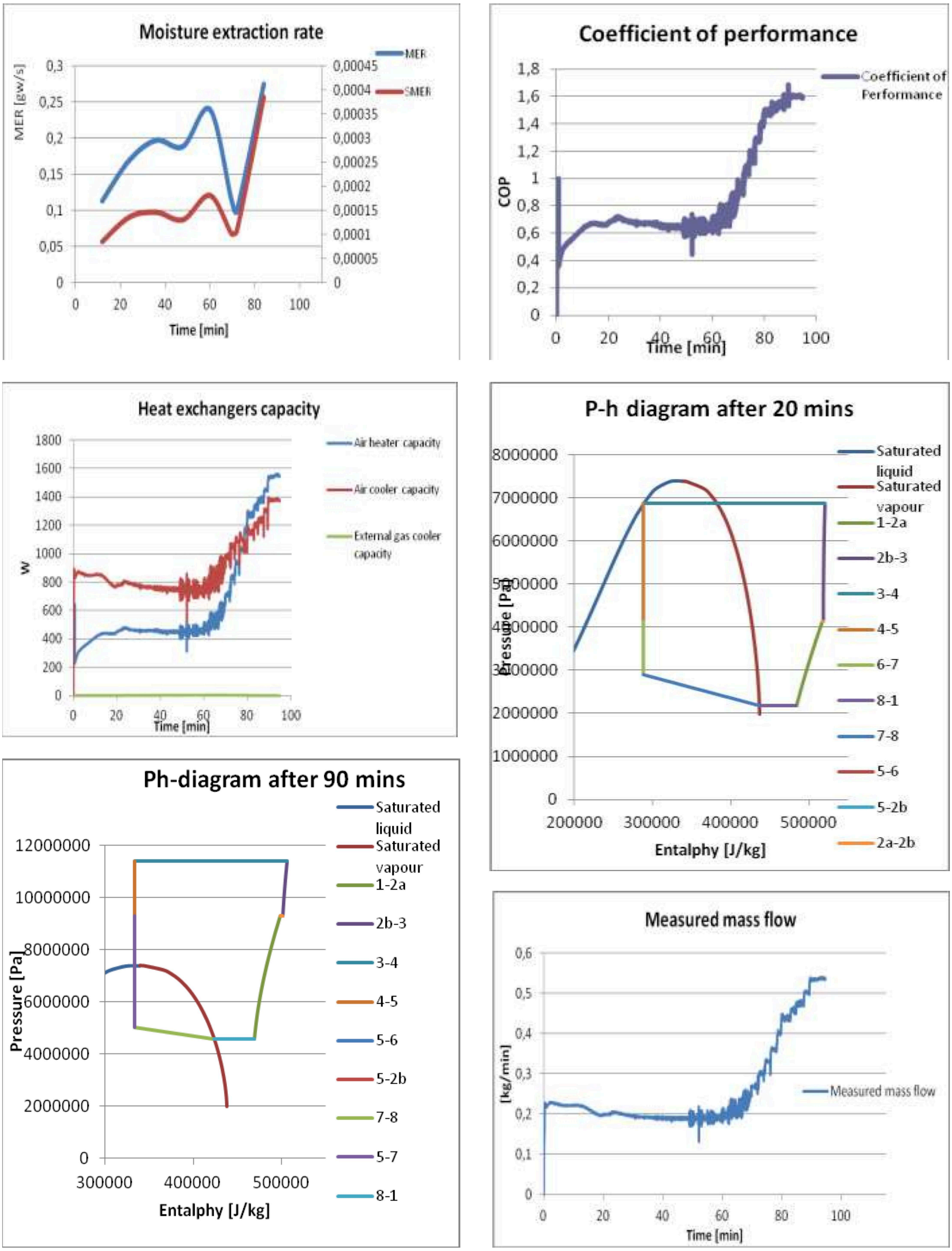

Figure 43: Results experiment 8

Table 16: Other results experiment 8

Other results		
Energy consumption:	1,49	kWh
Consumption per kg textile	0,29	$kWh/kg_{textile}$
Air leakage	19,44	%
Average COP	0,65	
Energy savings	-0,55	
Average water extraction	0,01037	kg/min
Moisture content end	26,42	%
Standstill pressure before run	55	barg
Weight dry textiles	5,11	kg
Weight incl. 60% moisture	8,176	kg
Weight after experiment	6,46	kg
Mass flow high-pressure side	0,0043	kg/s average

This experiment was executed with a closed valve at the intermediate pressure. The first 60 minutes of the experiment was run with a fixed amount of refrigerant and very low efficiency. CO_2 was then injected into the system in very small portions. This resulted in higher mass flow, lower superheat, higher moisture extraction rate and higher efficiency. The injection was continued until the pressure levels reached the predesigned levels: 115 barg in the gas cooler and 45 barg in the evaporator. At this point the mass flow was 0,00833 kg/s. After a short while at this state the compressor stopped without warning. There are no signs of liquid entering the compressor and the temperature of the compressor was stable well below limit temperature.

The trend during this experiment was very positive. The next experiment will be executed with a fixed amount of CO_2 equal to the amount of CO_2 present in the system at the end of experiment 8.

