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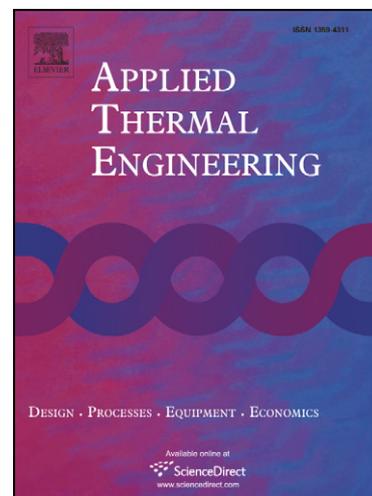
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Trigeneration in food retail: an energetic, economic and environmental evaluation for a supermarket application

N. Sugiarta^{*}, S.A. Tassou, I. Chaer¹, D. Marriott²
School of Engineering and Design, Brunel University, Uxbridge, Middlesex, UB8 3PH, UK

¹Department of Engineering Systems, London South Bank University, 103 Borough Road, SE1 0AA, UK

²Doug Marriott Associates, Kent, UK

Abstract

This paper presents results on the evaluation of energy utilisation efficiency and economic and environmental performance of a micro-gas turbine (MGT) based trigeneration system for supermarket applications. A spreadsheet energy model has been developed for the analysis of trigeneration systems and a 2,800 m² sales area supermarket was selected for the feasibility study. Historical energy demand data were used for the analysis, which considered factors such as the fraction of the heat output used to drive the absorption chillers, the chiller COP and the difference between electricity and gas prices. The results showed that energy and environmental benefits can be obtained from the application of trigeneration systems to supermarkets compared to conventional systems. The payback period of natural gas driven trigeneration systems and greenhouse gas emissions savings will depend on the relative gas and electricity prices and the COP of the vapour compression and absorption systems. It was also shown that operation at full electrical output gives a better performance than a heat

^{*} Corresponding author. Tel.: +44 (0)1895266938; Fax: +44 (0)1895256392
E-mail address: nyoman.sugiarta@brunel.ac.uk (N. Sugiarta)

load-following strategy due to the reduction of the electrical generation efficiency of the MGT unit at part load conditions.

Keywords: CHP, trigeneration, micro-gas turbine, absorption chiller, operating strategy, supermarket

1. Introduction

In the UK, over 40 % of the average electrical load in large supermarkets is used to drive the refrigeration systems [1]. Vapour compression chillers are normally used to produce the cooling required by the refrigerated display cabinets, cold rooms and air conditioning system. At the same time, supermarkets have a requirement for hot water and space heating in the winter in order to maintain comfort conditions. This heating load is normally covered by gas-fired boilers. The heat demand for space heating varies seasonally and this determines the heat load pattern of the supermarket. Demand for hot water is quite low and remains fairly constant throughout the year. The highest heating load normally occurs during the winter months. In the summer months, heat demand is fairly low.

A Combined Heat and Power (CHP) or cogeneration system is a potential method to reduce energy consumption and to improve energy system efficiency [2-6]. The simultaneous heat and electrical power generation from the CHP plant require a coincident site demand for the heat and electrical power in order to achieve economic operation. In supermarket applications, the low heat demand in the summer months makes it difficult to achieve high CHP plant utilisation. High and constant utilisation

can be achieved if some of the heat available from the plant is used to drive sorption refrigeration systems. This concept is known as Combined Heat, Refrigeration and Power (CHRP), or CHCP (Combined Heat Cold and Power). In this paper, we will refer to trigeneration which is the more modern definition of the concept.

Some examples of the application of trigeneration systems in the food industry have been presented by Bassols et al. [7] and Colonna and Gabrielli [8]. These involved ammonia-water absorption systems driven by gas engines and gas turbines.

In recent years significant attention has been placed on the assessment of tri-generation systems and their ability to improve energy utilisation efficiency, reduce greenhouse gas emissions and produce economic savings. A number of evaluation models have been developed for feasibility studies.

Heteu et al. [9] presented an energy model for the comparison between tri-generation systems and separate heat and power production in terms of quality indices for primary energy and carbon emission savings. The model assumes full utilisation of electricity and heat and was used to carry out sensitivity analyses of primary energy consumption for different quantities of heat used to drive an absorption chiller.

Maidment et al. [10] developed a model to evaluate primary energy and economic savings for a conventional CHP gas engine-based system and a CHP system coupled to a single-stage ammonia-water absorption chiller for a typical 2,000 m² total floor area supermarket. The authors used the BIN method to determine electricity and heat

consumption. The results showed that the trigeneration system could offer better performance than a conventional CHP system in terms of both payback and primary energy savings.

RETScreen [11] offers a spreadsheet model that enables the evaluation of CHP systems and allows the use of various combinations of power, heating and cooling plant. The model can be used to carry out performance and economic analyses and determine greenhouse gas emissions for separate energy production systems and combined heat and power generation. The model also allows the investigation of different operating strategies such as full power, power-following and heat-following modes but part load efficiency of prime movers is not considered.

Kong et al. [12] introduced a model to determine optimal strategies to minimise the overall cost of energy for trigeneration systems using a simple linear programming method. They characterised the optimal operation of the tri-generation system by considering fixed load sets in the form of ratios of cooling load to turbine electrical generation capacity. Variation of the cooling load while keeping heating and electrical loads fixed was used as a parametric variable to determine optimum values of turbine load factor and exhaust heat fraction allocated for cooling.

Cardona and Piacentino [13] developed a methodology for sizing trigeneration systems in the hotel sector based on thermal and cooling consumption data. The size of the prime mover was based on the variation of heating and cooling demand throughout the

year and the sum of the energy requirement to satisfy the maximum coincident heating and cooling demand.

Although trigeneration systems have been applied to the food manufacturing sector, interest in their application to the food retail sector is only recent. This has mainly been due to the unavailability of commercial size low temperature and low cost absorption refrigeration systems off-the-shelf. A number of recent applications in the retail food industry in the UK have been for space heating and cooling but retail food chains and equipment suppliers are now considering systems for refrigeration applications.

Ongoing research at Brunel University is aimed at accelerating this process through the development of assessment tools and the design of systems for retail food refrigeration applications [14].

This paper describes a methodology for the evaluation of the energy utilisation efficiency, economics and environmental performance of a trigeneration system in a supermarket. The model utilises monthly energy demand data for a 2,800 m² sales area supermarket and assumes a micro-gas turbine based trigeneration system able to provide refrigeration down to -12 °C. Test results on the performance of the micro-gas turbine and absorption refrigeration system obtained in the laboratory were used as inputs to the model. Two different control strategies were considered: full load continuous operation and heat load following operation

2. Energy Demand of Case Study Supermarket

The supermarket considered in the study is located in the South of England and has a sales area of 2,800 m². The energy flows in the supermarket are illustrated in Figure 1. The average electrical demand of the store is 395 kW_e with approximately 158 kW_e required for food refrigeration, of which 59 kW_e is for the low temperature frozen food display cabinets and cold rooms and 99 kW_e for the high temperature chilled cabinets and cold rooms. This represents 40 % of the total electrical demand of the whole supermarket. The rest of the electrical demand, 237 kW_e, is related to lighting, ventilation and air conditioning, bakery and various other auxiliary equipment. The average heat demand is 55 kW_{th}.

