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Thermodynamic efficiency of low-carbon domestic heating systems: heat pumps and micro-cogeneration

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Abstract:

Energy and exergy analysis is employed to compare the relative thermodynamic performance of low carbon domestic energy systems based on Air Source Heat Pumps and micro Combined Heat and Power (cogeneration) units. A wide range of current units are modelled under different operating conditions representative of the United Kingdom to determine the energy and exergy flows from primary energy inputs through to low carbon heating system and then to end use. The resulting performances are then analysed in order to provide insights regarding the relative merits of the systems under the different operating constraints that may be experienced both now and into the future. Although current mid-range systems achieve comparable performance to a condensing gas boiler, the state-of-art offers considerable improvements. Micro Combined Heat and Power units and Air Source Heat Pumps have the technical potential to improve the energy performance of dwellings. The relative performance and potential of the systems is dominated by the electrical characteristics: the grid electrical generation efficiency, the power-to-heat demand ratio and the availability of electrical export. For total power-to-heat demands below 1:1.5, Air Source Heat Pumps have greater improvement potential as their energy efficiency is not constrained. At higher power-to-heat ratios, micro Combined Heat and Power units offer the potential for higher overall efficiency and this generally occurs irrespective of whether or not the thermal energy from them is used effectively.

Keywords:

low carbon, heat pump, micro combined heat and power, cogeneration, heating, exergy analysis

1 INTRODUCTION

The carbon emissions and energy performance of the United Kingdom (UK) domestic sector must be improved. Domestic space heating accounted for 20% of total primary energy consumption of the UK in 2010, contributing 13% of CO₂ emissions [1]. It is estimated that a 29% reduction in the CO₂e emissions associated with domestic heating will be required as the UK meets its commitment to a total reduction of 34% relative to 1990 by 2020 [2].

To achieve this will require more efficient heating systems. Although ambitious improvements in building standards are envisioned, it is estimated that 80% of the UK's 2050 building stock is currently standing [3] and even with extensive refurbishment, domestic heat and power demand are likely to remain substantial [4]. Micro Combined Heat and Power units (mCHP) and Air Source Heat Pumps (ASHP) have both been suggested as devices with the potential to reduce carbon emissions and energy demand [5]. They are increasingly attracting attention, with major field trials of both devices conducted in recent years [6–9].

The approach taken in this study has been to model a selection of actual mCHP and ASHP units under a variety of different assumed conditions and then to analyse the resulting range of energy and exergy performances to draw general conclusions about the future applicability of these systems. Energy and exergy flows have been modelled in terms of three stages: (1) primary energy conversion and distribution, (2) mCHP or ASHP unit and (3) heat distribution in the dwelling (see Figure 1). In comparing the relative system performances, the ratio of power-heat consumed at the end use stage is shown to be a key determinant of which system has the greater potential efficiency.

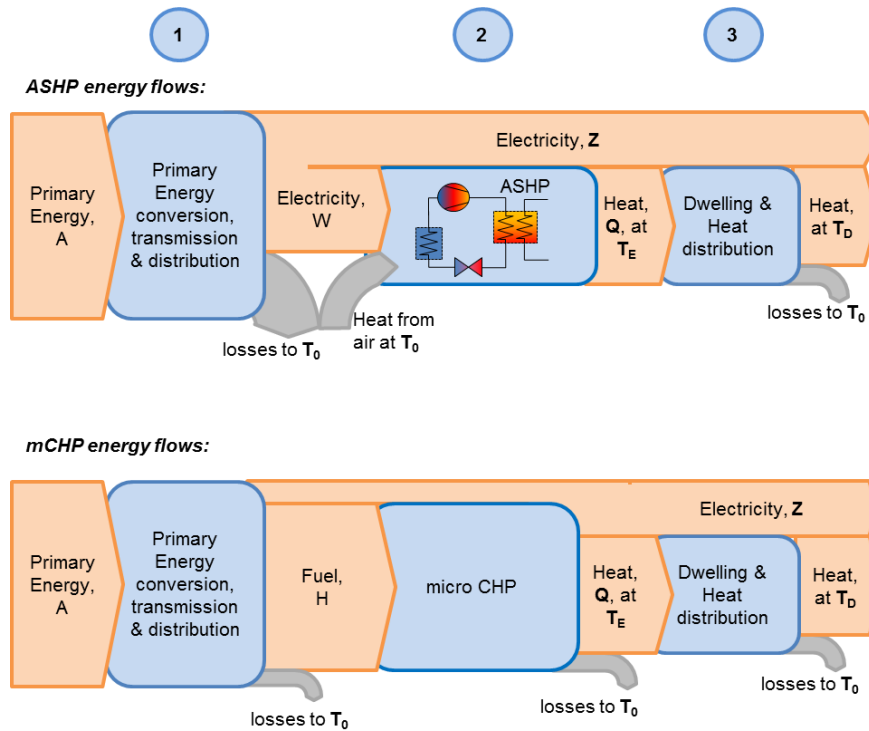


Figure 1: Representative energy flows

2 BACKGROUND

2.1 Combined Heat and Power Units

Combined Heat and Power units combine the generation of power with heat recovery in order to maximise efficiency. Stirling Engine mCHP (SE-mCHP) units are currently the most popular type of mCHP in the UK private domestic sector as Internal Combustion Engine based mCHP (ICE-mCHP) devices tend to be larger and pose more challenges to successful integration into the domestic environment. The higher capital cost of fuel cell based mCHP has contributed to limited adoption to date but they still attract interest due to their potential for high electrical efficiency [10]. The two leading fuel cell technologies are Polymer Exchange Membrane Fuel Cell (PEMFC) and Solid Oxide Fuel Cell (SOFC). Uptake of PEMFC units has been greater (e.g. [7]) but the higher electrical efficiency and simpler fuel reformer requirements of SOFC units means that they are potentially the more attractive option in the longer term [11].

