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Development of a reversible heating and cooling system using CO_2 as a refrigerant for industrial customers and residential areas with heating and cooling requirements

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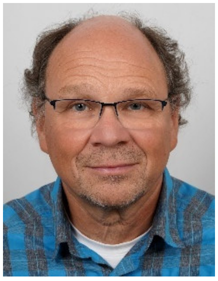

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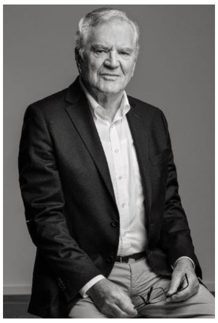

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Abstract

As part of a research and development project, BKW GmbH collaborated with Reutlingen University to develop a prototype reversible CO_2 heating and cooling system ***(CO_2 WP)*** with an output of 20 to 50 kW. enisyst GmbH provided the control and data acquisition system. To optimize the system, created Reutlingen University a digital twin ***(DT)***, which was used to validate the experimental results.

The ***CO_2 WP*** comprises a compressor, evaporator, gas cooler, internal heat exchanger, and expansion valve. To verify its functionality, extensive tests and simulations were conducted to determine the system's coefficient of performance ***(COP)*** and energy efficiency ratio ***(EER)*** in both heating ***(HM)*** and cooling modes ***(CM),*** with varying refrigerant mass flows $\dot{m}_{CO_2}$. The parameters for testing and simulation were set in accordance with DIN EN 14511-2. Additionally, a sensitivity analysis of the heat exchangers (evaporator, gas cooler, and internal heat exchanger) was performed to investigate how variations in the number of plates influence ***COP*** and ***EER*** in both modes.

The results demonstrate that the efficiency of the ***CO_2 WP*** can be significantly improved by adjusting the refrigerant mass flow rate $\dot{m}_{CO_2}$, the CO_2 charge, and by optimizing the number of plates in the heat exchangers. In particular, selecting the optimal number of plates in the gas cooler for heating mode and in the evaporator for cooling mode positively impacts both ***COP*** and ***EER***.

The project aims to enhance the efficiency and reliability of CO_2-based refrigeration and heat pump systems in the transcritical range using digital twin technology. It contributes to the development of expertise in sustainable refrigeration and heating solutions using natural refrigerants, which is especially relevant in the context of the European Unions current F-Gas Regulations 2006/2014. Another key objective is to enable real-time monitoring of similar systems via digital twins and innovative control strategies.

Content

Nomenclature and indices

CO_2 (R744)	Carbon dioxide
CO_2 WP	CO_2 heating and cooling system
DT	Digital twin
COP	Coefficient of performance
EER	Energy efficiency ratio
HM	Heating mode
CM	Cooling mode
GWP	Global warming potential
R&D	Research and Development
REZ	Reutlinger Energiezentrum - Reutlingen Energy Center
HP	High pressure
LP	Low pressure
EV	Evaporator
GC	Gas cooler
IHE	Internal heat exchanger
$T_{,GC,w,Out}$	Water outlet temperature at the gas cooler
$T_{GC,w,In}$	Water inlet temperature at the gas cooler
$T_{EV,w,Out}$	Water outlet temperature at the evaporator
$T_{EV,w,In}$	Water inlet temperature at the evaporator
$\dot{m}_{GC,w}$	Mass flow of water at the gas cooler
$\dot{m}_{EV,w}$	Mass flow of water at the Evaporator
$\dot{m}_{CO2}$	Mass flow of CO_2
lT	Low temperature
iT	Intermediate temperature
mT	Middle temperature
hT	High temperature

1 Introduction

Due to the negative environmental impact of fluorinated refrigerants and hydrofluoroolefins, there is growing interest in natural alternatives such as R744 (CO_2). CO_2 offers several advantages. It is non-flammable, non-toxic in low concentrations [1], has a global warming potential (GWP) of 1, and exhibits favorable thermodynamic properties in both normal and deep-freeze applications. However, challenges arise when using CO_2 in commercial and residential refrigeration and air conditioning applications due to its low critical temperature (31.1 °C) and high critical pressure (73.8 bar) [1]. These properties make it difficult to operate CO_2 heat pumps and CO_2-based refrigeration systems, especially in moderate to warm climates, a transcritical operation is required. This leads to increased compressor power and higher pressure losses.

An increase in performance and system efficiency in CO_2 applications could be achieved through the targeted use of specialized components that are optimized for CO_2 operation [2]. These include a different components as compressors, heat exchangers, and expansion valves. These components must be specially designed for the conditions of the CO_2 cycle and offer the potential for improved efficiency and lower pressure losses in the refrigeration or heat pump cycle for transcritical applications.

2 Objectives

In the project presented below, BKW GmbH and the Reutlingen Energy Center (REZ) at Reutlingen University have partnered to develop a novel CO_2 heating and cooling system (CO_2 WP) with innovative control technology and a digital twin (DT). The goal is to provide an efficient, sustainable, and economical solution for the simultaneous provision of heat and cooling using natural refrigerants in an industrial context.

This research and development project aims to develop extensive knowledge of the design, operation, and optimization of CO_2-based refrigeration and heat pump systems. The project focuses on improving energy efficiency, operational safety, and scalability in various application areas.

In the context of the current European F-Gas Regulation 2006/2014, the present project is highly relevant as its objective is to reduce industrial sector emissions by using environmentally friendly natural refrigerants, such as CO_2 (R744). The successful development of this system will contribute to a sustainable energy transition and strengthen the industry's competitiveness in environmentally friendly refrigeration technology. Moreover, the integration of such technologies into future smart energy solutions is facilitated, aiding in the realisation of climate targets.

3 Methods

As part of the R&D project, BKW GmbH developed and built a CO_2 refrigeration unit prototype (CO_2 WP). The project's objective was to provide heating and cooling in the 20 to 50 kW power range. The control system, data acquisition, and control of the CO_2 WP were integrated and supplied by Enisyst GmbH, a subcontractor.

The functionality of the CO_2 WP was tested in a series of extensive experiments. The primary focus of this study was on process stability, efficiency, and operating limits under various load conditions.

In order to optimise the CO_2 WP, Reutlingen University developed a digital twin DT. This model is designed to simulate the thermodynamic behaviour of the CO_2 WP, including the operation of the heat exchangers (evaporator, gas cooler, and internal heat exchanger) of this CO_2 WP. With the help of this digital twin, the experimental results were systematically compared with the simulation results. The objective of this comparison was to calibrate model parameters, identify deviations, and increase prediction accuracy. Based on the findings, optimization measures were defined to further improve efficiency and operational stability.

4 CO_2 WP

4.1 Components of the refrigeration circuit (primary side)

The CO_2 WP is a device that performs both cooling and heating functions. The design of the system is composed of various components that are employed in the field of refrigeration technology. The following section will describe the main components of the CO_2 refrigeration cycle. However, a detailed explanation of the heat source and heat sink is not provided.

In the heating or cooling mode, CO_2 is a cyclically compressed, cooled, and vaporized again. The CO_2 WP is composed of the following components:

- The compressor in the CO_2 WP increases the pressure of the CO_2 gas to the desired level, allowing it to circulate through the system.
- The gas cooler plays a pivotal role in the refrigeration cycle by facilitating the cooling of the high-pressure HP gas after compression, before it re-enters the evaporator.
- The expansion valve constitutes a significant component of the system, as it functions to reduce the elevated pressure of the CO_2-gas to the evaporator pressure (LP), thus permitting controlled evaporation.

- The evaporator absorbs heat from the heat source circuit (in the case of heating) or releases it (in the case of cooling), depending on the operating mode.
- The utilisation of an internal heat exchanger ensures that the evaporating of CO_2 is subjected to sufficient superheating to ensure stable operating conditions.
- The liquid separator functions to eliminate liquid droplets that have been agglomerated in the CO_2-gas phase,ensuring optimal efficiency of the CO_2 WP.

In heating mode, heat is transferred to the target area, while CO_2 is compressed, heated to a high temperature, and cooled in the gas cooler in order to transfer heat to the desired room or medium.

In cooling mode, heat is absorbed from the target area, CO_2 vaporizes in the evaporator at low pressure, and transfers heat to the water side (secondary side), thereby cooling the target area.

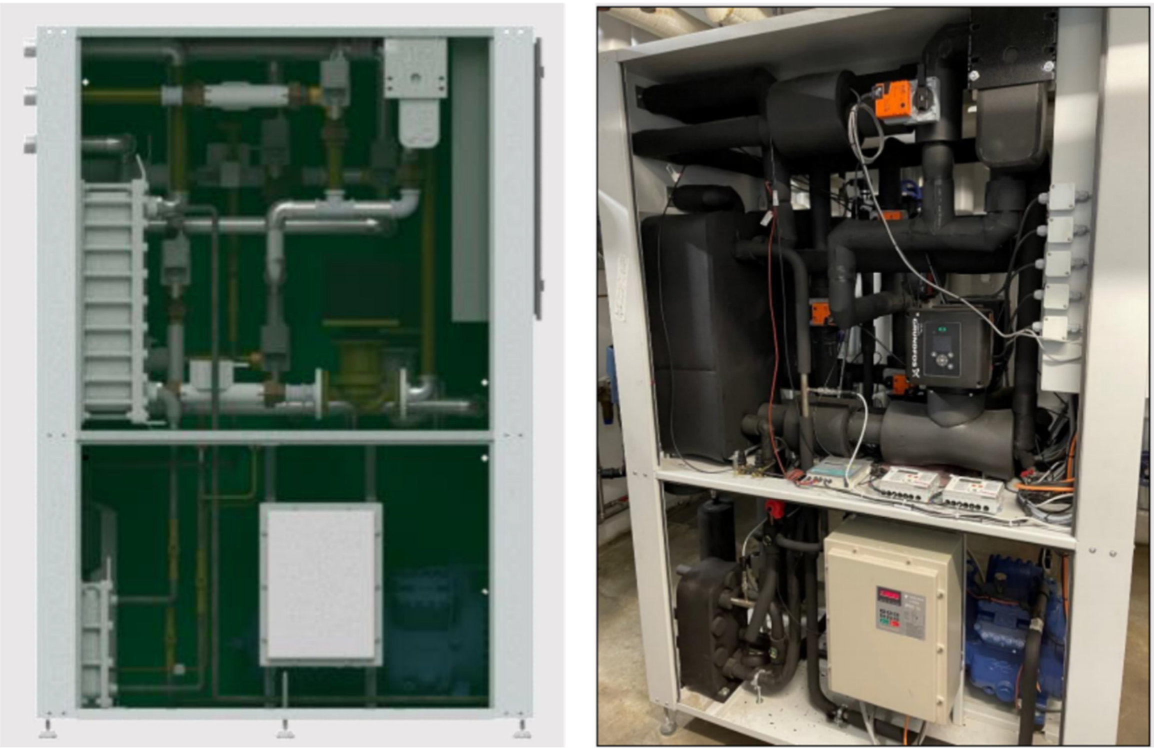

Figure 1: Design of the CO_2 WP

4.1.1 Compressor and Frequency Inverter

BKW GmbH selcted a BOCK compressor from Danfoss. BOCK compressors are regarded worldwide as the first choice for key components in commercial and industrial refrigeration technology, including air conditioning, cooling, heating, heat recovery, and heat pumps.

Danfoss is committed to a sustainable future and uses natural refrigerants such as CO_2 (R744) in transcritical and subcritical applications.

The selected semi-hermetic reciprocating compressor from BOCK has not experienced any failures to date, despite numerous oil pressure faults and temporary operation outside the operating limits (low pressure LP to a minimum of 12 bar). Contrary to the specifications, BOCK has granted special approval for evaporation temperatures down to -5 °C (29.5 bar) at a high pressure HP of 75 to 95 bar.

It is noticeable that the measured thermal and electrical outputs do not correspond to the expected values from the manufacturer's selection program.

According to the manufacturer, operation with a frequency converter is permitted from 20 to 70 Hz, but the test system is limited to 30-60 Hz.