The monthly variation of electrical and heat demand is shown in Figure 2. As can be seen from Figure 2(a), the average monthly electrical demand is fairly constant throughout the year, increasing slightly in the summer months due to the higher outdoor air temperature. Daytime demand (7.00 h to 24.00 h) is higher than nighttime demand primarily due to lower shopping activity and lower night time temperatures that lead to lower condensing temperatures and refrigeration system power consumption. Other loads that contribute to the higher daytime demand are air conditioning in the summer months that operates during opening hours and lighting. The monthly variation of daytime and nighttime heat demand is shown in Figure 2(b). It can be seen that the variation of heat demand during the year is significant with demand in the summer months being only a small fraction of the demand in the winter months.

The vast majority of supermarkets, including the supermarket considered in this study, use 'remote' refrigeration systems where the compressors and condensers are located

remotely from the direct expansion evaporator coils in the refrigerated display cabinets and cold rooms. Groups of compressors mounted on racks normally referred to as compressor packs provide capacity control to match the capacity of the system to the load (Figure 3). To reduce the quantity of volatile refrigerant employed and greenhouse gas emissions from refrigerant leakage, an alternative refrigeration system arrangement can be used which employs a volatile refrigerant in the central plant that chills a secondary fluid which is circulated to the coils in the display cabinets and cold rooms. If a trigeneration system is employed in a supermarket to satisfy part of the refrigeration load, an appropriate system arrangement would be to use a conventional direct expansion system for the low temperature frozen food cabinets and a secondary system for the chilled food cabinets. Depending on the sizing and operating strategy employed, the refrigeration capacity of the trigeneration system can be supplemented by a small direct expansion plant. This system arrangement is shown in Figure 4. To provide stable system operation a degree of storage will be required to balance the variation in the supermarket refrigeration system load and the output from the absorption chiller(s). The storage can be either 'sensible' using brine as the storage medium or 'latent' using ice or low temperature phase change materials (PCMs) as storage media. The selection of the most appropriate storage medium and optimisation of the storage capacity is beyond the scope of this paper.

3. Energy Model

Figure 5 presents the energy flow diagram for a 'conventional' supermarket and a supermarket employing the proposed trigeneration system. The trigeneration system design is based on a micro-gas turbine (MGT) CHP system capable of supplying up to

80 kW of electrical output and up to 163 kW of thermal output at 100% recuperation. The system utilises the heat rejected in the exhaust gases to drive modular absorption refrigeration units.

The refrigeration capacity of the trigeneration system will be dependent on the portion and quality of heat diverted to drive the absorption chillers and the COP of these chillers. Higher COP will result in higher refrigeration capacity and thus higher HT refrigeration load being displaced by the absorption chillers (Edisp) which will reduce the refrigeration load on the conventional vapour compression refrigeration plant.

The trigeneration plant can be sized to allow electricity export to other users when the site electrical demand is lower than the generation capacity of the system. Conversely, electricity can be imported from the grid when the site demand is higher than the generation capacity of the trigeneration plant. Depending on the operating strategy employed, the heat available from the tri-generation plant, if not sufficient to satisfy the heat demand of the site, can be supplemented by a standby gas fired boiler.

The trigeneration model can be used to identify the main factors affecting the overall energy system efficiency, economic viability and environmental impact for three different operation strategies: full electrical load continuous operation, heat load-following and electrical load-following. For full load continuous operation the MGT will run at full electrical power output irrespective of the site demand. Any heat generated over and above the heat demand will be rejected to the ambient. In the heat load-following strategy, the output of the MGT unit will be modulated in line with the

site heat demand. This will mean that the electrical power output of the unit will also vary and any deficiency will be made up from the national grid. In the event that the local generation capacity exceeds the site electrical demand the excess electricity will be exported to the grid. In the electrical load following strategy, the electrical output of the MGT unit will be modulated to match the site electrical demand. Any deficiency in heat output will be made up by a gas boiler and any excess heat will be rejected to the ambient.

4. Overview of the Experimental Investigations

The performance characteristics of a MGT based trigeneration system were obtained in a test facility established at Brunel University for CHP and trigeneration research with funding from the Department of the Environment Food and Rural Affairs (DEFRA) and support from a number of industrial partners. The CHP module is based on a 80 kW_e recuperative cycle gas turbine which was available from Bowman Power, one of the industrial partners in the project. MGT units with nominal electrical power capacities in the range 30 and 100 kW_e are available from a number of manufacturers. The 80 kW_e unit represents a good compromise between size and modularity for supermarket applications. The performance of the unit over a range of electrical power outputs between 10 and 80 kW_e is shown in Figure 6. From Figure 6(a) that shows the variation in electrical generation efficiency and exhaust gas temperature with power output it can be seen that both the electrical efficiency and exhaust gas temperature increase with electrical power output. At maximum power output of 80 kW_e, the electrical efficiency is just over 26 % and the exhaust gas temperature 275 °C.

The variation of the electrical generation efficiency with power output can be described by:

$$\eta_e = -0.002 * (E_{CHP})^2 + 0.4888 * (E_{CHP}) + 0.4161 \quad (1)$$

The thermal energy in the exhaust gases can be calculated based on the exhaust gas mass flow rate of the turbine, which is 0.83 kg/s and the difference between the exhaust gas temperature and an ambient reference temperature. This temperature was assumed to be 95 °C to avoid condensation of the exhaust gases in the heat recovery unit.

The heat to power ratio is an important performance parameter for CHP and trigeneration units when they operate on the heat load-following strategy. The variation of the heat to power ratio with thermal energy output for the 80 kW_e MGT unit is shown in Figure 6 (b). It can be seen that at full load and maximum thermal output the heat to power ratio is 2.0 but as the unit is modulated to reduce the thermal output the heat to power ratio increases exponentially. This will reduce significantly the electrical output available from the unit. The variation of the heat to power ratio of the MGT unit with thermal output can be described by:

$$H / P = 39.324 * e^{-0.0182 * Q_{CHP}} \quad (2)$$

The absorption unit employed is a gas-fired ammonia-water chiller of refrigeration capacity of 12 kW_c at ambient temperature of 35 °C and brine flow temperature of -5 °C [15]. The unit was installed and tested over a range of brine delivery temperatures between +3 °C and -11 °C at constant brine flow rate of 2.3 m³/h. The unit was sited outdoors and hence it was not possible to control the ambient temperature. During the

tests, however, the variation in ambient temperature was not significant and the results, for presentation purposes, were grouped in two temperature sets, 16.5 ± 1.5 °C and 20 ± 1.5 °C. The test results, presented in Figure 7, show the refrigeration capacity to increase from around 10 kW_c at a brine flow temperature of -10 °C to around 15 kW_c at brine flow temperature of +2 °C. For the same temperature range the COP of the unit determined as the ratio of refrigeration capacity to gas energy input increased from 0.4 to 0.6. For operation with the MGT unit in a trigeneration arrangement, the generator of the gas fired absorption unit was modified to operate in an indirect fired mode using a heat transfer fluid to transfer heat from the MGT heat recovery heat exchanger to the absorption unit [24].

5. Results and Discussion

Two different operating strategies for the trigeneration system were investigated and presented in this paper based on the 80 kW_e MGT-CHP units and a system design to displace a minimum of 50% of the supermarket's HT refrigeration demand. These are the i) full electrical load-continuous operation strategy and ii) heat load-following operation. Results for the third possible operating strategy, electrical load-following, are not presented, because if the system is not sized to satisfy the maximum electrical demand of the site this strategy will be the same as the full electrical load continuous operation strategy.