2.2 Heat Pump Systems

Heat pumps use a thermodynamic cycle to transfer heat from a source to a sink at higher temperature. In the context of domestic heating they extract heat from the outside air, ground or water. Most commonly, a vapour compression cycle is employed with an electrically driven compressor. Performance is measured by the heat output versus the consumed electricity and expressed as the Coefficient of Performance (COP):

$$C = \frac{Q}{W} \quad 2.1$$

where C is the COP, Q is the heat delivered and W is the total work input to the heat pump unit (excluding any component of power which is used for pumping hot water outside the heat pump).

Performance specifications and test conditions are standardised for ASHP application to space heating [12] and potable hot water supply [13]; these definitions are followed in this study. Although Ground Source Heat Pumps (GSHP, those that employ the ground as their heat source) are generally more efficient, the market potential for Air Source Heat Pumps (ASHP) is considered larger in the UK due to the additional installation requirements associated with GSHPs [14].

2.3 Exergy

To consider exergy, consideration must be made of the Second Law of Thermodynamics as well as the First Law; unlike energy, exergy is destroyed in any process that involves irreversibility:

$$\sum E_{in} - (\sum E_{out} + \sum E_{lost}) = I > 0 \quad 2.2$$

where $\sum E_{in}$ is the sum of exergy inputs to the system, $\sum E_{out}$ is the sum of desired exergy outputs from the system, $\sum E_{lost}$ is the sum of any other exergy outputs from the system and I is the amount of irreversibility associated with the process. The exergy efficiency is then defined [15]:

$$\psi = E_{out} / E_{in} \quad 2.3$$

Considering the case of heat transfer, Q , across the system boundary at constant temperature (T), the associated exergy transfer is given by the maximum work that could be obtained from that heat transfer:

$$E^Q = Q(1 - \frac{T_0}{T}) \quad 2.4$$

where T_0 is ambient temperature and T is the temperature of the heat sink (e.g. heat emitters or inside air).

Kotas (1980) provides a methodology for calculating the exergy content of chemical enthalpy. It is convenient to consider the exergy content of the fuel as related to its enthalpy of combustion, H , by a factor ϕ :

$$E_{fuel} = \phi.H \quad 2.5$$

Allen & Hammond [17] have collected values of ϕ for various fuels. These are adapted for inclusion in Table 1:

Table 1: Exergy-energy ratio of fossil fuels. Adapted from [17]

Fuel	ϕ
Coal	1.03
Fuel Oil	1.01
Natural Gas	0.94

3 METHODOLOGY

Performance data characterising 10 ASHP units and 11 mCHP units were gathered in order to provide an overview of the range of units currently available. Although nominal performance data are readily available ([10], [18]) they are generally specific to particular steady-state conditions. Therefore, to analyse the energy flows at each of the stages in Figure 1 in a way that enabled meaningful comparison between the units, a dynamic model was constructed. The model includes a component for each stage, calculating energy and exergy flows at each stage for each time step. Different permutations of building insulation, climate, heat emitter size and heating control system were selected as representative of the range of conditions that might be expected between now and 2030 (see Table 3). Once comparable performance results were determined for each unit, these were analysed for different ratios of final electrical power demand to heat demand, derived by assuming different domestic electrical power demands.

3.1 Grid electrical supply model

In order to calculate the primary energy and exergy requirements associated with each system, it is necessary to know how the electricity which either supplies the ASHPs or is displaced by the mCHP electrical production is generated. Two cases were considered in order to demonstrate the effect of assumptions regarding grid supply. Firstly, a Combined Cycle Gas Turbine (CCGT) achieving a constant 54% electrical efficiency [19], equivalent to 50.1% efficiency at point of use (taking transmission and distribution losses as 7.3% of electricity generated, [20]). Secondly, a dispatch model was constructed by using the electrical generation totals for 2030 from the “Market Rules” (“MR2030”) scenario developed by the Transition Pathways project (see Table 2, [21]) and historic generation data [22]. The dispatch model uses a modified merit order approach, similar to [23]; renewable and nuclear generation are prioritised, followed by industrial CHP, CCS plant and then CCGT and Coal for peaking. It is assumed that there is some overlap in the operation of the thermal plant; that is, CCGT and coal plant supply a proportion of demand when the CCS plant is operating above 80% capacity. This is in contrast to a strict merit order assumption where the conventional plant would only operate once the CCS plant was at maximum capacity. Hammond & Stapleton [24] were followed in taking the input to non-thermal renewable energy systems and imports to be equal to their electrical output. CCS plant electrical efficiency was taken from [25]. Transmission and distribution losses were assumed to remain the same.

As this study is comparative, the *marginal* primary energy requirement for electrical power was calculated each time step. The amount of electricity generated from renewable and nuclear sources changes less than the amount generated from fossil fuels when demand is affected by the heating systems and it is these changes which are taken into account. This has the interesting effect that the average marginal primary energy and exergy requirements for electricity are higher in the “MR2030” scenario than for the CCGT plant even though CO₂ emissions rates are lower. Marginal primary exergy requirements were calculated as the product of the primary energy requirements and the factors in Table 1.

Table 2: Electricity generation mix (output mix). Data calculated from [21] & [25]

Generating plant type	Transition Pathways, “Market Rules” 2030 mix	Average electrical generation efficiency
Gas CCGT	9.3%	43%
Coal CCS	16.5%	27%
Gas with CCS	16.4%	37%
Nuclear	17.1%	34%
Wind	20.2%	-
Other renewables	7.3%	-
Imports	4.9%	34%
Other (including CHP & coal)	14.3%	64%

Losses from natural gas pumping, leakage and other operator activities amounted to 2% of the total natural gas energy input in 2010 with a resultant energy requirement for natural gas supplied to mCHP units of 1.02. Losses upstream from input to the national transmissions system (e.g. extraction and processing) are not included in this study.