The limit values were set for the following reasons:

- 30 Hz: The mass flow $\dot{m}_{CO2}$ must not be too low in order to avoid oil displacement.
- 60 Hz: Depending on the operating point, the maximum operating current is exceeded at > 65 Hz.

4.1.2 Heat Exchanger

The CO_2 WP is equipped with three plate heat exchangers. The system comprises an evaporator, a gas cooler, and an internal heat exchanger (IHE). BKW GmbH has selected products from Alfa Laval for these plate heat exchangers. Alfa Laval's areas of expertise include heat transfer, separation technology, and fluid handling. The company places a high priority on the development of sustainable industrial processes and the responsible use of natural resources, such as CO_2, with the aim of improving energy efficiency and reducing emissions.

The three selected plate heat exchangers are mounted using a soldered design. The heat transfer surfaces are composed of thin, specially embossed stainless steel plates. The elevated turbulence of the media (CO_2 and water) within the channels leads to effective heat transfer and optimal media distribution within the heat exchangers. The integration of channel plates and connections is achieved through the application of copper solder, a specialized technique that facilitates the formation of a compact unit.

The operation of these three heat exchangers is in counterflow configuration. The gas cooler and evaporator were designed in accordance with the European standard conditions specified in DIN EN 14511-2 for the secondary side (water side). According to this standard, the nominal operating point of the water in the gas cooler is defined by an inlet temperature of $T_{GC,w,In}$ = +30 °C and an outlet temperature of $T_{,GC,w,Out}$ = +35 °C. The mass flow of water is

$\dot{m}_{GC,w}$ = 3 kg/s. The gas cooler has been engineered to achieve a maximum heat flow of $\dot{Q}_{GC}$ = 63 kW.

In the design of the evaporator, the water temperature at the inlet is $T_{EV,w,In}$ = +10 °C, and at the outlet, $T_{EV,w,Out}$ = +7 °C. The mass flow rate of the water, $\dot{m}_{EV,w}$, was determined to be 3.96 kg/s. The evaporator was engineered to attain a maximum cooling capacity of $\dot{Q}_{EV}$ = 50 kW. The IHE was designed to deliver a heat output of up to $\dot{Q}_{IHE}$ = 8.5 kW. The IHE has been designed to ensure superheating at the compressor inlet under all measured operating conditions.

4.1.3 Expansion Valve

To ensure precise control of the CO_2 refrigerant flow within the CO_2 WP, CAREL's electronic expansion valve was used. CAREL is an industry leader in highly efficient electronic expansion valves, particularly for natural transcritical CO_2 applications. CAREL's expansion valves are characterized by their high durability and ability to operate under extreme pressures of up to 140 bar and refrigerant temperatures between -40°C and +70°C. Additionally, these valves are user-friendly and easy to install.

The selected control valve (the expansion valve) receives information from temperature and pressure sensors. This information is then used to adjust the flow of CO_2 refrigerant dynamically based on the current heating or cooling capacity.

According to Literature [3], the control valve regulates the optimal high pressure using a simplified formula. The formula is as follows:

$$P_{HP} = 1 + 2{,}44 \times T_{GC,CO_2,Out} \qquad \textit{Eq 1}$$

In this formula (Eq 1), P_{HP} represents the pressure after the compressor in the transcritical range, and $T_{GC,CO2,Out}$ represents the measured outlet temperature of CO_2 at the gas cooler.

The CO_2 flow is defined by the measured P_{HP} and $T_{GC,CO2,Out}$, which determine the target heating or cooling output. This type of regulation has proven stable and effective up to this point.

4.1.4 Liquid Seperator

Because of delivery issues caused by the coronavirus pandemic, BKW was faced with the challenge of designing and constructing a liquid separator in the aftermath of the evaporator. The test results indicated that a significantly more liquid could be detected in the sight glass before the separator than after it. However, complete separation could not be confirmed. Due to the additional heat input from the internal heat exchanger (IHE), the sight glass in front of the compressor remained free of liquid. This process optimally separated CO_2 droplets from CO_2 gas, effectively protecting the compressor.

4.2 Digital twin DT – simulation model

In the context of this project, a digital twin of the CO_2 WP was developed. Digital twin DT technologies have already been implemented in various systems and industries, including manufacturing, construction, healthcare, aerospace, and transportation, among others. DT technologies facilitate the creation of a virtual duplicate of the existing system, providing a platform for the analysis of activities, interactions, and consequences of various decisions within the system. In the industrial sector, digital technologies DT are employed to enhance productivity, efficiency, availability, and, by extension, the quality of assets or services [4].

In this project, Dymola, developed by the French company Dassault Systèmes, was utilised for the modeling and simulation phases. This is a powerful simulation environment that is particularly suitable for modelling complex physical systems. Dymola is based on the Modelica programming language.

In this particular instance, Dymola will be employed in conjunction with the TIL library from TLK-Thermo-GmbH. This TIL library facilitates the efficient modelling and simulation of thermal processes, establishing a structured, modular basis for the developement of the digital twin or simulation model. Figure 2 illustrates the configuration of the digital twin.

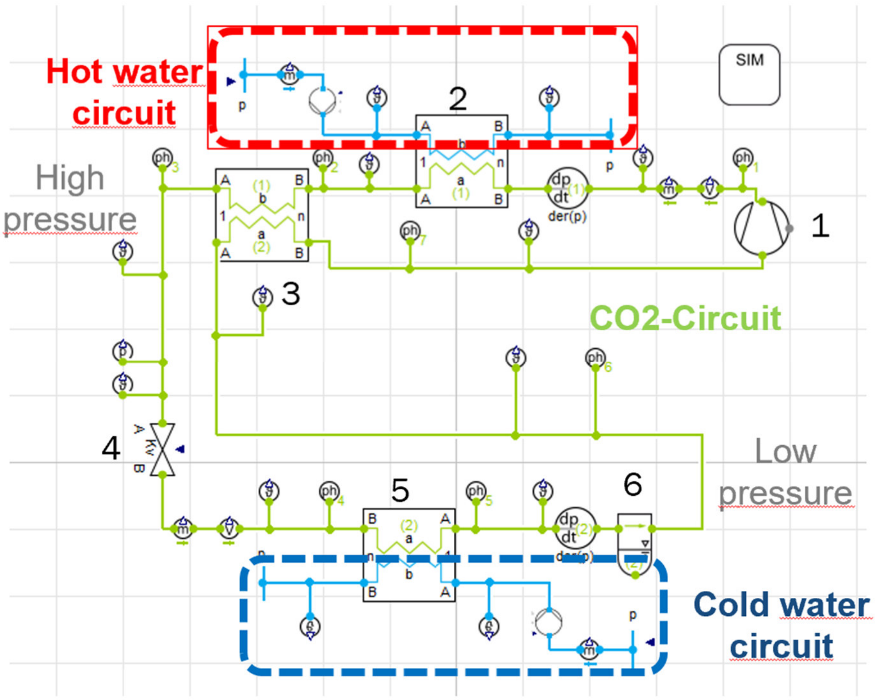

Figure 2: Digital Twin DT - CO_2 WP - Simulation model

The simulation model is presented in its most significant components. The three primary segments comprise: a. the CO_2 refrigeration circuit (green), b. the heat sink (red), and c. the heat source (blue), as well as the components integrated in the CO_2 WP. The main components are shown below:

1. Evaporator, 2. Gas cooler, 3. Internal heat exchanger, 4. Expansion valve, 5. Evaporator, 6. Liquid separator

5 Results and Discussion

5.1 Experiment vs. Simulation (COP / EER)

The efficiency of the CO_2 WP is determined through experiments and simulations in heating and cooling modes (HM and CM) with different CO_2 refrigerant mass flows $\dot{m}_{CO2}$. The specified temperature values for HM and CM come from the European standard DIN EN 14511-2. This standard establishes the testing conditions for evaluating the performance of air conditioners, liquid chillers, and heat pumps using air, water, or brine as heat transfer media and equipped with electrically driven compressors for space heating and/or cooling.

In the course of these experiments and simulations, the efficiency of the CO_2 WP is determined using performance figures. The ratios of the useful heat flow at the gas cooler $\dot{Q}_{GC}$ or the heat absorbed at the evaporator $\dot{Q}_{EV}$ to the electrical power P_{el} are also known as the coefficient of performance COP for heating mode (HM) and the energy efficiency ratio EER for cooling mode (CM) [5]. Figure 3 shows these physical parameters.

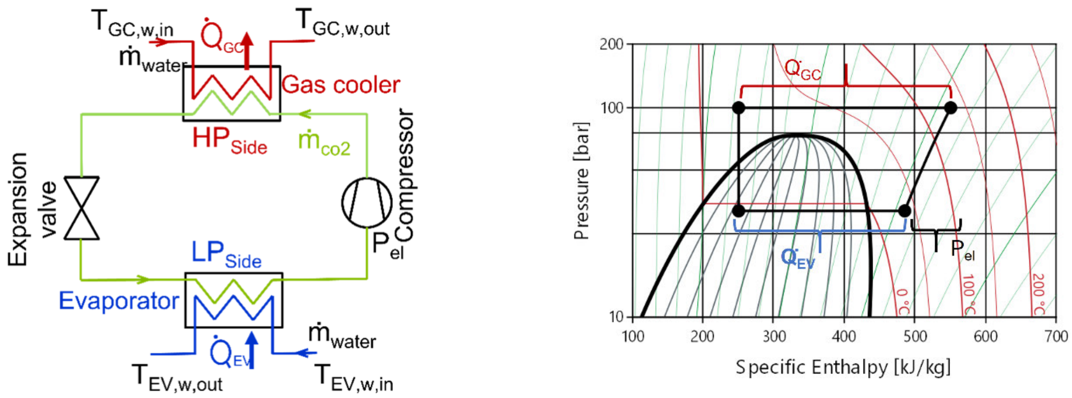

Figure 3: Physical parameters used to calculate COP and EER values

$$COP = \frac{\dot{Q}_{GC}}{Pel} \qquad \text{Eq 2}$$

$$EER = \frac{\dot{Q}_{EV}}{Pel} \qquad \text{Eq 3}$$

The experiments are being conducted at the CO_2 WP of BKW GmbH. They are being carried out using Eniserv, a platform that controls the CO_2 WP. This platform allows the experiments to be conducted remotely via the internet, regardless of location. To conclude, simulations are carried out using the digital twin DT created by Reutlingen University, and the simulation results are compared with the experiment results.

5.1.1 Heating mode HM

The heating mode values enumerated in *Table 1* have been derived from DIN EN 14511-2 and must be configured for the purpose of conducting simulations and tests. As demonstrated in *Table 1*, the inlet and outlet temperatures of the heat source at the evaporator, together with the outlet temperature of the heat sink at the gas cooler, have been established. The compressor frequency, ranging from 30 to 60 Hz, exerts a modification on the CO_2 mass flow $\dot{m}_{CO2}$ within the CO_2 WP.

Compressor frequency [Hz]			Evaporator		Gas cooler		Operating mode **HM**
			Inlet temperature $T_{EV,w,In}$ [°C]	Outlet temperature $T_{EV,w,Out}$[°C]	Inlet temperature $T_{GC,w,In}$ [°C]	Outlet temperature $T_{GC,w,Out}$ [°C]	
30	48	60	10	7	***30***	***35***	**HM** - low temperature **lT**
30	48	60	10	7	***40***	***45***	**HM** - intermediate temperature **iT**
30	48	60	10	7	***47***	***55***	**HM** - middle temperature **mT**
30	48	60	10	7	***55***	***65***	**HM** - high temperature **hT**

Table 1: Heating mode HM, values for simulations and experiments on evaporator and gas cooler in accordance with DIN EN 14511-2

Figure 4 supplements *Table 1* and shows the values for simulations and experiments.