Experiment 9: Closed valve at intermediate pressure, high refrigerant charge

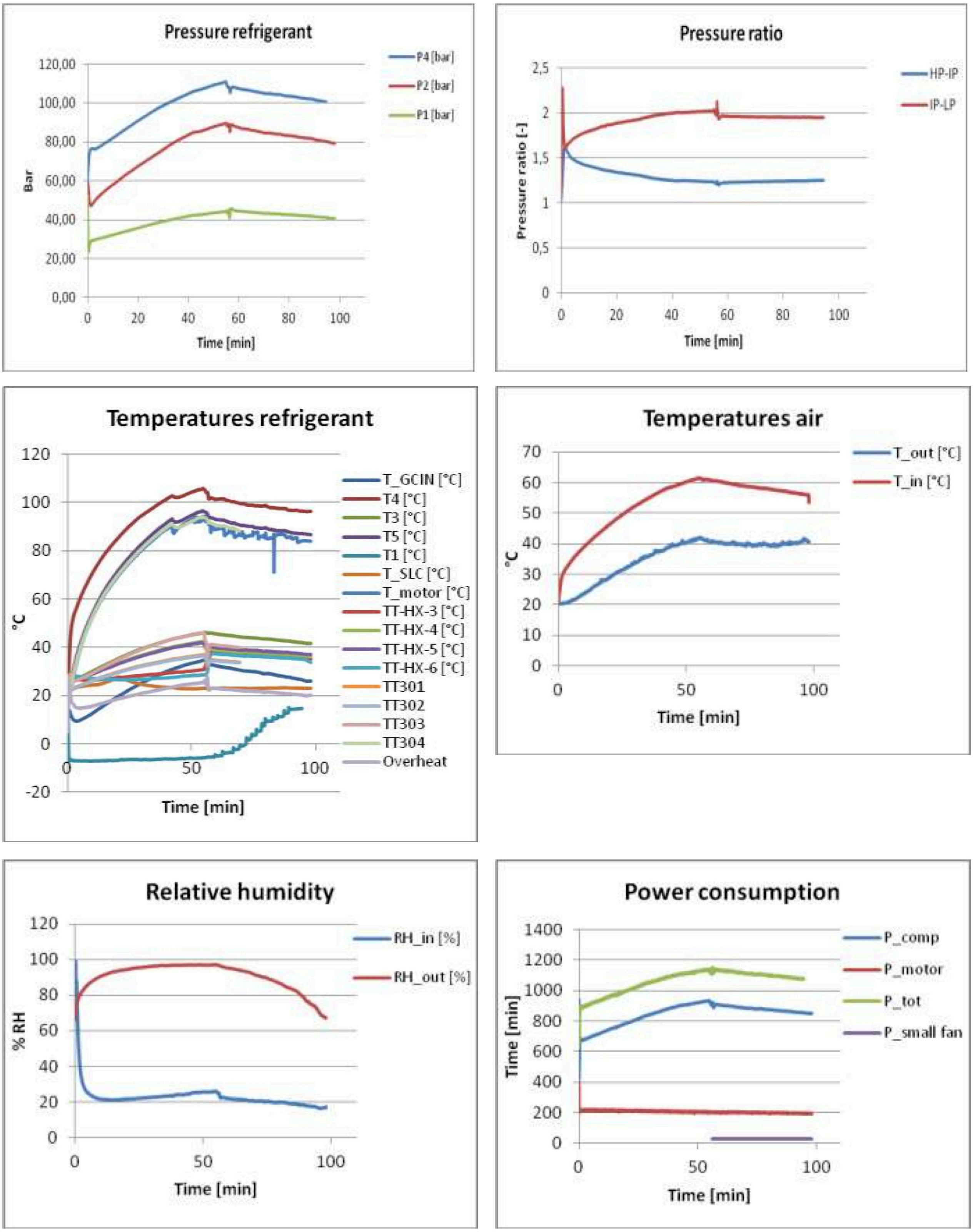

Figure 44: Results experiment 9

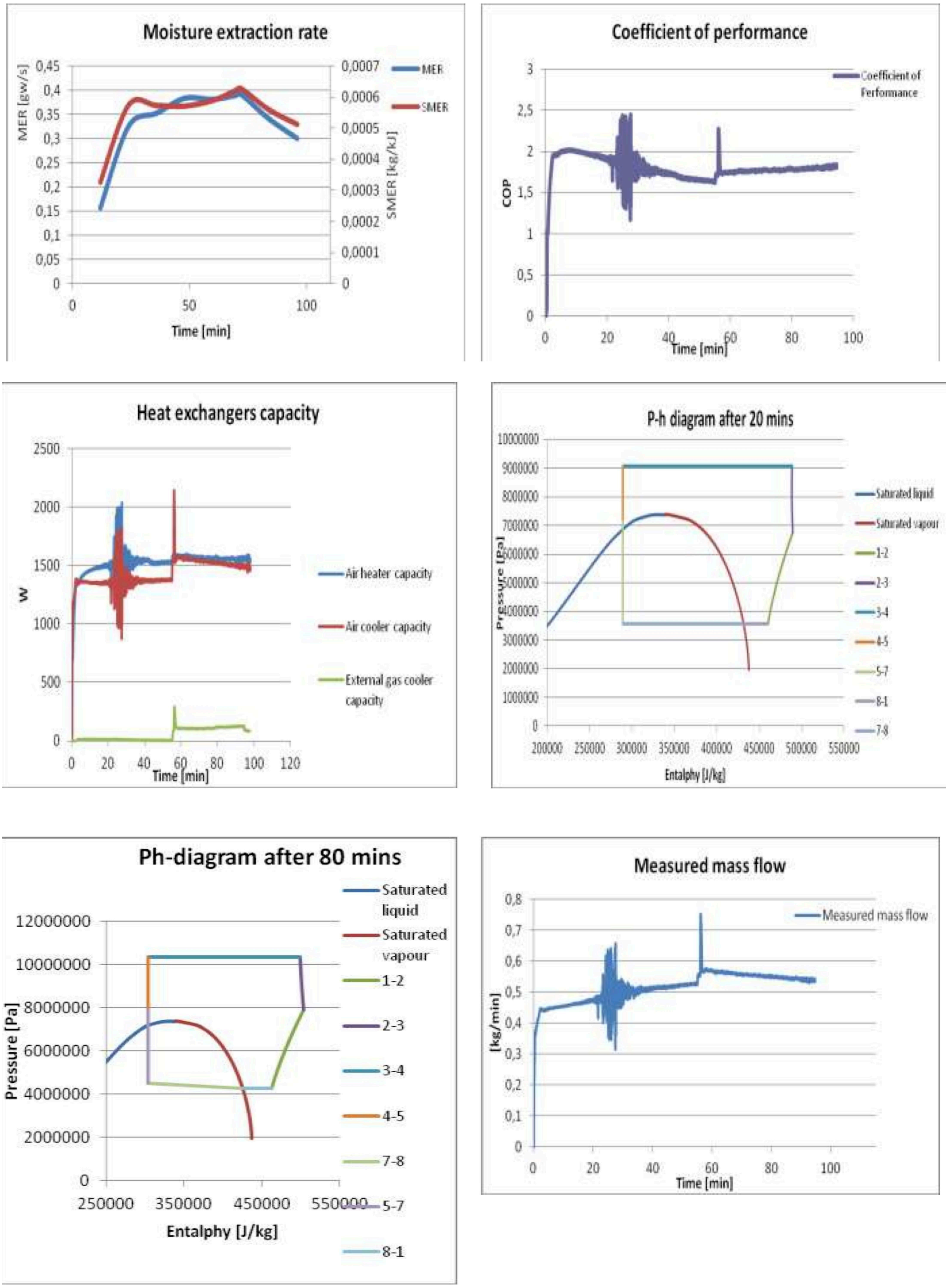

Figure 45: Results experiment 9

Table 17: Other results experiment 9

Other results		
Energy consumption:	1,74	kWh
Consumption per kg textile	0,34	$kWh/kg_{textile}$
Air leakage	9,36	%
Average COP	1,81	
Energy savings	0,46	
Average water extraction	0,0234	kg/min
Moisture content end	5,068	%
Standstill pressure before run	57	barg
Weight dry textiles	5,13	kg
Weight incl. 60% moisture	8,208	kg
Weight after experiment	5,39	kg
Mass flow high-pressure side	0,0086	kg/s average

The efficiency during this experiment is very good. The energy consumption of 0.34 kWh/kg textile is too high compared to the set goal of 0.26 kWh/kg textile, but very good compared to previous experiments. The mass flow is unstable for a few minutes at about 20 minutes and one large peak after almost 60 minutes. The reason for these instabilities may be a blocking that was found in the system shortly after this experiment. The average COP of 1.81 is among the best values achieved during the experiment series. The experiment ended with about 5% moisture still present in the textiles. This is too much but not disastrous. The trend is positive.

Experiment 10: Closed valve at intermediate pressure, 1110 g CO_2, hand valve throttle

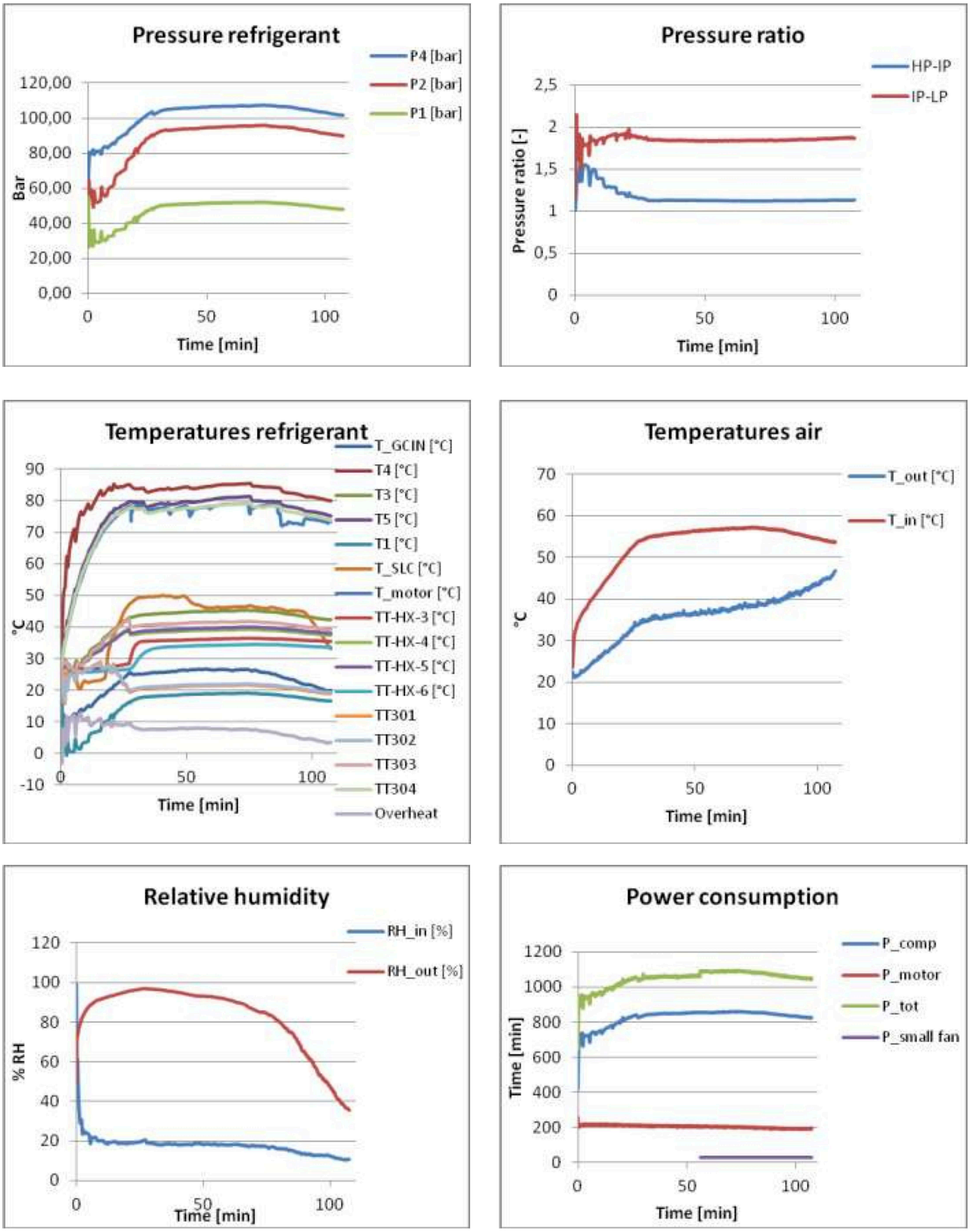

Figure 46: Results experiment 10

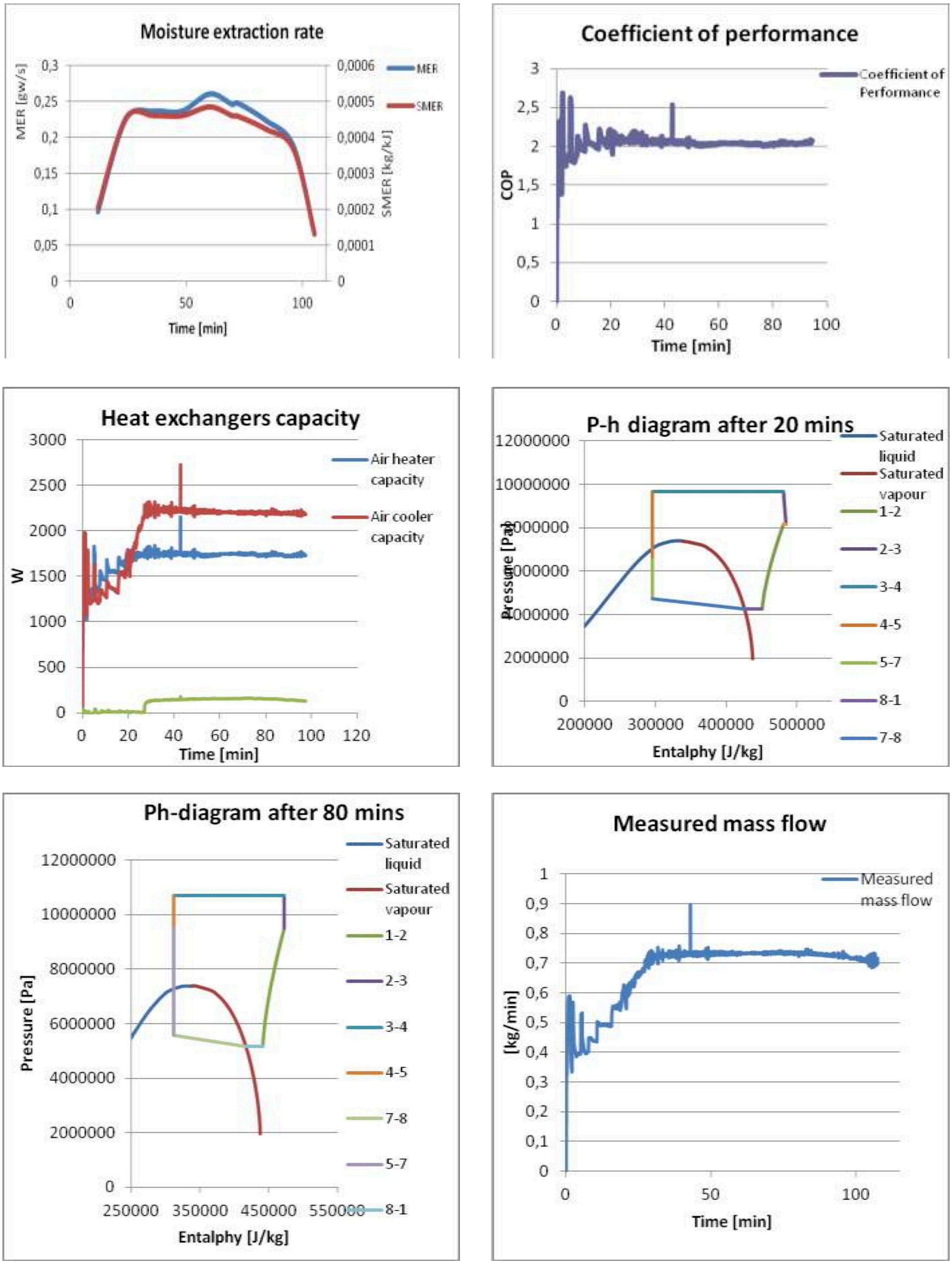

Figure 47: Results experiment 10

Table 18: Other results experiment 10

Other results		
Energy consumption:	1,87	kWh
Consumption per kg textile	0,37	$kWh/kg_{textile}$
Air leakage	13,04	%
Average COP	2,03	
Energy savings	0,51	
Average water extraction	0,0238	kg/min
Moisture content end	0,60	%
Standstill pressure before run	57	barg
Weight dry textiles	4,99	kg
Weight incl. 60% moisture	7,98	kg
Weight after experiment	5,02	kg
Mass flow high-pressure side	0,0112	kg/s average

In this experiment an adjustable throttling valve replaced the capillary tube between the intermediate pressure and the evaporator pressure. The idea is to adjust the evaporator pressure to keep the evaporator superheat between 5 and 10 K. The blocking valve at intermediate pressure stage is closed and the focus is on controlling the superheat out of the evaporator. The performance of the evaporator improved greatly compared to earlier experiments. The COP is the greatest achieved so far. The specific energy consumption is 0.37 $kWh/kg_{textiles}$. It is higher than the state of the art consumption for R134a og 0.26 $kWh/kg_{textiles}$, but with a moisture content of 0.60% in the end this is the best performing experiment so far.