3.2 mCHP models

Performance metrics were gathered for 11 different mCHP units. A “grey box” approach was taken, based on that developed by IEA ECBCS Annex 42 [26] [27], using two lumped thermal capacities. Parameters for two of the units (one SE-mCHP unit and one ICE-mCHP unit) were taken directly from Annex 42 calibration work, while parameters for two other ICE-mCHP units, one SE-mCHP and two PEMFC-mCHP units were calculated

based upon experimental data provided in the same report [28]. Steady-state performance was generally directly available (in this study, all values are converted to be relative to the higher heating value, HHV, i.e. the gross calorific value of fuel). However, in some cases, the relevant thermal inertias and heat transfer coefficients had to be inferred from the warm up and cool down characteristics of the unit and (in the case of the SE-mCHP) from the part-load electrical efficiency. It was not possible to model the variation in electrical efficiency of the ICE-mCHP units in this way and so an additional performance map was used. Similarly, in the case of fuel cell mCHP units, a full Annex 42 type model was not developed; instead, a similar approach was taken to model the thermal inertia of the units but the electrical and thermal efficiencies were interpolated between observed values at different fuel flows. This was done to enable the consistent use of reported performance data for other units (two PEMFC-mCHP units, one SE-mCHP unit and one SOFC-mCHP unit [29], [30], [31], [32], [33]).

Although these simplifications reduce confidence in the model's ability to predict the exact performance of a unit in specific circumstances, it is reasonable to assume that they will indicate the unit's performance under similar circumstances.

The units (apart from the SOFC-mCHP unit) are assumed to operate with a heat-led control methodology. It was assumed that there is no restriction to the export of electrical power and that this electricity displaces other demand locally (i.e. without distribution losses). This is not the case in locations where electrical export is not allowed (e.g. Japan) and may not be the case in locations where tariff structures disincentivise export (e.g. UK). The effect of this assumption is highlighted in the discussion section. In all results, net electrical power (i.e. net of inverter losses) is used. The SOFC-mCHP unit is constrained to a very low maximum rate of change of output. Therefore, rather than operating as a heat-led unit, four operating regimes were designed with different prioritisations between maximising electrical efficiency and minimising excess heat generation. These four regimes are treated as separate units. Two are constrained to always operate at the fuel flow corresponding to peak electrical efficiency or above while the other two can be turned down. One unit of each of these pairs is also oversized such that the fuel flow corresponding to peak heat demand also corresponds to maximum efficiency. In each case, excess heat is rejected and only the heat used is included in efficiency calculations.

The heat flows within each mCHP unit were calculated using a one second time step (to ensure modelling stability given the low ratio between the thermal inertia and the heat transfer coefficient in some cases). Heat flows outside the mCHP units were calculated using a one minute time step (see [34]).

3.3 ASHP models

Steady state performance data for 10 ASHP units were gathered from reports published by BRE Ltd [35] and the Swiss Wärmepumpen-Testzentrum (WPZ) [36]. Each performance measurement is specific to a given outside air (heat source) temperature and a flow (heat sink) temperature. Linear interpolation of the isentropic efficiency of the heat pump between these measured points was used to determine the COP of each ASHP when either the outside air temperature or the flow temperature differed from the relevant test temperature. A similar approach to that taken with the mCHP units was used to model the dynamic response of the ASHP units. However, given the low time constant of the units and the fact that they use modulating drives, their overall energetic performance is less sensitive to the value of their thermal inertia and so it was sufficient to estimate this from their size and the dynamic response of other ASHP units where available.

The ASHP units considered cover a range of nominal performances from the mid-range to the state-of-the-art currently available. Previous studies [37–39] have reviewed technical development and used exergy analysis at component level to identify that there are various aspects of ASHP design that can be optimised to improve the COP. An on-going trend towards higher performance is also observed in the units which are available and it is likely that the current state-of-the-art performance will be commonly achieved by mid-range units by the end of the decade.

