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Energy analysis of alternative CO_2 refrigeration system configurations for retail food applications in moderate and warm climates

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ABSTRACT

Refrigeration systems are crucial in retail food stores to ensure appropriate merchandising of food products. This paper compares four different CO_2 refrigeration system configurations in terms of cooling performance, environmental impact, power consumption and annual running costs. The systems studied were the conventional booster refrigeration system with gas bypass (reference system), the all CO_2 cascade system with gas bypass, a booster system with a gas bypass compressor, and integrated cascade all CO_2 system with gas bypass compressor. The weather conditions of London, UK, and Athens, Greece, were used for the modelling of energy consumption and environmental impacts to represent moderate and warm climatic conditions respectively. The control strategies for the refrigeration systems were derived from experimental tests in the laboratory on a conventional booster refrigeration system. The results from the analysis showed that the CO_2 booster system with gas bypass compressor can provide best performance with 5.0% energy savings for the warm climate and 3.65% for the moderate climate, followed by the integrated cascade all CO_2 system with gas bypass compressor, with 3.6% and 2.1% savings over the reference system for the warm and moderate climates respectively.

1. Introduction

Minimising the impact of climate change and reducing carbon emissions have been the key environmental objectives over the last few years for the food retail industry. Supermarket refrigeration systems are one of the largest consumers and emitters of high GWP refrigerants. GHG emissions from refrigeration applications can be split into two main categories, the "direct" and "indirect". Direct emissions are from refrigerant leakage that can arise from charging, normal operation, and repair or recovery of refrigerant from the system. The extensive pipe-work, large number of pipe joints and poor maintenance increase the possibilities for refrigerant loss. Indirect emissions arise from the generation of electricity used to power vapour compression systems.

A number of refrigeration system solutions for retail food applications using natural refrigerants have been developed to reduce environmental impacts. CO_2 has emerged as a credible natural refrigerant to replace HFCs in retail food applications. It is non-flammable and non-toxic, it has zero Ozone Depletion Potential (ODP) and negligible Global Warming Potential (GWP = 1). Alongside its excellent environmental characteristics, CO_2 has favourable thermophysical properties such as high density, specific heat, volumetric cooling capacity, latent heat and thermal conductivity. CO_2 is a high pressure refrigerant and depending on the ambient temperature the system can operate in transcritical or subcritical mode. In transcritical mode the heat rejection heat exchanger operates as a gas cooler and in subcritical mode as a condenser. Transcritical operation is less efficient than subcritical operation due to the high pressures on the gas cooler side of the system.

To increase the efficiency of CO_2 refrigeration systems for food retail refrigeration applications, a number of different system configurations have been considered that fall into three major categories: (i) indirect, (ii) cascade and (iii) all CO_2 transcritical systems. In indirect systems, the CO_2 is used in a similar manner as a secondary coolant in the low pressure, evaporator side, of the system. For the cascade solution, the system is divided into two different sub-systems which both exchange heat at the cascade heat exchanger which operates as condenser for the LP side and evaporator for the HP side. The HP side can operate with any HFC refrigerant or natural refrigerant such CO_2 or ammonia. The LP side uses CO_2 as the refrigerant which operates in the subcritical mode without any direct influence from ambient temperature as it is rejecting heat at constant temperature and pressure conditions to the cascade heat exchanger. For all-CO_2 systems, the CO_2 is the only refrigerant employed on the system. This offers the

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Nomenclature

a	recycling factor (%)
COP	coefficient of performance
DX	direct expansion
E	energy consumption (kW)
GHG	greenhouse gas emissions
GWP	global warming potential
HP	high pressure
L	annual refrigerant leakage (%)
LP	low pressure
LT	low temperature
m	amount of refrigerant charge (kg)
MT	medium temperature
n	operating lifetime of the refrigeration systems (years)
P_{cond}	condenser outlet pressure (bar)
P_{GC}	gas cooler outlet pressure (bar)
Q_{MT}, Q_{LT}	medium and low cooling capacity (kW)
T_{amd}	ambient temperature (°C)
T_{cond}	condenser outlet temperature (°C)
TEWI	Total Equivalent Warming Impact
T_{GC}	gas cooler outlet temperature (°C)
W	compressor power consumption (kW)
β	indirect emission factor ($kgCO_2$/kW h)
ΔT_{app}	approach temperature (K)

Subscripts

C/GC	condenser/gas cooler
com	compressor
HT	high temperature
LT	low temperature

advantage of elimination of the direct emissions from the refrigerant but the disadvantage that the system has to operate at higher pressures. All CO_2 transcritical systems have gained wide acceptance in retail food refrigeration applications with installations in Europe growing 117% from 2011 to 2013 from 1330 to 2885 stores [1]. This growth is continuing and it is foreseen that the numbers can increase to more than 32,000 by 2020 [2], linked to research and development aimed at improving the efficiency of the system particularly during operation at high ambient temperatures.