The energy consumption characteristics of the supermarket and trigeneration system performance data used in the analysis are summarised in Table 1. The trigeneration system assumes the use of 2 MGT units and a number of 12 kW_e absorption

refrigeration units that can vary depending on the amount of heat available from the MGT units to drive them. The economic variables and environmental parameters such as fuel emission factors are given in Table 2 and all the equations used for the calculation of the system performance are given in the Appendix.

In the full load operating strategy a fraction of the heat generated by the MGT units (α) will be diverted to drive the absorption chillers and the remainder will be used to satisfy the heat demand. Any excess heat will be rejected to the ambient. In the heat following strategy the output of the MGT units will be modulated so that the heat available exactly matches the heat demand.

Figure 8 shows the variation of the overall efficiency of the trigeneration system as a function of (α) for the two operating strategies and for two absorption refrigeration system COPs, 0.5 and 1.0. The overall efficiency was determined from:

$$\eta_{ov,tri} = \left[\frac{(E_{CHP} + Q_{heating} + CC_{HT,refrigeration})}{F_{CHP}} \right] \times 100\% \quad (3)$$

The reason that the two COP values of 0.5 and 1.0 were chosen is that the former is the average value obtained from laboratory tests (see Figure 7) and also delivered by commercially available low temperature ammonia-water systems operating with fluid delivery temperatures down to $-10\text{ }^{\circ}\text{C}$ whereas the latter is to demonstrate the potential energy and emission savings if more efficient ammonia-water systems are developed.

It can be seen that for both operating strategies, the absorption system COP has a significant impact on the overall system efficiency. This effect will vary with (α) and for high values of (α) can be as high as 30%. As (α) increases, the overall system efficiency increases up to a value of (α) of 80%. Above this value the overall efficiency stays almost constant for the refrigeration system COP of 1.0 and drops slightly for the COP of 0.5. The operating strategy has only a very small effect on the overall system efficiency. At low values of (α), full load operation (FLC) gives a slightly better efficiency than heat load following operation (HLF). This is because, the system will have excess heat capacity and to match the heat generated to the heat demand the MGT will operate at part load conditions with a consequent reduction in power output and efficiency (Figure 6). As (α) increases more heat will be diverted to drive the absorption chillers which will reduce the amount available for heating purposes. This will improve the load factor of the system which will lead to a higher efficiency.

Figure 9 shows the variation of the Fuel Energy Savings Ratio (FESR) with (α) for the two operating strategies. It can be seen that for values of (α) below 90%, the full load operating strategy leads to better performance than the heat load operating strategy. Above 90% the two strategies lead to the same FESR. The reason for the lower FESR at low values of (α) for the HLF strategy is the part load operation of the system at reduced heat demand. Two gas turbines are used for the supermarket under consideration. Because of the low heat demand of the supermarket in the summer months (Figure 2) if the HLF strategy is used and too much heat is diverted from the absorption chillers to provide space heating at low (α) values, then the speed of the turbine and its thermal output will be reduced to the point that just matches the space

heat load demand. The electrical power output will also be reduced and the power requirement from the electricity grid will increase to meet the total electrical and refrigeration load demand of the supermarket. This will lead to a lower FESR compared to the full electrical load (FLC) operating strategy.

For a refrigeration system COP of 0.5, the FESR of the trigeneration system is only marginally positive (only very small energy savings over the conventional system). For a COP of 1.0, the energy savings are of the order of 17% when (α) is above 80%. The corresponding greenhouse gas emissions savings are shown in Figure 10. Again the emissions savings are minimal for a COP of 0.5 but increase substantially to around 370 tonnes CO₂ per year for a COP of 1.0.

Figure 11 shows the variation of the payback period of the trigeneration system with (α). For the full load operating strategy, variation of (α) has very little impact on the payback period. For the heat following strategy, the payback reduces with increasing (α) until it becomes equal to that of the full load strategy at (α) above 85%. For the conditions considered, the payback period of the system is 4.0 years for a COP of 1.0 and 5.3 years for a COP of 0.5.

The relative cost of electricity and fuel used to generate it in a power plant has a significant impact on the economic viability of power generation systems. Various terms such as 'spark gap' and 'spark spread' are used by industry and researchers to refer to the difference in prices but the definition of these can vary. For a CHP system, spark spread is normally the difference between the offset cost of purchased grid power

and the cost of fuel for the CHP system to produce power and heat on site. Sometimes, however, spark spread is used to indicate the ratio between the market price of electricity produced and the cost of fuel for its production [25]. Figure 12 shows the effect of the ratio between electricity and gas prices on the payback period. It can be seen that the payback period is very sensitive to the difference between the electricity and gas prices particularly for low refrigeration system COPs. For the full load operating strategy and a COP for the refrigeration system of 0.5, the payback reduces from 15 to below 3 years as the electricity to gas price ratio increases from around 3 to around 7. For a COP of 1.0, the effect of electricity to gas price ratio reduces, with the payback period reducing from around 6 years to just below 3 years with an increase in the price ratio from 3.0 to 7.0.

6. Conclusions

Assessment of the feasibility of application of trigeneration to supermarkets has shown that:

1. For (α) below 85 %, the continuous full load operating strategy gives better energetic, economic and environmental performance than the heat load following strategy due to the reduction in the efficiency of the MGT units at part load conditions. For (α) above 85% the two operating strategies lead to similar performance.
2. For operation at refrigeration temperatures down to -12 °C, the COP of the absorption system should be greater than 0.5 for a micro-gas turbine trigeneration system to produce emissions savings. For a COP of 1.0 the

trigeneration system considered in the case study will produce emissions savings of 300 tonnes of CO₂ per year.

3. The economic viability of a gas driven trigeneration system is very sensitive to the relative prices of natural gas and grid electricity. The wider the gap between electricity and gas prices the lower will be the payback period of the trigeneration system. In the case considered, for the full load operating strategy, an electricity to gas price ratio of 4.0 will give a payback period of 5.7 years for an absorption system COP of 0.5 and 3.8 years for a COP of 1.0.
4. The model developed can be used to analyse and compare the performance of alternative trigeneration system designs and operating strategies in food and other industrial and commercial applications.

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Nomenclature

CC	cooling capacity
CCL	climate change levy
CHP	combined heat and power
CIT	compressor inlet temperature

COP	coefficient of performance
E	electrical power
EC	energy costs
EM	CO ₂ emissions
EMF	CO ₂ emission factors
ES	energy supply
F	fuel consumption
FESR	fuel energy saving ratio
FLC	full load continuously
GCV	gross calorific value
HLF	heat load-following
H/P	heat to power ratio
HT	high temperature
LT	low temperature
MGT	micro-gas turbine
NCV	net calorific value
PB	payback period
Q	heat energy
T&D	transmission and distribution
UC	unit cost
VAT	value added tax

Greek Symbols

α	fraction of micro gas turbine heat output diverted to drive absorption chillers
----------	---

η efficiency

Subscript

abs absorption system

aux, boiler auxiliary boiler

c cooling

conv conventional system

conv,grid grid electricity of the conventional plant

disp displaced

e electrical

el electric-driven vapour compression chiller

exh exhaust gas

f fuel

grid National grid

NG natural gas

ov,tri overall trigeneration system

ref reference

th thermal

tri trigeneration plant

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Appendix

Equations Used in the Model

The key equations used to model the energy and environmental performance of the conventional and trigeneration systems illustrated in Figure 5 are given below.