Experiment 11: Open valve at intermediate pressure, 1110 g CO_2, hand valve throttle

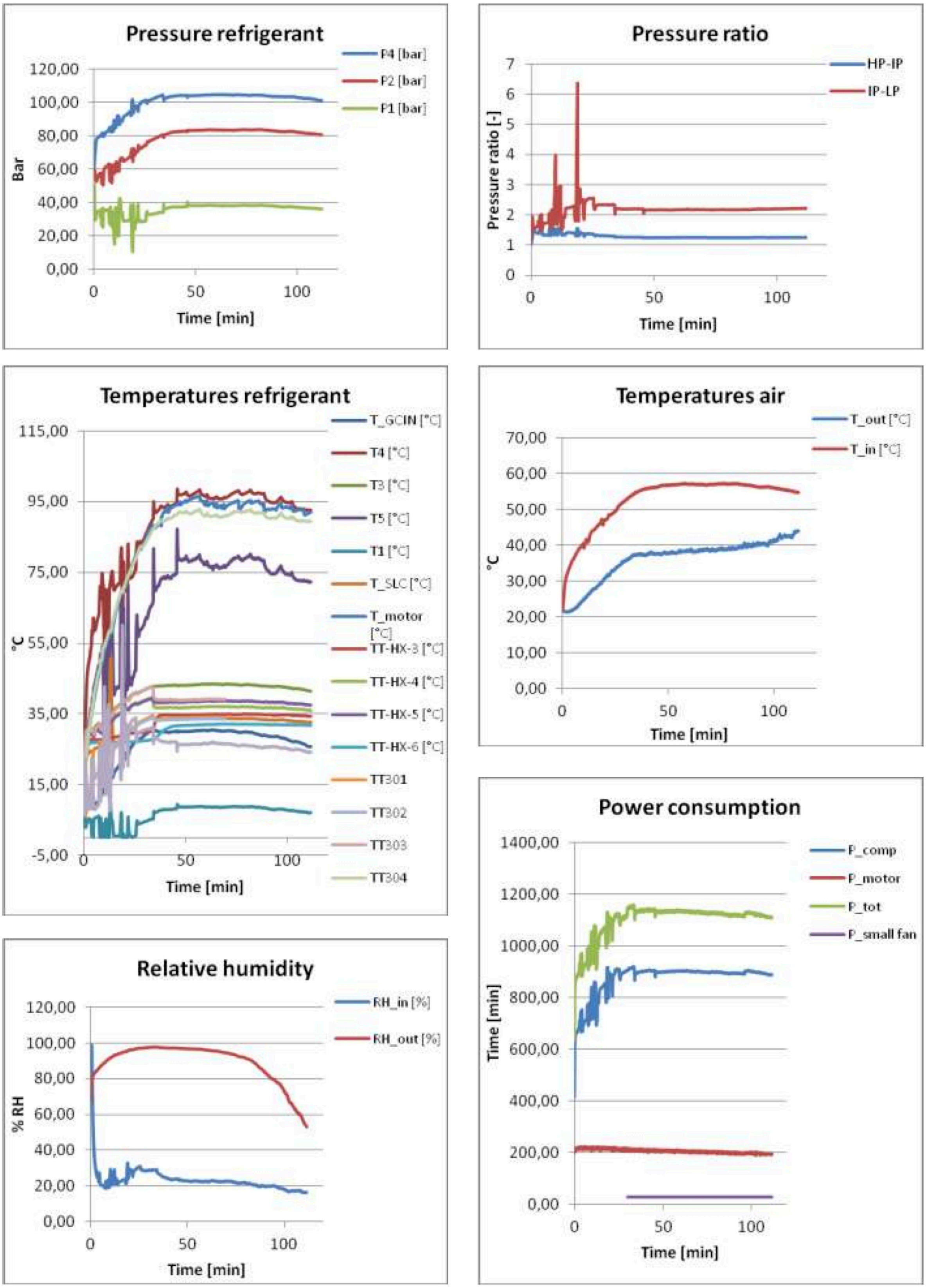

Figure 48: Results experiment 11

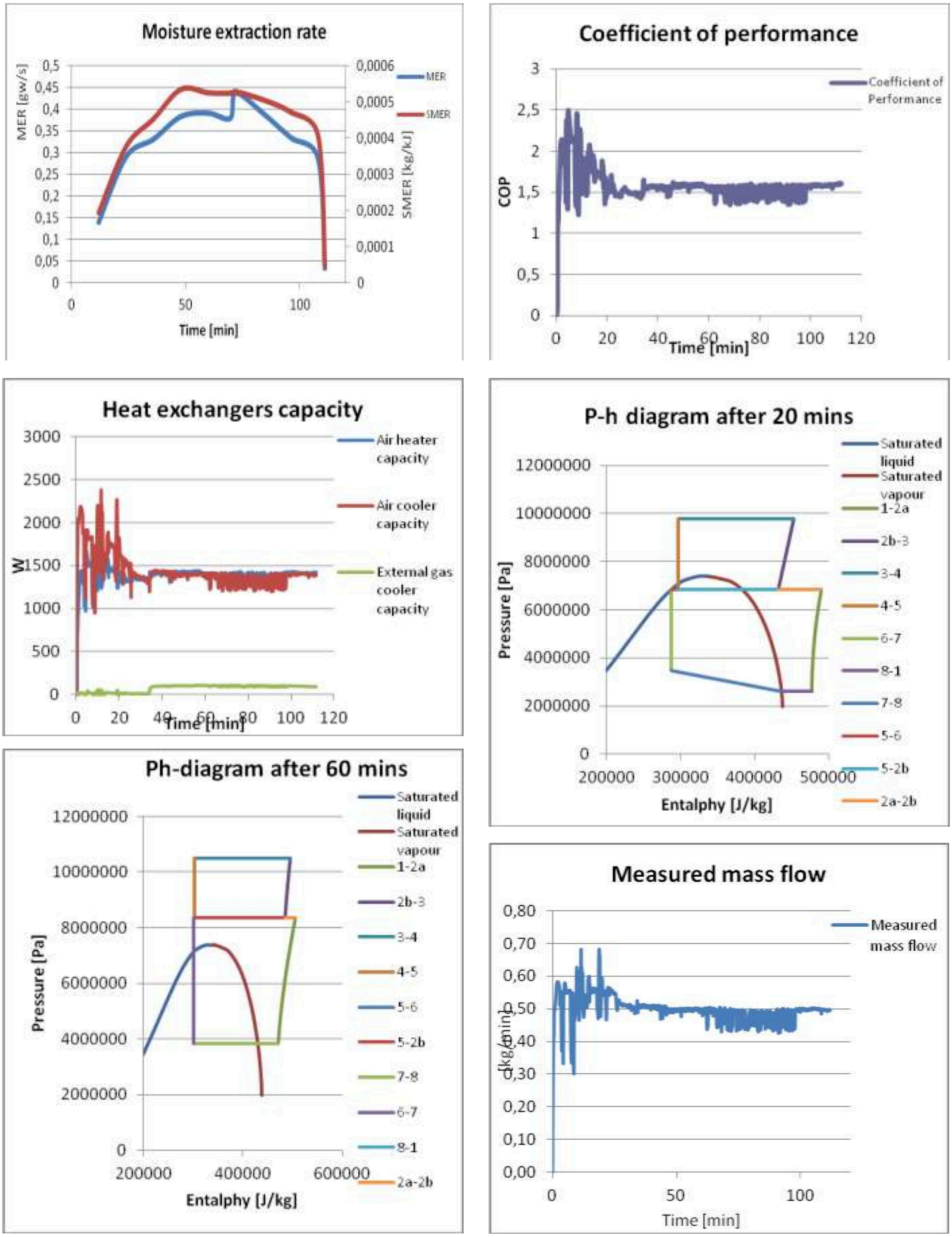

Figure 49: Results experiment 11

Table 19: Other results experiment 11

Other results		
Energy consumption:	2,045	kWh
Consumption per kg textile	0,41	$kWh/kg_{textile}$
Air leakage	6,85	%
Average COP	1,64	
Energy savings	0,39	
Average water extraction	0,0248	kg/min
Moisture content end	1,80	%
Standstill pressure before run	58	barg
Weight dry textiles	5,01	kg
Weight incl. 60% moisture	8,20	kg
Weight after experiment	5,10	kg
Mass flow high-pressure side	0,0083	kg/s average

In this experiment the valve at intermediate pressure was opened when the pressure levels stabilised. The hand-adjusted throttle valve was adjusted to maintain correct flow direction at the intermediate pressure stage. In order to have as little superheat as possible from the evaporator, the intermediate stage was run at the limit. The pressure in the flash tank was just barely higher than the pressure at the compressor inlet, and a small quantity of gas mixed with the compressor's 1st discharge gas. The superheat out of the evaporator was still too high, 25 K on average. The overall performance is poor. The specific energy consumption is too high and the COP is low compared to the previous experiment. It appears to be a compromise between the direction of the flow at intermediate pressure stage and superheat out of the evaporator. Ideally the flow direction would remain stable even with a high mass flow through the evaporator and accompanying low superheat.