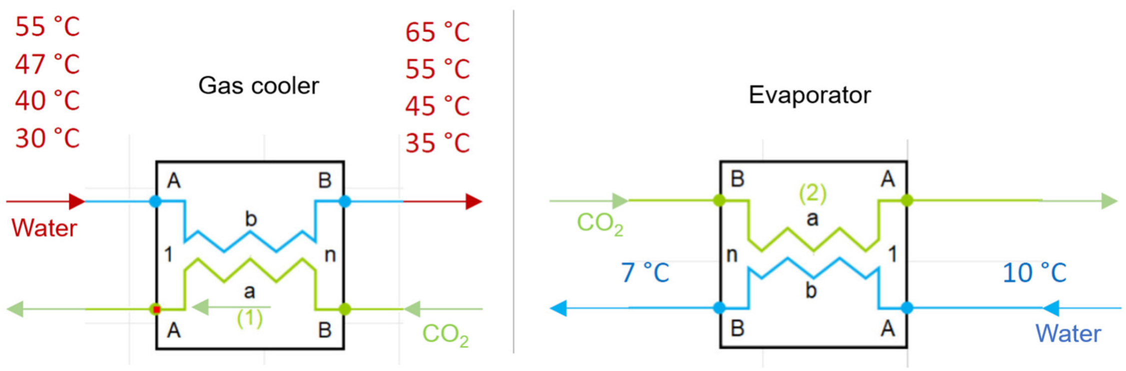

Figure 4: Heating Mode HM, Values for simulations and experiments on evaporator and gas cooler in accordance with DIN EN 14511-2

- HM – low temperature IT (Refer to Figure 9 at the end of this paper)

In applications where the CO_2 WP is utilized within the low temperature range IT (refer to *Table 1*), it is imperative to maintain the pressure on the high-pressure side (in Gas cooler) at 75 bar. This pressure limitation is a safety measure implemented by the CO_2 WP producer. In low temperature tests, a significant discrepancy was observed between results with minimal versus sufficient CO_2 inside the CO_2 WP compared to the simulation. The discrepancy is less pronounced when CO_2 is present in sufficient quantities, particularly within the low pressure range LP of the evaporator. This observation can be attributed to the replenishment of the CO_2 refrigerant in the CO_2 WP during the second test series.

Furthermore, the elevated pressure in the CO_2 WP's transcritical system has the potential to result in leakages and refrigerant losses. To obtain comparable pressure and temperature values at specific operating points, the refrigerant charge must be precise. This phenomenon has been observed at all three compressor speeds.

Additionally, a minor discrepancy was observed between the calculated COP from the simulation of 3.96 and the system value derived from the actual system of 3.77 (Figure 9). However, the target COP of at least 4.38 was not achieved.

For illustrative purposes, the following Figure 5 shows the thermodynamic cycle, the pressure and temperature deviations of the CO_2 WP with a compressor frequency of 30 Hz. The remaining compressor frequencies and the other thermodynamic cycles for heating HM and cooling modes CM are presented in figures at the end of this paper.

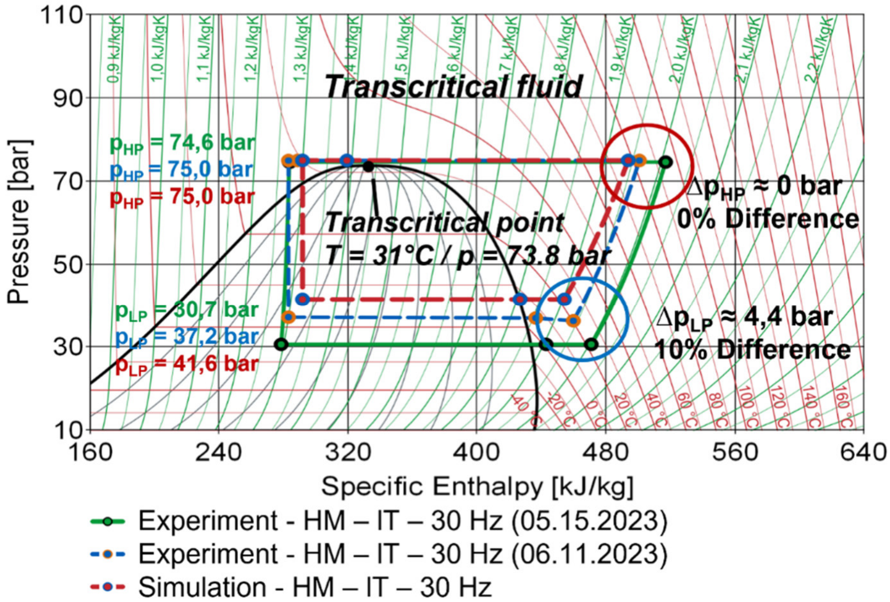

Figure 5: Heating mode HM, low temperature IT, 30Hz, experiment and simulation results with pressure limitation HP to 75 bar

- HM – intermediate temperatures iT (Refer to Figure 10 at the end of this paper) Figure 10

The utilization of CO_2 WP in the intermediate temperature range iT (refer to Table 1) modulates the pressure on the high-pressure side HP to 95 bar. The producer has established the pressure limit of 95 bar as a safety measure during the use of the CO_2 WP in higher temperature ranges.

Experiments have demonstrated slight pressure deviations of up to 8% in the HP range and large deviations of up to 25% in the LP range.

The evaporation process is generally carried out at an operating pressure of approximately 40 bar. This is the pressure employed in the simulation. However, within the low-pressure range, this phenomenon occurs at approximately 30 bar, attributable to the CO_2 WP's refrigerant insufficiency. Consequently, the compressor necessitates an augmentation in electrical power to compress the refrigerant and attain the requisite pressure and intermediate temperatures at the gas cooler inlet. This phenomenon also exerts a deleterious effect on the COP value of the CO_2 WP.

- HM – middle temperature mT (Refer to Figure 11 at the end of this paper)

As previously delineated, when operating the CO_2 WP in the meddle temperature range mT, the pressure on the high-pressure side was regulated to 95 bar.

In the HP range of the compressor outlet, with a compressor frequency of 30 Hz, there is only a minimal pressure deviation of 3% between the real CO_2 WP and the simulation. At a frequency of 48 or 60 Hz, no pressure deviation can be detected. The simulation and experimental results indicate that the pressure after the compressor is 95 bar.

In the LP range (at the evaporator), the process of evaporation in a real CO_2 WP occurs at a pressure ranging from 34.5 to 36.8 bar. However, in the simulation, this process takes place at 40 bar, which is equivalent to the real operating pressure of approximately 40 bar. This results in an augmented pressure deviation of 15% at a compressor frequency of 30 Hz, 11% at 48 Hz, and 10% at 60 Hz.

This deviation between the operating pressure and the actual pressure during the process of evaporation has been shown to result in the compressor requiring an increased quantity of electrical power when compressing the CO_2 refrigerant in order to reach the high pressure range of the middle temperatures. This circumstance exerts an influence on the COP value of the CO_2 WP.

It is plausible that repeating the experiments with a refilled CO_2 refrigerant will lead to optimized results.

The investigation revealed a discrepancy of 8% between the simulation's COP value of 2.12 and the real system's COP value of 1.96. The analysis demonstrated that the COP value exhibited an increase when the mass flow $\dot{m}_{CO2}$ within the CO_2 WP was elevated from 30 Hz to 60 Hz. This phenomenon can be attributed to the fact that friction losses at the compressor are reduced at higher mass flows $\dot{m}_{CO2}$, and less electrical power is required for compression.

- HM – High temperatures hT (Refer to Figure 12 at the end of this paper)

The objective of employing CO_2 WP is to attain elevated temperatures, hT (refer to Table 1), reaching up to 65 °C on the secondary side (the water side) of the gas cooler. The findings of this study demonstrate that, in this particular instance, the pressure on the HP side of the real CO_2 WP is maintained at 95 bar.

The experiments conducted within the high temperature range, designated as hT, and employing a compressor frequency of 30 Hz, were not successfully executed. This phenomenon can be attributed to the fact that, at a given compressor frequency, the CO_2 mass flows within the CO_2 WP are considered to be low.

The findings from the experiments and simulations conducted at 48 and 60 Hz indicate that there is no significant variation in the pressure deviations within the HP range. The simulation and experimental results indicate that the pressure after the compressor is 95 bar.

In the low-pressure range (at the evaporator), the process of evaporation at the actual CO_2 WP occurs at a pressure ranging from 36.8 to 37.8 bar. However, in the simulation, this process takes place at 41 bar, which is equivalent to the actual operating pressure of approximately 40 bar. This results in a pressure deviation. The deviation was found to be 10% at a compressor frequency of 48 Hz and 7% at 60 Hz.

The resulting pressure deviation between the operating pressure and the actual pressure during the evaporation process, as previously discussed in the mT range, leads to an increased electrical power requirement of the compressor to compress the CO_2 refrigerant and reach the HP range of the higher temperatures. This circumstance exerts an influence on the COP value of the CO_2 WP.

The repetition of the tests using refilled CO_2 refrigerant has the potential to result in more favorable outcomes.

The results of this investigation revealed a 5% deviation in the COP value between the simulation with 1.57 and the real system with 1.5.

An augmentation in the COP value can be discerned with an rise in mass flow $\dot{m}_{CO2}$ within the CO_2 WP, which signifies an increase from 48 Hz to 60 Hz. This can be attributed to the fact that friction losses at the compressor are reduced at higher mass flows, resulting in a decrease in the electrical power required for compression.

5.1.2 Cooling mode CM

The values indicated in *Table 2* for cooling mode CM are derived from DIN EN 14511-2 and must be configured for the simulation and the test. For both, the inlet and outlet temperatures of the evaporator and the gas cooler on the secondary side (water side) were set. The compressor frequency ranges from 30 to 48 and 60 Hz.

Compressor frequency [Hz]			Evaporator		Gas cooler		Operating mode **CM**
			Inlet temperature $T_{EV,w,In}$ [°C]	Outlet temperature $T_{EV,w,Out}$[°C]	Inlet temperature $T_{GC,w,In}$ [°C]	Outlet temperature $T_{GC,w,Out}$ [°C]	
30	48	60	***12***	***7***	30	35	**CM** - intermediate temperature **iT**
30	48	60	***23***	***18***	40	45	**CM** - low temperature **lT**

Table 2: Cooling mode CM, values for simulations and experiments on evaporator and gas cooler in accordance with DIN EN 14511-2

Figure 6 supplements *Table 2* and shows the values for simulations and experiments.

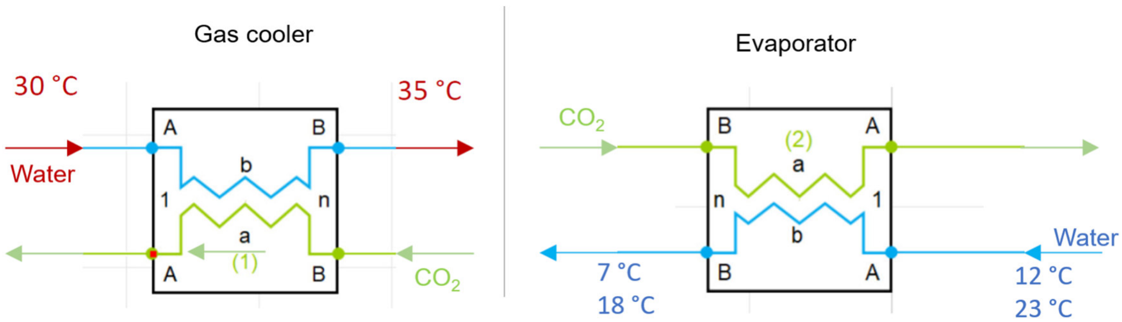

Figure 6: Cooling Mode CM, Values for simulations and experiments on evaporator and gas cooler in accordance with DIN EN 14511-2

- CM – intermediate temperature iT (Refer to Figure 13 at the end of this paper)

In the interest of safety, BKW imposes a limitation on the high-pressure side of the CO_2 WP, setting an upper limit of 75 bar during periods of cooling mode. The water temperature values at the evaporator and gas cooler (secondary side) were obtained from Table 2.

