Carbon dioxide refrigeration with heat recovery for supermarkets

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Abstract

This paper describes the outcomes of a research project that investigates sustainable heating and cooling solutions for retail applications using a carbon dioxide (CO_2) natural refrigerant (R744) for food refrigeration. The paper presents the findings from an applied research study on a booster CO_2 (R744) system with high and medium temperature heat recovery. The paper includes a description of the conceptual design and a computer model along with its validation based on some experimental results. The energy consumption and carbon emission reduction are investigated using this novel system based on an existing supermarket as a case study.

Keywords: sustainable refrigeration; supermarket; heat recovery; carbon; dioxide refrigeration

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1 INTRODUCTION

Food is an essential of life. The food industry is a crucial sector for the balance of an economy, especially in our modern world. However, the production of food also impacts on our environment. Recently, Beddington [1] reported that a large proportion of carbon emissions is attributed to food. In the UK, a large proportion of this is due to the retail food sector. Refrigeration plays an important role in retail stores to maintain the food at the required temperature, but in doing so, it significantly contributes to greenhouse gas emissions both directly and indirectly. Greenhouse gas emissions can occur directly through the leakage of high global warming potential (GWP) hydrofluorocarbon (HFC) refrigerants used in refrigeration systems, which can be as much as 30% of the system charge per year [2]. Indirect emissions are also significant as these systems are large consumers of electricity and are reported to consume around 4 $MtCO_{2e}$ per annum, where CO_{2e} is the carbon dioxide equivalent [3]. In addition to the costs associated with the leakage of refrigerants and energy, there are other reasons why reducing carbon emissions from the retail sector are important.

In recent years, natural refrigerants have been proposed as an environmentally friendly solution for the refrigeration industry due to the unavoidable future phase-out of HFCs [4]. These refrigerants do not contribute to ozone depletion and have low GWPs. These refrigerants include ammonia, hydrocarbons and CO_2. CO_2 (R744) offers a long-term solution suitable for many applications in refrigeration and heating, from domestic applications using heat pumps to industrial and commercial applications. CO_2 offers significant advantages as a refrigerant since it is non-toxic [5], non-flammable [6], environmentally benign (ozone depletion potential = 0 and GWP = 1) [7], has high refrigeration volumetric capacity [8] and has high heat transfer coefficients [9]. However, there are technical challenges to its application associated with its low triple critical points and high operating pressure [10].

The principle of a transcritical system is that sensible heat rejection occurs above the critical point at a constant pressure, resulting in gliding temperatures [10]; therefore, the refrigerant is not condensed by normal condensers or heat exchangers (HXs) but is cooled by gas coolers. Because of the low triple point, R744 refrigeration systems are suitable for freezing low temperatures (LTs) and chilling medium temperatures (MTs) at constant pressure. The combination of LT and MT in one system has been reported to be the most appropriate and efficient R744 application for supermarkets [11].

1.1 Combined MT and LT booster transcritical system

Most compressor manufacturers are now able to supply both MT and LT compressors with the same oil system, which makes booster systems possible [11]. The CO_2 booster system

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includes the combination of MT and LT evaporators, receivers to separate the liquid and gas, gas bypass (BP) accompanied by expansion valves and a gas cooler. Madsen [12] describes the design, construction, installation and monitoring of a system similar to that installed in a small supermarket. The system is a booster system with $-10°C$ MT, 30°C LT, 35 bar in the receiver and 90 bar as discharge pressure, with 32°C as the gas cooler exit temperature. The coefficient of performance (COP) values for the systems were not presented, but the electricity energy consumption was monitored and compared with that of stores using R404A [zeotropic blends [R-125/143a/134a (44/52/4)] systems; an R744/R410A [zeotropic blends [R-32/125 (50/50)] cascade system was also monitored. It was concluded that the transcritical system was more efficient than the R404A and R744/R410A systems by 4% and 2%, respectively.

1.2 Combined MT and LT enhanced booster transcritical system

The principle of an enhanced booster transcritical system is explained by Javerschek [13] and its operation is same as that of a booster transcritical system. The difference compared to the booster system described above is that the gas BP instead of being throttled to the MT pressure, mixed and then compressed by a transcritical compressor, it is compressed directly to the high temperature (HT) pressure by another transcritical compressor in parallel compression. The Bitzer ECO compressor [13] can provide parallel compression as this compressor has two suction ports (a gas BP port and an evaporator port), leading to a common discharge (transcritical) chamber. Theoretically, Javerschek [13] type of systems has been shown to be more energy-efficient than the cascade and booster systems by 16% and 12%, respectively. The COP of the enhanced booster system can be improved by increasing the receiver pressure, which reduces the work of the parallel compressor.

