THERMODYNAMIC AND ECONOMIC EVALUATION OF TRIGENERATION SYSTEMS IN ENERGY-INTENSIVE BUILDINGS

Chatzopoulou M.A.[1,*], Salvador A.[2], Oyewunmi O.A.[1], and Markides C.N.[1,2]
*Author for correspondence
[1]Clean Energy Processes (CEP) Laboratory, Department of Chemical Engineering,
Imperial College London, South Kensington Campus, London, SW7 2AZ, UK
[2]Centre for Process Systems Engineering, Department of Chemical Engineering,
Imperial College London, South Kensington Campus, London, SW7 2AZ, UK
E-mail: maria-anna.chatzopoulou11@imperial.ac.uk

ABSTRACT

Within the building sector, supermarkets are responsible for 3-5% of the electricity consumed in developed countries. To mitigate the associated environmental impact of this consumption, a growing interest has been developed in local combined heat and power (CHP) systems, due to their higher total efficiencies. However, CHP efficiency is highly dependent on the thermal output utilisation. In food retail buildings, where refrigeration dominates the building energy use, a promising means for utilising the thermal output is by using this to operate absorption chillers. This paper reports on a technical feasibility and financial viability study of an ammonia-water absorption chiller, coupled to a CHP unit, that is also compared to a conventional electrically-driven vapour-compression equivalent. A typical distribution centre located in the UK is selected as a case-study. Three alternative systems are considered: i) a conventional grid connected system; ii) a CHP system; and iii) a trigeneration system. Typical daily cooling, heating and hot-water demand data are provided on an hourly basis, and the system's ability to cover these loads is assessed. The results indicate that the trigeneration system can reduce the electricity demand by 16% compared to the baseline system, while offering a 48% annual energy cost saving. The system's primary energy utilisation rate exceeds 60%, while the power-to-heat ratio of the building demand improves from 7.0 to 0.9, thereby more closely matching the CHP system generation profile. Furthermore, the trigeneration system achieves CHPQA rating of 106, and it is qualified for enhanced capital allowance for the CHP plant. The results highlight the great energy and cost savings potentials of integrating trigeneration systems in energy-intensive buildings.

INTRODUCTION

The provision of heating and cooling in buildings and in the industry is responsible for more than 50% of the total primary-energy consumption in the EU [1]. Among the different building typologies, supermarkets have highly energy intensities and are responsible for 3-5% of the total electricity consumption in developed countries [2]. The electricity demand for refrigeration is the major energy consumer in this industry, corresponding to up to 60% of the total consumption [3]. To mitigate the environmental impact of retail food stores, retailers in the UK (Asda, Sainsbury's, Tesco, etc.) have published sustainability plans with commitments to reduce their energy consumption and emissions. Adding to this, major food retailers worldwide and members of the Consumer Goods Forum have committed to using natural refrigerants, or refrigerants with global-warming potential (GWP) lower than 150, effective from October 2016 [4]. This commitment is well-aligned with recent policies, e.g. the F-Gas regulations in Europe, which demand a gradual phase-out of refrigerants currently used in air-conditioning (AC) systems based on their environmental impact.

Traditional refrigeration and AC systems use electric-driven vapour-compression cooling systems, which have high energy consumption and employ high-GWP refrigerants such as chlorofluocarbons (CFCs) or hydrochlorofluorocarbons (HCFCs). Refrigerant leaks are the main source of direct emissions of such refrigeration systems in supermarkets. Therefore, the adoption of carbon dioxide (CO_2) has been gaining an interest as an alternative natural working fluid in these applications. Research to-date has shown that CO_2 refrigeration systems can save up to 90% of the direct emissions compared to conventional systems [2].

Another pathway for limiting the retail food sector's environmental impact is with local combined heat and power (CHP) generation. CHP has high overall efficiencies, and a significant cost savings potential. However, CHP efficiencies are highly dependent on the utilisation of the thermal output. In a supermarket setting, where the heating demand is very low in comparison to the demand for refrigeration, a promising means for utilising the thermal output of CHP engines is by supplying this to thermally-driven cooling technologies, such as absorption chillers, thus forming so-called trigeneration systems.

