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Carbon Dioxide Refrigeration with Heat Recovery for Retail Applications

by

I. Colombo*[a][b][c], P. Johal[b], L. Jordan[b], I. Chaer[a] , J.F Missenden[a] & G.G Maidment[a]

[a]Department of Engineering Systems, London South Bank University, 103 Borough Road, London, SE1 0AA, UK; email: Colombi@lsbu.ac.uk
[b]Space Engineering Services Ltd, Causeway Central, Pioneer Park, Bristol, BS4 3QB; email: Enquiries@space-engineering.co.uk
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This paper presents the findings from an applied research study on a booster carbon dioxide (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 upon some experimental results. The energy consumption and carbon emission reduction using this novel system is investigated based on an existing supermarket as a case study.

1.0 Background

Food is an essential of life. The food industry is 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 (2011) reported that large proportion of carbon emissions are 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 temperatures but in doing so significantly contributes both directly and indirectly to greenhouse gas emissions. Directly greenhouse gas emissions can occur through the leakage of high GWP HFC refrigerants used in refrigeration systems, which can be as much as 30% of the system charge per year (Bostock, 2007). Indirect emissions are also significant as these systems are large consumers of electricity and are reported to consume around 4 $MtCO_{2e}$ per annum (Tassou et al, 2007). As well as the costs associated with leakage of refrigerants and energy, there are other reasons why reducing carbon emissions from the retail sector are important. This includes meeting requirements such as the Carbon Reduction Commitment.

In recent years, natural refrigerants have been proposed as an environmentally friendly solution for the refrigeration industry; these refrigerants do not contribute to ozone depletion and have low global warming potentials. These refrigerants include ammonia, hydrocarbons and carbon dioxide. Carbon dioxide (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.

Carbon dioxide offers significant advantages as a refrigerant since it is non toxic (Pearson & Gillies, 2004), non-flammable (IoR, 2003), environmentally benign (ODP=0 and GWP=1) (Lorentzen, 1994), has high refrigeration volumetric capacity (Campbell et al, 2007) and has high heat transfer coefficients (Yang et al (2006), Mastrullo et al (2009)). However, there are technical challenges to its application associated with its low triple, critical points and high operating pressure (Pearson, 2004). Overcoming these barriers represents significant engineering challenges.

The application of R744 as a refrigerant in retail is the subject of this paper. It is a result of a 3 year PhD (EPSRC) Industrial Case Award programme supported by Space Engineering Services Ltd. 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 carbon dioxide as a refrigerant and the utilisation of the reclaimed heat from the R744 cycle. By using a low GWP gas and utilising the waste heat from the cycle the direct and indirect emissions are very significantly reduced compared to conventional systems.

The following were the objectives of this research project:

- The development of a R744 refrigeration system with heat recovery
- The development of a computer model of the system
- The investigation of the potential applications of the reclaimed heat
- The demonstration of the environmental impact of supermarkets
- The construction, commissioning and testing of the novel system

These are described below:-

2.0 Development of a R744 novel refrigeration system with heat recovery

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 as much of the heat normally rejected to the ambient and to use it efficiently for other building services applications within supermarkets. The experimental system developed is shown in Figure 2 and is a R744 enhanced booster transcritical system which provides low temperature (LT) cooling for cold room and frozen food cabinets and medium temperature (MT) cooling for chilled food cabinets. This system is enhanced because it is composed of suction liquid heat exchangers 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 (F1), the receiver (E) 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 (F2), the liquid enters the 4.5 kW MT evaporator (B) at a pressure of 26 bar. The saturated vapour is superheated by 20K via a suction/liquid heat exchanger (SLHE2) (G). At the LT stage, the liquid is sub-cooled and throttled by an expansion valve (F3) before entering the 5 kW LT evaporator coil (A) at 14 bar. After evaporation, the gas is superheated by 20K by SLHE1 (G) to ensure complete evaporation as well as increasing the refrigeration effect. The LT superheated gas is compressed sub-critically (C) to the medium pressure where it is mixed with the gas from the MT evaporator at same pressure. This mixture is then further mixed with by-pass gas from the receiver, superheated by 9K (SLHE3) and then compressed to a discharge pressure of 80 bar by the MT transcritical compressor (D). The discharge gas can reach high temperatures according to the discharge pressure. The gas is cooled through two heat exchangers $HE1_{HE}$ and $HE2_{HE}$ (H) that reclaim the heat rejected. The cooled gas exits the second heat exchanger at 30°C and returns to the receiver after being throttled.