Experiment 12: Closed intermediate pressure, 1160 g CO_2, hand-valve throttle

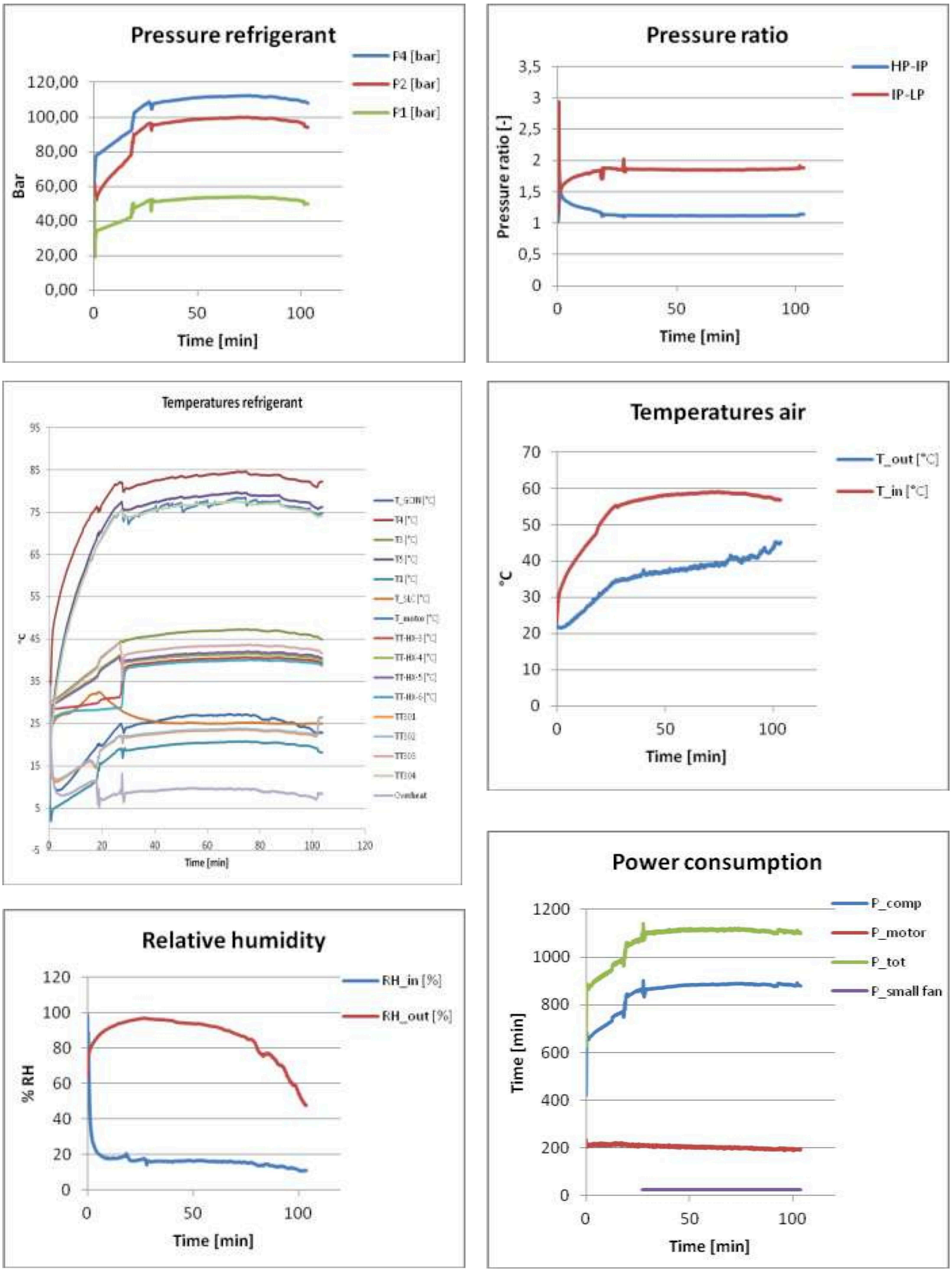

Figure 50: Results experiment 12

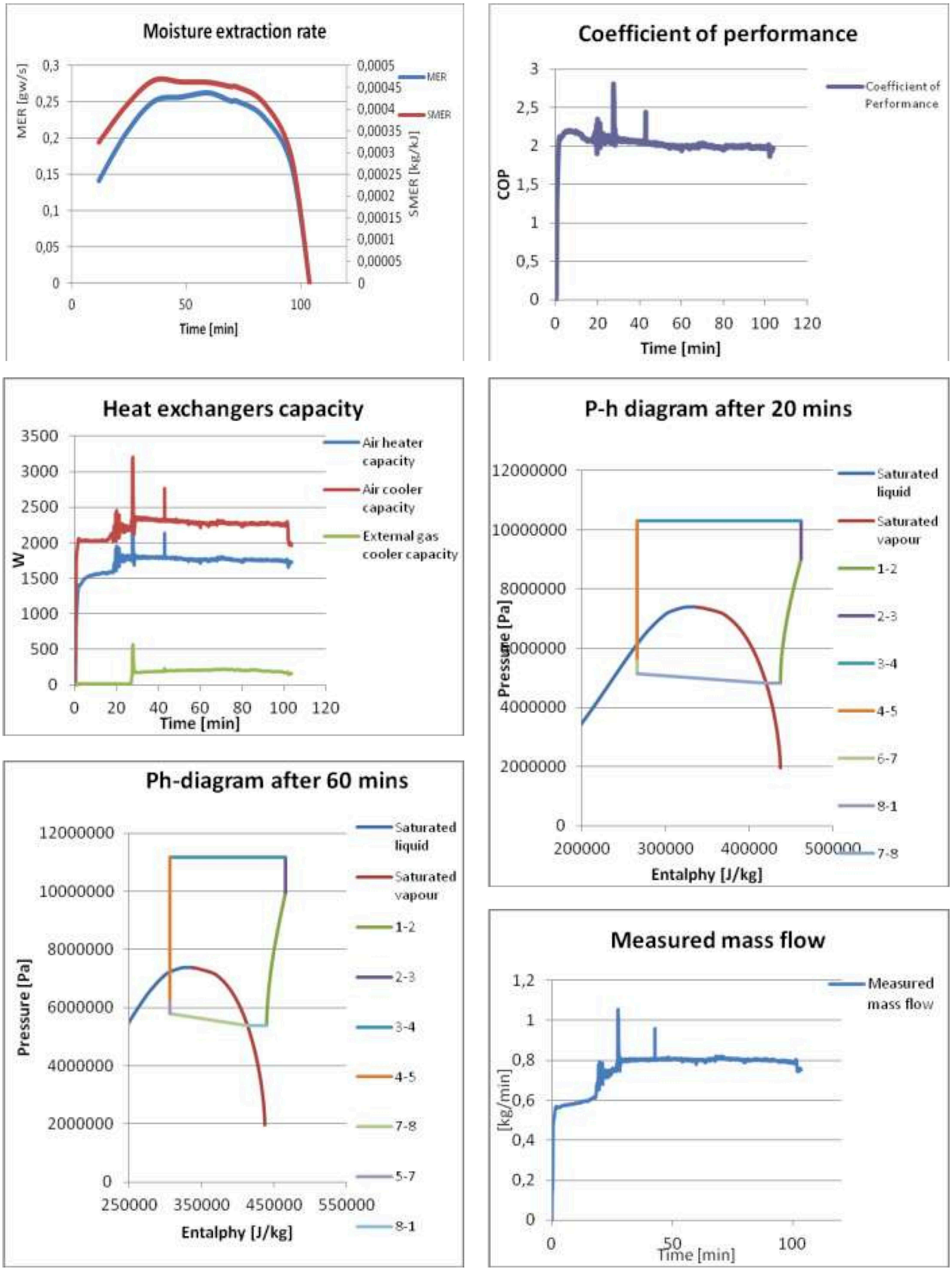

Figure 51: Results experiment 12

Table 20: Other results experiment 12

Other results		
Energy consumption:	1,86	kWh
Consumption per kg textile	0,37	$kWh/kg_{textile}$
Air leakage	6,13	%
Average COP	2,06	
Energy savings	0,51	
Average water extraction	0,026	kg/min
Moisture content end	0,79	%
Standstill pressure before run	59,5	barg
Weight dry textiles	5,06	kg
Weight incl. 60% moisture	8,10	kg
Weight after experiment	5,10	kg
Mass flow high-pressure side	0,0126	kg/s average

The setup of this experiment is similar to experiment 10, but the refrigerant charge is increased by 50g. The valve at the intermediate pressure stage is closed and the low-pressure stage is controlled to achieve a superheat between 5 and 10 K from the evaporator. The setup and results are almost identical to those of experiment 10, and proves that the results are possible to reproduce.

11. Discussion of Experimental Results

The average of the most important values from each experiment are included in Table 21 and Table 22.

Table 21: Results summary

Experiment #	Description	m_ref ave [kg/s]	Energy consump. [kWh]	Water removed [kg]	Energy per kg text. [kWh/kg]	Energy per kg water removed [kWh/kg]
1	Closed valve	0,0098	1,84	2,09	0,53	0,88
2	Open valve	0,0127	2,32	3,232	0,45	0,72
3	Long cap.tube, high charge	0,0098	1,23	1,82	0,24	0,68
4	Long cap.tube, closed valve	0,0051	1,79	2,582	0,35	0,69
5	Long cap.tube, Variable valve	0,0063	2,31	2,586	0,45	0,89
6	Long cap.tube, open valve, low charge	0,0097	1,83	1,674	0,36	1,09
7	Long cap.tube, throttle at IM P	0,0080	1,15	1,44	0,23	0,80
8	Closed valve increasing charge	0,0043	1,49	1,716	0,29	0,87
9	Closed valve, high charge	0,0086	1,73	2,818	0,34	0,61
10	Closed valve, hand-regulated valve	0,0112	1,87	2,57	0,37	0,63
11	Open valve, hand regulated valve	0,0083	2,04	2,71	0,41	0,70
12	Closed valve, hand-regulated valve, high charge	0,0126	1,86	2,996	0,37	0,61

Table 22: Results summary

Experiment #	Q_evap ave [W]	Q_gc ave [W]	Moisture cont. [%]	COP ave
1	1418,74	1470,27	-0,86%	1,97
2	1599,52	1711,87	-2,51%	1,99
3	1564,35	1427,78	24,31%	1,81
4	904,00	1071,30	9,57%	1,26
5	1466,08	941,20	9,39%	1,11
6	2130,95	950,04	27,43%	1,06
7	1373,93	1215,47	31,76%	1,32
8	894,97	651,78	26,42%	0,64
9	1430,86	1488,02	5,07%	1,80
10	2036,22	1654,93	0,60%	2,03
11	1440,24	1387,14	1,80%	1,64
12	2065,48	1696,76	0,79%	2,06

Experiments have earlier been executed on similar facilities. The results from these experiments are useful in order to compare with the results obtained from the experiments in this study. The results from the best experiments from different actors are given in Table 23.

Table 23: Overview results for comparison

Best results from	SEC textiles [$kWh/kg_{textiles}$]	SEC water [$kWh/kg_{removed\ water}$]
Me	0.37	0.61
Goal (R134a)	0.26	0.47 (calculated value)
Elnan	0.27 (calculated value)	0.534
Single	0.32	0.56
ASKO (CO_2)	0.39	N/A
ASKO (R134a)	0.31	N/A

11.1. Moisture extraction

The correlation between extracted moisture and energy consumption is displayed in Figure 52. As shown in the figure the lowest energy consumption is achieved in experiment 3 and experiment 7. Both these experiments ended too soon and the textiles were not sufficiently dry. Hence these results are misleading. It is desired to use as little energy as possible and still remove 100% of the moisture.