Conventional system

Energy supply:

$$ES_{conv} = [E_{lighting, etc.} + E_{LT, el} + CC_{HT, Re\ frigeration} + Q_{boiler}] \times 8,760 \quad [\text{kWh}] \quad (\text{a1})$$

where,

$$CC_{HT, Re\ frigeration} = E_{HT, el} \times COP_{HT, el} \quad (\text{a2})$$

Fuel energy consumption based on separate energy supply]:

$$F_{conv} = \left(\frac{E_{conv, grid}}{\eta_{e, grid}} + \frac{Q_{boiler}}{\eta_{boiler}} \right) \times 8,760 \quad [\text{kWh}] \quad (\text{a3})$$

Annual energy costs based on electricity and natural gas rates [£]:

$$EC_{conv} = \left[(E_{conv,grid} \times UC_{el}) + \left(\frac{Q_{boiler}}{\eta_{boiler}} \right) \times UC_{NG} \right] \times 8,760 \quad (a4)$$

Annual CO₂ emissions from the conventional plant:

$$EM_{conv} = \left[(EMF_{grid} \times E_{conv,grid}) + \left(EMF_{NG} \times \frac{Q_{boiler}}{\eta_{boiler}} \right) \right] \times 8,760 \quad [\text{tonnes}] \quad (a5)$$

Trigeneration system

Energy supply [kWh]:

$$ES_{tri} = \left[E_{CHP} + (1 - \alpha) \times Q_{CHP} + \alpha \times Q_{CHP} \times \left(\frac{COP_{abs}}{COP_{el}} \right) \right] \times AF \times 8,760 + (E_{grid,import/export} + Q_{aux,boiler}) \times 8,760 \quad (a6)$$

$$E_{grid,import/export} = E_{conv,grid} - (E_{CHP} + E_{disp}) \quad [kW_e] \quad (a7)$$

$$Q_{CHP} = \dot{m}_{exh} c_{p,exh} (T_{exh} - T_{ref}) \quad [kW_{th}] \quad (a8)$$

where, \dot{m}_{exh} is the exhaust gas mass flow rate, $c_{p,exh}$ is the specific heat of the exhaust gas, T_{exh} is the exhaust gas temperature exiting MGT (after recuperator), and T_{ref} is the ambient reference temperature in which the exhaust heat is available for recovery (limited by the dew point of the flue gas. In this paper, T_{ref} is taken as 95 °C, which is well above the exhaust gas acid dew point of 60 °C when natural gas is used as a fuel [21].

Heat to power ratio:

$$H / P = \frac{Q_{CHP}}{E_{CHP}} \quad (a9)$$

The CHP plant availability factor (AF) can be expressed as [22]:

$$AF = \left[\frac{T - (S + U)}{T} \right] \times 100\% \quad (\text{a10})$$

where:

S is annual hours of scheduled maintenance shutdown (120 h)

T is annual hours that the CHP plant is designed to operate (8,760 h)

U is annual hours of unscheduled maintenance shutdown (0 h)

The overall trigeneration system efficiency is the sum of the electrical power output, heat recovered for heating and cooling capacity produced by the absorption chillers divided by the fuel consumption [23]:

$$\eta_{ov,tri} = \left[\frac{(E_{CHP} + Q_{heating} + CC_{HT,refrigeration})}{F_{CHP}} \right] \times 100\% \quad (\text{a11})$$

Fuel consumption [kWh]:

$$F_{tri} = \left[\left(\frac{E_{CHP}}{\eta_{CHP}} \right) \times AF \times 8,760 \right] \pm \left[\left(\frac{E_{grid,import/export}}{\eta_{e,grid}} \right) + \left(\frac{Q_{aux,boiler}}{\eta_{aux,boiler}} \right) \right] \times 8,760 \quad (\text{a12})$$

Annual energy costs in accord with the fuel consumed by the CHP plant and auxiliary boiler and imported/exported electricity [£]:

$$EC_{tri} = \left[(F_{CHP} \times UC_{NG}) + (UC_{NG} \times \frac{Q_{aux,boiler}}{\eta_{aux,boiler}}) \right] \times AF \times 8,760 \pm (UC_{el} \times E_{grid,import/export}) \quad (\text{a13})$$

Annual CO₂ emissions [tonnes]:

$$EM_{tri} = \left[(EMF_{NG} \times F_{CHP}) \times AF \times 8,760 \right] \pm \left\{ (EMF_{grid} \times E_{grid,import/export}) + \left[EMF_{NG} \times \left(\frac{Q_{aux,boiler}}{\eta_{aux,boiler}} \right) \right] \right\} \times 8,760 \quad (\text{a14})$$

Fuel energy saving ratio [FESR]:

$$FESR = \frac{(F_{conv} - F_{tri}) \times 100\%}{F_{conv}} \quad (a15)$$

A simple payback time (PB) for the trigeneration plant investment [years]:

$$PB = \frac{\text{Installed costs of absorption chillers} + \text{Installed cost of the CHP plant}}{\text{Annual net savings}} \quad (a16)$$