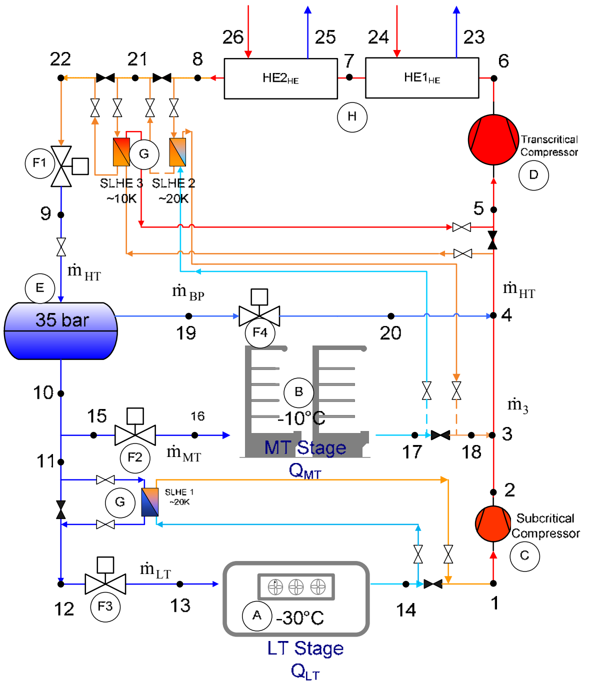

Figure 1: Conceptual design schematic

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:

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

The mass balance is:

$$\dot{m}_{HT} = \dot{m}_{BP} + \dot{m}_{LT} + \dot{m}_{MT}; \dot{m}_{BP} = \dot{m}_3 * \frac{(h_9 - h_{10})}{(h_{19} - h_9)}$$

Isentropic efficiencies of the transcritical and subcritical compressors are:

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

Compressor work equations are:

$$W_1 = \dot{m}_{LT} * (h_2 - h_1),$$
$$W_2 = \dot{m}_{HT} * (h_6 - h_5)$$

The standard CoP of is defined as:

$$CoP = \frac{Q_{MT} + Q_{LT}}{W_1 + W_2}$$

The heat reclaimed from $HE1_{HE}$ and $HE2_{HE}$ are calculated by:

$$HE1 = \dot{m}_{HT} * (h_6 - h_7), HE2 = \dot{m}_{HT} * (h_7 - h_8)$$

The overall CoP of the system including the heat reclaim is:

$$CoP_{overall} = \frac{Q_{MT} + Q_{LT} + HE1 + HE2}{W_1 + W_2}$$

3.0 Development a computer model of the novel system

The equations of the conceptual design described above 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 2. The calculated CoPs of the system with and without heat reclaim are 1.6 and 4.3 respectively. This is a steady state analysis based on the assumptions that all the heat can be recovered. However, section 3.0 considers the practical application of this system in a typical supermarket and the relative savings that can be practically achieved.