1.3 Transcritical system with heat reclaim

When operating transcritically, the discharge gas after compression can reach very HTs and this is a useful source of heat. This source of heat is valuable as the heat can be reclaimed for heating and hot water services (HWSs) or as the heat source for absorption or adsorption chillers. The concept of a CO_2 heat pump that uses the discharge heat has been reported for HWSs [14] and heating [15] and both are combined together in the same system [16]. This solution is more economical than the rejection of useful heat to ambient air using gas coolers. In commercial refrigeration, Nekså *et al.* [17] reported on a combined refrigeration and water heating system used in supermarkets. The waste of heat from the refrigeration system was utilized for space heating and domestic hot water, and a 37% reduction in overall energy consumption could be achieved compared with an R22 system (methane series chlorodifluoromethane) without heat recovery (HE) [17]. According to the system of Adriansyash [18], the COP was increased significantly by including the heat recovered, used for the HWS. From the compressor discharge, the system recovered heat from 90°C to 70°C and rejected the remaining heat at the gas cooler. It is noticed that without the suction liquid heat exchanger (SLHE) that increases the superheat of the gas after the evaporator, the discharge temperature would be $<90°C$. The cooling COP of this system is 2.5, but the total COP which includes the heat reclaimed for the HWS is 4.2 at 35 bar suction and 88 bar discharge pressures. The overall COP (COPT) has been calculated by Adriansyash [18] as: COPT = (Σ cooling capacities + Σ HEs)/Σ electrical power input of compressors. Reinholdt and Madsen [19] used the same equation to calculate the total COP of a booster transcritical CO_2 refrigeration system with HE. The study was a feasibility investigation into the HE possibilities of CO_2 refrigeration in supermarkets. The system is similar to the combined MT and LT booster transcritical system described earlier excluding the LT stage, and the HX is placed before the gas cooler. The energy reclaimed is used for air-conditioning via an absorption chiller, HWSs and underfloor heating, depending on the grade of the heat. At an optimum high discharge pressure of 95 bar, temperature out of gas cooler ($T_{GC,out}$) of 30°C and HE for hygienic cleaning hot water (15–75°C), the COP and COPT are equal to 1.9 and 4.6, respectively. However, if all the heat from the gas cooler had been used, then a further significant improvement in the COP is possible.

1.4 Objectives of research

The application of R744 as a refrigerant in retail is the subject of this paper. The aim of this work was to investigate a practical and low carbon solution for a novel refrigeration system for supermarket applications. The novelty of this practical system is both the use of CO_2 as a refrigerant and the utilization of the reclaimed heat from the R744 cycle. By using a low GWP gas and all the waste heat from the cycle, direct and indirect emissions are significantly reduced when compared with conventional systems.

2 DEVELOPMENT OF AN R744 NOVEL REFRIGERATION SYSTEM WITH HE

This section describes the concept developed and the thermodynamic equations used to describe the system.

2.1 The concept

The overall objective of this research is to investigate the improvement in COP of a transcritical R744 system, by recovering much of the heat normally rejected to the ambient and to use it efficiently for other building service applications within supermarkets. The experimental system developed is shown in Figure 1 and is an R744 enhanced booster transcritical system that provides LT cooling for cold room/frozen food cabinets and MT cooling for chilled food cabinets. This system is