Absorption refrigeration systems using the ammonia-water working pair are a viable alternative to conventional refrigeration systems. Despite having lower COPs than their electrically-driven compression counterparts, absorption systems offer a number of benefits: they can make use of waste- or low-grade heat; they have minimal electrical consumption; they are noise and vibration free; and they utilise environmental benign working fluids, with low GWPs and ozone-depletion potentials (ODPs) [5]. Some previous investigations of trigeneration systems with CO_2 refrigeration in supermarkets have shown payback periods of less than 4 years [6], and primary energy consumption reductions of 10%-30% [2].

Despite their benefits, trigeneration systems are not widely adopted, and it is not clear how to size, integrate and operate these systems efficiently. In this context, this paper performs a technical and financial viability study of deploying trigeneration systems in supermarkets, aiming to minimise a building's dependency on grid electricity imports (self-sustained system) and maximise the annual energy cost savings. A steady-state thermodynamic model is developed to evaluate the system energy performance. The aim of this study is twofold: (1) to compare the annual performance of trigeneration to conventional systems; and (2) to provide guidelines for the system design. The structure of this paper is as follows: first alternative refrigeration systems for supermarkets are presented, after which the trigeneration system considered in the present work is presented, along with the modelling methodology. Next, the case study of an energy intensive food distribution centre is introduced. The annual energy consumption and energy costs of the proposed trigeneration system are estimated and compared to a conventional system. Finally, a summary of the key findings from this work and recommendations for the design of such systems are presented.

SYSTEM DESCRITION AND MODELLING METHODOLOGY

A typical heating, ventilation and air-conditioning (HVAC) system, along with an electrically-powered refrigeration system in a supermarket facility are illustrated in Figure 1a. In this system, electricity for lighting, space cooling and refrigeration is provided by the grid. The demands for space heating and low-temperature hot water (LTHW) are covered by natural gas boilers. In Figure 1b, a similar system is presented, here driven by a CHP.

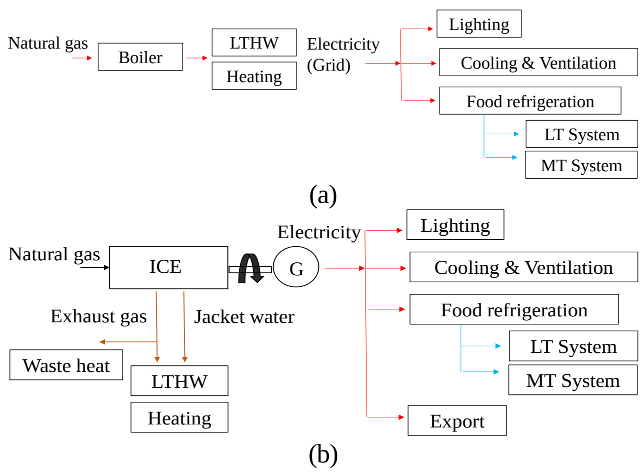

Figure 1 Block diagram of (a) conventional, and (b) CHP-driven, HVAC and CO_2 refrigeration system in a supermarket application

The refrigeration system in supermarkets comprises two sub-systems: (1) low temperature (LT) system (evaporation temperature of the refrigerant = -32 °C) for the frozen food cabinets; and (2) medium temperature (MT) system (evaporation temperature of the refrigerant = -7 °C) for the cold-food cabinets. As mentioned above, the use of CO_2 in refrigeration systems, especially when these feature food cabinets is gaining interest due to its environmental benign characteristics (natural refrigerant) and its relatively low cost per kg of charge.

A typical direct expansion (DX) CO_2 system is presented in Figure 2. The system operates at 3 pressure levels. The LT system corresponds to the lowest pressure (P_{LT}), in line with the low evaporation temperature required. The MT system operates at a higher pressure than the LT system (P_{MT}), and the heat rejection process in the condenser is performed at the highest pressure of the system (P_{cond}). Liquid CO_2 is stored in the liquid receiver tank, at slightly elevated pressure than the MT system. The refrigerant is then distributed to the MT and LT systems by expansion valves. The refrigerant from the LT system ($\dot{m}_{LT}$) flows thought the LT compressor (C1), which increases its temperature and pressure to P_{MT}. The MT refrigerant ($\dot{m}_{MT}$) is then mixed with $\dot{m}_{LT}$, and enters the high-pressure compressor (C2). Finally, the mixture enters the condenser, where it releases heat to the atmosphere (in air-cooled systems) or to a water circuit (in water-cooled systems).