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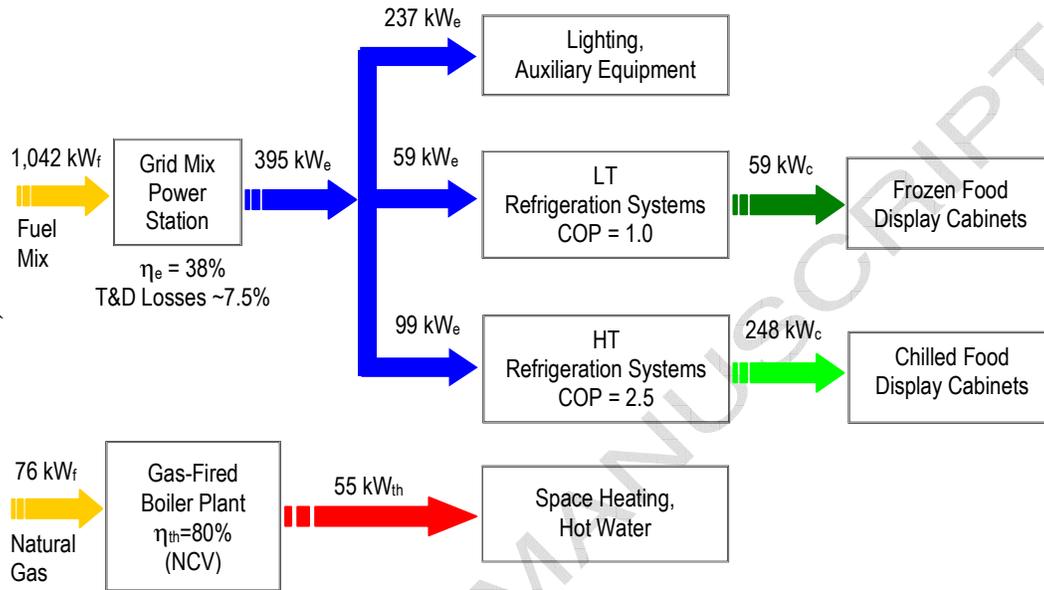
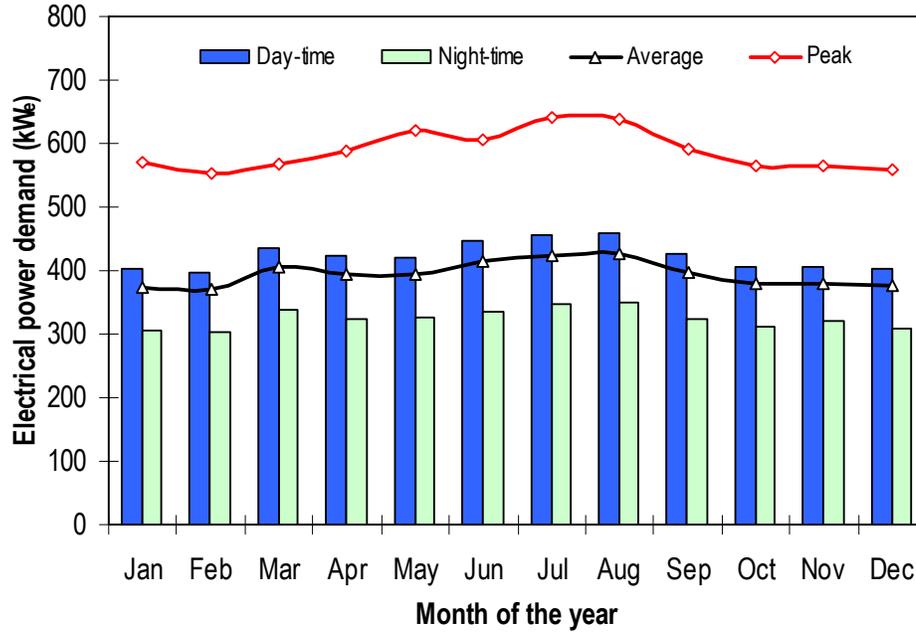
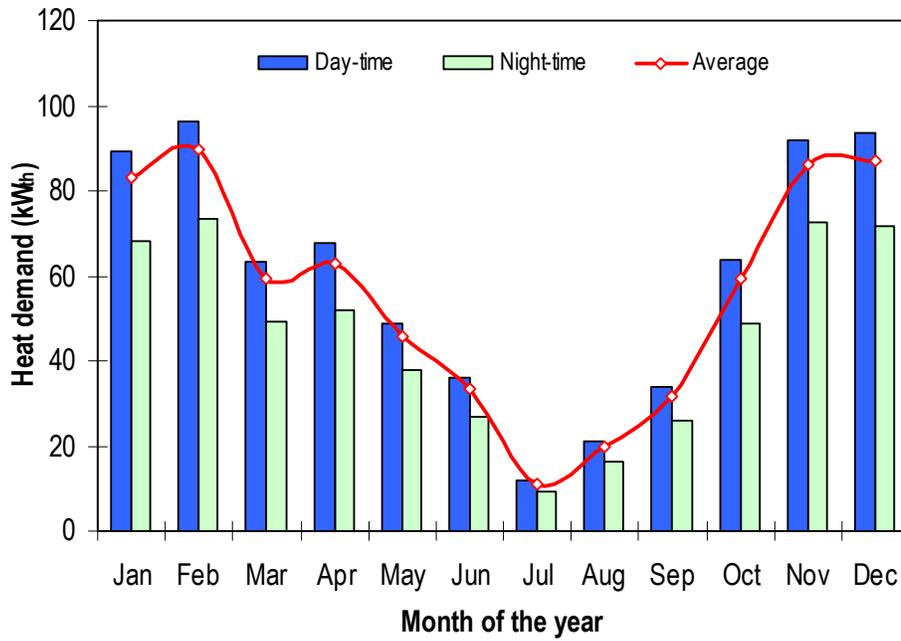


Figure 1 Flow diagram for conventional power, refrigeration and heating in a supermarket



(a)



(b)

Figure 2 Monthly energy demand profile for the supermarket

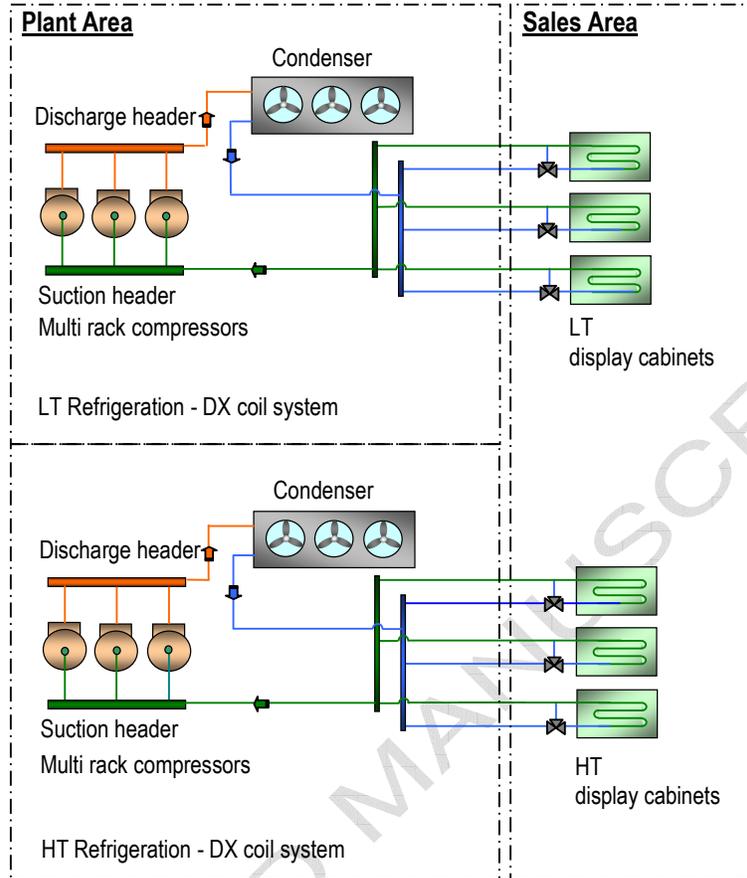


Figure 3 Conventional refrigeration system arrangement in a supermarket

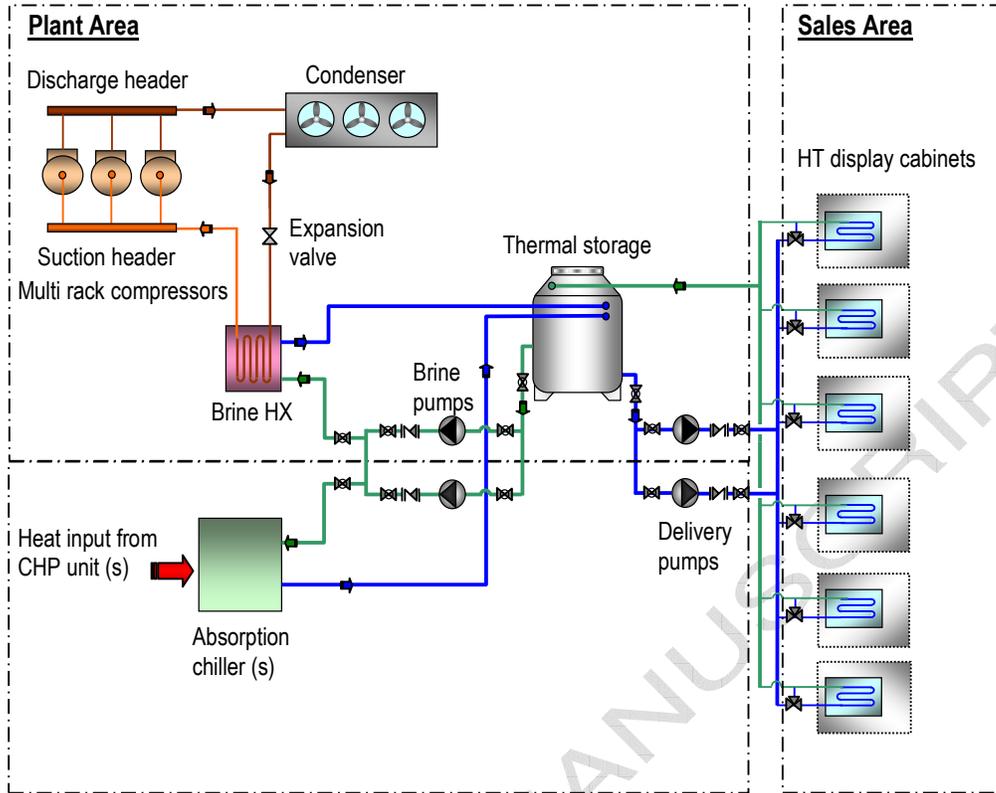


Figure 4 Proposed HT refrigeration system arrangement in a supermarket.