3.4 Dwelling model

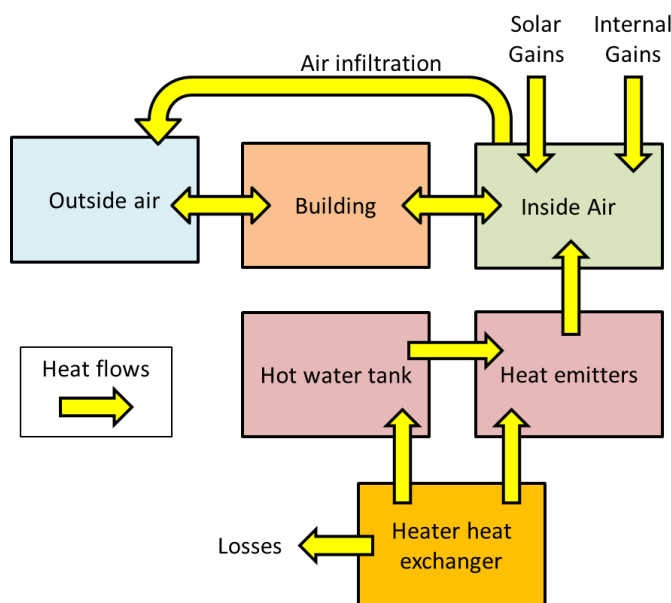


Figure 2: Dwelling heat flows simplified model

Given that the aim of this study is to analyse patterns and trends in the energetic performance of the domestic heating systems under consideration, it was decided that the most appropriate approach was to model a relatively high total number of permutations (see Table 3) with a dwelling thermal model sufficiently simplified to enable this. A lumped capacitance model was therefore selected; this includes a single building fabric thermal inertia with internal and external heat transfer coefficients, an internal air thermal inertia, internal gains, an effective solar gains area and an air infiltration rate (Figure 2). These parameters were calibrated against the thermal characteristics of a detailed model of a building created in ESP-r by Dr. N. Kelly and Dr. J. Hong at ESRU, University of Strathclyde to assist with a similar study (see [40], [41]). The effective building heat loss coefficient is 169W/K; lower than the average for the entire UK stock (247W/K) but similar to the average heat loss from flats (167W/K) [42]. A second dwelling with identical inertia and gains but with reduced external heat loss and air infiltration (equivalent heat loss coefficient of 118W/K) was used to represent improved building standards. Test Reference Year climate data was obtained from the PROMETHEUS project [43], based on UKCP09 climate modelling [44]. Two modelled climates were selected for the study: present day Glasgow (coastal) and central London in the 2030s. Two heat emitter systems were modelled; these were sized with heat transfer coefficients such that flow temperatures of 45°C and 35°C are required to maintain an indoor temperature of 20°C in the standard building when the outside air temperature is 0°C. A hot water tank of 200 litre capacity was used in each dwelling with insulation limiting its heat loss (to the inside air) to 2W/K. Domestic hot water demand was based on national averages [45]. Two heating control systems were used: (i) aiming for 55°C flow temperature using an on-off thermostat with a temperature dead-band of 2°C (although the actual flow temperature depends upon the conditions) and (ii) heat generation proportional to the difference in temperature between the inside air temperature and a control set point (20°C), flow temperature dependent upon the thermal characteristics of the heat emitters system and the ASHP / mCHP unit. These permutations are summarised in Table 3:

Table 3: Simulation permutations

Grid electrical supply	Heating unit	Control	Climate	Building	Other electrical demand
• Transition Pathways, “Market Rules” scenario for UK (2030) • CCGT	• 3 SE-MCHP • 3 ICE-mCHP • 4 PEMFC-MCHP • 1 SOFC-mCHP (in four configurations) • 10 ASHPs	• Fixed temperature (on-off thermostat controlled) • Variable temperature based on proportional controller	• London, 2030s modelled • Glasgow, 1970’s	• Current insulation • Improved insulation <i>and</i> • Standard heat emitters • Improved heat emitters	• 2230kWh/yr • 4460kWh/yr • 8920kWh/yr

The exergy transfers associated with the heat flows to the heat emitters and to the air inside the dwelling were calculated using equation 2.4 and the flow, return, inside air and outside air temperatures.

In order to consider the efficiencies of the whole system (from primary energy inputs to final use of energy) in terms of the ratio between thermal and electrical demand, three different annual electrical demands (i.e. appliances and lighting, excluding heating) were assumed based on the UK annual average (4460kWh/yr [1]), twice the UK annual average and then half of it.

4 RESULTS & DISCUSSION

4.1 mCHP unit performances

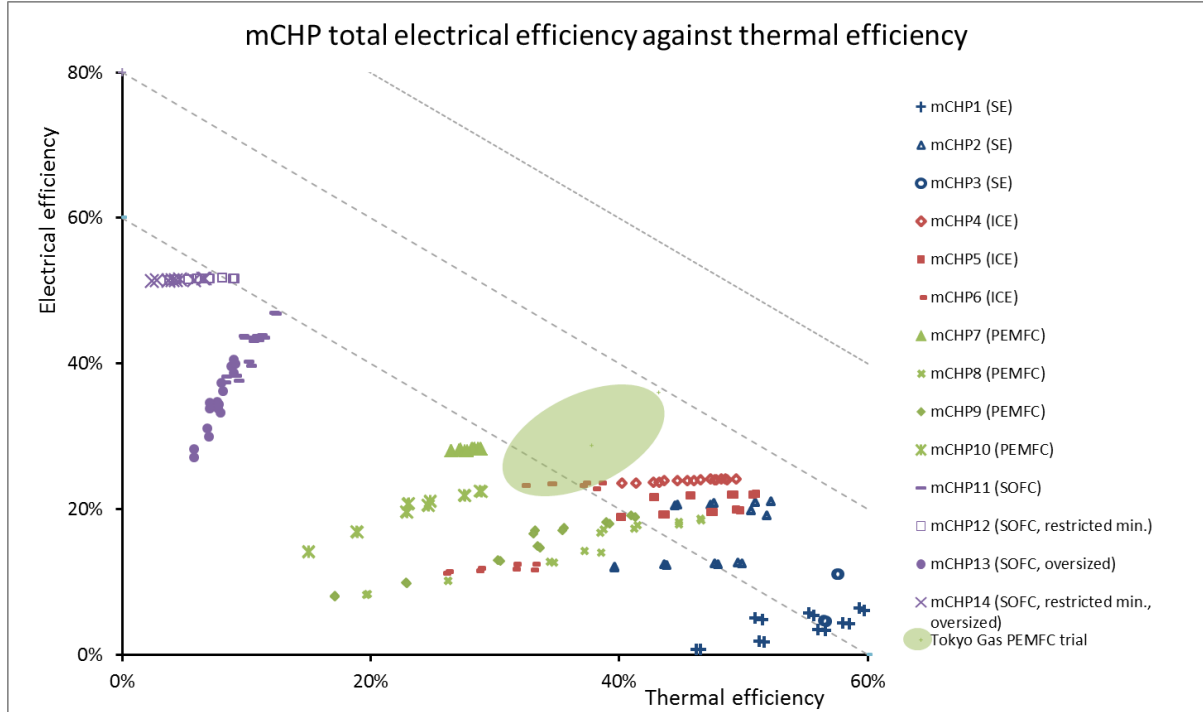


Figure 3: Variation in mCHP electrical efficiency with thermal efficiency.

