Design and optimisation of a small-scale tri-generation system

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Abstract This paper investigates design and theoretical analysis of a small-scale tri-generation system. The tri-generation system was made of a CHP unit and a heat-driven ejector cooling system. A TRNSYS mathematical tool was developed to simulate all components of the system in a dynamic process to understand the influence of various operating parameters. Different modes of operations were also considered including using a proportion of heat and/or electrical power output from the CHP to drive the cooling cycle to better assess the energetic and environmental performance of the system. Results of the simulation showed that fuel savings of about 15% could be achieved compared to using separate sources for heat, power and cooling.

Keywords CHP; tri-generation; ejector; TRNSYS

Nomenclature

\begin{align*}
A_g & \quad \text{CO}_2 \text{ emission factor for burning natural gas (kgCO}_2/\text{kWh)} \\
A_{gr} & \quad \text{CO}_2 \text{ emission factor of grid electricity displacement (kgCO}_2/\text{kWh)} \\
COP_{ej} & \quad \text{ejector coefficient of performance} \\
h_E & \quad \text{refrigerant vapour enthalpies at the exit of the evaporator (kJ/kg)} \\
h_G & \quad \text{refrigerant vapour enthalpies at the exit of the generator (kJ/kg)} \\
h_C & \quad \text{refrigerant liquid enthalpies at the exit of the condenser (kJ/kg)} \\
m_{CHP} & \quad \text{hot water mass flow rate (kg/s)} \\
m_E & \quad \text{refrigerant mass flow rate in the evaporator (kg/s)} \\
m_G & \quad \text{refrigerant mass flow rate in the generator (kg/s)} \\
Q_{CHP} & \quad \text{CHP unit heat output (W)} \\
Q_G & \quad \text{ejector cycle generator heat output (W)} \\
Q_E & \quad \text{ejector cycle evaporator cooling power (W)} \\
Q_{tri} & \quad \text{heat available from tri-generation for space heating (W)} \\
R & \quad \text{CO}_2 \text{ emission factor ratio of conventional to tri-generation system} \\
T_{CHP,i} & \quad \text{CHP unit return hot water temperature (°C)} \\
T_{CHP,o} & \quad \text{CHP unit supply hot water temperature (°C)} \\
W_{tri} & \quad \text{electrical power output of tri-generation system (W)} \\
y & \quad \text{ratio of heat used to drive the ejector cycle to heat used for heating} \\
\eta_T & \quad \text{tri-generation system overall efficiency}
\end{align*}

1 Introduction

Combined Heat and Power (CHP) and renewable energy technologies coupled with energy conservation measures form essentially the bulk for implementation of the UK government energy policy. One of the UK energy strategies is to achieve
10 MWe of electrical power generation from CHP plants by 2010, i.e. nearly doubling the existing CHP generation capacity of about 6 MWe. The government also aims to generate 10% of its total electricity consumption from renewable energy sources by 2010.

Compared to the conventional approach of producing heat and power from separate plants, CHP schemes have the potential to reduce fuel consumption by up to 30% while producing an equivalent amount of electricity and heat. CHP systems have been used mainly in district heating supplies of large urban areas using large steam turbines and in the form of packaged small-scale units in commercial, services and industrial facilities using internal combustion engines and gas turbines, where there is continuous demand for heat and power for long periods of the day. CHP systems often operate in parallel with the local grid whereby power flows in two directions for export and import purposes; this often has the advantage of sizing the CHP system to optimise heat load demand. However, the drawback of CHP systems is that high efficiency can be achieved only when electricity and heat is produced simultaneously in a single process. Since the deregulation of the energy industry in the UK in 1990s, the electrical power generators including CHP generators were forced to sell electricity at spot market prices, which vary constantly. This has created challenging implementation conditions for CHP systems and the task of identifying the optimal size become much more complicated as the option of exporting excess power to the grid may become uneconomical and force CHP plants to shut down instead [1, 2, and 3].