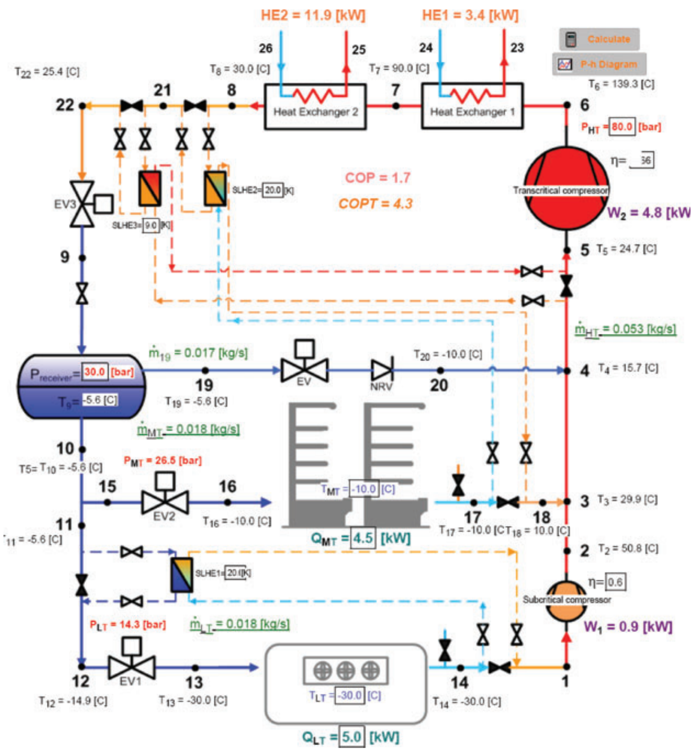

Figure 1. *Conceptual design and EES model of the novel system.*

enhanced because it is composed of SLHEs that increase the compressor discharge temperature and consequently provide higher potential for heat reclaim.

The conceptual design of the novel system detailed in Figure 1 is described as follows. After being expanded, the receiver separates the mixture of liquid/gas at a pressure of 35 bar. The liquid accumulates and is distributed to the LT and MT stages. At the MT stage after expansion, the liquid enters the 4.5 kW MT evaporator at a pressure of 26 bar. The saturated vapour is superheated by 20 K via SLHE2. At the LT stage, the liquid is subcooled and throttled by an expansion valve before entering the 5 kW LT evaporator coil at 14 bar. After evaporation, the gas is superheated by 20 K by SLHE1 to ensure complete evaporation as well as to increase the refrigeration effect. The LT superheated gas is compressed subcritically to the medium pressure where it is mixed with the gas from the MT evaporator at the same pressure. This mixture is then further mixed with BP gas from the receiver, superheated by 9 K (SLHE3) and then compressed to a discharge pressure of 80 bar by the MT transcritical compressor. The discharge gas can reach HTs according to the discharge pressure. The gas is cooled through two HXs, HX1 and HX2, that reclaim the heat rejected. The cooled gas exits the second HX at 30°C and returns to the receiver after being throttled.

2.2 Thermodynamic analysis

The thermodynamic equations used to describe the performance of the system are detailed in this section.

The capacity for MT and LT cabinets is

$$Q_{LT} = m_{LT} * \Delta h = m_{LT} * (h_{14} - h_{13}),$$
$$Q_{MT} = m_{MT} * \Delta h = m_{MT} * (h_{17} - h_{16}),$$

where h is the enthalpy.

The mass balance is

$$m_{HT} = m_{BP} + m_{LT} + m_{MT},$$
$$m_{BP} = m_3 * \frac{(h_9 - h_{10})}{(h_{19} - h_9)},$$

where m is the mass flow rate.

Isentropic efficiencies of the transcritical and subcritical compressors are

$$\eta_{TI} = \frac{h_{2'} - h_1}{h_2 - h_1},$$
$$\eta_{SI} = \frac{h_{6'} - h_5}{h_6 - h_5},$$

where η is the efficiency.

Compressor work equations are

$$W_1 = m_{LT} * (h_2 - h_1),$$
$$W_2 = m_{HT} * (h_6 - h_5),$$

where W is the power input of the compressor.

The heat reclaimed from HX1 and HX2 is calculated as:

$$\mathrm{HE1} = m_{HT} * (h_6 - h_7),$$
$$\mathrm{HE2} = m_{HT} * (h_7 - h_8).$$

The standard COP is defined as:

$$\mathrm{COP} = \frac{Q_{MT} + Q_{LT}}{W_1 + W_2}.$$

The COPT including the heat reclaimed is

$$\mathrm{COPT} = \frac{Q_{MT} + Q_{LT} + \mathrm{HE1} + \mathrm{HE2}}{W_1 + W_2},$$

where Q is the cooling capacity evaporator.

2.3 Development of a computer model of the novel system

The equations of the conceptual design described earlier have been used to create a steady-state model using the Engineering Equation Solve (EES) software, which has in-built R744 properties. The results are shown in Figure 1, and the calculated COP values of the system with and without heat reclaim are 1.6 and 4.3, respectively, at 80 bar. This is a steady-state analysis based on the assumptions that all the heat can be recovered. However, Section 3 considers the practical applications of this system in a typical supermarket and the relative savings that can be practically achieved.