The calculated mCHP electrical and thermal efficiencies are plotted in Figure 3. A large field trial of PEMFC units in Japan [7] reported higher performances than those modelled (average efficiencies of 28.8% electrical and 37.8% thermal). A lack of available detail regarding the units used and the conditions in the trial prevents direct comparison but the results are clearly significant and are approximated as an ellipse on the graph for completeness. The ICE-mCHP units tend to show quite consistent electrical efficiency whereas the smaller SE-mCHP units suffer greater electrical performance degradation with the higher cycling associated with less consistent heat demands [46]. One of the PEMFC units (“mCHP7”) does not show the same range of thermal efficiency variation. This is because the unit is undersized for most domestic heating duties and so operates more consistently; however, the use of an auxiliary heater is not taken into account and would significantly reduce the system electrical efficiency. Although intended to be indicative only, the modelling supports the suggestion that the SOFC-mCHP unit should be run continuously, rejecting heat as necessary. The slow ramp rate of the unit makes it difficult to fully utilise the 20% thermal recovery that is theoretically possible.

The exergy efficiencies of the mCHP units are plotted against the power fraction they supply, F_p , in figure 4, where:

$$F_p = P / (P + Q)$$

4.1

where P is the electrical power generated by the mCHP unit.

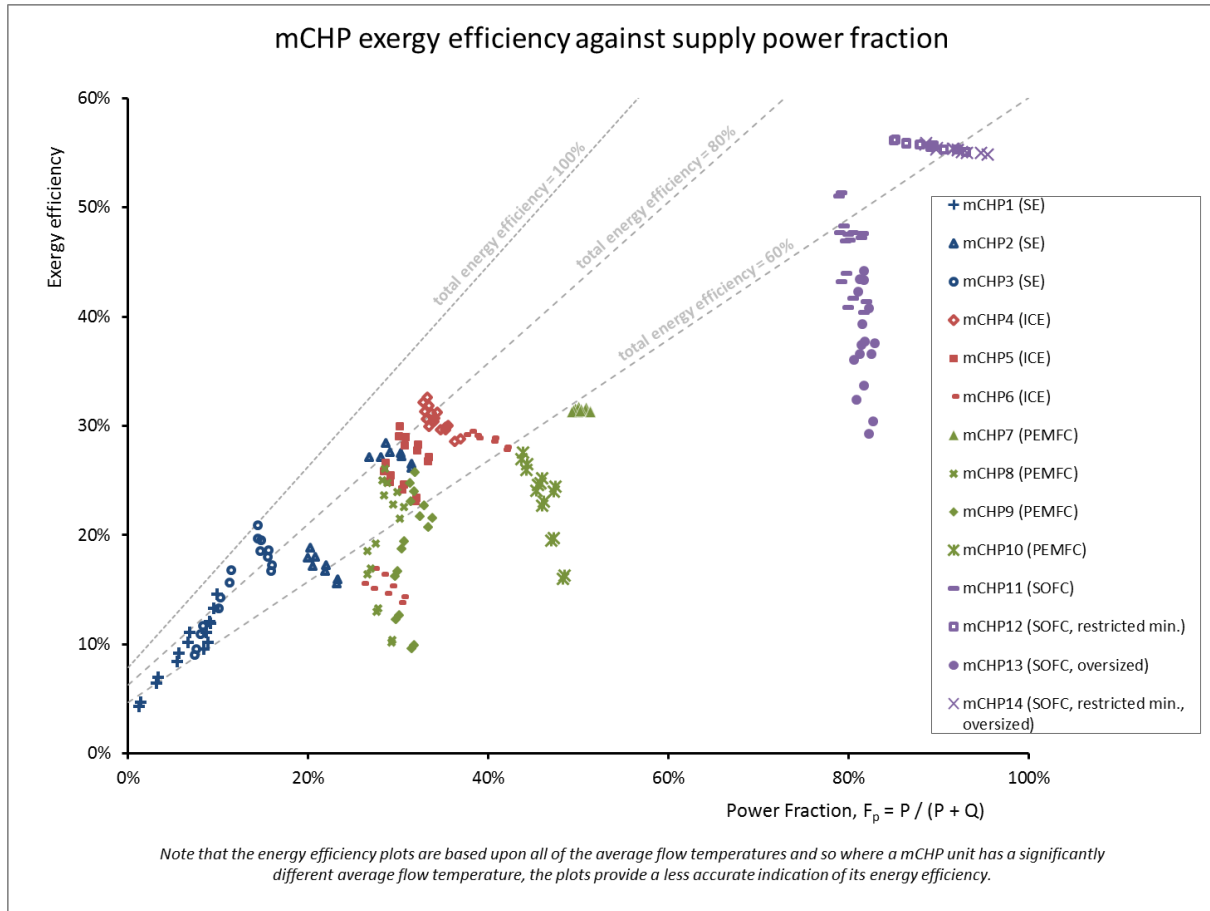


Figure 4: Variation in mCHP exergy efficiency with power fraction.

It can be seen that although the total energy efficiency of most of the mCHP units being considered is high, the relatively low exergy value (c. 8%) of heat delivered at this temperature means that the exergy efficiency of units is low unless their power-heat output ratio is high. The proximity of the 100% energy efficiency plot at lower power fractions constrains any significant improvements in exergy efficiency such that it must involve an increased electrical efficiency. Despite achieving a lower total energy efficiency, the exergy efficiency of the SOFC-mCHP units is therefore much higher.

4.2 ASHP unit performance

The annual average COP of each ASHP under the various conditions is plotted in Figure 5.