Dubey et al. [3] presented a thermodynamic analysis of a R744/R1270 cascade refrigeration system for different design and operating parameters. The authors reported that the R744/R1270 system provided better performance compared to the subcritical cascade CO_2 cycle and N_2O/CO_2 transcritical cycle. Ommen and Elmegaard [4] used the exergy cost based method of thermoeconomic analysis to investigate exergy losses of a CO_2 refrigeration system. The identified the cost required to provide refrigeration to the frozen food cabinets to be double the cost required for chilled food cabinets. Goodarzi and Gheibi [5] studied the performance of a two-stage transcritical CO_2 system with different design modifications including internal heat exchanger, intercooler, ejector and separator. The authors reported that the COP can be improved by 27% and exergy destruction decreased by 15% compared to conventional systems. Singh et al. [6] provided comparison between a number of CO_2 system configurations including the basic CO_2 transcritical cycle, the transcritical CO_2 refrigeration cycle with internal heat exchanger (IHX), a transcritical cycle with a work recovery expander, a transcritical CO_2 cycle with both IHX and work recovery expander and a multi-stage compressor CO_2 transcritical system with intercooler between the compressors. The authors compared the systems under different parameters such as compressor discharge pressure and temperature, refrigerant mass flow rate and interstage pressure. They reported that the systems with IHX provided better performance at higher ambient temperatures. At high ambient temperatures the compressor efficiency dropped significantly and impacted negatively on the overall system COP.

Ge and Tassou [7] presented a thermodynamic analysis of a conventional CO_2 booster refrigeration system. Sensitivity analysis was performed to identify the main parameters that affect the performance of booster systems when they operate in the transcritical mode at ambient temperatures between 25 and 40 °C. The optimal gas cooler pressure was determined as a function of the ambient temperature, effectiveness of the internal heat exchanger located downstream the condenser/gas cooler and isentropic efficiency of the high stage compressors [14]. Gullo et al. [8] presented the energy and environmental performance of different CO_2 system configurations including a cascade R134a/CO_2 solution, a conventional booster system, a CO_2 booster system with dedicated mechanical subcooling, and a CO_2 booster system with parallel compression and mechanical subcooling. The evaluation of these systems was made for two different climate zones of Athens, Greece and Valencia, Spain. The system which combined parallel compression with dedicated subcooling showed best performance. Beshr et al. [9] presented a comparative study based on the environmental impact of supermarket refrigeration systems using low GWP refrigerants. The authors used an open-source Life Cycle Climate Performance (LCCP) framework to compare the systems. The EnergyPlus simulation tool was used to simulate the hourly performance of the systems. The authors reported that the booster CO_2 system provided the lowest CO_2 equivalent emissions when operating in cold climates. Sharma et al. [10] performed a comparative analysis of various CO_2 system configurations for different climatic conditions in the USA. They found the performance of the CO_2 booster system with bypass compressor to be similar to that of R404A direct expansion systems.

The energetic, exergetic, economic and exergoeconomic analyses of a CO_2 refrigeration machine for ambient conditions higher than 40 °C were investigated by Fazelpour and Morosuk [11]. The exergetic and exergoeconomic analyses showed that the most important exergy loss components in the system are the expansion valves. Reducing the exergy destruction during the expansion process should lead to better thermodynamic performance and decrease the overall operating cost of the system. The use of an ejector to replace the high stage expansion valve of a transcritical CO_2 system was investigated by Bai et al. [12]. Exergy analysis of the system was carried out by splitting the exergy destruction into unavoidable/avoidable and endogenous/exogenous. It was identified that 43.4% of the total exergy destruction could be avoided by improving the performance of system components such as the compressor, ejector, evaporator and gas cooler. Mosaffa et al. [13] investigated the performance of CO_2/NH_3 cascade refrigeration system. They considered the effect of CO_2 evaporating temperature on the low pressure cascade and NH_3 condensing temperature on exergy destruction, COP and GHG emissions of the system. They identified that increasing evaporating temperature and reducing condensing temperature improved the exergy efficiency and the COP of the system.

Tsamos et al. [15] reported experimental results from a CO_2 booster system which highlights the effects of different condenser/gas cooler sizes and designs. The CO_2 booster system was experimentally tested under various ambient conditions, air flow

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rates and approach temperatures. Santosa et al. [16] presented an investigation of the local heat transfer coefficients in a condenser/-gas cooler coil using experimental data and CFD modelling. They showed that coil design enhancements such a slit fin can lead to smaller heat exchanger coils.