During the tests and simulations in the iT range, no pressure deviations between the simulation and the test were found in the HP range at the gas cooler. However, certain discrepancies were observed in the LP range (at the evaporator).

The process of evaporation within a real CO_2 WP deviations. These values are equivalent to 11% at a compressor frequency of 30 Hz, 10% at 48 Hz, and 6% at 60 Hz.

In this instance, as well, when employing the CO_2 WP in cooling mode, the CO_2 charge constitutes a pivotal criterion for optimal performance of the CO_2 WP. This factor exerts a significant impact on the overall efficiency of the system.

The higher pressure required for CO_2 WP in comparison to refrigeration systems employing conventional refrigerants necessitates significantly higher pressures. This phenomenon results in the occurrence of leaks within the system, consequently leading to refrigerant losses. In order to obtain comparable pressure and temperature values at certain operating points, it is essential that the refrigerant be at the correct fill level. This phenomenon is evident at all three operating points.

The EER value of the real system ranges from 2.4 to 2.36 and 2.32 for 30 to 48 and 60 Hz, respectively.

- CM – low temperature IT (Refer to Figure 14 at the end of this paper)

As mentioned previously, the pressure exerted by the CO_2 WP on the HP side is regulated to 75 bar during cooling mode. The values entered for the water temperatures at the evaporator and gas cooler (secondary side) for the low temperature IT range were taken from Table 2.

In the context of cooling mode CM in the IT, the utilization of a CO_2 WP typically results in the evaporation of the CO_2 refrigerant at elevated temperatures. Consequently, a pressure increase at the evaporator from 40 bar (operating pressure in HM) to 46 bar is observed.

The evaluation of the simulations and tests demonstrated that the pressure in the gas cooler did not deviate and remained constant at 75 bar in both the simulation and the test. Minor pressure deviations were identified at the evaporator.

The pressure at the evaporator of the real CO_2 WP varies between 39 and 45 bar, while in the simulation it is between approximately 44 and 46 bar. This phenomenon gives rise to pressure deviations. At a compressor frequency of 30 Hz, the value is 11%, at 48 Hz it is 4%, and at 60 Hz it is 3%.

The EER value of the actual system is as follows: The values 3.4, 3.17, and 3.13 are associated with 30, 48, and 60 Hz.

5.2 Sensitivity analysis

As part of the project, a sensitivity analysis was performed for all heat exchangers (evaporator, gas cooler, and internal heat exchanger). The objective of this study was to analyze the impact of variations in the number of plates in the heat exchangers on the coefficient of performance (COP) and efficiency ratio (EER) values when employing CO_2 WP across various temperature ranges and at different compressor frequencies (30, 48, and 60 Hz). This analysis was conducted in both heating and cooling modes.

The number of plates is a critical factor in the design of heat exchangers. This variable is equivalent to the heat transfer area A of the heat exchanger. The dimensioning of each heat exchanger can be represented by the specific thermal output Q̇, which is defined by the product of the heat transfer coefficient k, the associated heat transfer area A (number of plates), and logarithmic mean temperature difference LMTD between the primary and secondary sides at the inlet and outlet of the heat exchanger LnΔT (Eq 6) [6].

$$\dot{Q} = k \times A \times \mathrm{Ln}\Delta T \qquad \text{Eq 4}$$

$$\Delta T_1 = T_{W,In} - T_{CO_{2Out}} \qquad \text{Eq 5}$$

$$ln\Delta T = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \qquad \text{Eq 6}$$

$$\Delta T_2 = T_{W,Out} - T_{CO_{2In}} \qquad \text{Eq 7}$$

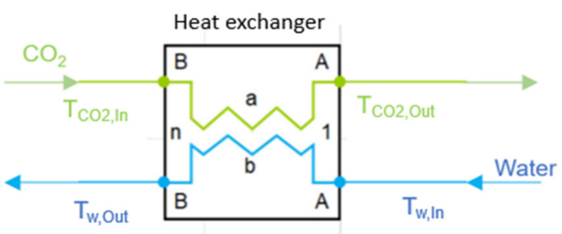

Figure 7: Temperatures of fluids at the heat exchanger

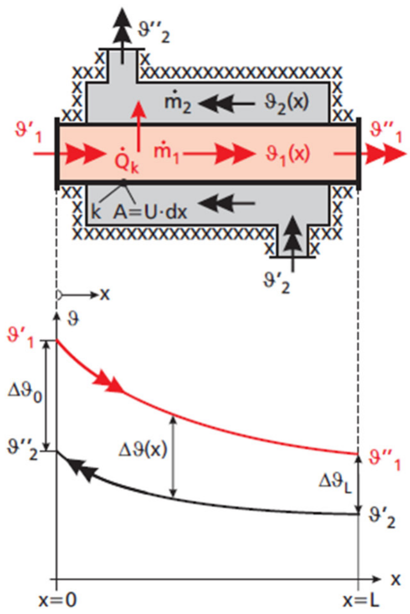

Figure 8: Fluid flows (top) and temperature profile (bottom) in a counterflow heat exchanger [7]

According to equation (Eq 4), increasing the heat transfer area (A), characterised by an increase in the number of plates, results in an increase in $\dot{Q}$ at constant k and Ln ΔT. This optimises the COP and EER values. If the COP or EER is to be maintained with a variation in A and a constant Ln ΔT, the effect of the change in A must be compensated for by a change in the heat transfer coefficient k (Eq 4).

Sensitivity analysis

5.2.1 Sensitivity analysis – Heating mode HM

Table 1 presents the temperature ranges of HM.

- Variation in the number of plates on the evaporator up to 120 (Refer to Figure 15)

The coefficient of performance (COP) curve demonstrates a continuous increase across all temperature ranges and compressor frequencies as the number of plates in the evaporator increases, up to a certain number of plates (approximately 64). This indicates that the desired thermal output of the gas cooler cannot be achieved with fewer than 64 evaporator plates. It is apparent that the required heat transfer area, A, is not sufficiently dimensioned with fewer than 64 plates.

It is evident that, With 64 evaporator plates, the COP value remains almost constant. This means that using more than 64 plates on the evaporator is inefficient.

This is due to the fact that the thermal output achieved at the gas cooler remains unchanged, irrespective of the number of plates utilised on the evaporator.

In the context of the HM, it has been demonstrated that the deployment of a CO_2 WP allows for a reduction in the number of plates on the evaporator from 92 to 64. This has no effect on COP of the CO_2 WP.

- Variation in the number of plates on the gas cooler up to 98 (Refer to Figure 16)

In the context of employing CO_2 WP within the IT range, the number of plates on the gas cooler assumes considerable significance across all compressor frequencies (30, 48, and 60 Hz). The COP value demonstrates a substantial increase within the IT range up to a plate count of approximately 86, after reaching which it remains nearly constant.

The utilization of CO_2 WP in heating mode at intermediate to higher temperatures (iT, mT, and hT) becomes inefficient above a plate count of approximately 42. The utilization of multiple plates on the gas cooler is no longer relevant in this case.

In order to optimize the COP values in the IT range, it is necessary to employ an increased number of plates on the gas cooler. The augmentation of the actual number of plates from 54 to 86 has been demonstrated to be advantageous for the COP of the CO_2 WP. Conversely, a reduction in the number of plates from 54 to 42 has been shown to be advantageous for the utilization of CO_2 WP in the iT, mT, and hT ranges.

- Variation in the number of plates on IHE up to 28 (Refer to Figure 17)

The simulations have demonstrated that a reduction in the number of plates within the internal heat exchanger (IHE) has a negative effect on the COP value of the CO_2 WP. In the context of employing the CO_2 WP across the high-temperature range hT, a potential reduction in the number of plates can lead to a 40% decline in efficiency or COP. In addition, incomplete superheating of the CO_2 refrigerant may occur.

In contrast, increasing the number of plates in the IHE has minimal effect on the coefficient of performance, with the exception of the high-temperature range hT. The present study concludes that increasing the number of plates in hT to 28 leads to a 12% improvement in the coefficient of performance.

The simulation has demonstrated that the primary function of the IHE in CO_2 WP is to attain complete superheating, and that decreasing the number of plates has a negative impact.

5.2.2 Sensitivity analysis – Cooling mode CM

Table 2 presents the temperature ranges of CM.

- Variation in the number of plates on the evaporator up to 120 (Refet to Figure 18)

The EER value increases significantly when using CO_2 WP in the iT range at all compressor frequencies with the use of multiple plates. It has been demonstrated that, after a certain number of plates (approximately 64), the EER value remains almost constant.

It is evident that the desired cooling capacity at the evaporator cannot be achieved with fewer than 64 plates. In addition, it can be concluded that the required heat transfer area, A, is not sufficiently large with this number of plates. In cooling mode CM, it is possible to reduce the number of plates on the evaporator from 92 to 64, provided that the CO_2 WP is used in the iT temperature range. This has no effect on the EER value of the CO_2 WP.

In contrast, simulations have demonstrated that the number of plates on the evaporator exerts a substantial influence on cooling mode within the low temperature lT range, particularly at low compressor frequencies of 30 Hz (with low $\dot{m}_{CO2}$). The EER value was found to improve with an increase in the number of plates on the evaporator, from 92 to 120.

The present study aims to determine the optimal temperature range for the operation of the CO_2 WP in cooling applications. In the field of iT, it is recommended to minimise the number of plates. Conversely, within the IT range, it is imperative to augment the number of plates on the evaporator to ensure the attainment of superior EER values.

- Variation in the number of plates on the gas cooler up to 98 (Refer to Figure 19)

It was demonstrated that a reduction in the number of plates in the gas cooler had a significant effect on the COP value in heating mode. The sensitivity analysis of the cooling mode has been conducted, yielding consistent results. In the context of employing the CO_2 WP in cooling mode within the temperature range iT and IT at all compressor frequencies, it has been demonstrated that a reduction in the number of plates on the gas cooler is disadvantageous. This has a detrimental effect on the EER value of the CO_2 WP, resulting in the failure to achieve the desired cooling capacity at the evaporator.

In contrast, the EER value can be optimised through an increase in the number of plates. It is evident that the EER value can be augmented to approximately 19% when employing the CO_2 WP within the IT range and a compressor frequency of 60Hz with a plate count of 98 plates.

In order to optimise the EER values in the iT and IT ranges, it is necessary to use an increased number of plates in the gas cooler. It is evident that an augmentation in the number of plates from 54 to 98 results in a substantial enhancement in the EER values.

- Variation in the number of plates on IHE up to 28 (Refer to Figure 20)

The simulations in cooling mode have demonstrated that a reduction in the number of plates in the internal heat exchanger IHE has a negative effect on the EER value of the CO_2 WP. This can result in incomplete superheating of the CO_2 refrigerant.

In contrast, an increase in the number of plates in the IHE has been shown to have a positive effect on the EER value. The present study concludes that increasing the number of plates in the IT to 28 leads to an 8% improvement in the coefficient of performance.

The simulation has demonstrated that the primary function of the IHE in CO_2 WP is to achieve complete superheating, and that reducing the number of plates is counterproductive.

6 Conclusion

In this collaborative project, BKW GmbH and Reutlingen University have developed a prototype of a reversible CO_2 refrigeration system (CO_2 WP) with an output capacity ranging from 20 to 50 kW, capable of simultaneous heat and cooling generation. The regulation, data acquisition, and control were carried out by enisyst GmbH, while the optimization was performed by Reutlingen University. The performance of the COP and EER in heating and cooling mode at varying $\dot{m}_{CO_2}$ levels was validated through a combination of experiments and simulations, in accordance with the standards established in DIN EN 14511-2. A detailed sensitivity analysis was performed on the evaporator, gas cooler, and IHE to ascertain the impact of plate number on COP and EER. The following results were achieved during the course of the project:

The fill quantity of CO_2 exerts a substantial influence on the efficiency of the CO_2 WP. The addition of a sufficient amount of CO_2 refrigerant to the CO_2 WP results in better COP and EER values.