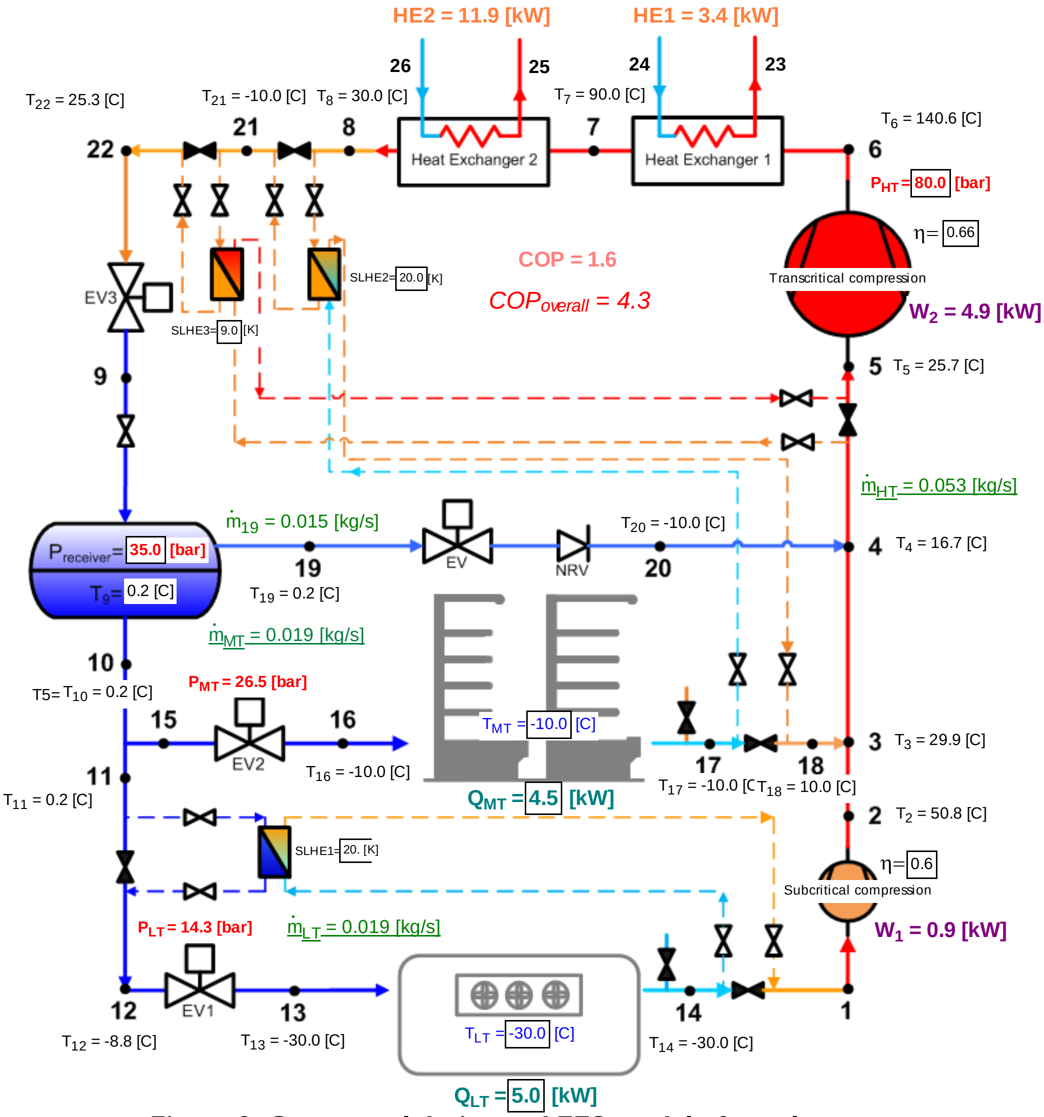

Figure 2: Conceptual design and EES model of novel system

3.0 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 (AM&T) tool. This tool provides automatic meter readings and data collection of mains and sub-metered electricity and gas and provides half hourly, weekly and monthly energy readings. The 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 sub-metered 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 (ie refrigeration LT and MT packs, fan & electronics and service shop services represent 33% of the total electricity use 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 tool and assuming CoPs of 2.00 for the MT and 1.00 for the LT packs, the calculated average cooling capacities obtained are 111 kW and 28 kW respectively.