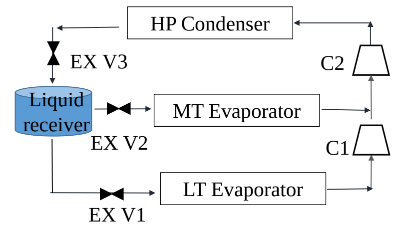

Figure 2 Schematic diagram of a conventional DX CO_2 refrigeration system in a supermarket application

Typically, the use of a high pressure compressor (C2) will lead to a high energy consumption. Looking at the electricity demand of the entire refrigeration system, this is split: 10% for the cabinets, and the remaining amount as 25% for the LT system and 75% for the MT system [2]. Therefore, investigating ways of reducing the energy consumption of the MT system will have major impact on the overall system performance and costs in this type of refrigeration system. The use of absorption chillers is becoming increasing of interest in this context, since these systems have significantly lower power requirements than their electric counterparts. A schematic of the proposed trigeneration system with an absorption chiller is shown in Figure 3.

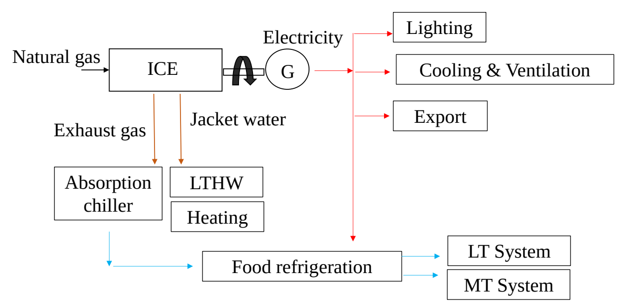

Figure 3 Block diagram of a CHP-driven HVAC system, coupled to an absorption chiller and DX/pumped CO_2 refrigeration

A schematic of the proposed refrigeration system is presented in Figure 4. The high-pressure condenser of the refrigeration system has been replaced by the absorption chiller evaporator, where ammonia absorbs heat from the cooling cycle. The saturated liquid refrigerant leaves the evaporator at P_{MT}, and enters the liquid receiver. From there the refrigerant is distributed to the MT system and the LT system via control valves. The LT system design is a conventional DX CO_2 system, operating at P_{LT}, with its associated low-pressure compressor C1. The key difference in the proposed system, relative to the conventional one, is the use of pumped refrigeration for the MT system. Saturated/sub-cooled refrigerant enters the MT evaporator and exits as saturated vapour/liquid mixture. The mixture returns to the liquid receiver and from there gets to the absorption chiller evaporator. The return $\dot{m}_{LT}$ is also mixed with $\dot{m}_{MT}$ prior to entering the absorption chiller.

The benefit of the proposed design (Figure 4) is the reduction of the power requirements of the system due to the omission of compressor C2 (if the absorption chiller is sized to cover the full cooling demand) which is the major energy consumer of a conventional system. If the absorption chiller is sized to cover part of the refrigeration demand, when the load exceeds the chiller capacity, the control valve prior to C2 opens (Figure 4) to allow for a proportion of CO_2 to enter the high-pressure compressor and air-cooled condenser, to reject the remaining heat.

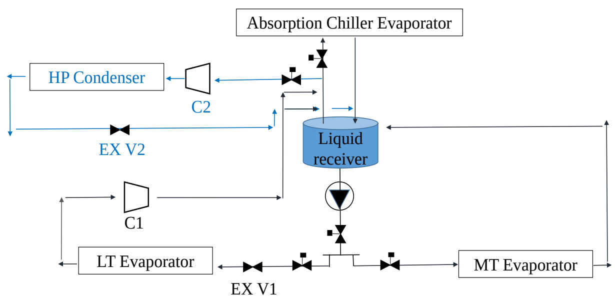

Figure 4 Schematic diagram of an absorption chiller and DX/pumped CO_2 refrigeration system

CASE STUDY

A distribution centre located in the Bristol, UK has been selected as the building case study. The current design is a conventional DX CO_2 refrigeration system, similar to the one presented in Figures 1a and 2. The recorded case-study distribution centre demand for electricity for refrigeration, non-refrigeration electricity, and heating are presented on half-hourly basis in Figure 5, for typical weekdays (WD) and weekend days (WE).