This paper presents results from simulation models that have been developed to investigate different CO_2 refrigeration system configurations in supermarkets. The systems studied were, (i) the conventional booster refrigeration system with gas bypass, (ii) all CO_2 cascade system with gas bypass, (iii) a booster system with a gas bypass compressor, and (iv) integrated cascade all CO_2 system with gas bypass compressor. These are referred to as Systems 1, 2, 3 and 4 respectively. The weather conditions of London, UK and Athens, Greece were used for the modelling of energy consumption and environmental impacts to represent moderate and warm climatic conditions respectively. The control strategies for the refrigeration systems were derived from experimental tests in the laboratory on a conventional booster refrigeration system and the modelling approach was validated against experimental test results.

2. Experimental facilities

The experimental investigations were carried out on a CO_2 transcritical booster system developed at the Centre for Sustainable Energy Use in Food Chains (CSEF) at Brunel University London. The system is shown schematically in Fig. 1. The main components of the CO_2 refrigeration system test rig are describe in Table 1. A specially designed gas cooler/condenser test unit, Fig. 2, was used to enable the variation of the flow and temperature of the air entering the heat exchanger to simulate different ambient conditions. The refrigeration system serves a chilled and a frozen food display cabinet in an environmental chamber which provides controlled conditions of air temperature, humidity and flowrate.

A number of tests were carried out to evaluate the cooling performance of the CO_2 booster refrigeration system with gas bypass. A schematic of the system is shown in Fig. 4 (System 1). A comprehensive instrumentation and data logging system was employed to record pressures and temperatures at different points in the system and refrigerant mass flowrate. The intermediate pressure was kept constant at 35 bar_g during all the experimental tests. The same intermediate pressure level was set for the simulations.

Table 1
Main components of the CO_2 test rig.

1	Condenser/gas cooler test unit
2	Additional cooling load
3	Condensing unit – safety protection for CO_2 receiver
4	MT display cabinet
5	LT display cabinet
6	CO_2 compressor rack

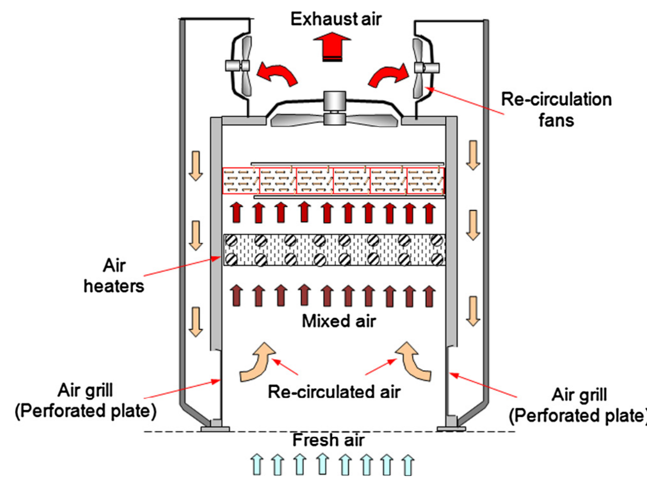

Fig. 2. CO_2 condenser/gas cooler test unit.

The MT and LT evaporating temperatures were set at −8 °C and −32 °C respectively with ±1 °C tolerance. The transition air temperature between subcritical and transcritical operation was found to be 26.8 °C.

3. Investigated climate conditions

The weather conditions for London, UK, considered as moderate conditions, and Athens, Greece, considered as warm, were used for the modelling of the alternative system configurations. The ambient temperature variation for these locations was obtained from Weather Underground [17]. The monthly average temperatures for the period July 2015-June 2016 are shown in Fig. 3 with Athens