Systems monitored	kWh/year
Bakery	611,974
Chicken Rostisserie	99,280
Lighting	899,550
HVAC Fans + Electronics	292,894
HVAC Refrigeration	27,162
Refrigeration HT packs	487,304
Refrigeration LT packs	243,420
Refrigeration Fan + Electronics	128,951
Refrigeration Service Shop Panel	387,806
Unsubmetered	530,967
Total Electricity Consumption	**3,709,307**
Heating	659,187
HWS	155,074
Total Gas Consumption	**1,017,826**

Table 1: Store energy consumption

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 2,498,543 kg/yr of CO_{2e}, equivalent to 442 $kgCO_{2e}/m^2$ (using carbon factors of 0.544 and 0.184 $kgCO_{2e}$/kWh respectively for electricity and gas). Also a refrigerant leakage rate of 90 kg/year (Carbon Trust, 2010) of R404A for the typical store is assumed. Therefore 88% of the store's total global warming impact is from energy use to run the store and 12% is due to refrigerant leakage.

Using the EES model developed with annual average loads Q_{MT} and Q_{LT} respectively of 111 and 28 kW and 80 bar discharge pressure: W_I= 5 kW; W_2= 67 kW; HE1= 40 kW; HE2= 171 kW; and CoP= 1.9 and $CoP_{overall}$= 4.8 were calculated.

3.1 Potential application

Figure 3 shows a suggested application of the heat reclaimed from the CO_2 system using two heat exchangers ($HE1_{HE}$ and $HE2_{HE}$). HE1 is the heat reclaimed by $HE1_{HE}$ from the high discharge temperature to 90°C of the CO_2 system. $HE2_{HE}$ 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 heat exchangers.

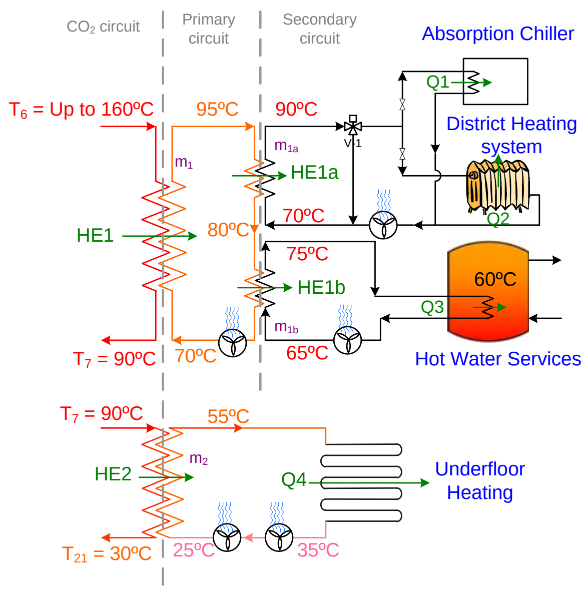

Figure 3: Schematic diagram of potential heat recovery systems

In the analysis of the system, the following assumptions about the destination of recovered heat have been made (Figure3):

- The heat reclaimed HE1a can provide a heat source (Q1) for an absorption chiller to provide cooling for the store, in summer period. When the air-conditioning is not required, Q2 can be used for export to district heating. During winter additional heating demand will be available and an assessment of relative values will be made for prioritisation.

- The heat reclaimed HE1b can provide heat for the domestic hot water services (Q3) in the store.
- HE2 heat reclaimed can provide heat (Q4) for underfloor heating system in the store according to seasonal demand.
- Any heat not utilised is rejected to atmosphere, although 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.

3.1 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 heat exchangers HE1 and HE2. These reclaim heat for the provision of absorption chilling, hot water services and underfloor heating systems for the store as described in Figure 3. The results from this analysis using energy balances over the individual components are described below:

- The heat reclaimed from HE1a (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 from HE1a (Q2) can provide an average of 22 kW and can provide up to 15,000 kWh/month for district heating for export during winter.
- The heat reclaimed from HE1b (Q3) can provide an average of 18 kW of heat for domestic hot water services which can save 160,000 kWh of the hot water demand.
- The heat reclaimed from HE2 (Q4) can provide up to 118 kW of heat output for underfloor heating of the store in winter. This would save 776,000 kWh per year which is currently provided by gas burners within the AHU.