Recently, interest in CHP systems has shifted towards exploring systems with power output of a few kW_e for potential applications in residential and small size buildings. Available prime mover technologies such as internal combustion engines is a mature technology as it has undergone decades of development for the automobile industry while novice and more efficient technologies under development include fuel cells and Stirling engines as the most likely future candidates technologies [4, 5].

It is well known that one of the prerequisites for a CHP system to be economically viable is to match its thermal and electrical outputs with the heat and power demand of the site in order to maximise the number of operating hours of the plant. In many small-scale applications however, running hours are limited to heating seasons. Hence, excess heat would be produced in periods of low demand for heating which would be unacceptable to reject it to the environment. To remedy this operational restriction, excess heat from the CHP unit could be used to operate a heat driven air conditioning or refrigeration system to form what is commonly known as a tri-generation system and thereby enhancing the efficiency of CHP and increase its economical and environmental benefits [6, 7].

Tri-generation systems ranging from tens of kW_e to several MW_e have been operated in several sectors, and their use is set to increase with the development of packaged products [8, 9]. In this range, most CHP systems use internal combustion engines, large gas turbines or micro-gas turbines as prime movers, whereas absorption chillers are the preferred option for cooling and refrigeration. Interest in micro-scale CHP systems has led extensive field trials of a number of micro-CHP systems.
using Stirling engine as prime mover as well as fuel cells with absorption, adsorption and ejector cycles as cooling systems are also trialled [10, 11, 12]. However, limited available experimental results and issues of costs and efficiency of the cooling cycle are even more challenging at the lower end of small-scale systems with power outputs in the range of a few kWe. Therefore, this paper is an attempt to investigate a small tri-generation system combining a commercially available CHP unit, known as Dachs CHP unit, and an in-house built ejector cooling cycle.

2 Description of the tri-generation system

2.1 The Combined Heat and Power (CHP) Unit

The present research was conducted on two separate Dachs CHP units that were installed as a retrofit heating system at the School of the Built Environment, University of Nottingham. Fig. 1 shows a schematic diagram of the CHP units installation where it operates as lead heating system and connected in parallel to the building power grid. The CHP unit is built around a natural gas-fuelled internal combustion engine (0.578 litre single-stroke four-cylinder engine) capable of delivering 5.5 kW of electrical power and 12.5 kW of heat in form of hot water at maximum temperature of 83°C. The CHP units were extensively monitored to establish continuous operation performance and this is given in Table 1 where efficiencies are based on fuel Net Calorific Value.

It can be seen from Table 1 that data obtained from the monitoring exercise is largely in accordance with the manufacturer’s energy performance specification. However, as stated earlier, the economical viability would essentially depend on matching closely the site demand for heat and power to the CHP energy outputs to maximise the number of operating hours per year. In most small size buildings, there is simply not enough heat loads in summer time for the CHP to operate without dumping excess heat to the environment. This situation however is changing with the tightening of building regulations to minimise energy loss through conduction and ventilation making buildings’ indoor environment uncomfortable and hence the need for space air conditioning during the hot season. This would be an ideal opportunity to extend the number of operating hours of a CHP unit year round by using rejected heat to drive a cooling system.

2.2 The heat driven cooling cycle

A number of heat driven technologies are often associated with CHP systems. Absorption systems are commercially available for large scales of MW cooling capacities. For the present application of a few kW cooling capacity, the ejector cycle technology was selected for its mechanical simplicity, low cost, and ability to operate from low-grade heat [13, 14]. The development of such a cooling cycle would also be of particular interest in applications where low-grade heat is generated from renewable energy sources and fuel cells [11, 15].