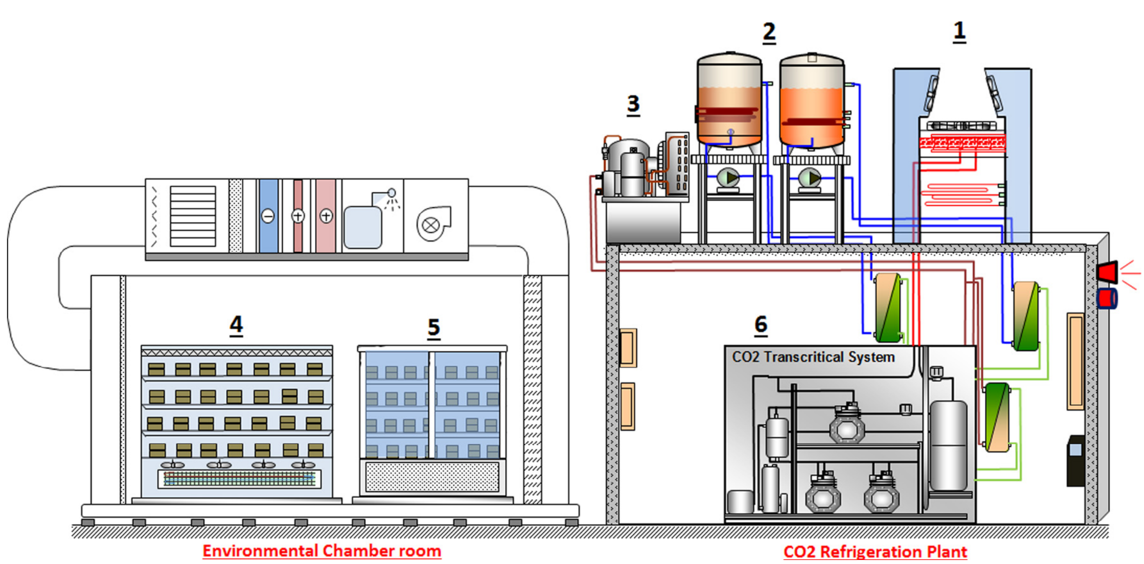

Fig. 1. Simplified drawing of CO_2 test facilities.

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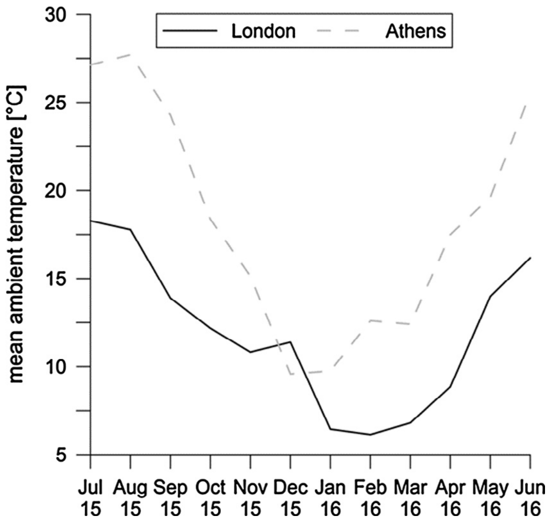

Fig. 3. Monthly mean temperatures for London and Athens.

showing higher average temperatures throughout the year apart from a few days in December 2015.

4. Investigated system configurations

Table 2 illustrates the four different supermarket refrigeration system configurations investigated in this work.

System 1 refers to a typical layout of a conventional booster CO_2 system. The system can operate in both subcritical and transcritical modes depending on the ambient temperature. Two-stage compression is commonly used. The refrigerant from the LT evaporator outlet is drawn into the low-stage compressor suction line. The discharge from the LP compressor mixes with the outlet of the MT evaporator, point 1. Before the mixed refrigerant enters to the suction line of the high stage compressors is mixed with the gas bypass refrigerant from the CO_2 liquid receiver at point 2. The mixed refrigerant enters the suction line of the HP compressor and is compressed to the gas cooler/condenser pressure. The pressure is controlled by the HP expansion valve and the variable speed fan of the gas cooler/condenser. System 2 is a cascade arrangement. The high pressure section is similar to the booster system, System 1. A separate LT section satisfies the LT cooling load and evaporates the refrigerant in the HT system in the cascade heat exchanger as shown in Fig. 4 (System 2). This configuration has been implemented in a number of UK supermarkets [18].

For both Systems 1 and 2, as the ambient temperature rises the pressure in the gas cooler/condenser will increase. The higher pressure will also result in greater flash gas quantities produced in the receiver which will need to be handled by the HP compressor. The higher pressures and the higher quantities of flash gas will lead to higher electrical power consumption by the HP compressor leading to a reduction in the COP of the system.

The relatively poor energy performance of booster systems at high ambient temperatures can be addressed to a certain extent