5.0 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 produce significant energy savings of 94% of HWS, 72 % of heating, and 56% of air-conditioning energy use. This equate to 15% reduction in the refrigeration systems electricity consumption which is in line with the findings of Madsen (2008). This resulted in a total gas consumption reduction of 70%. Assuming the same leakage of 90 kg/year for the carbon dioxide system, the direct emission represents 0% of the store CO_{2e} emission compared to 102% of the indirect emission from energy used (2% gained from the space heating export). Combining the direct and indirect impacts it can be shown that the total CO_{2e} emissions of the store can be reduced immediately by 22% as the store CO_{2e} emission has reduced from 2,498,543 kgCO_{2e} to 1,952,624 kgCO_{2e}, equivalent to 346 kgCO_{2e}/m^2.

6.0 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. The LSBU team has provided the conceptual design and Spaces Engineering Services team has produced technical drawings, selected components and built the prototype. After assembling, the rig was delivered to LSBU's laboratory to be finally installed and connected to ancillary components and prepared for commissioning.

Figure 4: Picture of the experimental rig

6.1 Testing of system

During the commissioning, it was found that there was a small obstruction in the LT line that created a large pressure drop which prevented the system stabilising at the desired LT pressure/temperature. Consequently, the LT stage was 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 analysis was collected during steady state initially for a period of 120 minutes. The following are important set-up parameters to take into consideration in order to understand the operation of system:

- The expansion valve of the receiver was set up to maintain the pressure at 38 bar in the receiver.
- The expansion valve of the by-pass 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 below 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 heat transfer occurred in the evaporator, the pressure continues to decrease and when the pressure reached 24 bar the compressor started running again.
- When the compressor switched off because the MT evaporating pressure was reached, this affected on the rest of the system such as the mass flows, the discharge temperature and consequently the heat reclaimed.

6.2 Results

Table 2 compares the results from the model and the testing of the CO_2 system and demonstrates that the results are very similar. There are 2 temperatures for the discharge temperature (point 6) because there are 2 thermocouples to measure the discharge temperature. One is located at the discharge of the compressor and the other is located before HE1 heat exchanger at a distance of 2 meters. Over this distance 13K has been lost through pipeline heat transfer to the ambient. The variables are refers to Figures 1 and 2.

Variables		Units	Design/Model Results	Testing Results
Pressures	P_{MT}	Bar	26	27
	P_{HT}		75	75
	$P_{receiver}$		38	38
Temperatures	3	°C	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	m_{MT}	kg/s	0.019	0.021
	m_{HT}		0.024	0.025
Powers	Q_{MT}	kW	4.5	4.7
	W_2		2	2
	HE1		1.7	1.7
	HE2		4.5	4.5
CoP			2.2	2.3
$CoP_{overall}$			5.2	5.5

Table 2: Comparison of results

The temperatures of the heat recovery system (points 23, 24, 25 and 26)) do not match with the design temperatures. Point 23 was designed for 95°C compared to 74°C from the test. The reason for this difference is that the return temperature from heat exchanger HE1 could not reach 90°C because lack of load on the MT compressor caused the discharge compressor to modulate and this impacted on the discharge temperature consequently and the heat reclaimed. This also affected the temperatures recovered by heat exchanger $HE2_{HE}$ (point 25 and 26) because the CO_2 temperature at point 8 entering $HE2_{HE}$ is 25K below the design temperature. The reason for is when the compressor cut 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 the low $HE2_{HE}$ heat recovery temperatures (point 25 and 26).

7.0 Conclusion

This paper describes a R744 based system with heat recovery and its investigation in a supermarket application. An experimental and modelling investigation has been described and this has been shown to give large potential reductions in CO_{2e} emissions compared to a conventional HFC based systems.