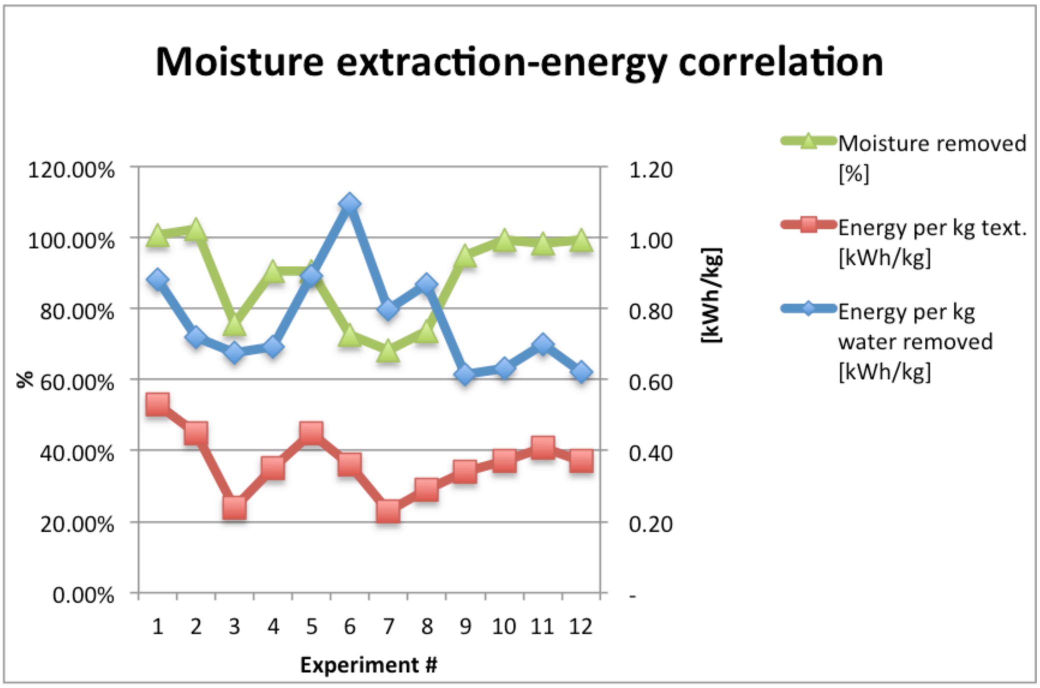

Figure 52: Correlation between moisture removed from textiles and energy consumption

The moisture content after the experiment is not equal for each experiment. This makes the specific energy consumption per kg load an inaccurate measurement of performance. However, the amount of removed water is known. The specific energy consumption per kg water removed is a more accurate measurement of the dryer's performance in this case. The red curve in Figure 52 displays this parameter. Experiment 3 and 7 consumed very little energy to remove the water, but a lot of the water was still present in the end. The last few per cents of moisture content take most energy to remove, so this is not a surprising result. The last four experiments performed best overall, in particular experiment 10 and 12. They both achieved a COP above 2 and good specific energy consumption values. The best specific energy consumption based on mass water removed was 0.61 $kWh/kg_{removed\ water}$. This value was achieved in both experiment 9 and 12, but experiment 9 had higher moisture content after the experiment. This result is about 16% above the average value that Elnan [22] got in his experiments and about 38% more than a similar system would have used with R134a, calculated based on a SEC of 0.26 kWh/kg$_{\text{dry textiles}}$.

11.2. COP and superheat

The COP is related to the gas cooler performance, which in turn is related to the mass flow. A higher mass flow increases the amount of transferred heat, but does also require more work from the compressor. The measured mass flows in the experiments were all lower than the calculated value of 0.0161 kg/s. The introduction of a longer capillary tube forced the flow in the correct direction at intermediate pressure stage, but the mass flow and COP was generally reduced. A lower mass flow will cause a higher superheat out of the evaporator. Less CO_2 can evaporate and the evaporated refrigerant will be superheated before it enters the compressor. This is considered inefficient because evaporation occurs at a lower temperature than superheating, thus more heat can be transferred. The refrigerant is also meant to cool the compressor, which of course is more effective the colder the refrigerant is. Ideally the superheat could be 0 K, but usually the refrigerant is superheated 5-10 K to make sure that all the refrigerant evaporates. Figure 53 displays the relation between superheat and COP. There is a solid correlation between superheat and COP. Experiment 2, 10 and 12 stand out as the most efficient experiments, they all have low average superheat and high COP. The common denominator for these experiments is that they were run with a closed valve at the intermediate pressure.

The worst performing experiments were 5,6 and 8. Experiment 8 started with a very low refrigerant charge, which led to very poor heat exchanger performances. Experiment 5 and 6 used a variable valve and an open valve at intermediate pressure respectively. In experiment 5 the flow is throttled into the two-phase area, which is neither effective nor healthy for the compressor. In experiment 6 the valve is open and the refrigerant flows in the opposite of what is desired. This leads to a massive loss in the system and the COP is equivalent low.

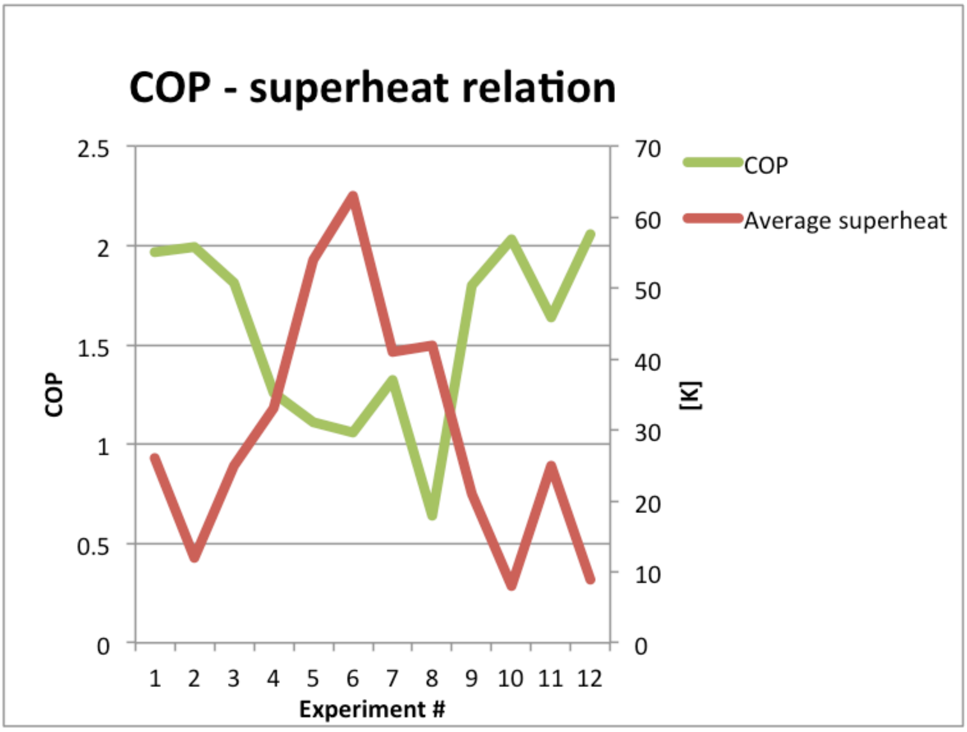

Figure 53: COP-superheat relation

11.3. Assessment of new compressor and external heat exchanger

A new two-stage compressor is installed in the system. This compressor enables recycling of flash gas, and should generally be more efficient than the previous single-stage compressor due to higher isentropic and volumetric efficiency. However, the energy consumption is not lower than previous results. With a closed valve at the intermediate pressure, the system setup is not very different from what Single [23] used in his experiments. The main difference is the separation tank and the two-stage throttling. Single's results are 19% better with respect to energy consumption based on mass of the load and 8.9% better with respect to energy consumption based on mass water removed.

In the experiments performed in this study it has been aimed to cool the refrigerant down to 40°C in the external heat exchanger before throttling. This has been achieved without difficulties by using a third of the external heat exchanger and turning on the fan when necessary. For commercial purposes it would be beneficial to include a smaller external heat exchanger. A smaller heat exchanger would both save space and be cheaper than the current heat exchanger.

12. Control Strategy

A proper control strategy is essential for the system to operate efficiently without the need of constant monitoring. There are several elements that require controlling, an important one being the drying time. During the tests the system has been shut down when humidity of the exit air from the drum is sufficiently low. This can be done automatically by using a humidity receiver. However, this is an expensive device. A simpler method is to use a pre-set drying time for a given weight of the load. This requires thoroughly testing of the drum dryer with various loads, but is a cheaper solution per unit sold. A possible side effect of this solution is reduced energy efficiency or moist clothes due to too long or too short drying time respectively.

The drum inlet temperature should be limited to avoid damaging the textiles. This could be controlled by a small external heat exchanger in front of the gas cooler with a fan that turns on when the temperature is too high. During the experiments in this study the drum inlet temperature has not exceeded 60°C, which should be all right for most textiles.

The gas cooler pressure should not exceed 130 bara, which is the maximum pressure the compressor can produce safely. The pressure can be controlled by a pressure-limiting valve if there is a chance that the system will work with pressures close to the limit over time. If the system is designed to work on a much lower pressure, the pressure will only exceed the limit if there is something wrong. In this case a pressure safety valve would be sufficient. This will open if the pressure gets too high and discharge refrigerant into the surroundings, thus reducing the internal pressure.

The pressure levels could be controlled to achieve the desired superheat from the compressor. This will be very costly and advanced both in terms of throttling devices and the control of them. Capillary tubes are cheap and demands no control if they are designed correctly. However, calculating the optimal tube length is difficult and variable throttle valves should be used to find this length.

13. Conclusion

The main purpose of this study is to introduce injection of flash gas into the compressor at the intermediate pressure stage. This process has proved to be hard to control. In some experiments hot refrigerant flows from the compressor's 1st discharge into the separation tank. This is waste of energy and reduces the system's efficiency greatly. Increasing the pressure in the separation tank relative to the compressor's 1st discharge pressure can turn the direction of the flow. In this study this was executed by increasing the length of the capillary tube between the separation tank and the evaporator. This led to lower mass flow through the evaporator followed by excessive superheat. An option is to reduce the length of the capillary tube from the gas cooler to the separation tank instead, but this could introduce other challenges like supercritical separation tank pressure and too high gas cooler pressure. While using the current setup there are more negative effects from increasing superheat than positive effects from recycling the flash gas. The result is that the system works better without the flash gas recycling. More work should be done to recycle the flash gas whilst keeping the evaporator superheat between 5 and 10 K.

The secondary goal of this study is to investigate the influence of a two-stage compressor rather than a single stage compressor. The overall results from the experiments are not better than previous studies. Single consumed 19% less energy based on mass of the load and 8.9% less energy based on mass water removed. These results are compared for the case when the valve is closed at the intermediate pressure stage. The energy consumption per kg dry textiles is 5% less than ASKO got in their tests, but the energy consumption per kg removed water is 15% higher than ASKO[2]. The new two-stage compressor has not had a big influence on the energy consumption.

The best results from these experiments are 0.61 $kWh/kg_{removed\ water}$ and 0.37 $kWh/kg_{dry\ textiles}$. These results were achieved by running the system without injecting the flash gas at intermediate pressure. The best result from an experiment where the

[2] The results from ASKO are included in appendix B

flash gas injection process was included consumed 0.41 $kWh/kg_{dry\ textiles}$ and 0.70 $kWh/kg_{removed\ water}$. Compared to the case where the flash gas recycling is disabled, this experiment consumes 14.7% more based on the weight of the load and 10.8% more based on the weight of the removed water.

Experiments 10 – 12 were executed using a manually adjustable throttle valve before the evaporator. This feature made it easy to control the process during the drying cycle. The results from these experiments were generally better than earlier experiments. The possibility to keep the superheat low throughout the cycle contributed to improve the efficiency by a great deal.