It was determined that, in the heating mode, the number of plates on the evaporator can be reduced to a specific number in all temperature ranges from low lT to higher temperatures hT. However, in the cooling mode, this reduction is only applicable at iT. This has no effect on the COP and EER values. In contrast, within the CM at lT configuration, it is imperative to augment the number of plates to attain an enhanced EER.

It is possible to increase the number of plates on the gas cooler to certain number, both in HM for the low temperature range lT and in CM for all temperature ranges. This results a significant improvement in the COP and EER values. Conversely, the number of plates in HM can be reduced to a specific value at other temperatures (iT, mT, and hT).

It is imperative to ensure the maintenance of the correct number of plates within the internal heat exchanger. It has been established that reducing the number of plates is generally not advisable. This has a negative impact on the COP and EER values. The augmentation of the number of plates has been demonstrated to optimize the efficiency of the CO_2 WP in the heat and cooling modes.

The sensitivity analysis demonstrated that the COP and EER values are dependent on the temperature ranges. The optimal number of plates for each individual heat exchanger (evaporator, gas cooler, and IHE) is determined by taking into account the specific application (heating or cooling) and the temperature range. The corresponding methodology was described in detail above.

It can be deduced that in order to achieve enhanced COP and EER, an increase in the number of plates in specific values of plates is required in all temperature ranges for the evaporator, the gas cooler and the IHE when utilising CO_2 WP in both heating and cooling directions.

Acknowledgement

We would like to acknowledge and express our profound gratitude to our project partners, BKW GmbH + Enisyst GmbH and Reutlingen University, for their invaluable collaboration, extensive expertise, and unwavering support throughout the project lifecycle. Their contributions have been instrumental in achieving project milestones, facilitating knowledge transfer, and disseminating innovations. We would also like to express our profound gratitude to ZIM for providing financial support through the project ZIM-KK5007203, which has facilitated critical research activities, development, and validation efforts.

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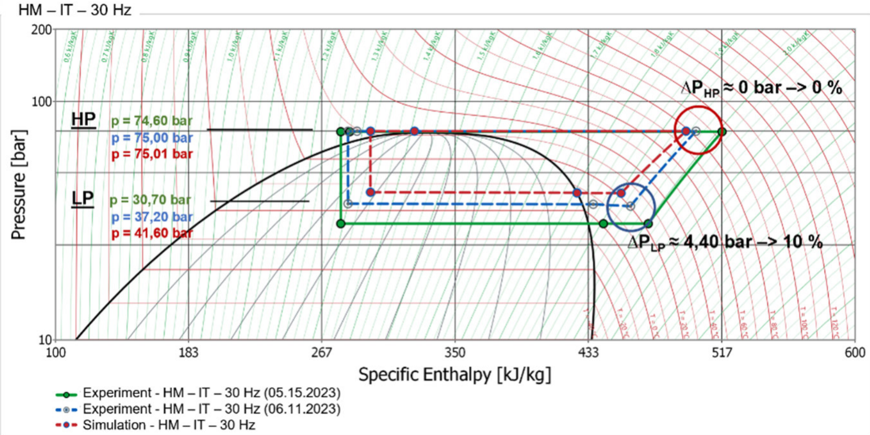
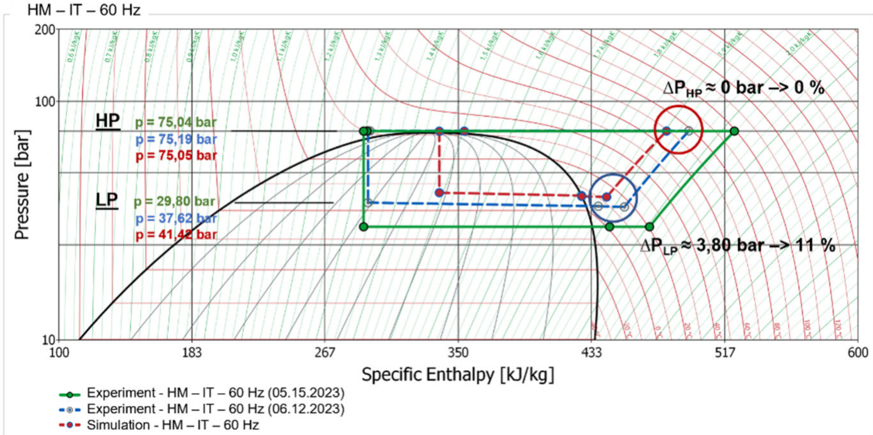
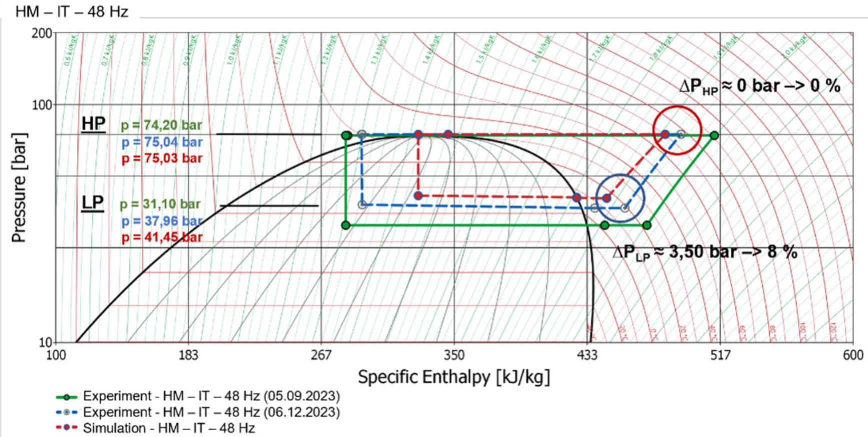
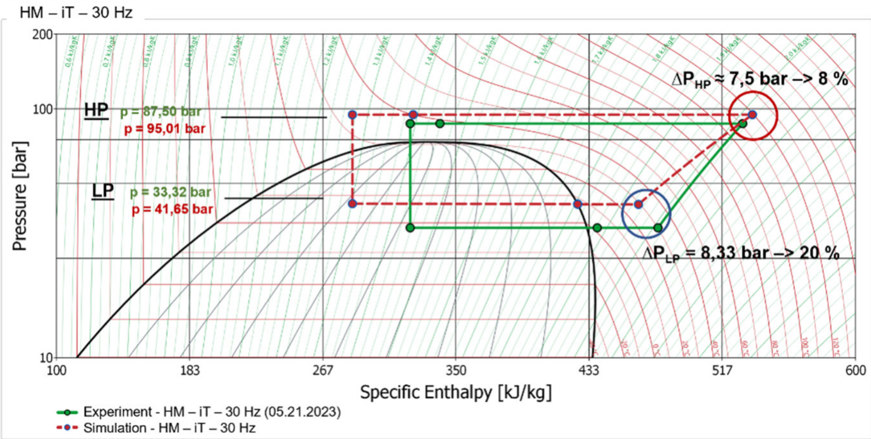


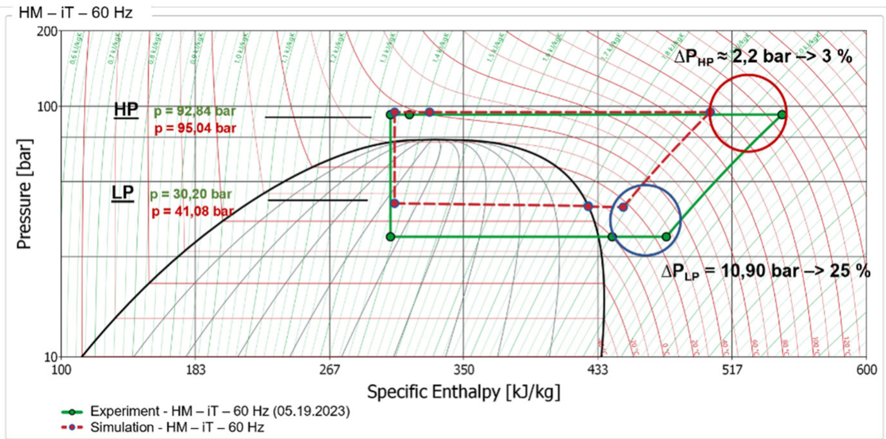


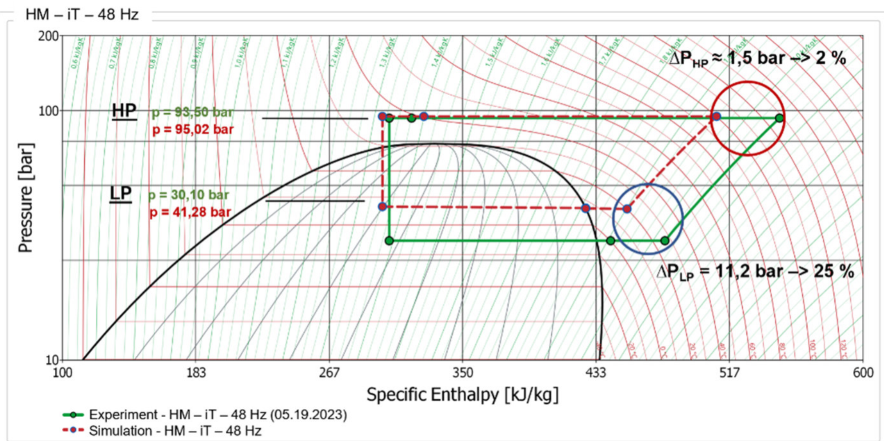


Compressor-frequency [Hz]	Temperature [°C]		Simulation	Experiment
30	IT	COP	3,96	3,77
48			3,51	3,64
60			3,2	3,58

Figure 9: Heating mode HM, low temperature IT, experiment and simulation results with pressure limitation HP to 75 bar

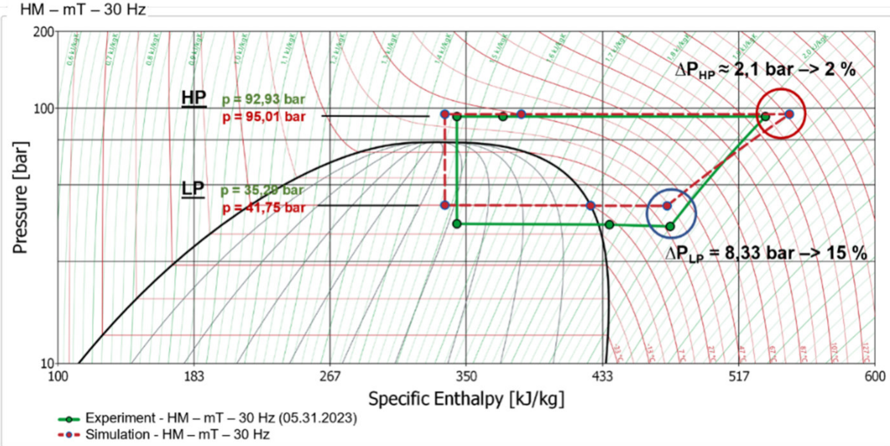


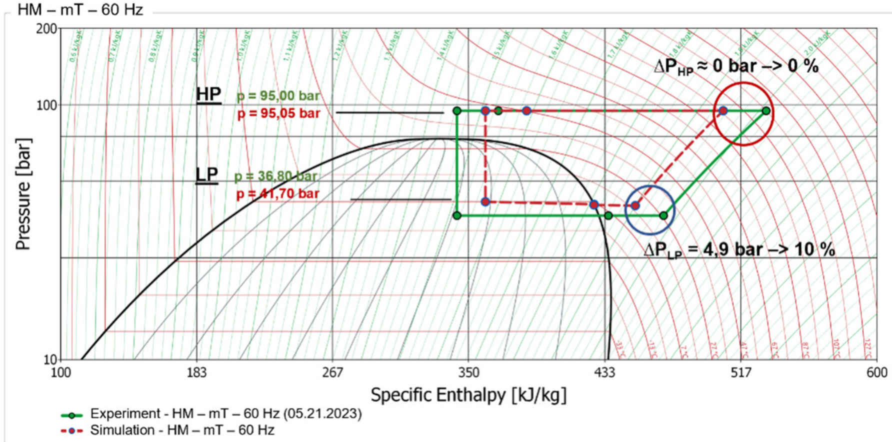


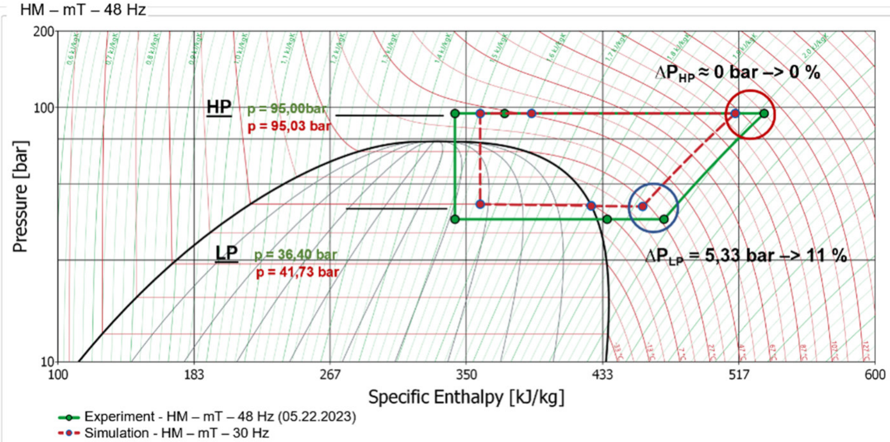


Compressor-frequency [Hz]	Temperature [°C]		Simulation	Experiment
30	iT	COP	2,82	2,21
48			3,1	2,57
60			3,06	2,69