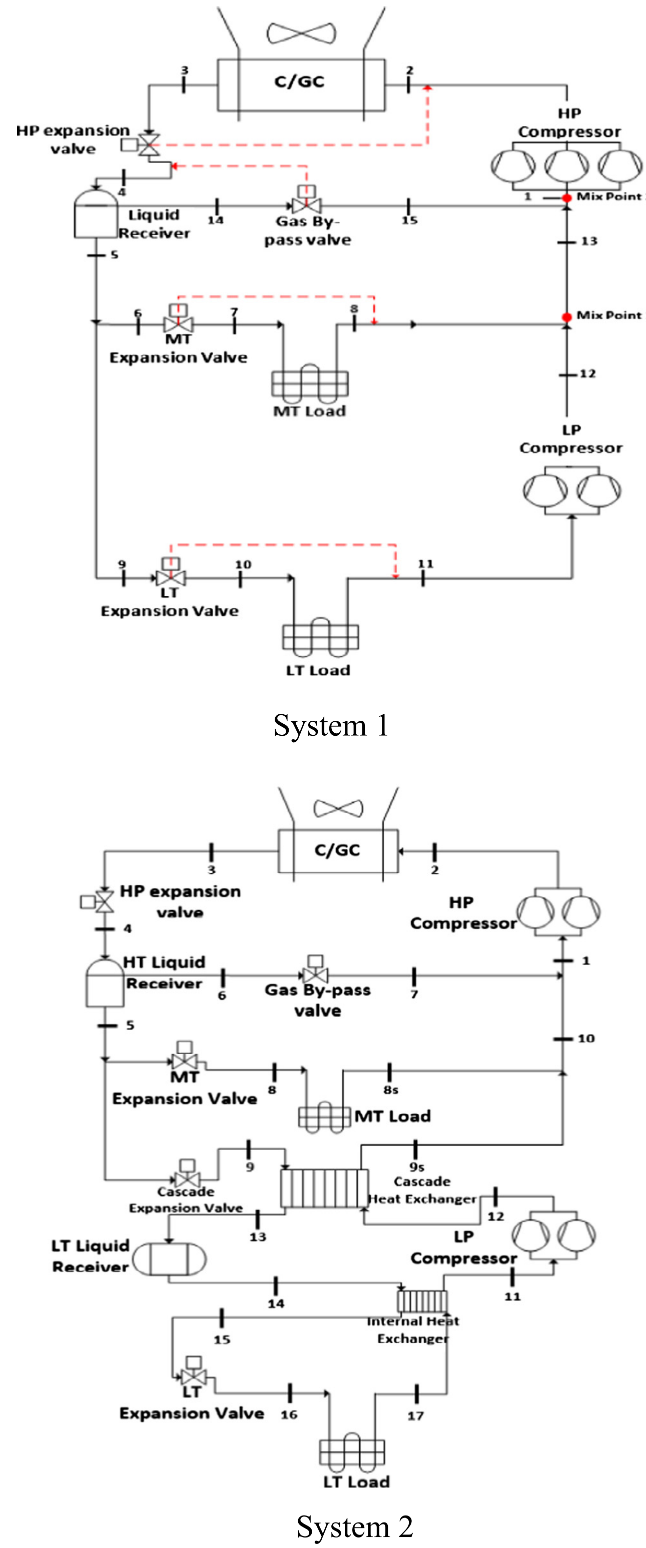

Fig. 4. Conventional booster and CO_2/CO_2 cascade configurations with gas bypass.

Table 2
Investigated refrigeration system configurations.

Investigated systems	System application
1	CO_2 booster system with gas by pass
2	CO_2/CO_2 cascade system with gas bypass
3	CO_2 booster system with bypass compressor
4	CO_2/CO_2 cascade system with bypass compressor

by dealing with the flash gas, that does not do any useful cooling, separately from the refrigerant flowing through the MT and LT evaporator coils. This can be achieved through an additional compressor to work in parallel to the HP compressor and handle the excess flash gas as shown in Fig. 5 (System 3). A similar modification can be applied to the cascade all CO_2 system with the parallel compression system arrangement shown in Fig. 5 (System 4).