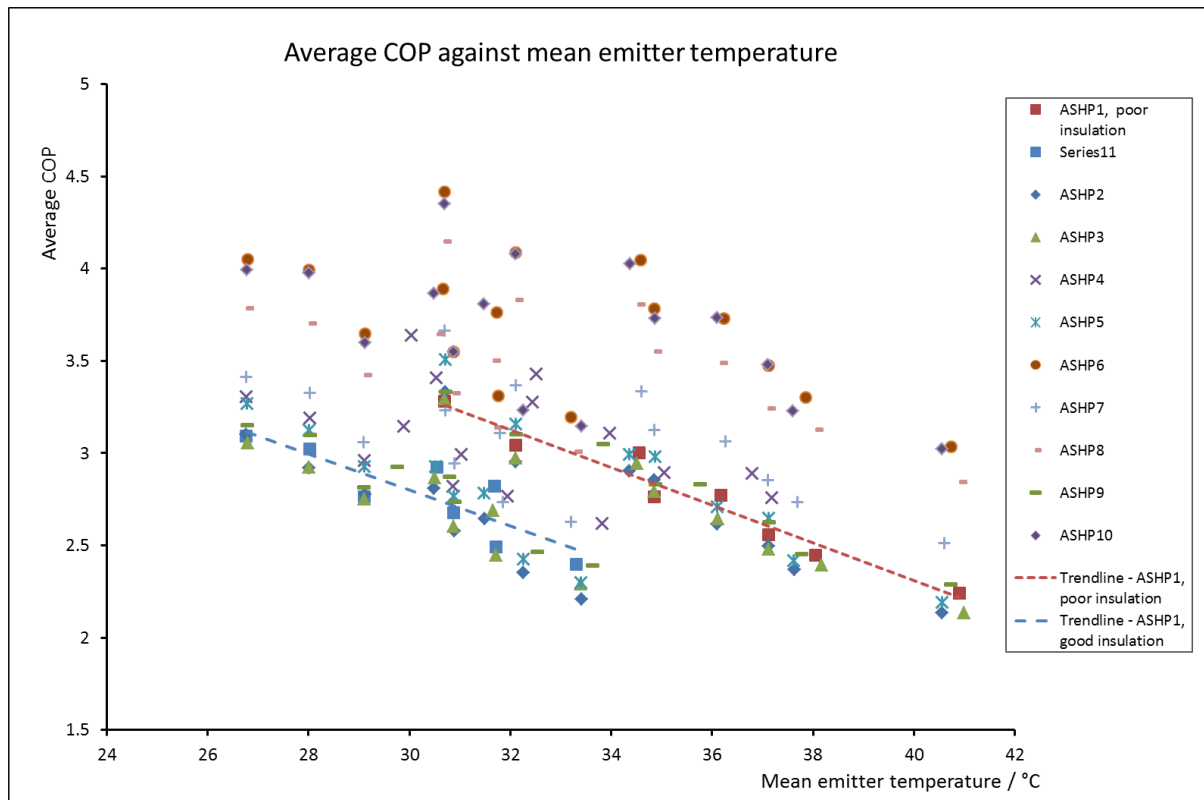


Figure 5: Variation in ASHP annual average COPs with average (heat-demand-weighted) emitter temperatures.

There are clear groupings of results with each of the 16 permutations of building construction, emitter size, climate and control system resulting in a different mean emitter temperature. Within each of these permutations there is a range of about 35% between the performance of the most efficient ASHP (usually “ASHP6”) and the least efficient ASHP (usually “ASHP2”). As expected, for each ASHP unit, higher mean emitter temperatures are associated with a lower average COP. However, there are separate trends for the two types of building (illustrated for “ASHP1”); the average COP in the well insulated building is typically similar to that in the less-well insulated building despite a lower average flow temperature. This is probably due to the better insulated house exhibiting a shorter heating season; far less heat is required but it is required when the outside temperature is colder. Notably, using a control methodology with constant flow temperature can cause an increase in power consumption of more than 30%, a similar effect to that observed by Kelly & Cockroft [47]

who calculated a comparable average COP (c. 2.7) using a detailed model without “weather-compensation” control.

The exergy delivered as heat to the heat emitters from each ASHP is plotted against the electrical power (exergy) consumed by each ASHP in Figure 6. In contrast to the energy efficiency, the exergy efficiency of each ASHP (i.e. the gradient of a trend line for that unit) does not vary greatly with ambient air temperature (typical variation of +/- 3% across working range). For example, the exergy efficiency of “ASHP1” averages about 27% for the conditions modelled.