Systems Description

Three energy systems have been considered and modelled in MATLAB. They are compared in terms of their annual energy performance and operating costs for the selected case study:

System 1: Conventional (baseline) system connected to the electricity grid, with electric-driven CO_2 refrigeration, and natural gas boiler for heating and LTHW provision (refer to Figures 1a and 2 for the system's design).

System 2: CHP-driven system connected to electric-driven CO_2 refrigeration system (see Figures 1b and 2 for the system's design).

System 3a/b: CHP-driven system coupled to an absorption ammonia-water chiller, and CO_2 refrigeration system (refer to Figures 3 and 4 for the system's design). Two different capacity absorption chillers have been investigated for this system.

The operational strategy deployed for System 2, is to follow the electricity load, while meeting the heating and LTHW demand. Since the CHP units operate to cover the full electricity demand, the available thermal output is higher than the building demand, resulting in significant excess waste heat. The operational strategy followed for System 3a/b is firstly for the system to cover the electricity for non-refrigeration uses, followed by an attempt to cover the refrigeration demand (either partially or the full load) with the use of the absorption chiller. When the CHP waste-heat stream is not enough to cover the thermal demand of the absorption chiller, the CHP unit ramps up. Finally, when the cooling demand exceeds the absorption chiller capacity, the remaining refrigeration load is covered by enabling the electric driven high-pressure compressor and air-cooled condenser units. Because the coverage of the refrigeration demand of the distribution centre is critical to the business continuity, the CHP units in System 3 are sized to be able to cover the full electricity demand of the refrigeration system, in case there is a failure of the absorption cooling system.

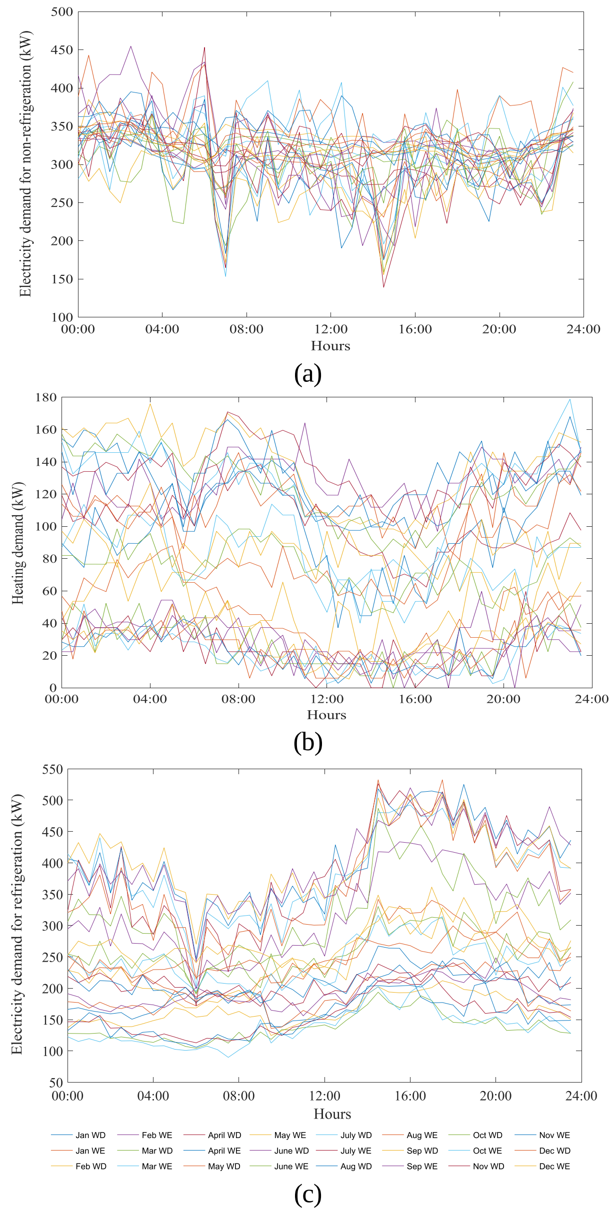

Figure 5 Half-hourly demand profile of a typical WD and WE day per month, for (a) non-refrigeration electricity; (b) electricity for refrigeration; and (c) heating