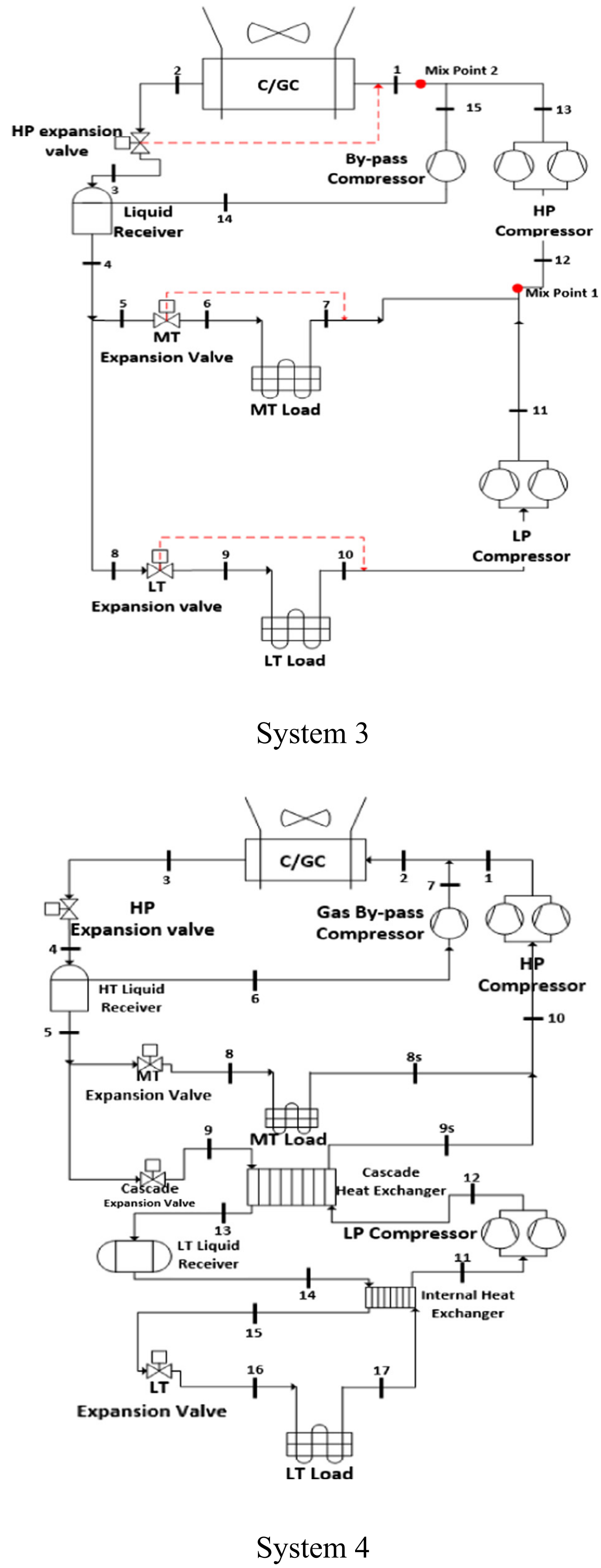

Fig. 5. Booster system with bypass compressor and CO_2/CO_2 cascade with bypass compressor.

5. System simulation

To assess the performance of the proposed gas by-pass (parallel compression) with the performance of more conventional CO_2 refrigeration systems data from a typical European supermarket as presented by Tassou et al. [19] of total sales area of 1400 m^2 was used. The refrigeration capacity was assumed to be 100 kW MT and 30 kW LT. Table 3 illustrates the parameters used in the simulation of the four systems load.

The pressure drop across the Gas Cooler/Condenser and evaporators were found to be small compared to the high pressures in the system and was not included in the simulations. For all the investigated systems, the liquid receiver was used as refrigerant storage device and gas-liquid separator. The intermediate pressure in the systems was set constant at 35 bar_g. In real system applications, the intermediate pressure is determined by the system settings and controls.

All the valves in the systems were treated as isenthalpic devices in the simulations. The control strategies for the high pressure side, including pressure control of the gas cooler/condenser were derived from the experimental test results as well as the transition temperature between subcritical and transcritical operation. Tables 4 and 5 show the control parameters and correlations developed from the experimental tests and used in the simulations.

The isentropic efficiency of the compressor can be expressed as a function of the pressure ratio (R_p) and was calculated from [20].

$$\eta_{isentropic} = 0.00476R_p^2 - 0.09238R_p + 0.89810 \tag{1}$$

where,

$$R_p = P_{discharge}/P_{suction} \tag{2}$$

The model of System 1 (conventional booster system) was validated against experimental results from the CO_2 test facilities at Brunel as discussed earlier. Fig. 6 shows a comparison between the COP of the system predicted with the model and calculated from the experimental test results.

It can be seen that the model provides a good prediction of the variation of the COP with ambient air temperature. Maximum difference between experimental and simulation values was 7.1%. Fig. 6 also indicates the dramatic reduction of the system COP with ambient temperature and the more rapid decrease at temperatures above the transition point between subcritical and transcritical operation at 26.8 °C.

6. Results and discussion

Eq. (3) was used to determine the COP of the four alternative systems for comparison purposes.

$$COP = \frac{Q_{MT} + Q_{LT}}{W_{com,HT} + W_{com,LT} + W_{MT,fans} + W_{LT,fans} + W_{\frac{C}{GC},fans}} \tag{3}$$

The cooling performance of the investigated refrigeration configurations is a function of the MT and LT loads divided by the total electrical power consumption of the system (HP and LP compressors, evaporator fans and condenser/gas cooler fan power). These values are given in Table 3. The variation of the COP of the four systems with ambient temperature is shown in Fig. 7. Comparing the booster system configurations, it can be seen that the system with the by-pass compressor (system 3) has a higher COP across the whole range of ambient temperatures with the percentage increase increasing with ambient temperature. This arises from the use of the by-pass compressor which deals only with the flash gas and operates across a smaller pressure differential compared to the case where the flash gas is handled together with the rest of the refrigerant flow in the system by the HP compressor.