In an ejector cooling cycle, the refrigerant compression is achieved by a static Venturi type device known as the ejector. A high-pressure refrigerant vapour contained in a generator is released through the ejector nozzle where its speed exceeds
the speed of sound sucking in the process a cooling fluid contained in the evaporator. The resulting flow mixing then slows down in the diffuser section of the ejector and undergoes a compression process through a pressure shock at the condenser pressure. The refrigerant liquid in the condenser is then pumped back to the generator and evaporator to complete a closed thermodynamic cycle. Fig. 2 shows a schematic
Table 1. Comparison of manufacturer’s and monitored CHP unit performance

<table>
<thead>
<tr>
<th>Specified</th>
<th>Monitored</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical output (kW)</td>
<td>5.34</td>
</tr>
<tr>
<td>Thermal output (kW)</td>
<td>12.5</td>
</tr>
<tr>
<td>Electrical efficiency (%)</td>
<td>23.4</td>
</tr>
<tr>
<td>Thermal efficiency (%)</td>
<td>54.8</td>
</tr>
<tr>
<td>Overall efficiency (%)</td>
<td>78.2</td>
</tr>
<tr>
<td>Maintenance intervals</td>
<td>3,500 hours</td>
</tr>
<tr>
<td>Operation life</td>
<td>&gt;80,000 hours</td>
</tr>
</tbody>
</table>

Figure 2. Layout diagram of tri-generation system.

diagram of the system, where the generator is composed of the heat exchanger interfacing the CHP unit and the cooling system.

3 Simulation of tri-generation system

The interaction of different components making the tri-generation system was studied by building a TRNSYS computer simulation model to optimise operational performance of the system. A user-friendly graphical interface of TRNSYS allows the connecting of the input/output parameters of different blocks of the tri-generation system, that is the Dachs CHP unit, the ejector cycle, vapour separator vessel, and properties of the refrigerant HFE 7100, as shown in Fig. 3. The interaction of inputs and outputs of these components defines a system of mathematical equations, which is then solved by TRNSYS to find, where possible, a converging steady state
solution. The simulation model of the Dachs CHP unit was built using both the manufacturer’s specifications and monitoring data. The computer modelling of the ejector cycle was obtained using a modified Keenan model by adding refrigerant HFE 7100 tabulated properties.

3.1 Combined Heat and power system

The thermal generation capacity of the CHP system, $Q_{CHP}$, was assumed to be constant but the hot water flow rate, $m_{CHP}$, is variable and depends on temperature of hot water return from the building, $T_{CHP,i}$. The CHP output hot water temperature is then expressed as:

$$T_{CHP,o} = T_{CHP,i} + \frac{Q_{CHP}}{m_{CHP} C_p}$$  \hspace{1cm} (1)

The built-in control function was set up to mimic accurately the manufacturer’s specification of the CHP operation conditions so that the unit would shut down if temperature of hot water from the building, $T_{CHP,o}$ is greater than 73°C and switch on when the temperature is lower than 70°C.

The TRNSYS simulation model was developed to allow for the following combinations of operational scenarios:

i) The cooling system could use only part of the heat from the CHP.

ii) Part of the electrical output from the CHP could be used to boost the cooling cycle generator temperature, allowing higher condenser temperatures and/or higher COP values.
iii) The size of the interface heat exchanger between the CHP unit and ejector cooling system was designed to obtain large overall heat transfer coefficient and high cooling capacity.

3.2 Ejector cooling system
The simulation of the ejector using TRNSYS was carried out by developing FORTRAN program codes based on the Keenan model and embedded properties of HFE7100. The program codes were developed as follows:

i) Calculation of the operating pressures and enthalpies from the input temperature.

ii) For an initial entrainment ratio, an iterative calculation process of critical pressure is carried until the value of the critical pressure is equal that of the condenser.

iii) Calculation of the generator heating capacity, evaporator cooling capacity and COP of the cooling system.

The coefficient of performance of the cycle, \( \text{COP}_{ej} \), is the ratio of heat energy available from the evaporator and generator respectively and can be expressed as follows:

\[
\text{COP}_{ej} = \frac{Q_E}{Q_G} = \frac{m_E(h_E-h_C)}{m_G(h_G-h_C)}
\]  

(2)

where \( h_E \) and \( h_G \) are the saturated vapour enthalpies of the refrigerant at the exit of the evaporator and generator respectively and \( h_C \) is the saturated liquid enthalpy at the exit of the condenser.