3 INVESTIGATION OF POTENTIAL APPLICATIONS OF THE HEAT RECLAIMED

In order to demonstrate the energy consumption and CO_{2e} reduction using the proposed system, a 5600 m^2 store has been simulated as a base store and its energy consumption has been monitored using an Automatic Monitoring and Targeting tool. This tool provides automatic meter readings and data collection of mains and submetered electricity and gas and provides half hourly, weekly and monthly energy readings. Sub-meters can provide detailed data for systems such as bakery, chicken rotisserie, lighting, refrigeration, air handling plant and petrol services of the supermarket. Table 1 shows the annual submetered energy consumption of the supermarket. The total annual electricity consumption of the store is 3.71 GWh and the gas consumption is 1.02 GWh. The food refrigeration systems (i.e. refrigeration LT and MT packs, fan and electronics and service shop panel) represent 34% of the total electricity used by the store. The store currently uses conventional R404A HFC refrigeration packs to provide MT cooling to chilled cabinets and LT cooling to frozen cabinets. The monthly average power input to the MT and LT refrigeration packs was established by the monitoring the tool and by assuming COPs of 2.00 for the MT and 1.00 for the LT packs, and the calculated average cooling capacities obtained are 111and 28 kW, respectively.

The performance indicators for this store are 657 kWh/m^2 for electricity and 180 kWh/m^2 for gas. The total annual CO_{2e} emission of the store is 3 025 348 kg/year, equivalent to 536 kg CO_{2e}/m^2 (using carbon factors of 0.544 and 0.184 kg CO_{2e}/kWh, respectively, for electricity and gas). The refrigerant leakage rate is 252 kg/year from the *F*-gas logs of installations. Therefore, 73% of the store's total global warming impact is from energy use to run the store and 27% is due to refrigerant leakage.

Table 1. *Store energy consumption.*

Systems monitored	kWh/year	%
Bakery	611 974	16
Chicken rotisserie	99 280	3
Lighting	899 550	24
HVAC fans + electronics	292 894	8
HVAC refrigeration	27 162	1
Refrigeration HT packs	487 304	34
Refrigeration LT packs	243 420	
Refrigeration fan + electronics	128 951	
Refrigeration service shop panel	387 806	
Unsubmetered equipment	530 967	14
Total electricity consumption	3 709 307	100
Heating	823 984	81
HWS	193 841	19
Total gas consumption	1 017 825	100

Using the annual average loads Q_{MT} and Q_{LT}, respectively, of 111 and 28 kW, Figure 2 shows the heat outputs HE1 and HE2 with the variation in the discharge pressure (P_{HT}) from 75 to 120 bar from the EES model developed.

3.1 Potential application

Figure 3 shows a suggested application of the heat reclaimed from the CO_2 system using two HXs (HX1 and HX2). HE1 is the heat reclaimed by HX1 from the high discharge temperature of 90°C of the CO_2 system. HX2 recovers the heat rejection of the system from 90°C to 30°C. As illustrated in Figure 3, $HE1_a$ and $HE1_b$ are the heat reclaimed from HE1 by other HXs.

In the analysis of the system, the following assumptions have been made about the destination of the recovered heat.

- The heat reclaimed HE1a can provide a heat source (Q1) for an absorption chiller to provide cooling for the store in the summer period. When the air-conditioning is not required, Q2 can be used for district heating for export. During winter, additional heating demand will be available and an assessment of relative values will be made for prioritization.
- The heat reclaimed HE1b can provide heat for the domestic HWSs (Q3) in the store.
- The heat reclaimed HE2 can provide heat (Q4) for the underfloor heating system in the store according to the seasonal demand.
- Any heat not used is rejected to atmosphere, although it could be used offsite.

In order to satisfy the demand of the arrangement of Figure 3, this innovative system will need to be supported by an intelligent control system able to alternate with the conventional backup systems when required.

3.2 Excel model of supermarket with novel system

The conceptual system in Figure 1 has been modelled in Excel using the hourly energy demand of the store with the hourly heat reclaimed by HXs HE1 and HE2. These reclaim heat (HE1 and HE2) are for the provision of absorption chilling, HWSs and underfloor heating systems for the store as described in Figure 3. The results from this analysis using energy balance over the individual components are described below.

- The heat reclaimed (Q1) can provide an average of 21 kW of heat in summer and save 15 259 kWh of electricity, if the air-conditioning is provided by vapour compression systems.
- The heat reclaimed (Q2) can provide an average of 22 kW and can provide 145 213 kWh of heat for district heating for export during winter.