The introduction of a by-pass compressor to the all CO_2 cascade configuration, system 4, also shows improvement in COP at ambient temperatures above 5 °C compared to conventional all CO_2 systems. This benefit again arises from the reduction in overall

Table 3
System parameters used in simulations.

	MT load				LT load			
Systems	1	2	3	4	1	2	3	4
Load (kW)	100	100	100	100	30	30	30	30
Evaporating temperature (°C)	−8	−8	−8	−8	−32	−32	−32	−32
Superheat (°C)	10	10	10	10	10	10	10	10
Subcooling (°C)	2	2	2	2	–	–	–	–
Cascade heat exchanger approach temperature (°C)	–	–	–	–	5	5	5	5
Evaporator fans, lights, defrost power consumption (kW)	10.5	10.5	10.5	10.5	7	7	7	7
Condenser/gas cooler fan power consumption (kW)	7.5	7.5	7.5	7.5	–	–	–	–

Table 4
Gas cooler/condenser outlet temperature.

	Ambient temperature range	Gas cooler/condenser outlet temperature
a	$T_{\_amb} < 0$	8 °C
b	$0 \le T_{\_amb} \le 10$	$T_{\_amb} + \Delta T_{\_app}$
c	$10 < T_{\_amb} < 26.8$	$T_{cond,\ \text{“c”}} = (-0.0144 \cdot (T^2_{amb})) + (1.1264 \cdot T_{amb}) + 9.8272$
d	$T_{\_amb} = \geq 26.8$	$T_{\_amb} + \Delta T_{\_app\_GC}$

Table 5
Gas cooler/condenser outlet pressure.

	Temperature range	Condenser/gas cooler temperature outlet
a	$T_{\_amb} < 0$	$P_{out} = f(T_{out}, \Delta T_{sub})$
b	$0 \le T_{\_amb} \le 10$	$P_{out} = f(T_{out}, \Delta T_{sub})$
c	$10 < T_{\_amb} < 26.8$	$P_{condenser\ \text{“c”}} = (0.0522 \cdot (T_{cond,\ \text{“c”}})^2) - (1.0178 \cdot T_{cond,\ \text{“c”}}) + 60.798$
d	$T_{\_amb} \le 26.8$	$P_{gas\ cooler,\ \text{“d”}} = (2.3426^{*}\ T_{GC,\ \text{“d”}}) + 11.541$

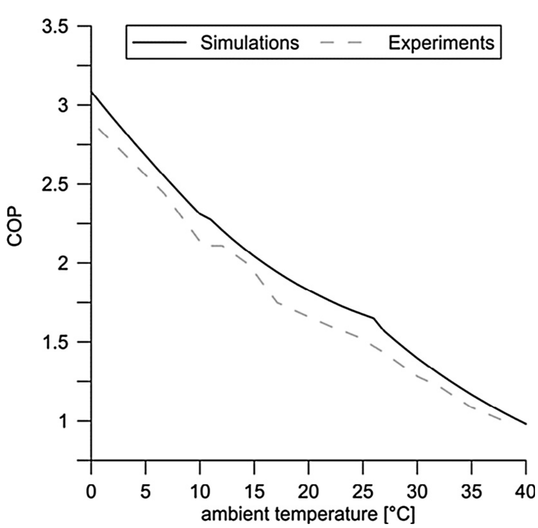

Fig. 6. Comparison between simulation and experimental results.

compressor power consumption of the HP compressors in the system for similar reasons as system 3.

The energy consumption of the four system configurations for London and Athens is shown in Table 6. It can be seen that system 3 has the lowest annual energy consumption for both climatic conditions, 5% for Athens and 3.6% for London, compared to system 1.

The environmental impacts of a refrigeration system can be determined as the sum of direct and indirect GHG emissions. The direct carbon dioxide emissions are a result of refrigerant leakage from the system. The indirect emissions arise from the emissions

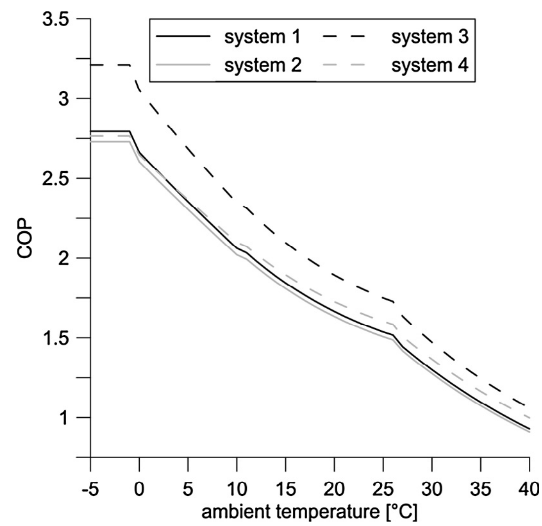

Fig. 7. Performance among the four investigated cases.