To select the absorption chiller units, the refrigeration demand of the building is required. However, the available meter readings provide only the electricity demand utilised for refrigeration. Therefore, the coefficient of performance (COP) is required to extract the cooling demand of the building. To obtain the system's COP, a detailed description of the mechanical equipment is required, along with the respective weather data of the locality at the time of measuring the electricity demand for refrigeration. In terms of technology selection, the building is assumed to deploy the conventional System 1 design. The annual weather data for the Bristol area is presented in Figure 6, on an hourly basis. This data has been obtained from Meteonorm [7].

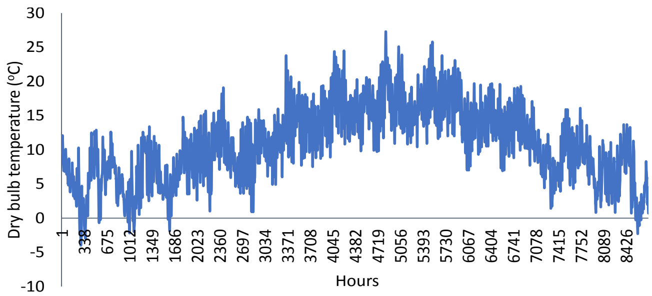

Figure 6 Hourly weather data for the Bristol area [7]

The annual weather data in hourly intervals is processed to determine representative daily temperature profiles, to align with the typical day demand data available monthly. Hence, 12 typical days have been used in the modelling exercise (one typical day per month) to represent the weather conditions. In Figure 7, the typical daily profiles are illustrated. The COP of the system is then calculated using the typical day weather data, to obtain the cooling demand of the building, from the electricity consumption for refrigeration meter readings.

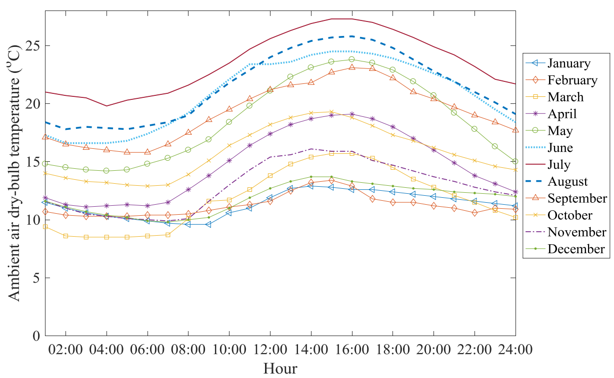

Figure 7 Typical ambient air dry bulb temperature profile, for a 24-hr period per month, for the Bristol area

The peak electricity load of the building is 841 kW, and the peak heating demand is 180 kW. Over the year, the total electricity base load of the building is 380 kW, which is approximately 43% of the peak load. To investigate the potential and feasibility of operating the distribution centre with no electricity imports from the grid, 2 CHP units of 500 kW_{el} have been selected, to ensure that the minimum PL of the CHP units is not lower than 55%. At very low PL conditions (<50%) the CHP performance deteriorates, therefore it is good practice to have 2 smaller units in operation (at 500 kW_{el}), instead of a single large unit of 950 kW_{el}. Adding to this, the capital investment cost of 1 MW_{el} CHP unit as provided by ENER-G is equal to £1,000,000, whereas the cost of one 500 kW_{el} unit is £258,000 [8], making the purchase of 2 smaller units a more cost-effective solution.

Regarding the absorption chiller unit, an ammonia-water system has been selected, due to the lower cooling temperatures this can achieve, without the risk of crystalisation faced by the Lithium/Bromide systems [9]. Ammonia as a working fluid is classified as a natural refrigerant with low GDP and ODP values, however, it is classified as a B2L fluid based on the ASHRAE classification [10], which means it has low flammability, and it is moderately toxic. Therefore, supermarkets do not use ammonia in their refrigeration distribution system and in the food cabinets. An intermediate refrigeration circuit is required to make use of the cooling effect of an ammonia-water chillers in the food industry. As illustrated in Figure 4, System 3a/b, uses ammonia only in the internal circuit of the absorption chiller, whilst the entire refrigeration distribution system uses CO_2. The absorption chiller selected in this study is provided by AGO, and it can deliver cooling to temperatures as low as -28 ºC [11]. The chiller capacity is selected to cover the building cooling base load, to maximise the chillers performance and utilisation rate. The building cooling load estimation is presented in the following section.