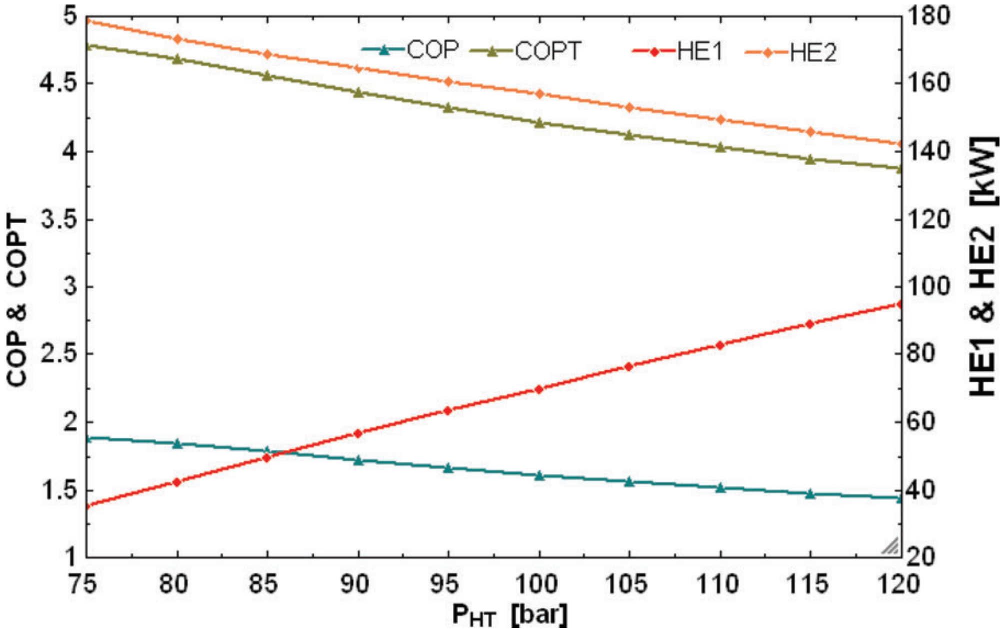

Figure 2. *COPT, COP, HE1 and HE2 versus discharge pressures.*

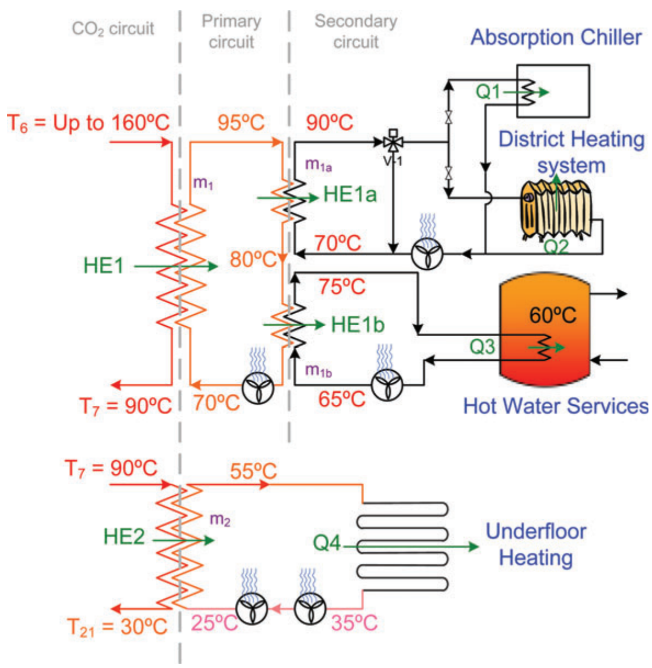

Figure 3. *Schematic diagram of a potential HE system.*

- The heat reclaimed (Q3) can provide an average of 18 kW of heat for domestic HWSs and can save 158 631 kWh of the hot water demand.
- The heat reclaimed (Q4) can provide up to 118 kW of output for underfloor heating of the store in winter. This would save 620 353 kWh per year, which is currently provided by gas burners within the air-handling unit.

4 THE ENVIRONMENTAL IMPACT OF SUPERMARKETS WITH THE SYSTEM

The environmental impact of the store was investigated using the Excel model. It was shown that the heat reclaimed has the potential to offset existing energy provision and to produce significant energy savings of 94% of HWS, 61% of heating and 56% of air-conditioning energy use. This equates to 13% reduction in electricity consumption of refrigeration systems. This resulted in a total gas consumption reduction of 70%. Assuming the same leakage rate for the CO_2 system, the direct emission represents 0% of the store CO_{2e} emission, compared with 102% of the indirect emission from the energy used (100% from indirect emission and 2% gained from the space heating export). Combining the direct and indirect impacts, it can be shown that the total CO_{2e} emission of the store can be reduced immediately by 34% as the store CO_{2e} emission has reduced from 3 025 348 to 1 987 010 kg CO_{2e}, equivalent to 355 kg CO_{2e}/m^2.