In Equation 2, the power consumption of fluid pumps has not been taken into account, as this is usually negligible compared to the generator capacity [16]. One of the important parameter in ejectors design is the critical condenser pressure, \( P_{C^*} \). This is determined by the ejector geometry and size, properties of the refrigerant, and operating conditions of the evaporator and generator. The COP of an ejector would remain broadly constant for a given evaporator and condenser temperature, \( T_E \), and, \( T_G \), respectively so long as the condenser operating pressure is kept below the critical pressure \( P_{C^*} \). However, the ejector performance would drop sharply if the condenser pressure is higher than the critical pressure.

3.3 Refrigerant HFE 7100
Refrigerant HFE 7100 was selected for its performance at low driving temperatures, as well as operational and environmental criteria of low toxicity and low flammability, charge pressure above 10 mbar and below 3 bar, zero ozone depletion potential, and low global warming potential. The temperature of the heat source (i.e., CHP) of 83°C is adequate for use with HFE 7100. The saturation temperature of HFE 7100 in the CHP/ejector heat exchanger was assumed to average 72°C. Fig. 4 shows design mapping of the ejector COP as a function of the critical pressure. With increasing critical pressure, the COP of the cooling cycle increases with increasing evaporator temperature, and decreases with increasing generator temperature [14, 16].
4 Simulation results

4.1 Performance analysis

The performance of the tri-generation system was analysed for the operational case scenarios outlined earlier. In cases where the tri-generation system provides heat, power and cooling, a fraction, \( y \), of the available heat supply from the CHP unit, \( Q_{CHP} \), was used to drive the cooling cycle (i.e., \( yQ_{CHP} \)) and the remaining heat, \((1 - y)Q_{CHP}\), was used for hot water and space heating purposes. Considered in this analysis also was the diversion of part of the electrical power output from the CHP unit, \( W_{tri} \), to superheat the refrigerant (a heat booster) of the cooling system, so that higher COP can be achieved at higher condenser temperatures. The influence of ratio, \( y \), and heat transfer coefficient, \( U_A \), of the heat exchanger interfacing between the CHP and the ejector cooling system on the cooling capacity, \( Q_E \), and overall efficiency of the tri-generation system, \( \eta_T \), is shown in Fig. 5 and Fig. 6 respectively.

It can be seen from Fig. 5 and Fig. 6 that increasing the amount of generated heat from the CHP unit to drive the cooling cycle (i.e. increasing ratio, \( y \)) increases the cooling capacity while the overall efficiency of the tri-generation system decreases drastically. On the other hand, diverting part of the CHP electrical power output to boost the cooling cycle upper end temperature increases the cooling capacity by up to 49% for \( y = 1 \). The effect on the overall efficiency of the tri-generation system though is negligible. It was also found that the interface heat exchanger optimum heat transfer coefficient ranges from 1000 W/K for \( y = 0.3 \) to 2000 W/K for \( y \geq 0.6 \).
4.2 CO₂ emissions analysis
The environmental impact of this energy efficient tri-generation system can be quantified in terms of CO₂ emission savings when compared to a more traditional method of providing heat, power and cooling from separate sources using a boiler, grid and mechanical vapour compression system respectively.
The efficiency of the tri-generation system can then be written as:

\[
\eta_{\text{tri}} = \frac{W_{\text{tri}} + Q_{\text{tri}} + Q_{E}}{Q_{G}}
\]

(3)

where \(Q_{G}\), \(W_{\text{tri}}\), \(Q_{\text{tri}}\), and \(Q_{E}\) are the gross calorific value of fuel energy input, electrical power output, heat output and cooling output respectively.