Furthermore investigations need to be carried out on the implementation of this type of system in the retail sector. For instance:

- The financial and economical aspects of implementing in new build or retrofit stores.
- The CO_{2e} emission further 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 housings, shopping, leisure centres, tourism facilities and new housing (Bird, 2011).

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References

Beddington, J., 2011. *The Future of Food and Farming: Challenges and choices for global sustainability*. London, UK, The Government Office for Science.

Bird, B., 2011. *Energy saving in supermarkets*. CIBSE Home Countries South East, 19/01/2011, CIBSE.

Bostock, D., 2007. *Designing to minimise the risk of refrigerant leakage*. Institute of Refrigeration Annual Conference, 29th November 2007, UK.

Campbell. A., Missenden, J.F., Maidment, G.G., 2007. *Carbon Dioxide for Supermarkets*. Institute of Refrigeration, London, UK, 5th April 2007.

Carbon Trust, 2010. Refrigeration road Map. London, March 2010.

Department Environment Food and Rural Affairs (DEFRA), 2006e. *Primarily environmental*. Food industry sustainability strategy. Norwich, Stationery Office, pp. 30.

Institute of Refrigeration (IoR), 2003. *Code of practice for refrigeration systems utilising carbon dioxide systems*. 1st edition. London, IOR.

Madsen, K.B., 2008. *Transcritical CO_2 system in a small supermarket*. Proceeding 8th Gustav Lorentzen Conference on Natural Working fluids, Copenhagen, Denmark, 7-10th September 2008.

Mastrullo, R., Mauro, A.W., Rosato, A. and Vanoli, G.P., 2009. *Comparison of R744 and R134a heat transfer during flow boiling in horizontal circular smooth*. International conference on renewable energies and power quality, Valencia, Spain, 15-17th April 2009.

Pearson, A.B., and Gillies, A., 2004. *Safety considerations for carbon dioxide systems*. Proceeding 6th Gustav Lorentzen conference on natural working fluids, International Institute of Refrigerant, Glasgow, August 2004.

Pearson, A.B., 2004. *The critical point and transcritical refrigeration*. African heating and cooling, Issue September 2004, pp. 36-39.

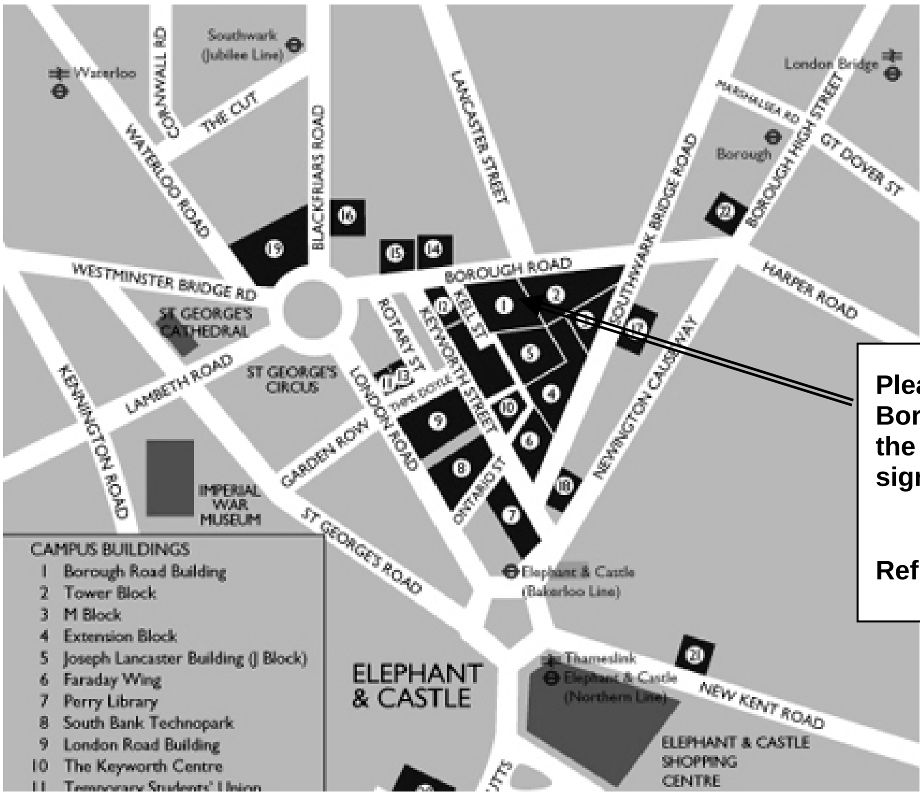
Tassou, S.A., Ge, Y., Hadawey, A., Evans, J., 2007. *Energy consumption and conservation in food retailing*. Research paper, Brunel University.