generated in the production and transmission of electricity consumed by the system. The TEWI (Total Equivalent Warming Impact) methodology illustrated below was used to assess the environmental impact of different refrigeration systems due to direct and indirect carbon dioxide emissions.

$$TEWI = TEWI_{Direct} + TEWI_{Indirect} \tag{4}$$

$$TEWI_{Direct} = GWP \cdot L \cdot n \cdot + GWP \cdot m \cdot (1 - a) \tag{5}$$

$$TEWI_{Indirect} = E \cdot \beta \cdot n \tag{6}$$

GWP for CO_2 is 1. “L” is the annual leakage rate which was assumed to be 15% [21]. “n” is the operating lifetime of the refrigeration systems, assumed for all configurations to be 10 years [21]. “m” is the refrigerant charge amount in kilograms for each system. For System 1 and 3 the refrigerant charge was assumed to be 1.2 kg/kW cooling load [22]. For Systems 2 and 4 the refrigerant charge of the HP cascade side was assumed to 1.2 kg/kW load and for the LP cascade 1 kg/kW load. The direct TEWI calculations take into account the recycling factor of the refrigerant which was assumed to be 95% [21]. The indirect TEWI calculations take into account the annual energy consumption (kW h/year) which calculated from the model. The indirect emission factor “β” was taken as 0.72 $kgCO_2$/kW h for Athens [23] and 0.53 $kgCO_2$/kWh for London [24].

Table 7 illustrates the results of the TEWI calculations for the four system configurations and the two climate conditions.

It can be seen that System 3 has the lowest life cycle TEWI from all the systems considered, arising primarily from its lowest energy consumption. The total lifecycle emissions from operation in the warm climate of Athens are much higher than those for operation

Table 6
Annual energy consumption.

System	1	2	3	4
Annual energy consumption –London (MW h)	577.40	589.08	556.33	565.17
Annual energy consumptions-Athens (MW h)	672.99	684.14	639.13	648.80

Table 7
TEWI results.

System	1	2	3	4
TEWI emissions – London (tonnes CO_2)	3061.3	3122.5	2948.9	2995.8
TEWI emissions – Athens (tonnes CO_2)	4845.9	4926.2	4602.1	4671.8

Table 8
Annual running cost for the different climate conditions.

System	1	2	3	4
Annual running cost – London (£)	83,166	84,828	80,112	81,385
Annual running cost – Athens (£)	104,987	106,726	99,067	101,214

in London due to the higher ambient temperatures and compressor power consumption. The annual electricity running costs of the four systems are shown in Table 8. Electricity prices were assumed to be £0.144/kW h, for London [25] and £0.156/kW h for Athens [26].

It can be seen that system 3 will result in annual cost savings of around £3000 compared to system 1, which is the most common system currently in use, for the London climatic conditions and around £6000 for the climatic conditions of Athens. For the cooling capacity of the systems under consideration in this paper, a reasonable estimate of the additional cost for a by-pass compressor system over the conventional booster refrigeration system will be in the region of £10,000. The system with the by-pass compressor will therefore have a payback period of the order of 3 years for operation at moderate climatic conditions and 2 years for warm climatic conditions.

7. Conclusions

In this paper, comparative studies of four different CO_2 commercial refrigeration system configurations have been performed. The evaluation was carried out for moderate and warm weather conditions and ambient temperatures. The weather conditions of London and Athens were used to represent the moderate and warm conditions respectively. The following conclusions can be drawn from the study.

- From the alternative system configurations considered, the CO_2 booster system with by-pass compressor (parallel compression) was found to be the most energy efficient system for both the moderate and warm climates. Energy efficiency improvement over the conventional CO_2 booster system was found to be in the region of 5.0% for the warm climate and 3.6% for the moderate climate.
- Operation of the CO_2 refrigeration system in warm weather conditions such as those of Athens, Greece, will result in up to 16% higher energy consumption compared to the London weather conditions.
- The Greenhouse gas emissions from the operation of the systems in Athens will be up to 50% higher than those in London due to the higher electrical energy consumption and the higher electricity generation emission factor for Greece.
- By-pass compression in all CO_2 booster systems is economically more effective in warm weather conditions. The study showed that payback periods of around 2 years can be achieved in warm weather conditions and 3 years in moderate weather conditions.

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