14. Suggestion of Further Work

Further experiments can be executed including the adjustable throttle valve with some additional modifications to the system. The length of the capillary tube from the gas cooler to the separation tank could be reduced. This way the pressure in the tank would most likely increase and the refrigerant would easier flow into the compressor's 2nd suction inlet. With this setup the adjustable throttle valve could adjust the superheat without reversing the flow at the intermediate pressure. However, there is a chance that other problems could arise. Possible challenges could be too high gas cooler pressure and supercritical separation tank pressure. An option is to replace the remaining capillary tube by an adjustable throttle valve as well. This would make it possible to try out several pressure levels before determining suitable capillary tube lengths.

In order to make the results more comparable, all experiments could be run for an equal amount of time. If all experiments were stopped after e.g. 100 or 120 minutes comparing would be easy and self-explaining. In this case the industrial partner specifically wanted the experiment to end at exactly 0% moisture content, thus the total time varies for each experiment.

For further development, external cooling of the compressor should be considered. The compressor have occasionally stopped because of overheating, and in most experiments external cooling with a wet cloth has been necessary to avoid overheating. A fan or cooling coil would keep the compressor cool and able to operate without the risk of taking any damage from the heat.

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Appendix A: Heat exchanger properties

Table 24: Main gas cooler specifications [22]

Main gas cooler					
Main dimensions		**unit**	**Tube bundle and lamellas**		**Unit**
Core length	0,302	m	Tube diameter(s)	7,00/5.00	mm
Finned tube length	0,248	m	Fin thickness	1,2	mm
Core height	0,085	m	Fin spacing	2,69	mm
Core depth	0,204	m	Fin material	Aluminium	
Air side area	2,91	m2	Tube material	Copper	
Tube inner area	0,351	m2	Tube arrangement	Staggered up	
Area ratio	10,37	-	Number of vertical tubes	6	
			Vertical tube pitch	13	mm
Total tube length	24,74	m	Number of horizontal tubes	12	
			Horizontal tube pitch	17	mm

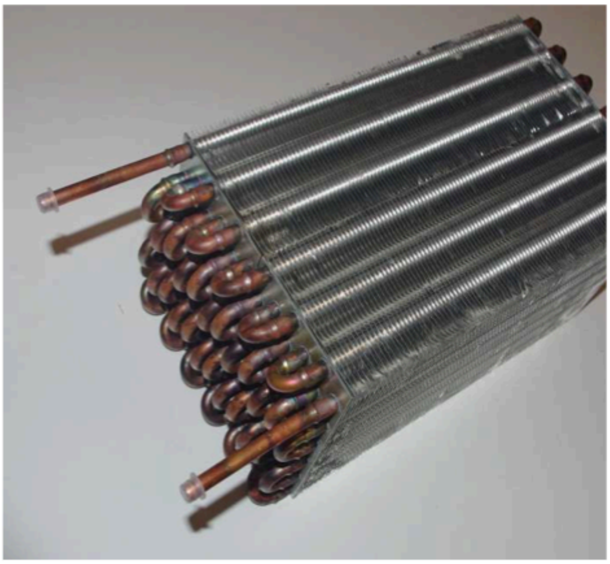

Figure 54: Main gas cooler [22]

Table 25: Evaporator specifications [22]

Evaporator					
Main dimensions		unit	**Tube bundle and lamellas**		Unit
Core length	0,302	m	Tube diameter(s)	7,00/5.00	mm
Finned tube length	0,25	m	Fin thickness	0,15	mm
Core height	0,095	m	Fin spacing	2,69	mm
Core depth	0,136	m	Fin material	Aluminium(coated)	
Air side area	2,3	m^2	Tube material	Copper	
Tube inner area	0,245	m^2	Tube arrangement	Staggered down	
Area ratio	12,19	-	Number of vertical tubes	6	
			Vertical tube pitch	15,75	mm
Total tube length	14,5	m	Number of horizontal tubes	8	
			Horizontal tube pitch	17	mm

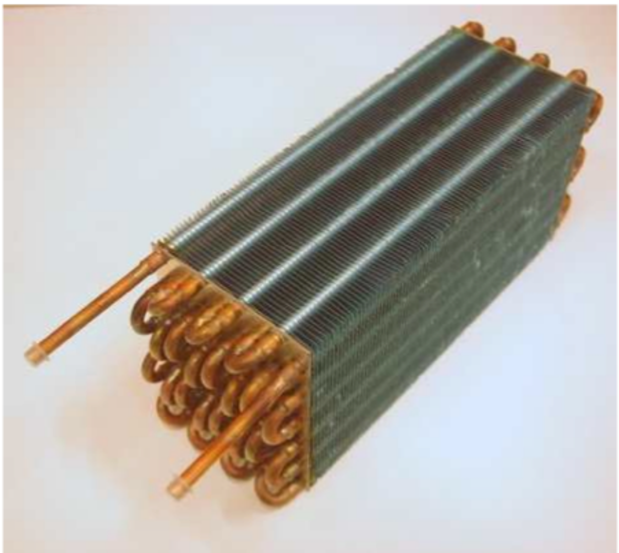

Figure 55: Evaporator [22]

MAIN DIMENSIONS:		TUBE BUNDLE AND LAMELLAS:	
Core length	: 0.116 m	Tube diameter(s)	: 7.20 / 5.80 mm H
Finned tube length:	0.064 m		: 7.20 / 5.80 mm V
Core height	: 0.063 m	Fin thickness	: 0.12 mm
Core depth	: 0.152 m	Fin spacing	: 3.88 mm
Air side area	: 0.31 m²	Fin material	:Aluminum
Tube inner area	: 0.098 m²	Tube material	:Copper
Area ratio	: 7.42 -	Tube arrangement	:Staggered_down
		Number of vertical tubes	: 3
Core weight	: 0.895 kg	Vertical tube pitch :	21.00 mm
Tube weight	: 0.853 kg	Number of horizontal tubes	: 12
Fin weight	: 0.042 kg	Horizontal tube pitch:	12.70 mm

Figure 56: External gas cooler specifications [24]

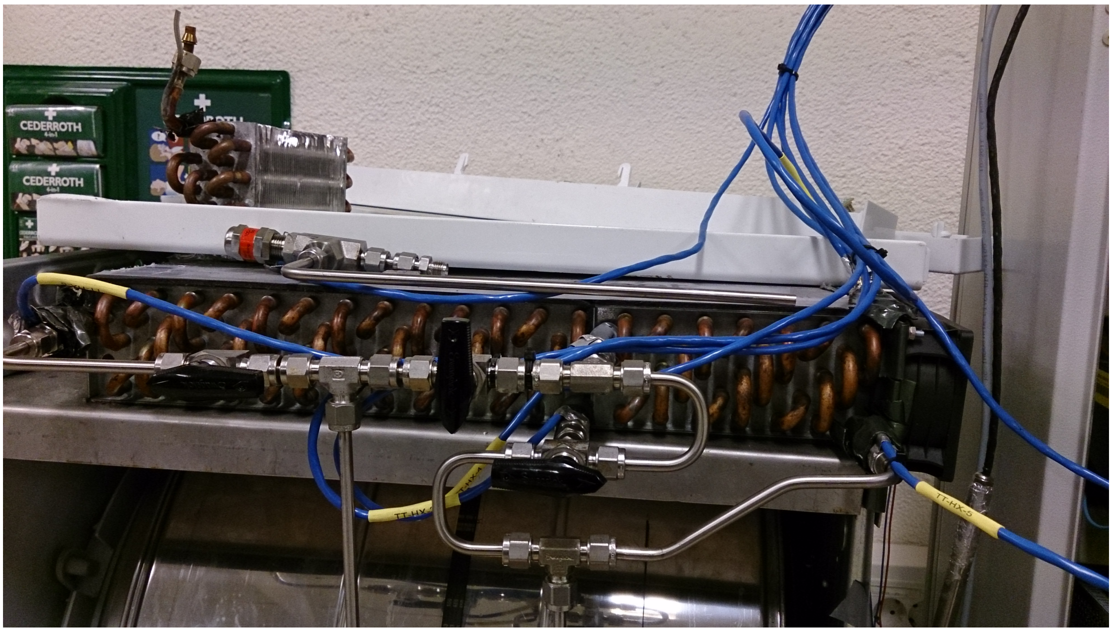

Figure 57: External gas cooler

Appendix B: Test results ASKO R134a

Test results from measured data of the R134a system from the manufacturer, which shows the measured data from the testing on the previous system. 4 experiments have been performed on this system.

TORKTUMLARTEST TD70 prototyp

Program			
	Bomull	Syntet	Akryl
	P 1 Extra torrt	Temp °C	Normal
	P2 Skåp torrt		Låg
	P3 Normal torrt		
	P4 Stryktorrt	Tidsprogram	

Klimat % RH	55	Temperatur °C	23
Tolerans	± 5	Tolerans	± 2
Serienr.	Prototypserie 1-1		
Int.nr.	13051	Luftflöde (m^3/h)	
Provserie	788	Elementeffekt (W)	

Nominell vikt, W (g)	8000	Nominell fukt start, μ_{i0}	60 %	Nominell fukt slut, μ_{f0}	0 %
Vikt tom kondensvattentank (g)	325				
Konditionerad vikt, W_0 (g)	7991				

Datum	2009-08-17	2009-08-17	2009-08-18	2009-08-18						
Starttid	10,15	13,30	8,40	11,55						
Körning nr.	**10072**	**10073**	**10074**	**10075**					Medelvärden	
Centrifugerad vikt, W_i (g) Starting load + w ater	12786	12786	12786	12786					12786	
Torkad vikt, W_f (g) Dry Load after drying	8099	8090	8249	8239					8169	
Programtid, t_m (min)	145	145	145	145					145	
Energiförbrukning, E_m (kWh) Energiconsumption	2,110	2,118	2,035	2,091					2,089	
Vikt kondensvattentank slut (g) Endgew icht Kondensatbehälter	4271	4315	4162	4139					4222	
	Resultat									
Returvatten i tank, W_w (g) Return Water	3946	3990	3837	3814						
Fuktinnehåll Start, μ_i (%) Feuchtigkeitsgehalt Start	60,0%	60,0%	60,0%	60,0%					60,0%	
Fuktinnehåll Slut, μ_{fi} (%) Feuchtigkeitsgehalt Ende	1,4%	1,2%	3,2%	3,1%					2,2%	μ (%)
Förbrukning kWh / liter borttorkat vatten Verbrauch kWh / Liter Wasser zu entfernen getrocknet	0,45	0,45	0,45	0,46					0,45	
Kondenseringseffekt, C (%)	84 %	85 %	85 %	84 %					84 %	
	Korrigerrade resultat till nominell fukt									
Korrigerad energiförbrukning, E (kWh)	2,16	2,16							2,16	
Korrirerad programtid, t (min)	148	148							148	
	Korrigerade resultat till EN61121:2005									
Korrigerad energiförbrukning, E_{corr} (kWh)	2,46	2,47							2,47	
Korrirerad energiförbrukning (kWh/ kg gods)	0,31	0,31							**0,31**	
Energiklass									**A**	

Figure 58: Test results R134a, ASKO

Appendix C: Test results ASKO CO_2

Test results from measured data on the CO_2 system; the table shows the drying and energy use from the given input for the 8 experiments. They are not directly comparable due to that there are changing factors in the 8 different experiments.