Energy Prices

Energy prices for both electricity and gas are required to calculate the annual energy costs, and potential savings among the systems investigated. The electricity and gas prices data used in this study are based on previous work done by Refs. [12,13]. The gas price used is 2.4p/kWh [6]. The Bristol area belongs to DNO WPD South West England, which provide electricity costs for commercial and industrial users on a half-hourly basis. In Figure 8, the electricity prices profile variation for typical WD and WE is presented for 2017.

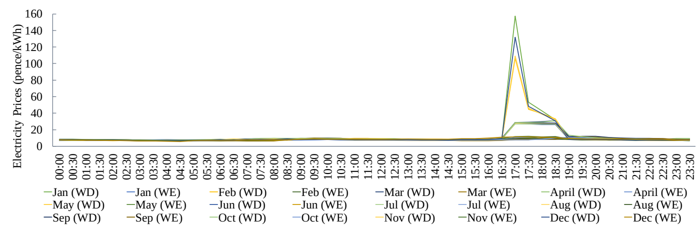

Figure 8 Half-hourly electricity prices for typical days per month, for 2017; data obtained from Refs. [12,13]

Energy and Environmental Policy

Government policies and incentives affect significant the attractiveness of investments in distributed CHP and trigeneration systems. In the UK, the Good Quality CHP scheme (CHPQA) offers exemption from the Climate Change Levy (CCL) of all fuel inputs and electricity outputs consumed, or supplied to a third party. The scheme also offers eligibility for Enhanced Capital Allowances (ECA) for the CHPQA plant and machinery, reducing significantly the payback period of the investment for the producers qualified for the scheme.

RESULTS AND DISCUSSION

The CHP unit selected for the case study is a 500 kW_{el} unit by ENER-G. The system's electric and thermal efficiency at different load conditions were obtained directly from manufacturer's data sheets and presented in Figure 9 as function of load/power output.

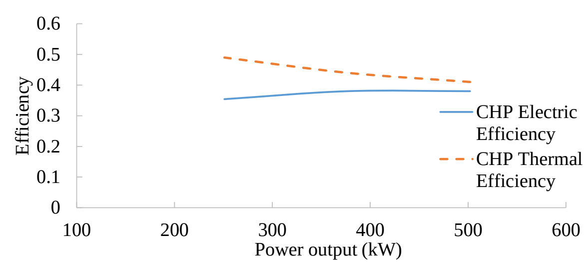

Figure 9 CHP electrical and thermal efficiency curves; data obtained from ENERG CHP-500 data sheet [8]

The cooling system COP for the baseline scenario is obtained using the weather data for the Bristol area, and the performance curves obtained by Ref. [14]. Depending on the ambient air conditions the system operates with a COP between 1.5 and 2.8. Using the COP, the electricity demand data for refrigeration provided in Figure 6c, has been converted into the cooling load demand of the building. In Figure 10, the cooling load demand profile is illustrated for one typical WD and one WE per month.

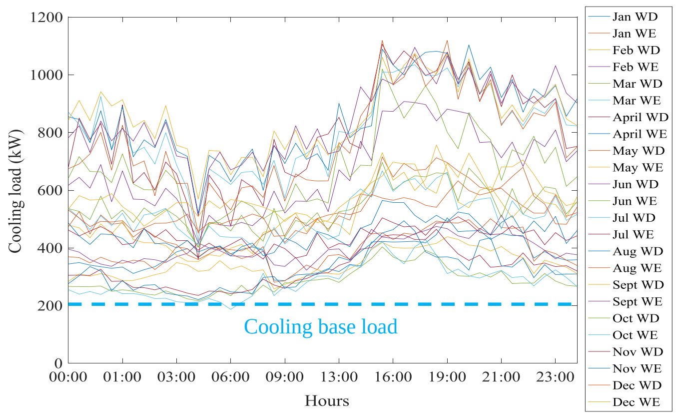

Figure 10 Half hourly cooling demand using the COP of the electric driven DX CO_2 refrigeration system for the Bristol area