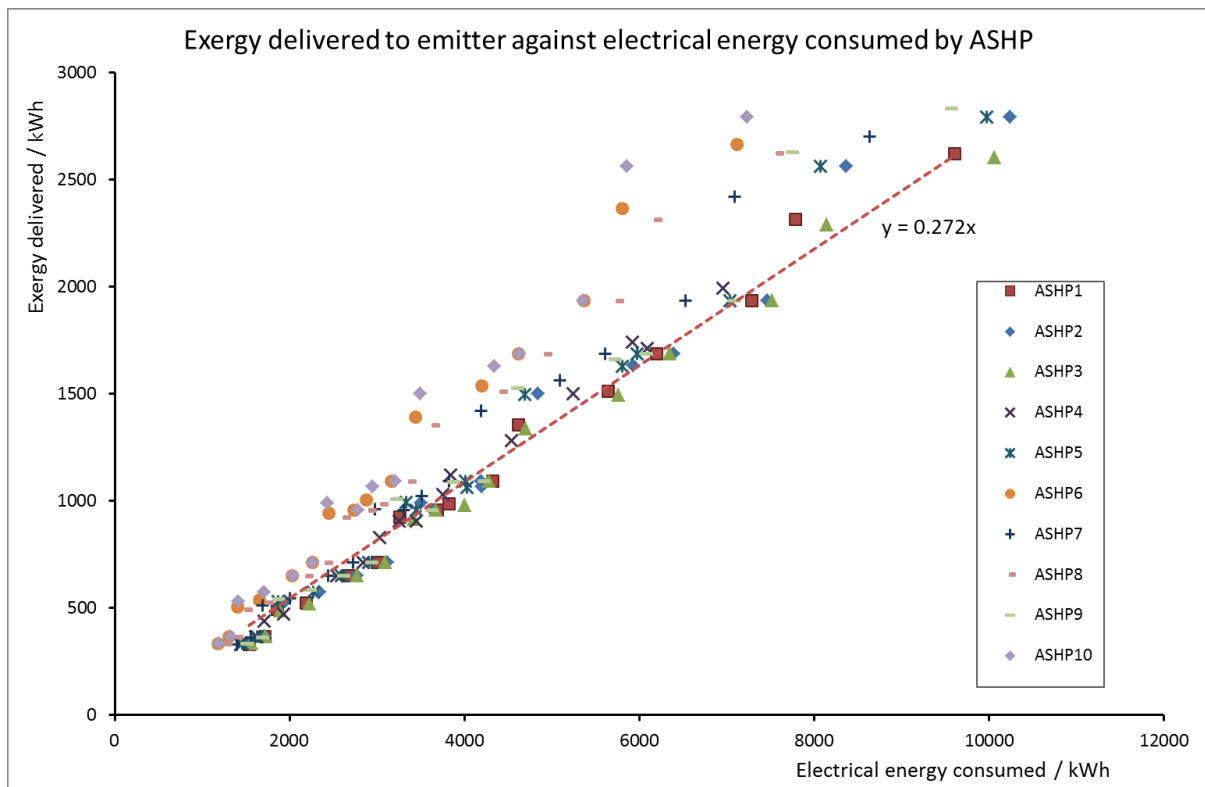


Figure 6: Exergy delivered against power consumed for ASHPs under varying conditions

4.3 Overall system

The variation of the energy and exergy efficiencies of the whole energy system (i.e. primary energy inputs through to final energy use) with the power fraction can be considered by reference to Figure 7 (energy efficiencies plotted for $T_D = 20^\circ\text{C}$). The power fraction (x-axis) referred to is the fraction of electrical *demand* (appliances and lighting, Z), compared to total energy demand in the dwelling except in the case of the SOFC-mCHP units which generate more electricity than is demanded. In these cases, it refers to the ratio of electricity *generated* (P) compared to electricity generated and heat used (equation 4.1).

The data associated with the mCHP units is similar to that in figure 4 but includes losses in the distribution of natural gas and the exergy loss as heat is transferred from the heat emitter system to the air in the room (at lower temperature). The SOFC-mCHP unit achieves the highest exergy efficiency but is limited to high power fractions. The dashed line shows the effect on the SOFC-mCHP whole system exergy efficiency of adding an auxiliary boiler to reduce the power fraction; it is likely to remain higher than the alternatives until the power fraction is below about 65%.

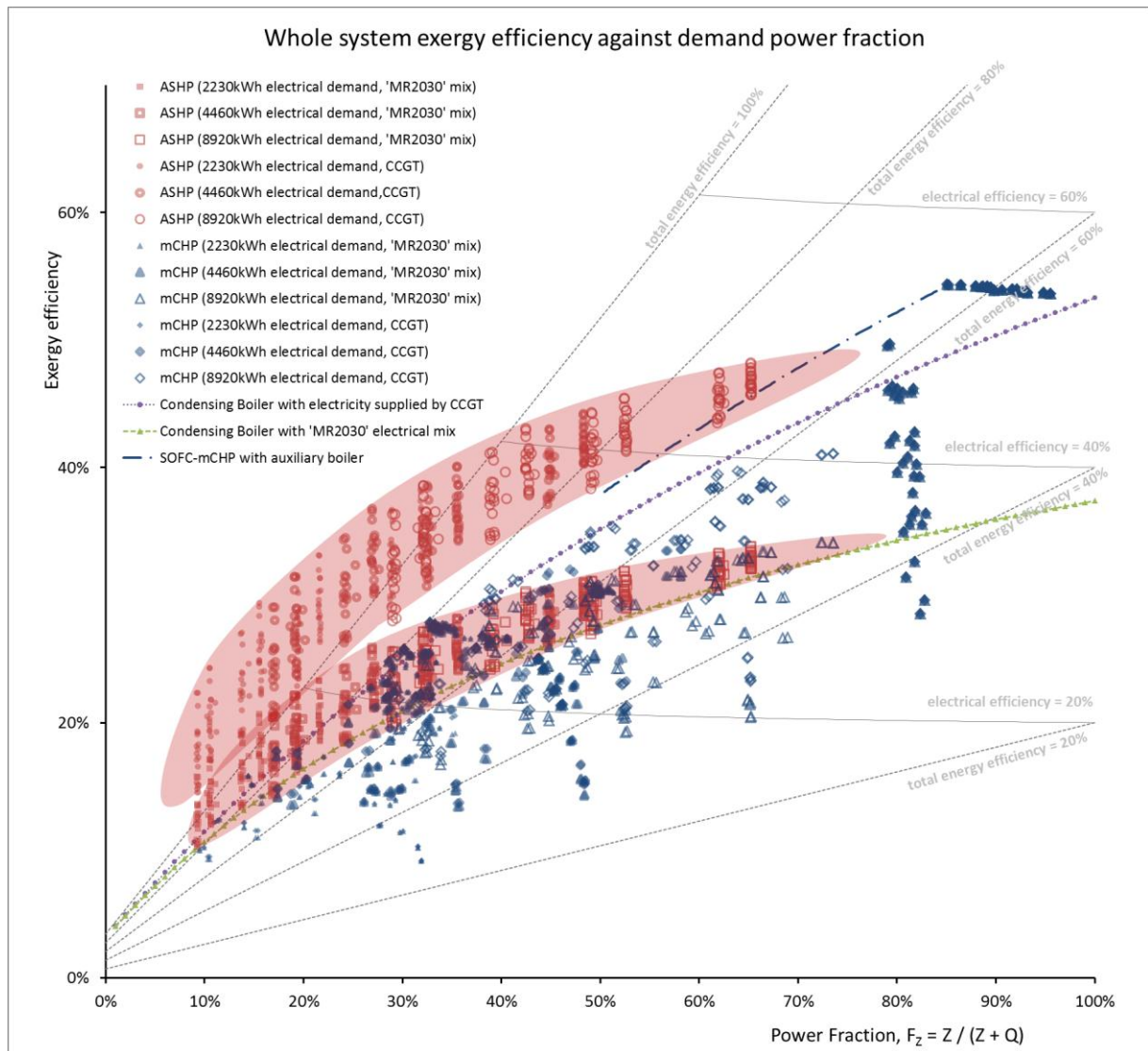


Figure 7: Exergy and energy efficiency of ASHP and mCHP based whole systems against power fraction