TORKTUMLARTEST TD60.3 Kondens

Program			
	Bomull	*Syntet*	*Akryl*
	P 1 Extra torrt	Temp °C	*Normal*
	P2 Skåp torrt		*Låg*
	P3 Normal torrt		
	P4 Stryktorrt		

Klimat % RH	55	Temperatur °C	22
Tolerans	± 5	Tolerans	± 2
Serienr.	Prototype 1		
Int.nr.		Luftflöde (m^3/h)	
Provserie		Elementeffekt (W)	

Nominell vikt, W (g)	6316	Nominell fukt start, μ_{i0}	60 %	Nominell fukt slut, μ_{f0}	0 %
Vikt tom kondensvattentank (g)	0				
Konditionerad vikt, W_0 (g)	**6316**				

Datum	2011-08-11	2011-09-11	2011-11-11	2011-15-11	2011-17-11	2011-18-11	2011-23-11	2011-24-11	
Starttid	10:30	12:00	09:00	13:00	14:00	10:30	09:00	12:30	
Körning nr.	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	Medelvärden
Centrifugerad vikt, W_i (g)	9500	9500	9500	9972	9500	9908	9500	9810	9649
Torkad vikt, W_f (g)	6310	6326	6315	6328	6323	6362	6325	6332	6328
Programtid, t_m (min)	126	115	100	116	116	116	100	110	112
Energiförbrukning, E_m (kWh)	1,96	1,99	1,89	2,00	1,87	1,87	1,69	1,85	1,890
Vikt kondensvattentank slut (g)	1970	1928	2247	2537	1611	2314	1988	2094	2086
	Resultat								
Returvatten i tank, W_w (g)	1970	1928	2247	2537	1611	2314	1988	2094	
Fuktinnehåll Start, μ_i (%)	50,4%	50,4%	50,4%	57,9%	50,4%	56,9%	50,4%	55,3%	52,8%
Fuktinnehåll Slut, μ_{fi} (%)	-0,1%	0,2%	0,0%	0,2%	0,1%	0,7%	0,1%	0,3%	0,2% μ (%)
Förbrukning kWh / liter borttorkat vatten	0,61	0,63	0,59	0,55	0,59	0,53	0,53	0,53	0,57
Kondenseringseffekt, C (%)	62 %	61 %	71 %	70 %	51 %	65 %	63 %	60 %	63 %
	Korrigerade resultat till nominell fukt								
Korrigerad energiförbrukning, E (kWh)	2,33	2,38	2,25	2,08	2,23	2,00	2,02	2,02	2,16
Korrigerad programtid, t (min)	150	137	119	121	138	124	119	120	129
	Korrigerade resultat till EN61121:2005								
Korrigerad energiförbrukning, E_{corr} (kWh)	2,65	2,71	2,56	2,37	2,54	2,28	2,30	2,30	2,46
Korrigerad energiförbrukning (kWh/ kg gods)	0,42	0,43	0,41	0,38	0,40	0,36	0,36	0,36	**0,39**
Energiklass									**A**

Appendix D: Draft Scientific Paper

Development of an energy efficient and environmentally friendly drum dryer using a heat pump with CO_2 as working fluid

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ABSTRACT

Tumble dryers for household use are in constant development. Heat pumps are now common in conventional tumble dryers. The next step is to use natural refrigerants, which are more environmental friendly. The heat pump will operate in a transcritical cycle where heat is transferred by cooling of supercritical CO_2. In the evaporator heat from hot humid air is absorbed to evaporate the refrigerant. The refrigerant is throttled in two steps, which makes it possible to inject flash gas into the compressor at an intermediate pressure stage. This process has been challenging to control and has not increased the efficiency of the system in the current setup. However this matter should be researched more and some modifications to the system have been suggested.

1. INTRODUCTION

Drum dryers have been used for a long time in homes for drying of laundries. These dryers have been developed over years from the simple type with direct electric heaters and rejection of the humid air to the ambient. The next generation included a heat exchanger between the inlet air and the exit air from the drum (condensation units). In this case it is possible to reduce the electric consumption. More modern system have been developed with a heat pump for cooling of the air to a temperature below the dew point (condensation and removal of water) and then reheating of the inlet air to the drum dryer. In this case we have a closed loop of the air in the dryer. Typical working fluid in this type of dryer is R134a. This refrigerant has a GWP factor of 1300. The industry of these type of dryers like to reduce the environmental impact factor of their systems and have looked into using CO_2 as the working fluids in the heat pump system. CO_2 is a natural refrigerant with a GWP of 1. In this project it is used a two-stage compressor with a liquid separator at the intermediate pressure.

2. SYSTEM PRINCIPLE AND DESIGN

CO_2 is a natural refrigerant, which means that it is already present in the atmosphere. The Global-Warming Potential (GWP) is a relative measure of how much heat the greenhouse gas traps in the atmosphere. CO_2 is the reference with a GWP of 1, while the conventional refrigerant R134a has a GWP of 1300. The Ozone Depletion Potential (ODP) is zero for both R134a and CO_2. The biggest difference between these refrigerant concerning the heat pump process it that the heat transfer in the gas cooler is a supercritical process for CO_2. This means that the heat is transferred at a constant pressure but at a gliding temperature. R134a delivers heat during condensation at constant pressure and temperature. The gliding temperature curve of CO_2 is actually a better fit to the air heating curve than the condensation curve for R134a. Low-temperature CO_2 is used to heat low-temperature air and high-temperature CO_2 is used to heat high-temperature air. This results in considerably lower exergy losses than for R134a. This can be visualised as the area between the dotted line and the solid-drawn line in the t-h diagrams in Figure 1 and Figure 2 for CO_2 and R134a respectively.

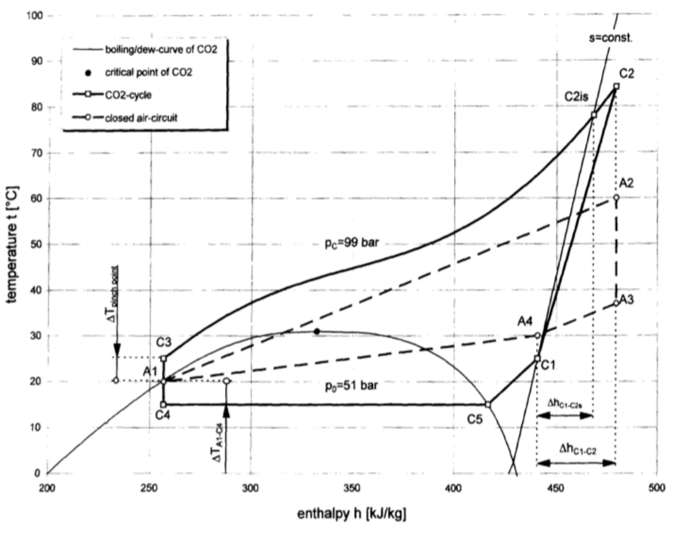

Figure 1: T-h diagram CO2 [1]

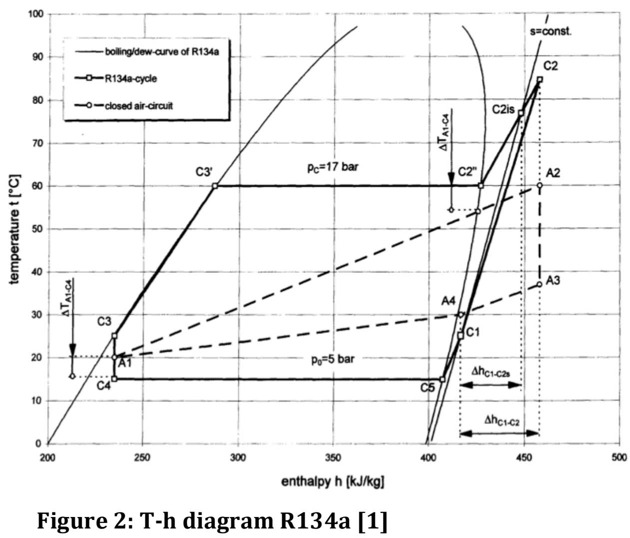

Figure 2: T-h diagram R134a [1]

The process of a tumble dryer suits the heat pump process excellent. The warm side of the heat pump dryer, the gas cooler, heats the circulating air before it enters the drum. The hot, dry air heats and extracts moisture from the clothes. The humid air exits the drum and meets the cold side of the heat pump, the evaporator. The air is then cooled below dew point and the vapour condenses. This results in cold dry air ready to be heated at the gas cooler. The air cycle and refrigerant cycle are combined as explained in Figure 3.

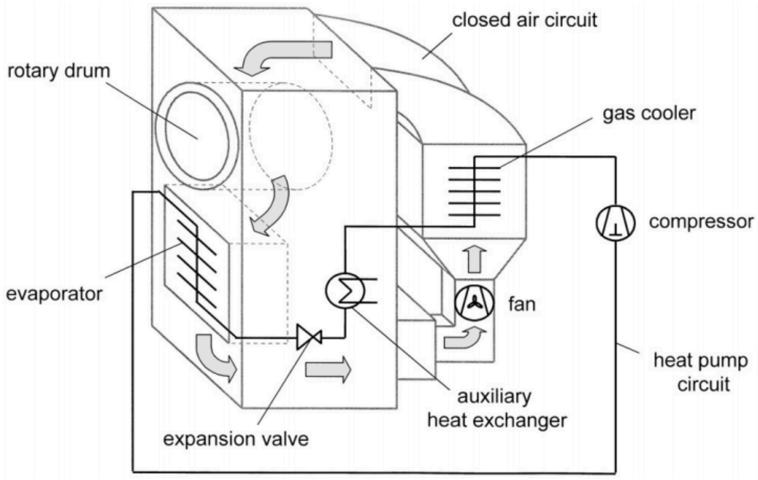

Figure 3: Tumble dryer with heat pump process[2]

Both the air cycle and the refrigeration cycle work as independent closed cycles. A fan drives the air cycle and a compressor drives the refrigeration cycle. The refrigerant is throttled in two steps using capillary tubes. This enables the possibility to separate and inject the flash gas into the compressor. The separation process occurs in a small tank, the flash gas exits through the top while the liquid flows from the bottom of the tank to the next capillary tube. The process chart is included in Figure 4. The two-stage compressor combined with the expansion tank makes up a quite complicated process. A block valve at the intermediate pressure stage enables the opportunity to run without this process. This is useful in order to determine whether the flash gas injection increases the efficiency of the system.