Figure 10: Heating mode HM, intermidiate temperature iT, experiment and simulation results with pressure limitation HP to 95 bar

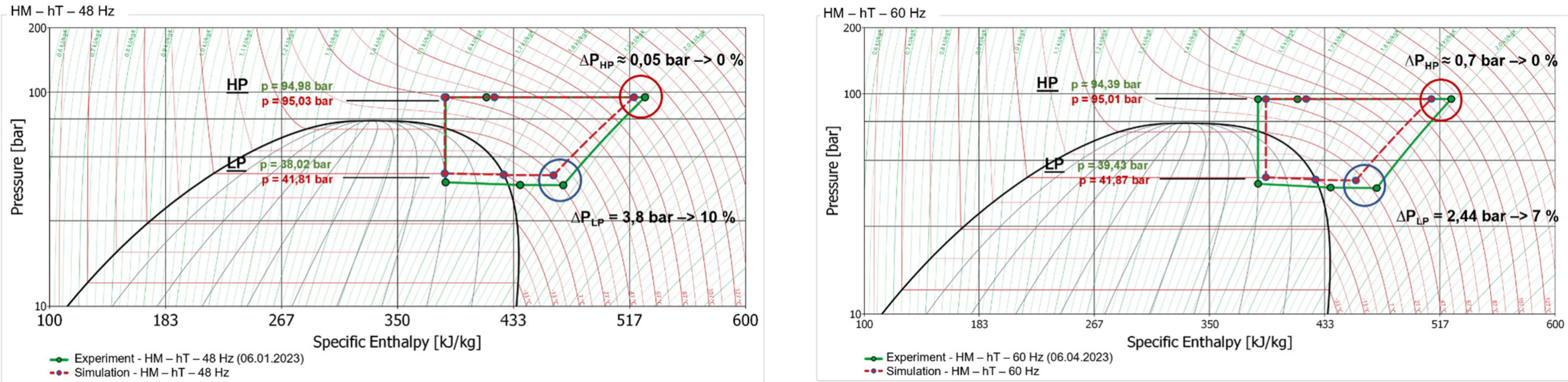






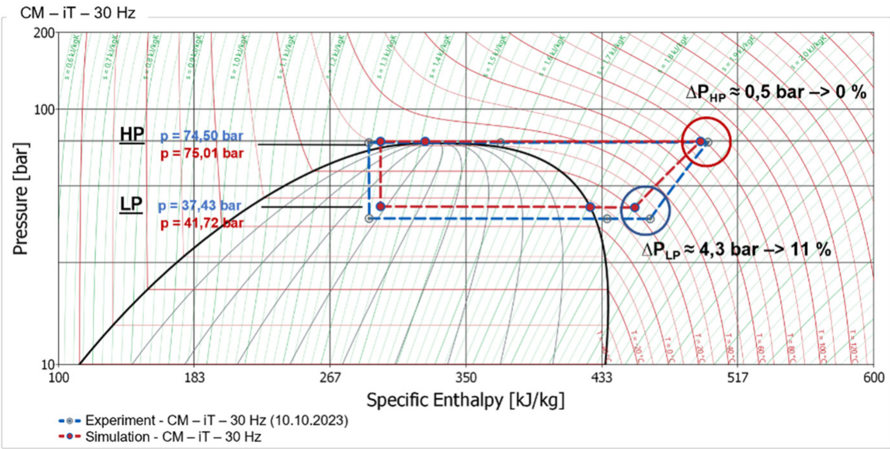
Compressor-frequency [Hz]	Temperature [°C]		Simulation	Experiment
30	mT	COP	2,08	1,46
48			2,08	1,97
60			2,12	1,96

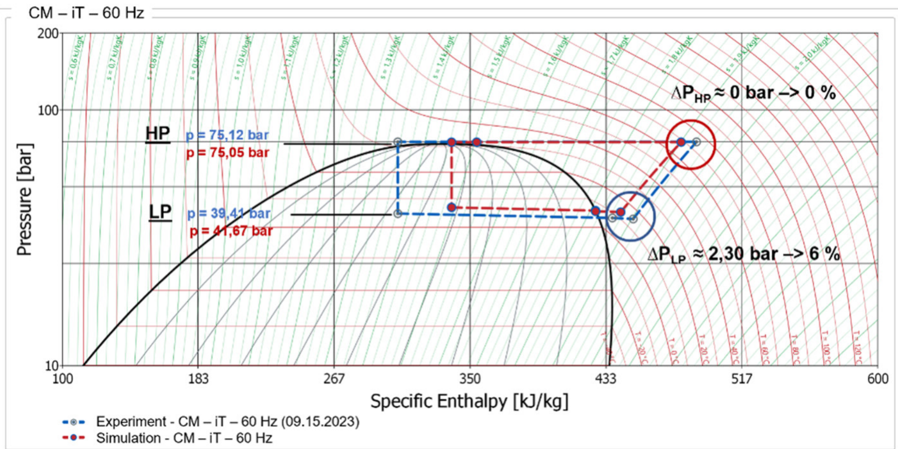
Figure 11: Heating mode HM, middle temperature mT, experiment and simulation results with pressure limitation HP to 95 bar

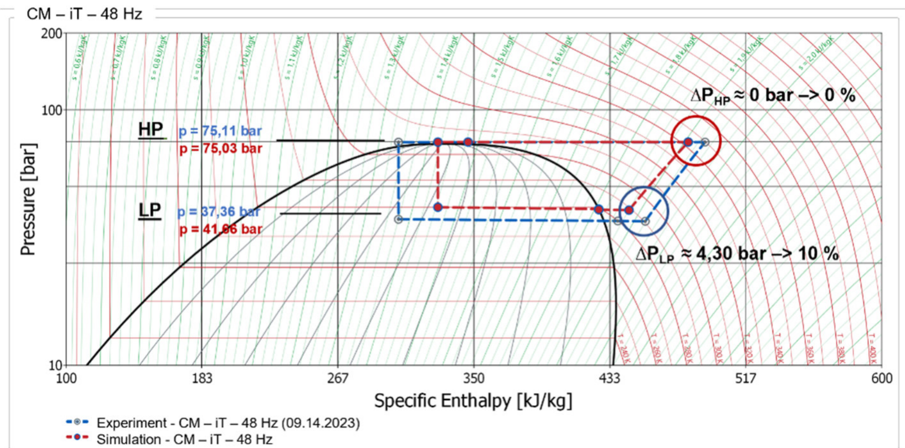


Compressor-frequency [Hz]	Temperature [°C]		Simulation	Experiment
30	hT	COP	-	-
48			1,64	1,45
60			1,57	1,50

Figure 12: Heating mode HM, high temperature hT, experiment and simulation results with pressure limitation HP to 95 bar

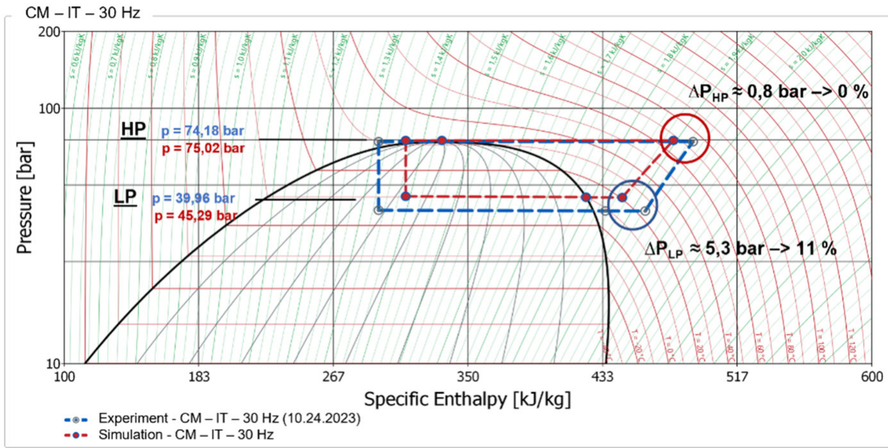


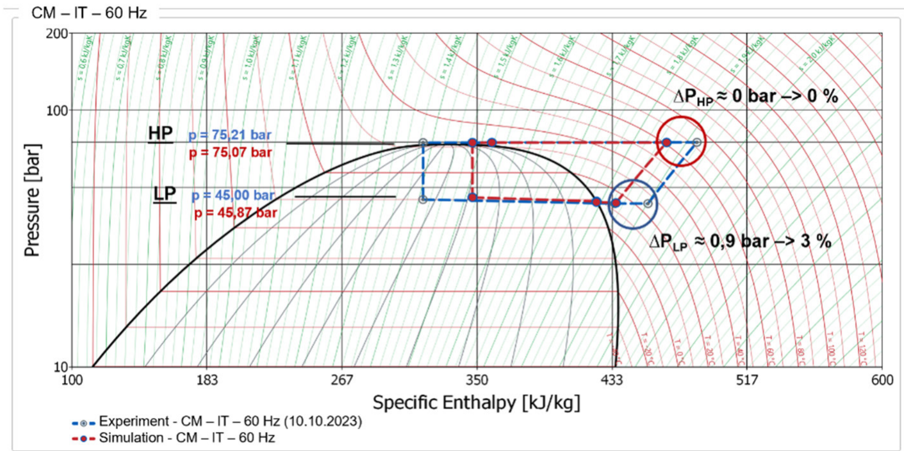


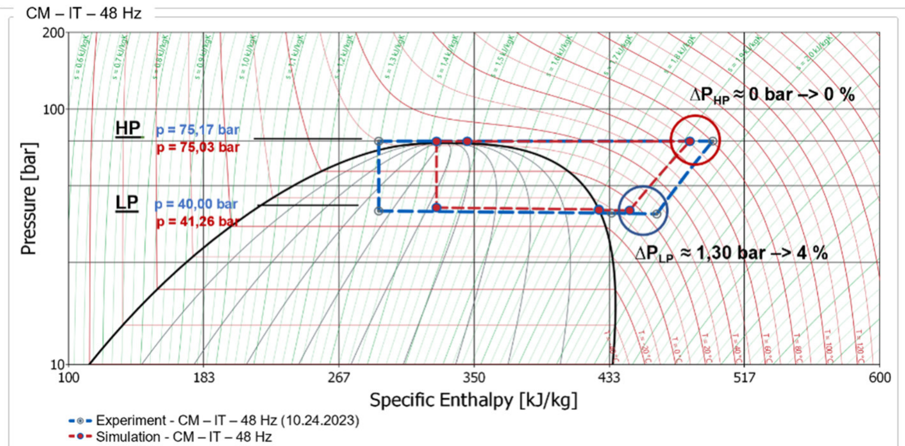


Compressor-frequency [Hz]	Temperature [°C]		Experiment
30	iT	EER	2,40
48			2,36
60			2,32