Yang, J., Ma,Y., Liu, S., Zeng, X., 2006. *Comparison investigation on the heat transfer characteristics for supercritical CO_2 fluids and conventional refrigerants*. Proceeding 7th IIR Gustav Lorentzen Conference on Natural Working Fluids, Trondheim , Norway, 28-31th May 2006.

Nomenclature

AHU	Air Handling Unit	kWh/m^2	Kilo Watt Hour per Square Meter
AM&T	Automatic Monitoring and Targeting tool	kWh/month	Kilo Watt Hour per Month
BP	By-Pass	kWh/year	Kilo Watt Hour per Year
CO_2	Carbon Dioxide	LSBU	London South Bank University
CO_{2e}	Carbon Dioxide Equivalent	LT	Low Temperature
CoP	Coefficient of Performance	$\dot{m}$	Mass flow rate (kg/s)
$CoP_{overall}$	Overall Coefficient of Performance	m^2	Square meter
EES	Engineering Equation Solve	MT	Medium Temperature
EICT	Energy Information and Communication Technology	$MtCO_{2e}$	Million Tonne of Carbon Dioxide equivalent
ESPRC	Engineering and Physical Sciences Research Council	ODP	Ozone Depletion Potential
GWh	Giga Watt Hour	P	Pressure (bar)
GWP	Global Warming Potential	PhD	Doctor of Philisophy
h	Enthalpy (kJ/kg)	Q (1,2,3,4,5)	Heat reclaimed of $HE1_{HE}$ and $HE2_{HE}$ (kW)
Δh	Delta enthalpy (kJ/kg)	Q_{LT}	Cooling capacity of LT evaporator (kW)
$HE1_{HE}$	Heat exchanger 1	Q_{MT}	Cooling capacity of MT evaporator (kW)
$HE2_{HE}$	Heat exchanger 2	R404a	Zeotropic Blends [R-125/143a/134a (44/52/4)]
HFC	Hydrofluorocarbon	R744	Carbon Dioxide refrigerant
HT	High Temperature	SES	Space engineering Services
HWS	Hot Water Services	SLHE	Suction Liquid Heat Exchanger
IoR	Institude of Refrigeration	SSP	Shop Service Panel
K	Kelvin	UK	United Kingdom
kg/year	Kilo Gram per year	W_1	Power input of subcritcal compressor (kW)
$kgCO_{2e}$	Kilo Gram of Carbon Dioxide equivalent	W_2	Power input of transcritcal compressor (kW)
$kgCO_{2e}/kWh$	Kilo Gram of Carbon Dioxide equivalent per Kilo Watt Hour	η_{Sisent}	Isentropic efficiency of subcritical compressor
$kgCO_{2e}/m^2$	Kilo Gram of Carbon Dioxide equivalent per Square Meter	η_{Tisent}	Isentropic efficiency of transcritical compressor
kW	Kilo Watt	%	Percentage
kWh	Kilo Watt Hour	°C	Degree Celsius

MAP AND DIRECTIONS



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Refreshments served from 5.15pm