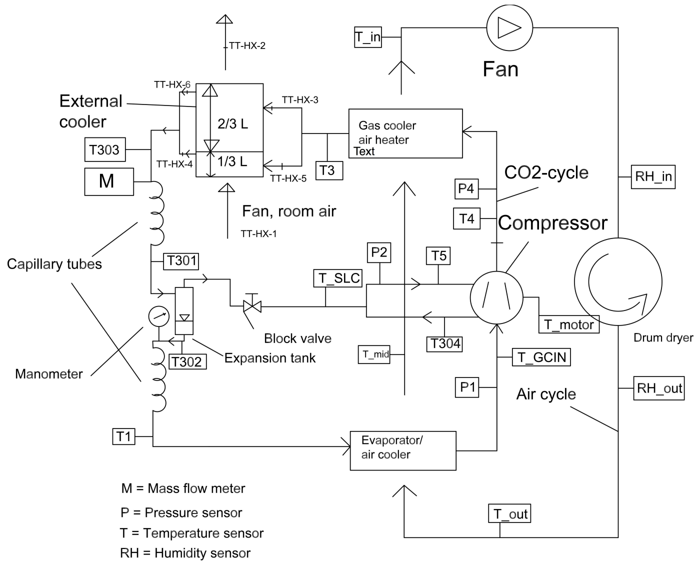

Figure 4: Process chart

An external gas cooler is included in the system to reduce the temperature of the refrigerant before it is throttled. When the refrigerant is throttled, the liquid fraction will be larger due to the extra cooling. The ph-diagram of the process is included in Figure 5. When the intermediate pressure stage-process is used, the mass flow through the evaporator increases with lower temperature before throttling. If the blocking valve at intermediate pressure stage is closed, the refrigerant will have a lower enthalpy when it enters the evaporator. In both cases the evaporator performance increases with lower temperature before throttling.

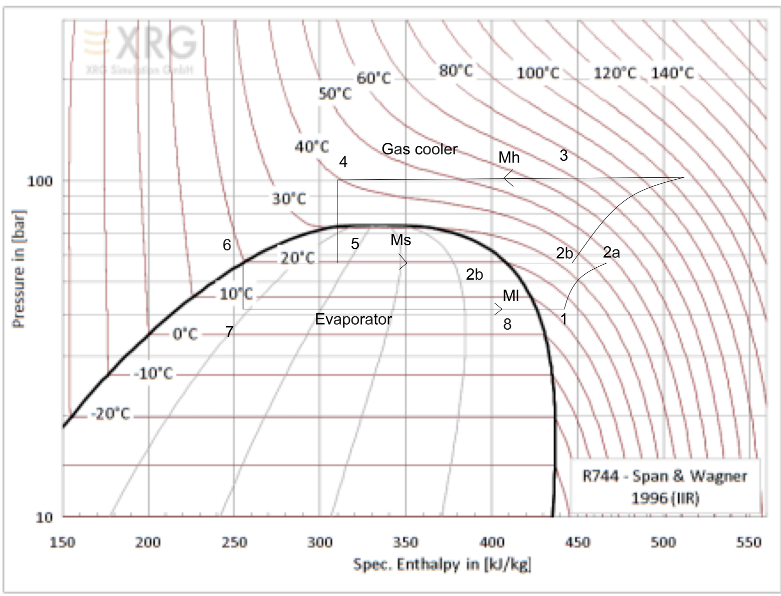

Figure 5: Ph-diagram of the process

3. EXPERIMENTAL METHOD AND SETUP

Temperature, pressure and humidity sensors are placed at appropriate points in the system. All measurement points are included in the process diagram in Figure 4. The properties are monitored during the experiments and the data are processed after the experiment to calculate ph-diagrams, COP, heat exchanger performance and more. There are a few modifications that have been made to the system between the experiments to produce some comparable results.

- Open/closed blocking valve at the intermediate pressure stage
- Two different lengths of capillary tube between expansion tank and evaporator
- Hand-regulated throttle valve between expansion tank and evaporator
- Varying amount of refrigerant charge

The results should be compared with similar experiments using R134a. The goal is to consume 0.26 $kWh/kg_{textiles}$ or less. ASKO has run some experiments where the system consumed 0.31 $kWh/kg_{textiles}$ for R134a and 0.39 $kWh/kg_{textiles}$ for CO_2.

4. RESULTS AND DISCUSSION

The most important results are given in Table 1. The description of each experiment is kept short to save some space. The first two experiments are executed using a capillary tube of length 32.7 cm between the expansion tank and the evaporator. This is then enlarged to 40.0 cm, which is referred to as "long cap.tube" in Table 1. The open/closed valve refers to the blocking valve at intermediate pressure. In the last few experiments the capillary tube between the expansion tank and the evaporator is substituted by a hand-regulated throttle valve.

Table 1: Results summary

Exp. #	Description	m_ref ave [kg/s]	Energy per kg water removed [kWh/kg]	Moisture content [%]	Energy per kg text. [kWh/kg]	COP ave.
1	Closed valve	0,0098	0,88	-0,86%	0,53	1,97
2	Open valve	0,0127	0,72	-2,51%	0,45	1,99
3	Long cap.tube, high charge	0,0098	0,68	24,31%	0,24	1,81
4	Long cap.tube, closed valve	0,0051	0,69	9,57%	0,35	1,26
5	Long cap.tube, Variable valve opening	0,0063	0,89	9,39%	0,45	1,11
6	Long cap. tube, open valve, low charge	0,0097	1,09	27,43%	0,36	1,06
7	Long cap.tube, throttle at intermediate pressure	0,0080	0,80	31,76%	0,23	1,32
8	Closed valve increasing charge	0,0043	0,87	26,42%	0,29	0,64
9	Closed valve, high charge	0,0086	0,61	5,07%	0,34	1,80
10	Closed valve, hand-regulated valve	0,0112	0,63	0,60%	0,37	2,03
11	Open valve, hand regulated valve	0,0083	0,70	1,80%	0,41	1,64
12	Closed valve, hand-regulated valve, high charge	0,0126	0,61	0,79%	0,37	2,06

Experiments 9,10 and 12 stand out as the most efficient experiments based on specific energy consumption and COP. Some of the earlier experiments achieve a lower specific energy consumption based on weight of load, but they finish before all the moisture is removed from the textiles. Hence they are inaccurate measurements of performance.

The relation between the percentage of moisture removed and specific energy consumption based on both the weight of the load and weight of water removed is given in Figure 6. It is quite obvious that experiment 9, 10 and 12 performed well in these three categories.

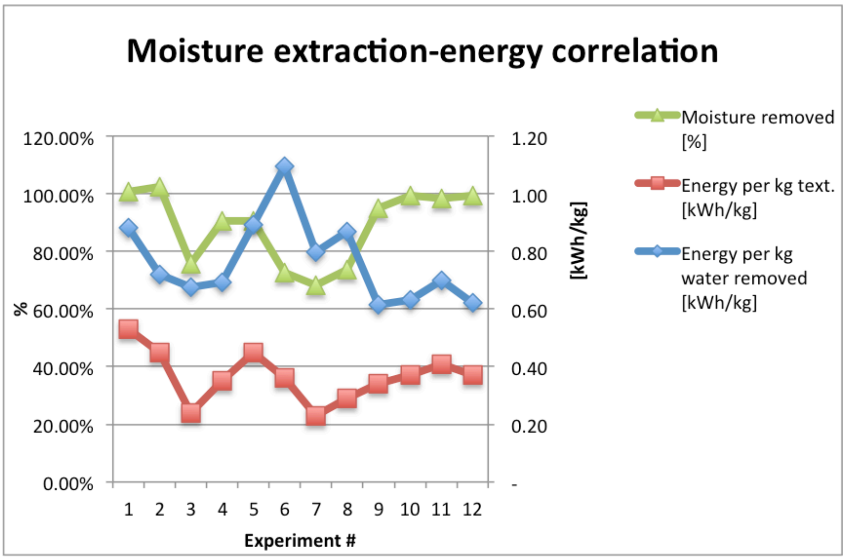

Figure 6: Correlation between moisture removed from textiles and energy consumption

A key element in order to achieve high efficiency is the superheat out of the evaporator. This is a direct correlation that can be concluded from Figure 7. The superheat should be limited to 5-10 K. This has only been possible during the last few experiments were it has been possible to regulate the throttle, and hence the superheat, continuously throughout the experiment. Less throttling before the evaporator gives a higher mass flow and a lower superheat. However the process at the intermediate pressure stage requires that the pressure in the expansion tank is higher than the pressure at the compressor's 1st discharge. With the current layout, this can only be achieved by keeping a certain pressure difference between the expansion tank and the evaporator. In experiment 11 the pressure ratio were kept at the limit to force the process in the right direction at the intermediate pressure stage. This led to an average superheat of 25 K and reduced overall efficiency.

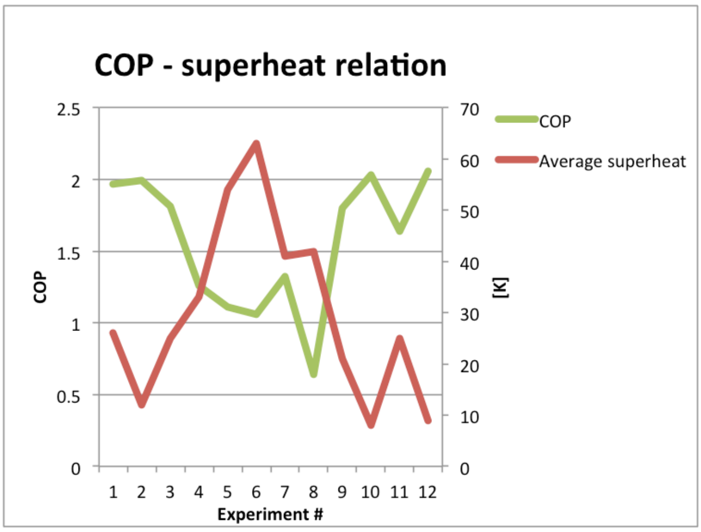

Figure 7: COP-superheat correlation

The current setup makes it difficult to keep the superheat low and at the same time inject the flash gas at the intermediate pressure stage. More studies should be done on this topic. It could be interesting to substitute the first capillary tube by an adjustable hand valve as well. This way different pressure ratios can be investigated and an optimal capillary tube length can be found.

5. CONCLUSION

Thoroughly testing of the heat pump system shows that it is essential to keep the superheat out of the evaporator low. The easiest way to obtain this is to constantly regulate the mass flow through the evaporator. There has been a great challenge to run the process of flash-gas injection at intermediate pressure in the correct direction. It is indeed possible but it increases the evaporator superheat a great deal, which in turn reduces the overall efficiency. Further work should be done to solve this issue, and suggestions have been made to test different pressure ratios at both throttling stages. With the current setup the best experiment consumes 0.37 kWh/kg_{load}. This is 19% more than the set goal of 0.26 kWh/kg_{load}, but 5% less than average result ASKO achieved in their tests.

6. REFERENCES

[1] E. L. Schmidt, K. Klöcker, N. Flacke, and F. Steimle, "Applying the transcritical CO2 process to a drying heat pump," *International Journal of Refrigeration,* vol. 21, pp. 202-211, 5// 1998.

[2] K. Klöcker, E. L. Schmidt, and F. Steimle, "Carbon dioxide as a working fluid in drying heat pumps," *International Journal of Refrigeration,* vol. 24, pp. 100-107, 1// 2001.