The amount of heat available from the tri-generation system for heating purposes can be given by:

\[
Q_{\text{tri}} = (1 - y)Q_{\text{CHP}}
\]

(4)

Hence, equation 3 can be re-written as follows:

\[
\eta_{\text{tri}} = \frac{W_{\text{tri}} + (1-y)Q_{\text{CHP}} + y\text{COP}_{Q}Q_{\text{CHP}}}{Q_{G}}
\]

(5)

The amount of emission of CO\(_2\) depends on the type of fuel use. For instance, CO\(_2\) emission factor, \(A_{g}\), for burning natural gas is set to 0.194 kgCO\(_2\)/kWh and emission factor, \(A_{gr}\), for using grid electricity is 0.568 kgCO\(_2\)/kWh when electricity is being displaced through on site generation or 0.42 for simple consumption of electricity [17].

Hence, the overall CO\(_2\) emission factor, \(E_{\text{tri}}\), for the tri-generation system could be evaluated as:

\[
E_{\text{tri}} = A_{g}Q_{G}
\]

(6)

Similarly, if electrical power, \(W_{\text{tri}}\), heat output, \(Q_{\text{CHP}}\), and cooling power, \(Q_{E}\), were provided separately from grid, local gas boiler of efficiency, \(\eta_{B}\), and a mechanical vapour compression cooling system of coefficient of performance, \(\text{COP}_{\text{comp}}\), then the total CO\(_2\) emission factor would be calculated as follows:

\[
E_{\text{conv}} = W_{\text{tri}}A_{g} + (1-y)\frac{Q_{\text{CHP}}}{\eta_{B}}A_{g} + yQ_{\text{CHP}}\frac{\text{COP}_{Q}}{\text{COP}_{\text{comp}}}A_{gr}
\]

(7)

Comparing equation 6 and equation 7, it can be seen that savings on CO\(_2\) emission could be realised using the tri-generation system only when the ratio \(R = \frac{E_{\text{conv}}}{E_{\text{tri}}}\) is higher than 1.

Tables 2(a) and (b) show the effect of varying parameter, \(y\), which expresses the amount of heat used to drive the cooling system on the overall CO\(_2\) emission savings of the tri-generation system. In this calculation, it was assumed that if heat and cooling power are to be supplied from a condensing boiler and a mechanical vapour compression then the efficiency, \(\eta_{B}\), and COP, \(\text{COP}_{\text{comp}}\), are set to 0.9 and 0.3 respectively. It is interesting to notice from Tables 2(a) and (b) that where most of the CHP heat is used to drive the cooling cycle (i.e., \(y \geq 0.6\)), CO\(_2\) emission will actually increase compared to conventional systems (i.e., \(R \leq 1\)). The best emission savings were obtained for \(y = 0.3\) with overall reduction in CO\(_2\) emission of 16\% and 13\% for cooling system without a booster and with a booster respectively. The selection
of either design options should then be based on electrical and cooling requirements of the targeted site. Diverting part of the CHP electrical power output into the cooling cycle does not affect the overall efficiency of the tri-generation system, however the electrical efficiency of the system is decreased. Hence, Table 2b shows that for \( y \geq 0.6 \) the resulting environmental performances are worse than in the case where no electrical power was used to drive the cooling system.

5 Conclusion

A small-scale tri-generation system to provide heat, electrical power and cooling was investigated and a TRNSYS computer simulation model was developed. An ejector cooling system was designed to work with available low-grade heat from the CHP system. The design was based on an improved Keenan analysis with real refrigerant HFE 7100 properties. The tri-generation system design was evaluated for its energetic and environmental performance.

The analysis showed that the overall energy efficiency of the tri-generation system could approach 70% and would save as much as 15% in \( \text{CO}_2 \) emission when a heat to cooling power ratio, \( y \), was selected. It was also found that while using part of the CHP electrical power output to improve the refrigerant vapour quality of the cooling cycle boosts the cooling capacity, it does not improve the overall efficiency of the tri-generation and in effect, it led to higher \( \text{CO}_2 \) emissions.

One area that would benefit future research in this area to improve the ejector cycle COP is by using the heat available in the exhaust gases from the CHP unit to boost the generator temperature of the cooling cycle.

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References