5 CONSTRUCTION, COMMISSIONING AND TESTING OF THE NOVEL SYSTEM

To prove the concept of the system, an experimental rig has been built for the purpose of validation and will be tested to compare the outputs with the predicted results. After assembling, the rig was delivered to the laboratory of London South

Figure 4. *Picture of the experimental rig.*

Bank University to be finally installed and connected to ancillary components and prepared for commissioning (Figure 4).

5.1 Testing of system

During commissioning, it was found that there was a small obstruction in the LT line that created a large pressure drop which prevented the system from stabilizing at the desired LT pressure/temperature. Consequently, the LT stage has been isolated and the system has been operated and tested with the MT stage alone. The calculated results from the steady-state experiment were analysed and compared with the predicted results from the EES model. The raw data available for the analysis were collected during the steady state for a period of 120 min. The following are the important setup parameters to take into consideration in order to understand the operation of the system.

- The expansion valve of the BP circuit was set up to drop the pressure to the MT pressure.
- The HT compressor operated according to its suction pressure, which was the MT evaporating pressure. In order to obtain an evaporating pressure of 26 bar as designed, the compressor cycled between pressures of 24 and 37 bar. When the evaporating pressure was <24 bar, the compressor started running at full load until it reached 37 bar and then switched off. The pressure decreased gradually to 26 bar. As the heat transfer occurred in the evaporator, the pressure continues to decrease and when it reached 24 bar, the compressor started running again.
- When the compressor switched off because the MT evaporating pressure was reached, this affected the rest of the system such as the mass flows, the discharge temperature and consequently the heat reclaimed.

Table 2. *Comparison of results.*

Variables		Design/model results	Testing results
Pressures (bar)	P_{MT}	26	27
	P_{HT}	75	75
	$P_{receiver}$	38	38
Temperatures (°C)	3	15	15
	4	9	14
	5	22	26
	6	116/130	109/124
	7	65	64
	8	30	30
	10	3.5	3.5
	18	15	25
	19	3.5	6.5
	22	20	26
	23	95	74
	24	70	62
	25	50	38
	26	25	28
Mass flows (kg/s)	m_{HT}	0.019	0.021
	m_{HT}	0.024	0.025
Powers (kW)	Q_{MT}	4.5	4.7
	W_2	2	2
	HE1	1.7	1.7
	HE2	4.5	4.5
COP		2.2	2.3
COPT		5.2	5.5

6 RESULTS

Table 2 compares the results of the model and the testing of the CO_2 system and demonstrates that the results are similar.

Temperature 6 from Table 2 shows two temperatures, this is because there are two thermocouples to measure them. One is located at the discharge of the compressor and the other is located before the HX (HE1) at a distance of 2 m. Over this distance, 13 K has been lost through pipeline heat transfer to the ambient. The variables are referred to in Figure 1.

The temperatures of the HE system (points 23–26) do not match with the design temperatures. Point 23 was designed for 95°C compared with 74°C from the test. The reason for this difference is that the return temperature from the HX, HE1, could not reach 90°C because of the lack of MT cooling demands that cause the transcritical compressor cut-off when the suction pressure is reached, and this impacted on the discharge temperature consequently and the heat reclaimed. This also affected the temperatures recovered by the HX, HX2 (points 25 and 26), because the CO_2 temperature at point 8 entering HX2 is 25 K below the design temperature. The reason for this is that when the compressor cuts off, the temperature at point 24 rapidly decreases below 50°C, which directly impacts on the temperature at point 8 from 90°C (design) to 64°C (test). This results in low HX2 HE temperatures (points 25 and 26).

7 CONCLUSION

This paper describes an R744-based system with HE and its investigation in a supermarket application. Experimental and modelling investigations have been described, and this has been proved to provide a large potential reduction in CO_{2e} emissions compared with a conventional HFC-based system.

Furthermore, the following investigations need to be carried out when implementing this type of system in the retail sector:

- the financial and economical aspects of implementing in new build or retrofit stores;
- further carbon emission reduction if the system is electrically self-sufficient, energy provided by wing turbines or photovoltaic systems;
- the potential use of the heat reclaimed in the new regeneration plan for new eco-stores, which includes the development of new housing, shopping, leisure centres and tourism facilities [20].

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