Based on the figures, the building base load for refrigeration is approximately 210 kW. Therefore, the absorption chiller selected has 250 kW of capacity (System 3a), to maximise the chiller running hours and efficiency. System 3b deploys a slightly larger 300 kW chiller, to evaluate the impact of recovering higher percentage of the available thermal output of the CHP units, on the system performance and costs. The absorption chiller COP varies between 0.55-0.75 [11]. For the purposes of this feasibility study a conservative COP of 0.6 has been used. When the cooling load exceeds the absorption chiller capacity, electricity will be used to drive the high-pressure compressor of the system and the air-cooled condenser to provide the remaining cooling (refer to Figure 4 for the system schematic).

The LT system evaporation temperature is set to -32 ºC as per BS EN ISO 23953 [15], and the vapour exits the evaporator with 8 K superheat. The pressure of the LT system is P_{LT} = 1,330 kPa. The MT system evaporation temperature is set to -7 ºC [15], and exits the evaporator as saturated vapour. The vapour/liquid mixture conditions entering the absorption chiller evaporator is calculated for every hour and load condition. The refrigerant exits the absorption chiller evaporator as saturated liquid at P_{MT} = 2,870 kPa and at a saturation temperature of -7 ºC. The evaporator heat exchanger pinch point (PP) method has been used to obtain the ammonia circuit entering and leaving conditions. The PP temperature difference is set to 5 K. Based on these conditions the ammonia saturation temperature is -12 ºC, corresponding to saturation pressure of 267 kPa.

Annual Energy Performance and Costs

In Figure 11, the annual energy demand for heat and electricity is presented for Systems 1, 2, 3a and 3b. The annual electricity demand for the baseline case (System 1) is about 5,000,000 kWh, whilst the thermal demand is roughly 760,000 kWh. The high electricity load is attributed to the high electricity demand for refrigeration, since in System 1 an electric driven DX CO_2 refrigeration system is deployed. In Figure 12, the annual energy costs of the baseline system are also illustrated, corresponding to £555,000 per year.

System 2 has similar electricity and heating demand to System 1, because both systems have the same DX refrigeration system. For System 2, the CHP units are sized to make the distribution centre self-sustained, eliminating any vulnerabilities against power shortages of the grid.

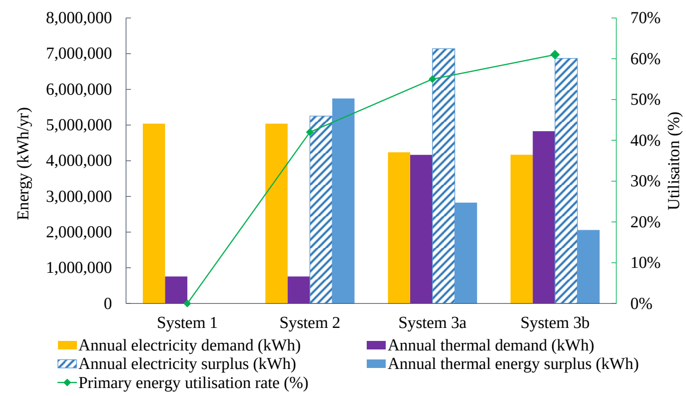

Figure 11 Annual energy consumption breakdown, excess energy streams, and primary energy utilisation on-site for the three systems investigated

When a CHP unit is required to operate, it runs at a minimum part load of 55% in line with the manufacturer's guidelines. Therefore, the CHP units in System 2, after covering the building demand, generate approximately 5,000,000 kWh/yr of electricity as surplus, which for this case study is assumed to be exported to the grid at rate of approximately 3-5 p/kWh as provided by the DNO. This surplus could be also sold to other customers in the proximity of the installation in so-called micro-grid developments. System 2 also has a high thermal-energy surplus, since there is not enough heating demand on-site for full utilisation. Looking at the annual energy costs (Figure 12), these are 36% lower than the baseline scenario when the revenues from the electricity exports are included. It is approximately 31% lower when no exports are considered. However, the CHPQA rating achieved is low (90), which means that the system is not qualified under the scheme.