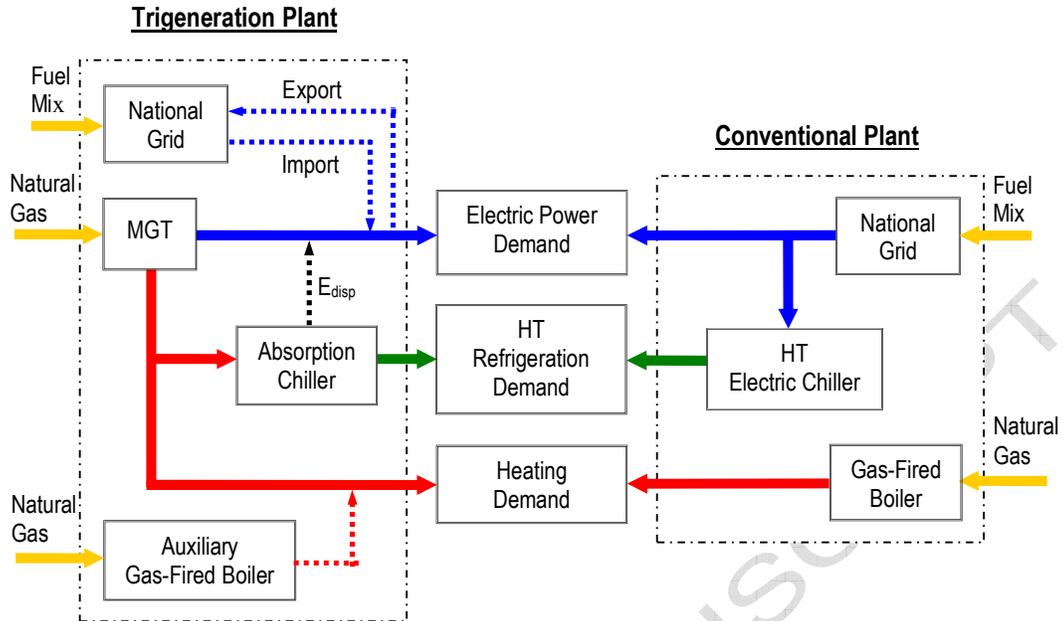
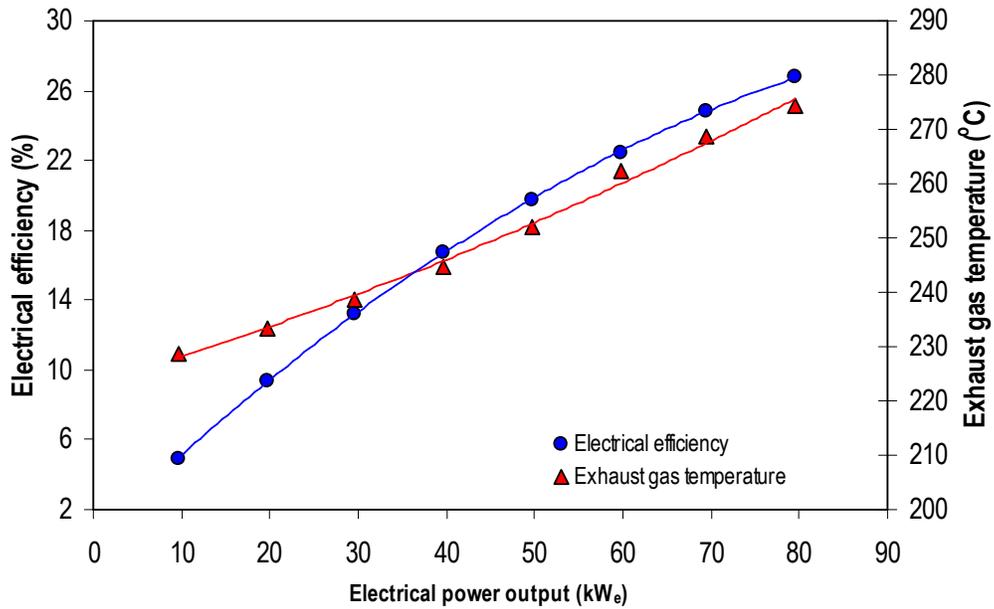
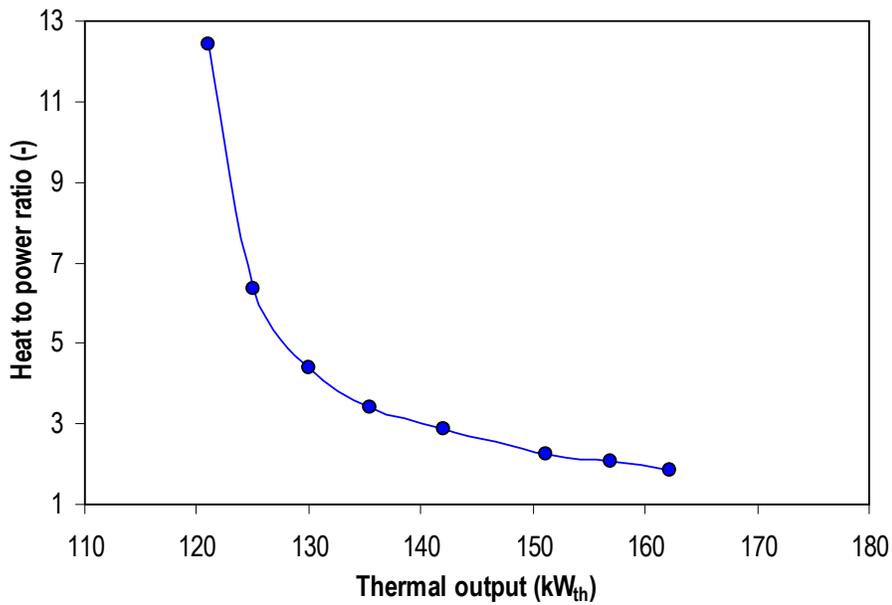


Figure 5 Energy flow diagram for conventional and trigeneration systems



a) Variation of electrical efficiency and exhaust gas temperature with power output



b) Variation of heat to power ratio with thermal output

Figure 6 Part load performance of micro-gas turbine CHP system (GCV of 39.2 MJ/m^3 , CIT = $26\text{-}28 \text{ }^{\circ}\text{C}$)