Figure 13: Cooling mode CM, intermidiate temperature iT, experiment and simulation results with pressure limitation HP to 75 bar







Compressor-frequency [Hz]	Temperature [°C]		Experiment
30	IT	EER	3,40
48			3,17
60			3,13

Figure 14: Cooling mode CM, low temperature IT, experiment and simulation results with pressure limitation HP to 75 bar

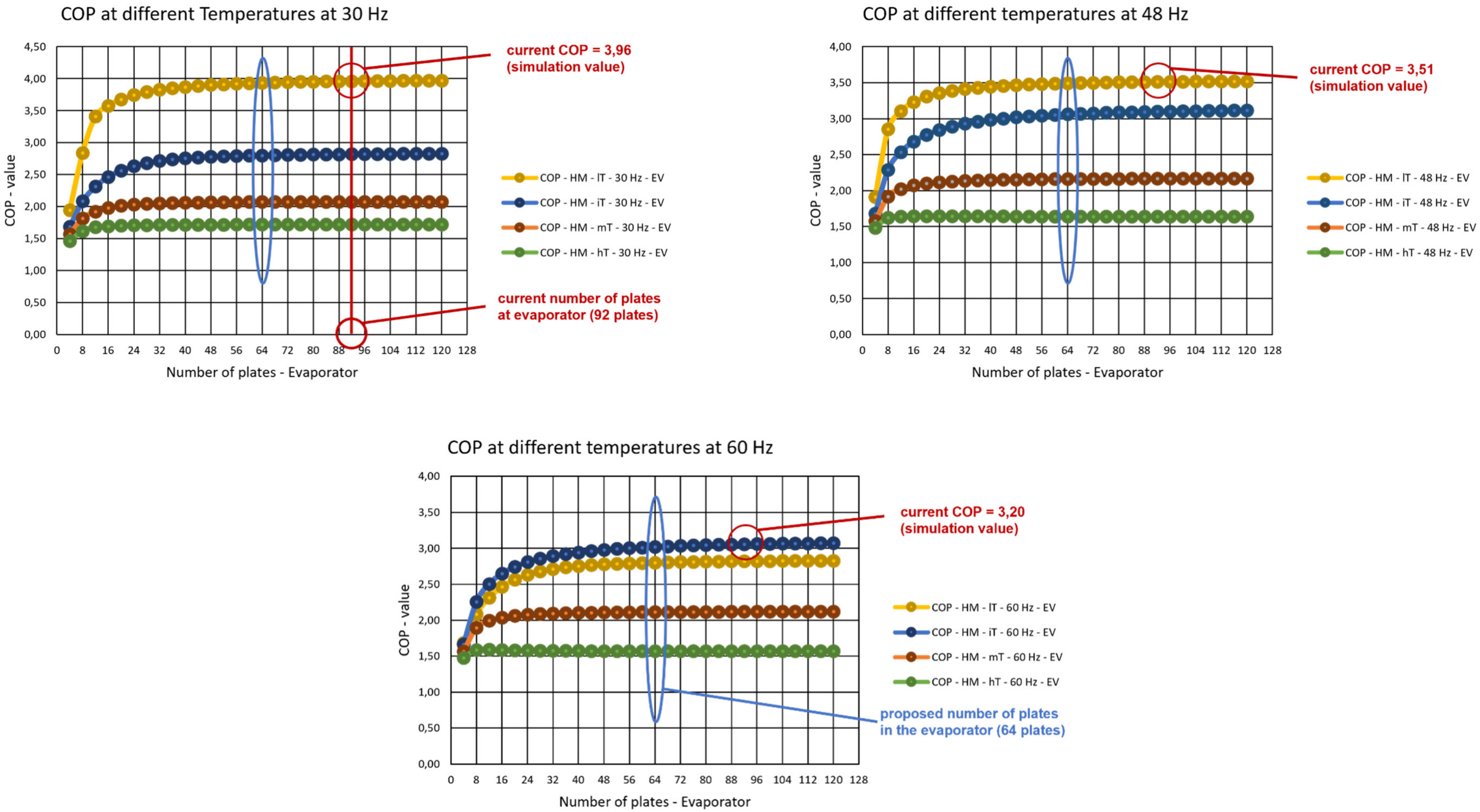


Figure 15: Sensitivity analysis, heating mode HM, Evaporator

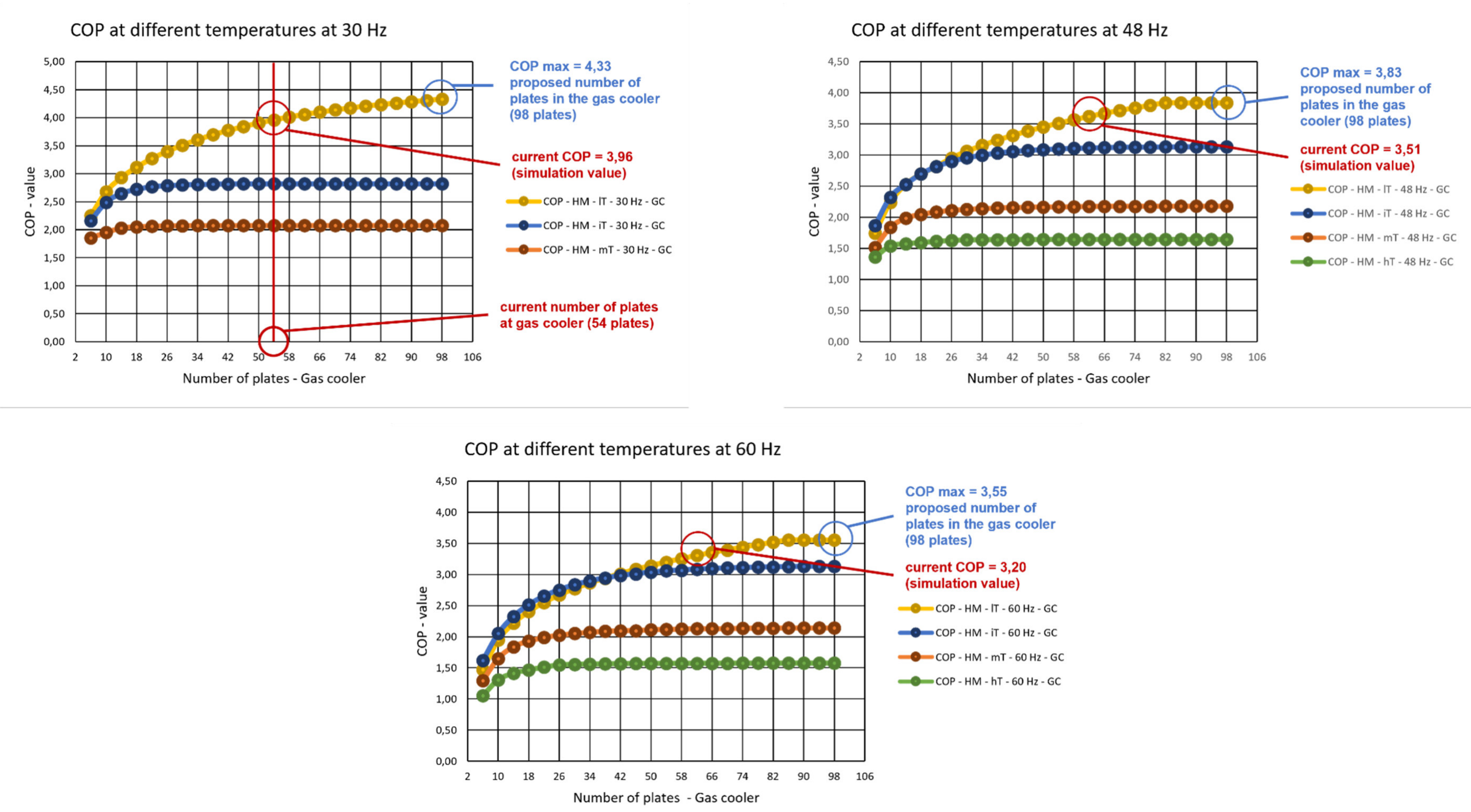