The exergy efficiency of the ASHP systems is highly dependent on the efficiency of grid electrical generation. This is shown clearly on the graph by the two distinct groupings of ASHP system performances relating to the two electrical supply scenarios considered. For comparison, plots are provided for a 90% efficient condensing

gas boiler with each of electrical supply alternatives; if electricity is supplied by the “MR2030” mix, the system performances are generally similar but with electricity supplied by the CCGT, the ASHP systems show higher efficiency in almost all cases.

In the cases with power fractions between about 35% and 60% (i.e. heat to power ratios of 2:1 to 0.7:1), both types of unit offer comparable performance (for the “MR2030” mix), with operating conditions and constraints determining the highest exergy efficiency available. At the higher end of this range of power fractions, the PEMFC-mCHP units from Tokyo Gas field trial would represent the highest performance but they are not plotted on this graph as discussed earlier.

At lower power fractions (i.e. heat to power ratios above 2:1), most of the ASHP systems have the potential to achieve greater than 100% energy efficiency and it is impossible for a conventional mCHP system to exceed this. For the higher performing systems with electricity supplied by CCGT, 100% primary energy efficiency can occur with a power fraction as high as 40% (i.e. heat to power ratio of 1.5:1). Conversely, if unrestrained electrical export is allowed, the high electrical efficiency achieved by using the SOFC-mCHP unit means that energy is saved even if the thermal energy is unused. Under these conditions, the assumed reduction in electrical distribution losses achieved by the SOFC-mCHP unit is more significant than the heat recovered or the electricity used by the more efficient ASHPs. The SOFC-mCHP system efficiency would therefore be higher than that of an ASHP system regardless of the average COP achieved. For high performing systems, the overall energy performance is largely dictated by the relative grid electrical efficiency and restrictions on power export, not on the provision of heat.

The largest exergy losses are due to inefficient generation of electricity and to the low exergy value of the heat flow. Because of the low exergy content of the thermal energy, relatively small changes in the exergy efficiency of the systems have the potential for a larger impact on their energy performance; this sensitivity partly explains the wide range of system performances reported in the literature. Although outside the scope of this analysis, it should also be noted that savings in the electrical demand of appliances will have a proportionally larger effect on the exergy and energy required earlier in the system than savings at other points. This analysis considers the energetic and exergetic performance of the units; for a holistic assessment of their relative merits, consideration should also be given to other factors such as their whole-life environmental impacts and the economic aspects of their adoption.

Technical constraints such as local grid infrastructure may limit power export. It is feasible that a clustering approach (i.e. several ASHPs with some fuel cell mCHP units) could be employed to maximise overall efficiency in situations involving multiple low power to heat ratio demands. It has been suggested that this would also minimise daily variation in grid power demand [48] and it is likely that this effect would also be seasonal.

5 CONCLUDING REMARKS

Energy and exergy analysis has been conducted for a range of mCHP and ASHP devices, modelled as discrete units but within the context of the whole energy system. The current mid-range ASHP and SE-mCHP units have broadly comparable performance to a condensing boiler with grid supplied electricity for normal power to heat demand ratios. However, appropriately installed, state-of-art ASHP units and SOFC-mCHP units have the potential to achieve 40 – 50% primary energy savings.

Efficient SOFC-mCHP units can achieve higher efficiency than new CCGT units (once grid losses are accounted for). Although energy savings are clearly possible, economic and environmental considerations must be fully taken into account to provide a holistic appraisal [49].

The relative performance and potential of the systems is dominated by the electrical characteristics: the grid electrical generation efficiency, the power-to-heat demand ratio and the availability of electrical export. For total power-to-heat demands below 1:1.5, ASHPs have greater improvement potential as their energy efficiency is not constrained. At higher power-to-heat ratios, mCHP units do offer the potential for higher overall efficiency and this generally occurs irrespective of whether or not the thermal energy is used effectively. In practice, it is likely that a combination of the two systems would provide the best performance.

6 ACKNOWLEDGEMENTS

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7 ABBREVIATIONS AND NOMENCLATURE

ASHP Air Source Heat Pump

COP Coefficient of Performance

ICE Internal Combustion Engine

mCHP Micro Combined Heat and Power

PEMFC Proton Exchange Membrane Fuel Cell

SE Stirling Engine

SOFC Solid Oxide Fuel Cell

A Total non-renewable primary energy input

C Coefficient of Performance

E Exergy

F_p Fraction of energy at end use delivered as electrical power

H Enthalpy of combustion (higher heating value)

P Power output from mCHP unit

Q Heat flow

T_D Room temperature

T_E Heat emitter temperature

T_0 Ambient temperature

W Work input to heat pump

Z End use electrical demand

η	Energy efficiency
ϕ	Ratio of chemical exergy to enthalpy of combustion
ψ	Rational (Exergy) efficiency

8 FIGURES

Figure 1: Representative energy flows

Figure 2: Dwelling heat flows simplified model

Figure 3: Variation in mCHP electrical efficiency with thermal efficiency.

Figure 4: Variation in mCHP exergy efficiency with power fraction.

Figure 5: Variation in ASHP annual average COPs with average (heat-demand-weighted) emitter temperatures.

Figure 6: Exergy delivered against power consumed for ASHPs under varying conditions

Figure 7: Exergy and energy efficiency of ASHP and mCHP based whole systems against power fraction

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