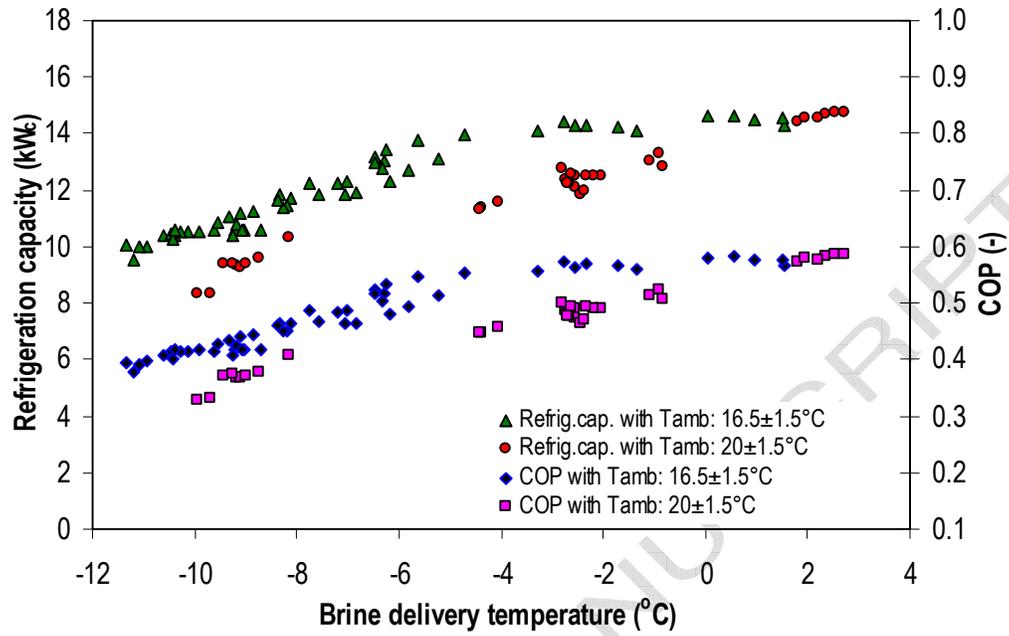


Figure 7 Performance of absorption chiller as a function of brine delivery temperature

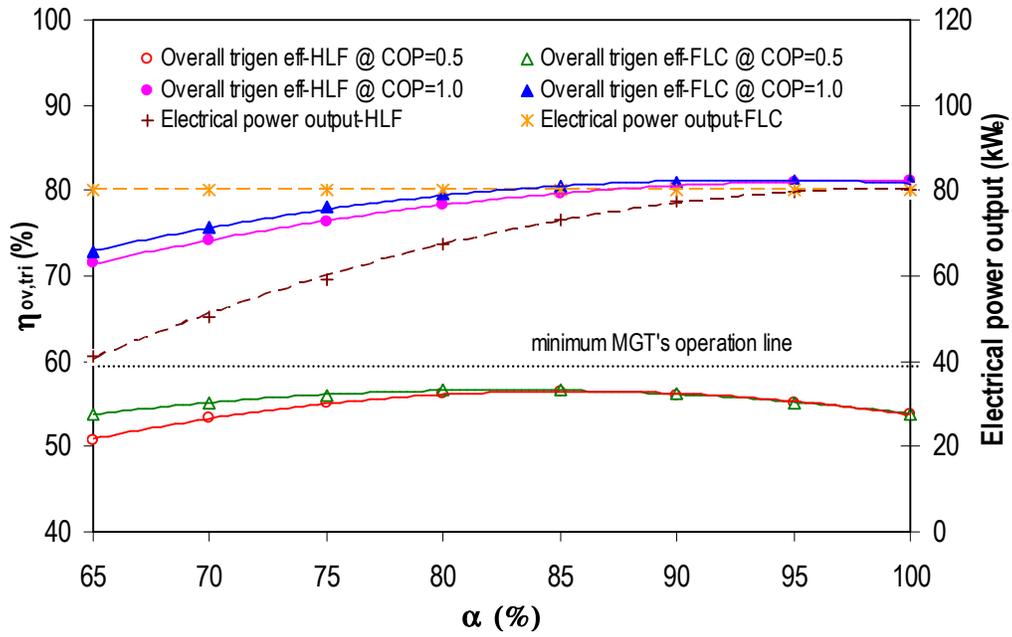


Figure 8 Impact of percentage of heat used to drive the absorption system on overall efficiency

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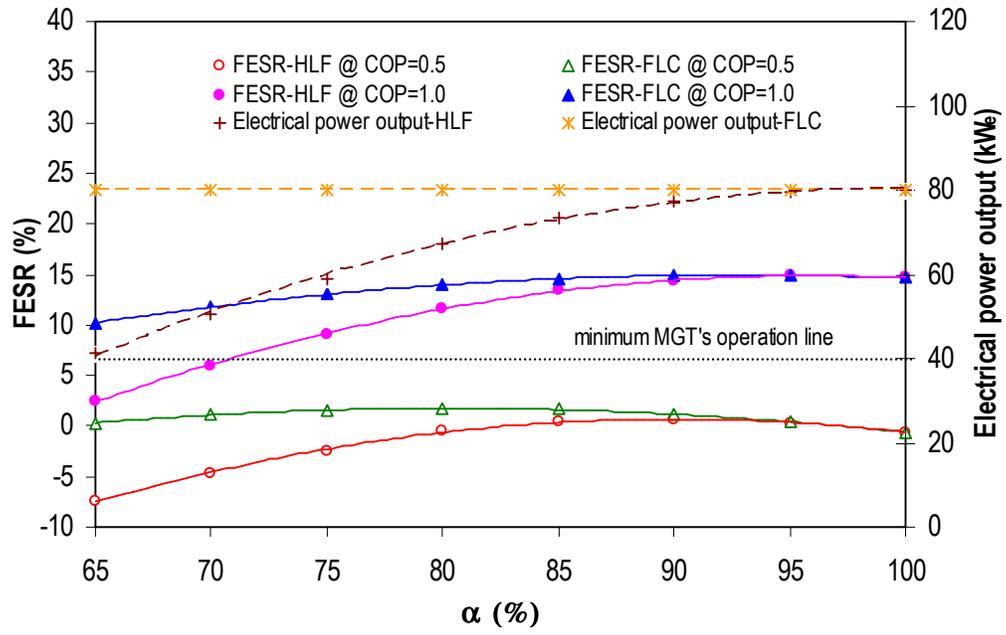
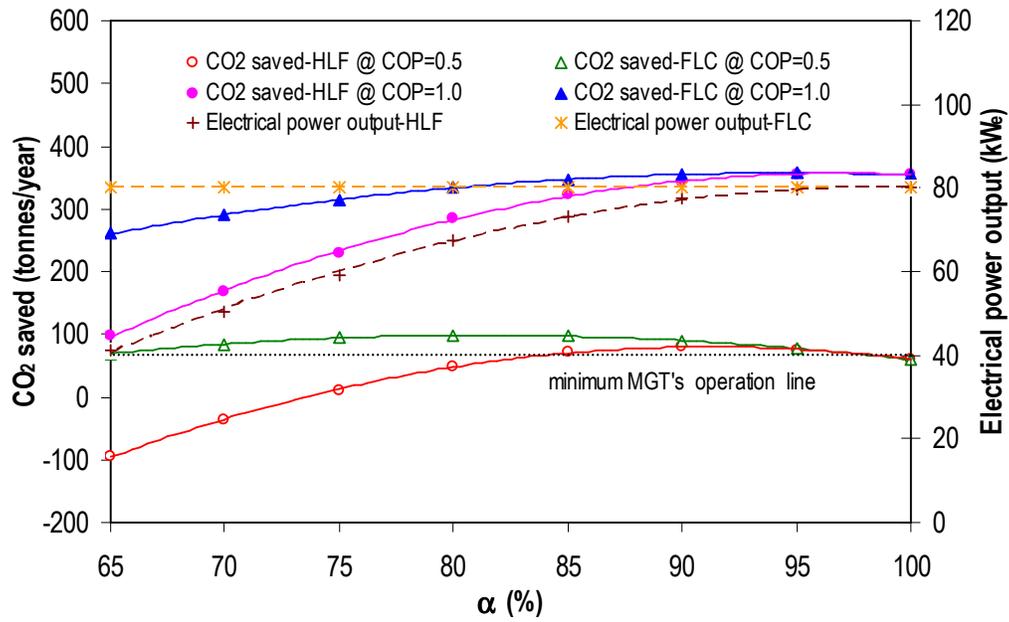
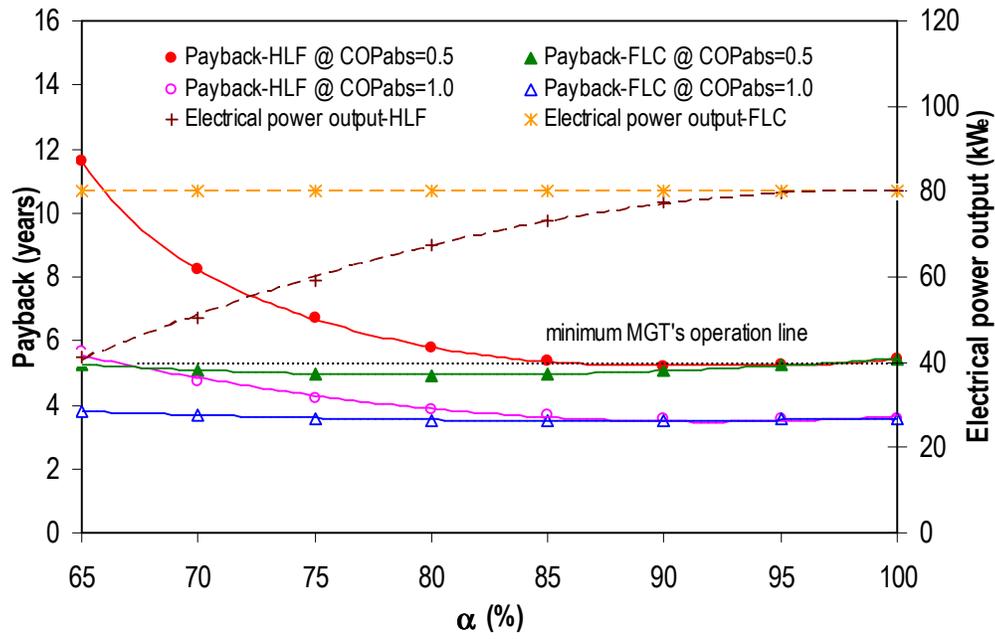