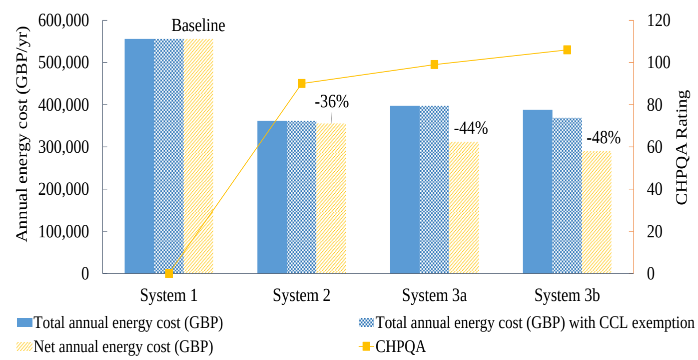

Figure 12 Annual energy costs, and CHPQA rating of the three systems investigated, including discounted rates (where applicable) and revenues from electricity exports

These results illustrate the great potential of using a thermally driven chiller to cover part of the refrigeration demand by using the excess waste-heat stream from local CHP engines, while also improving the CHPQA rating achieved. Considering the performance of System 3a with a 250 kW absorption chiller, the CHP still generates all the electricity required on-site, however, the electricity demand is lower by 16% (4,250,000 kWh/yr) in comparison to the baseline case, due to the use of the absorption chiller (Figure 11). The peak electricity demand for System 3a is also reduced to 720 kW, in comparison to the baseline of 841 kW. The thermal output surplus is reduced by almost 40% in comparison to System 2. The primary energy utilisation on-site increases from 44% (System 2) to 55% for System 3a. In terms of annual costs System 3a annual energy cost is 44% lower than the baseline case, when the revenues from electricity exports is included (Figure 12). The CHPQA rating obtained is higher than System 2, and equal to 98. However, still the system does not meet the threshold of 105 to be qualified under the scheme.

To meet the CHPQA threshold (>105), a slightly larger 300 kW chiller is considered so as to further increase the CHP thermal output utilisation (System 3b). The thermal energy surplus for System 3b decreases significantly (-28%) in comparison to System 3a, while the electricity surplus on-site slightly drops (Figure 11). The utilisation of primary energy is 61%, improved significantly from Systems 2 and 3a. This improvement is reflected on the CHPQA value, which corresponds to 106, making the scheme qualified for enhanced capital allowance. Entering the CHPQA scheme can decrease significantly the system payback period, increasing the attractiveness of the investment. Finally, in terms of annual energy costs, System 3b energy bill is 48% lower than the baseline, when the CCL exemption due to the good quality CHP achieved is considered, and when the electricity surplus is exported to the grid (Figure 12).

SUMMARY AND CONCLUSIONS

The results of this study suggest that the integration of trigeneration systems in supermarkets enable these buildings to minimise their dependence on the electricity grid, while achieving annual energy system costs that are 48% lower than those of the baseline design. By selecting the appropriate size of absorption chiller unit, the scheme also qualifies with CHPQA, and it has significant capital allowance reductions, which can decrease significantly the payback period of this investment. For cases where the thermal output of the CHP units is not utilised, the CHPQA rating is lower than 100 (System 2 and 3a). However, there are still significant cost savings on an annual basis of between 36% and 44% compared to the baseline scenario.

The selection of the absorption chiller capacity to be slightly higher than the cooling demand base load (300 kW) maximises the chiller running hours and efficiency. This design increases the primary energy utilisation of the system, exceeding 60%. Adding to this, the demand power-to-heat ratio reaches 0.9 (System 3b) matching the building load profile to the generation profile.

It should be noted that the profitability of such schemes is reliable on the forecasted electricity prices and on the existence of incentives. Further research should be done to understand the sensitivity of each system performance against fluctuations on the gas/electricity prices, and potential incentives. Further analysis could be also done to evaluate how potential customers for electricity and heat in the proximity of the trigeneration facility, can influence the primary energy utilisation rate of the systems, the CHPQA rating and the annual revenues of the owners. Finally, a full optimisation study should be conducted, to identify the best suited technology selection and sizing with respect to alternative criteria of interest to the interested stakeholders.

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