Figure 16: Sensitivity analysis, heating mode HM, Gas cooler

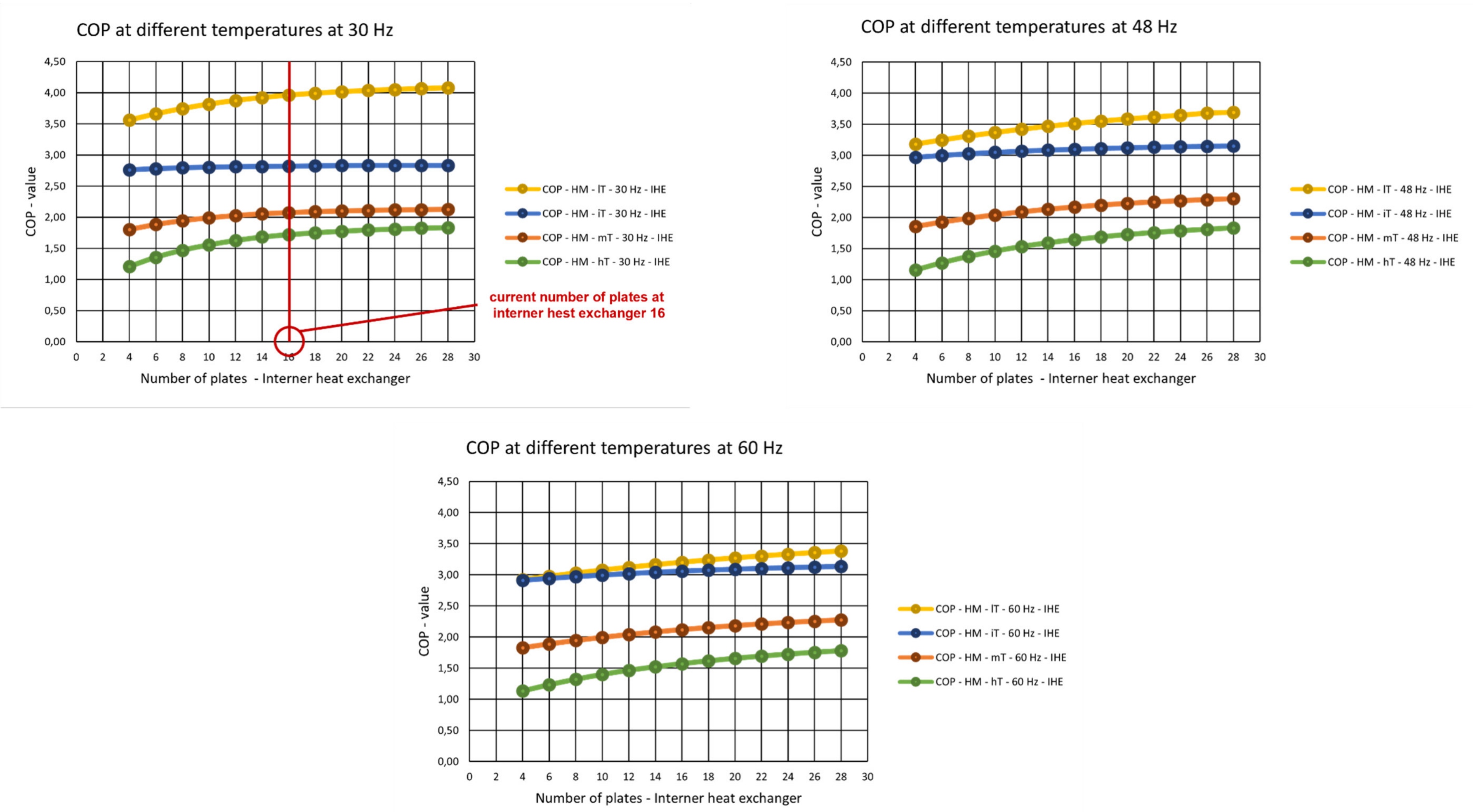


Figure 17: Sensitivity analysis, heating mode HM, Interner heat exchanger

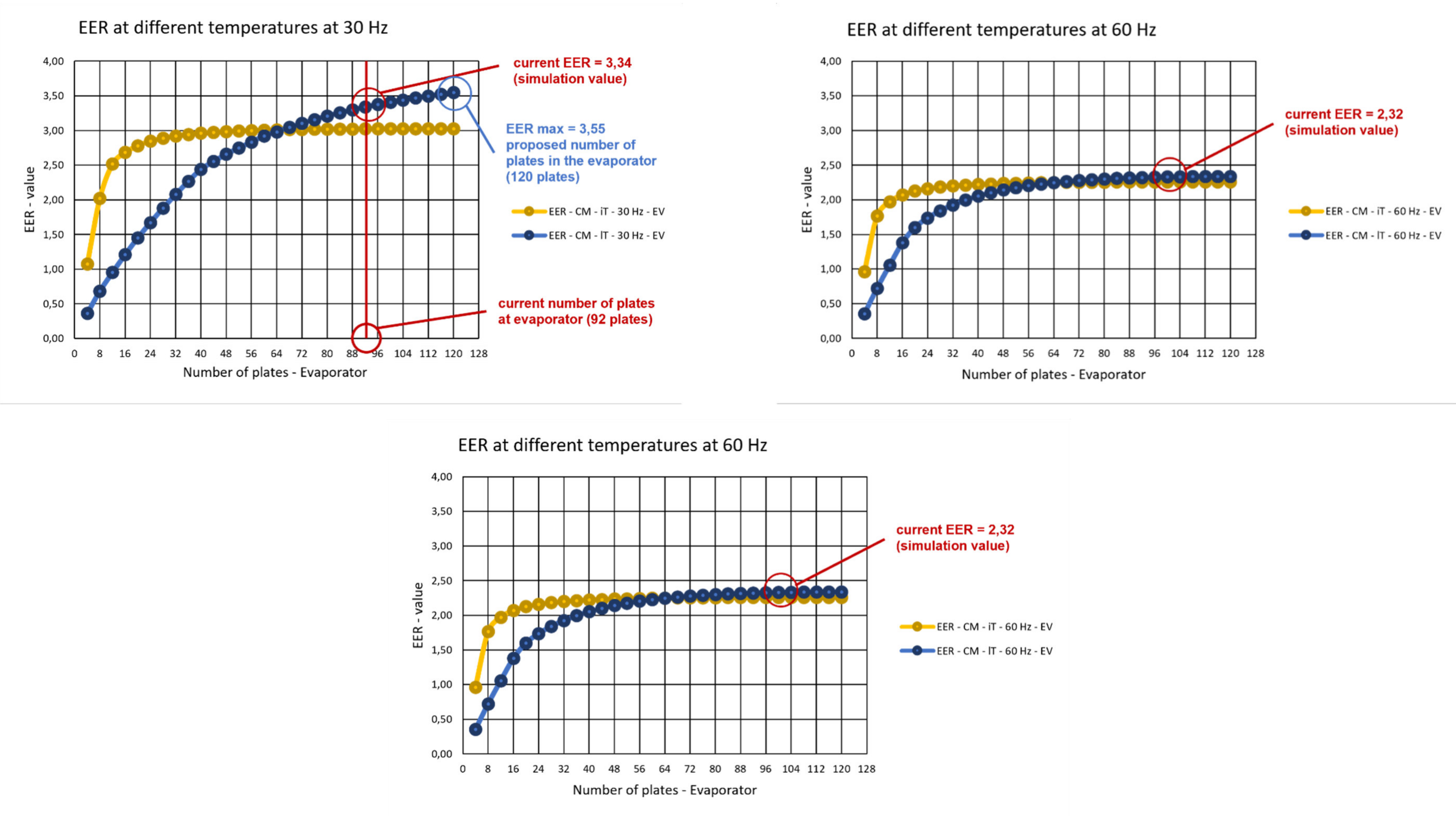


Figure 18: Sensitivity analysis, cooling mode CM, Evaporator

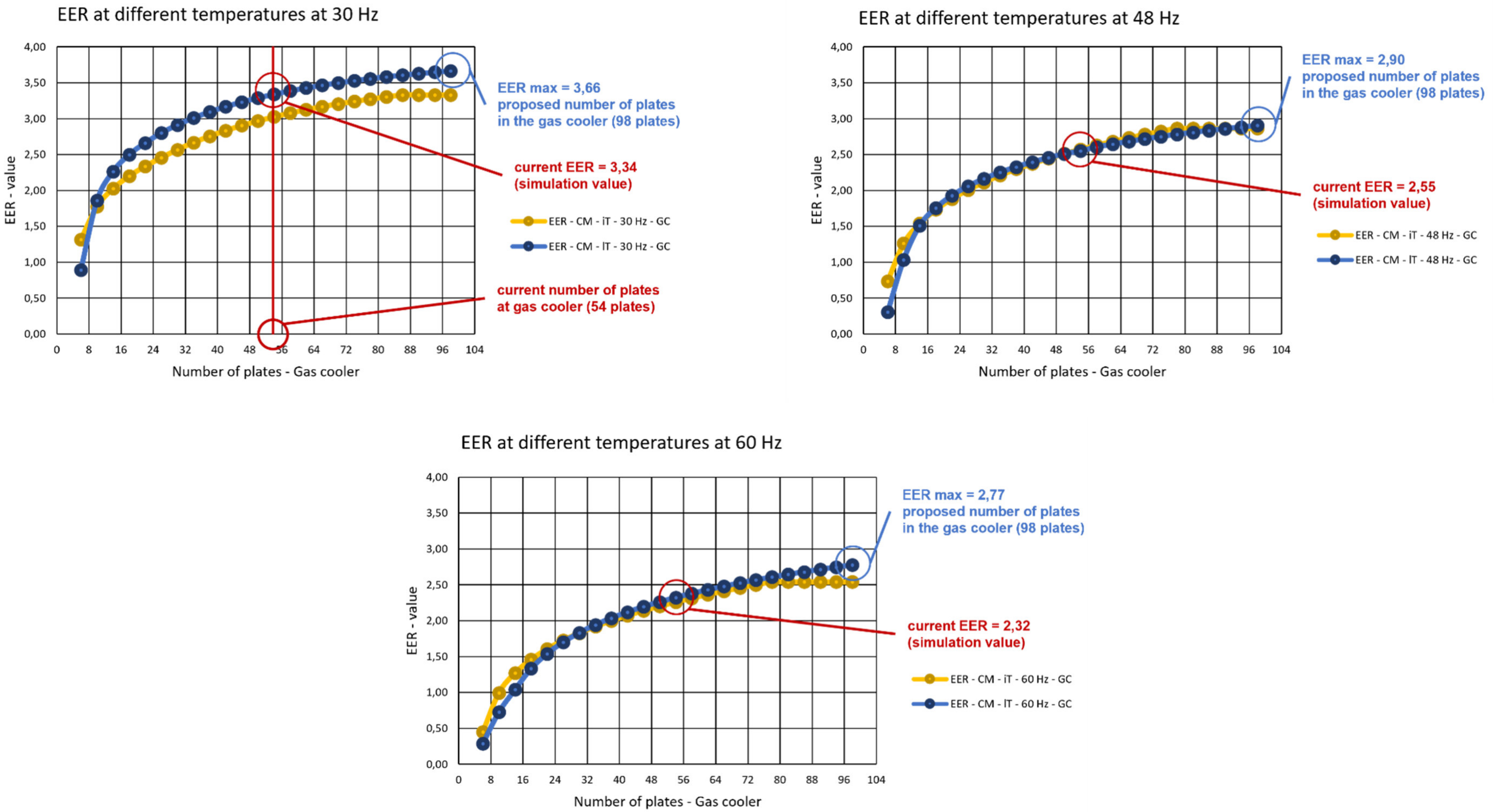


Figure 19: Sensitivity analysis, cooling mode CM, Gas cooler

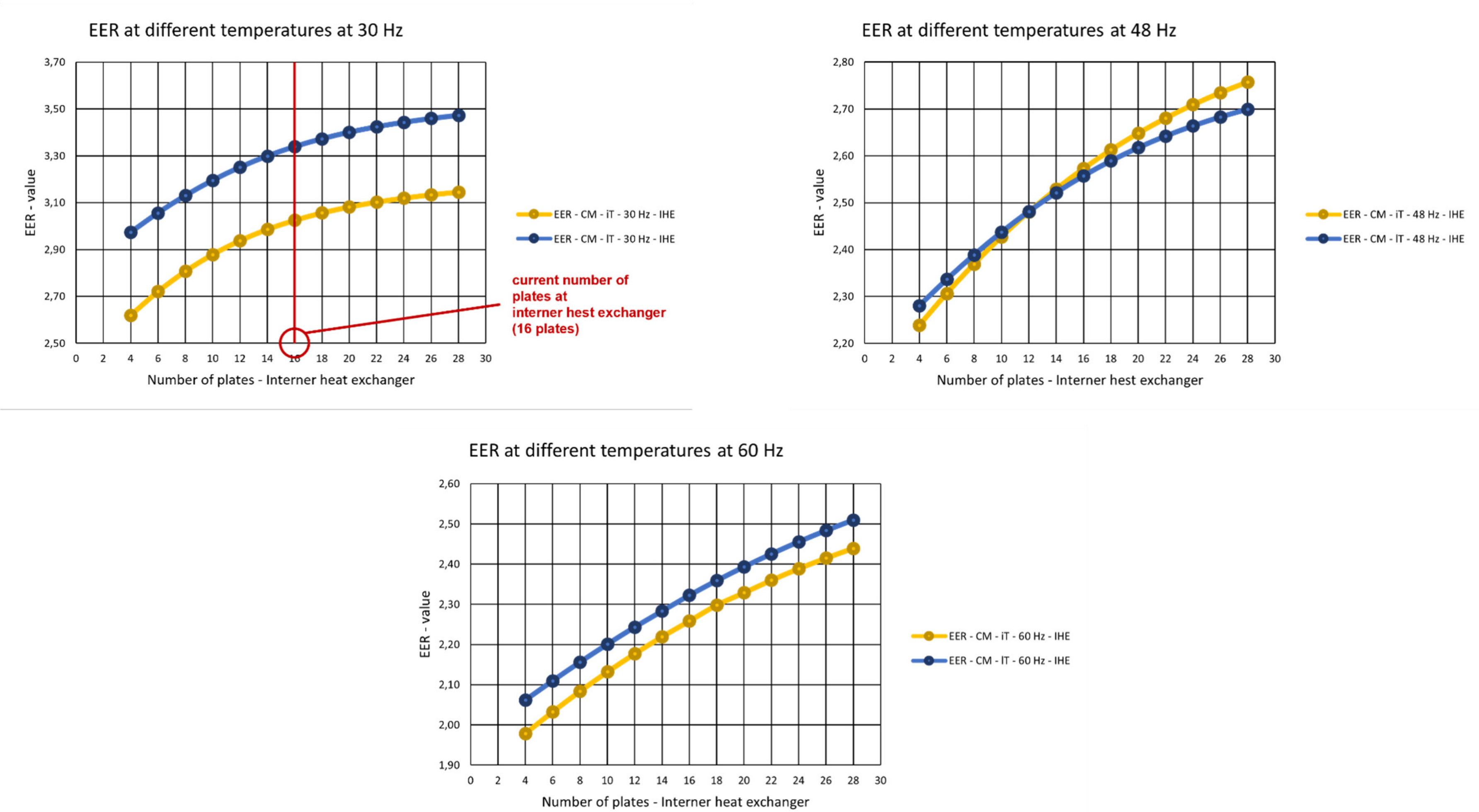


Figure 20: Sensitivity analysis, cooling mode CM, Interner heat exchanger