Figure 9 Variation of Fuel Energy Savings Ratio (FESR) with (α)

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Figure 10 Variation of CO₂ emissions savings with (α)

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Figure 11 Variation of payback period with (α)

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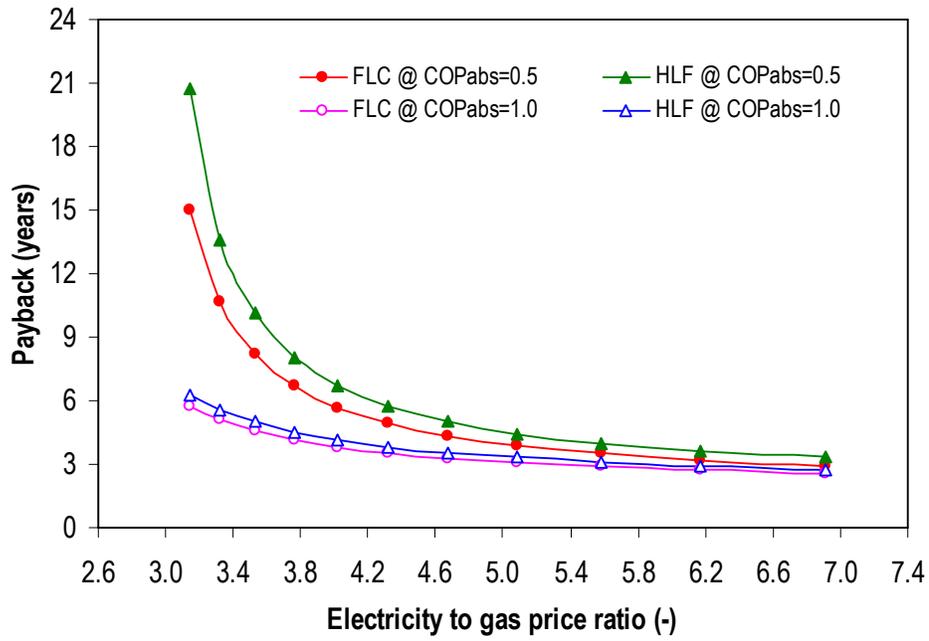


Figure 12 Influence of electricity to gas price ratio on payback period at (α) = 80%.

Table 1 Parametric data for system comparison

Conventional System	
Electrical demand	
- Day-time (07:00h-24:00h) [kWh]	2,628,444
- Night-time (00:00h-07-00h) [kWh]	829,189
- Total [kWh]	3,457,633
Average grid mix electrical efficiency (GCV) [%]	38 [16,17]
Grid T&D losses [%]	7.5 [16,17]
Heat demand	
- Day-time (07:00h-24:00h) [kWh]	486,282
- Night-time (00:00h-07-00h) [kWh]	369,460
- Total [kWh]	486,282
Gas-fired boiler	
Fuel	Natural gas
Thermal efficiency (NCV) [%]	80
Electric-driven vapour compression chiller	
COP _{el} [-]	2.5
Installed cost [£/kW _c]	160
O&M costs [£/(kW _c -year)]	70
TRIGENERATION Scheme	
MGT	
Fuel	Natural gas
Power capacity per unit [kW _e]	80
Number of units	2
Electrical efficiency (NCV) [%]	26.7
Thermal efficiency (NCV) [%]	54.4
Availability factor (AF) [%]	98.6 (8,640 h/year)
Thermal output per unit [kW _{th}]	163
Exhaust gas temperature [°C]	275
Exhaust gas mass flow rate [kg/s]	0.83
Ambient reference temperature [°C]	95
Installed cost [£/kW _e]	1,009
O&M costs [£/kWh]	0.0051
Absorption chiller	
Cooling capacity per unit [kW _c]	12
COP _{abs} [-]	0.5 to 1.0
Installed cost [£/kW _c]	569
O&M costs [£/(kW _c -year)]	40

Table 2 Energy rates and emissions factor

Energy prices (excluding CCL & VAT) [18]	Notes	
Electricity purchased from National grid [p/kWh]:		
- Day-time (07:00h-24:00h)	6.6	Annual consumption: 880 to 8,800 MWh (medium size user)-average prices of three quarters in 2007 * Assuming 50% of the day-time electricity purchased price
- Night-time (00:00h-07-00h)	3.96*	
Electricity sold back [p/kWh]:		
- Day-time (07:00h-24:00h)	3.3**	** Assuming 50% of the electricity purchased price for day-time and night-time, accordingly
- Night-time (00:00h-07-00h)	1.98**	
Natural gas [p/kWh]	1.527	Firm contract (uninterruptible supply)-average prices of three quarters in 2007
Standing charge [£/month]	89	
Availability charge [£/(kW _e -month)]	0.78	
CCL rates and tax [19]		
Electricity [p/kWh]	0.441	
Natural gas [p/kWh]	0.154	
VAT [%]	17.5	
CO₂ emissions factor [20]		
Grid electricity [kg/kWh]	0.523	
Natural gas [kg/kWh]	0.185	