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# THE INFLUENCE OF THERMAL STORAGE ON MICROGENERATION FLEXIBILITY

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## ABSTRACT

In a future power system, the ability to manipulate generation and load will be a critical factor in providing a secure and stable supply of electrical energy to consumers.

Using a simulation-based approach, this study assesses the ability of thermal storage to help deliver flexibility in the operation of domestic micro-generation technologies without sacrificing householder comfort and convenience. A typical UK detached dwelling is modelled along with its heating system, which features a retro-fitted air source heat pump (ASHP). The model is used to determine the maximum possible temporal shift for different capacities and configurations of thermal storage, taking into account the influence of climate, building fabric, control settings and occupancy. The limits of time shifting are dictated by the living space temperature and the hot water temperature delivered to the occupants. The storage mechanisms examined are: the basic thermal inertia of the building fabric; increasing the space heating set point temperatures to increase fabric storage and inserting a dedicated thermal buffer between the ASHP and the heat distribution system.

The simulation results indicate that back-shifting of the ASHP start/stop times of between one and two hours are possible without causing serious discomfort or inconvenience to the occupants.

## INTRODUCTION

In order to maintain a reliable supply of electrical power to consumers in a future, more highly distributed power system, a key requirement will be the ability to control and co-ordinate the operation of local sources of low-carbon electrical generation and electrical demands. This ability to manipulate supply and demand is required for the following reasons:

Firstly, at the local level, current policy is to encourage the proliferation of large numbers of small-scale renewable generators such as photovoltaics (PV), micro-scale wind turbines (SWECS), solar thermal collectors, etc., by offering financial inducement through the feed-in-tariff (FIT) and renewable heat incentive (RHI) (DECC, 2009). To ensure that the local supply of electricity is secure and of high quality, there must be sufficient flexibility in both supply (through controllable low-carbon generation such as micro-combined-heat-and-power [μ-CHP]) and demand to accommodate the inevitable fluctuations from local non-despatchable sources.

Secondly, at a national level, it is likely in the coming decades that electricity distribution networks will need to be actively managed to accommodate significant variations in electrical generation from a grid featuring significant penetrations of large scale-renewables, e.g. offshore wind and other inflexible low carbon plant. Such a scenario is likely if the UK's current target for an 80% reduction in greenhouse gas emissions by 2050 is to be realised (OPSI, 2008).

This paper explores making use of the thermal capacity of buildings and their systems to facilitate flexibility (i.e. time shifting) in the operation of microgeneration and thus enable greater co-ordination and control over both supply and demand. The case studied in detail in this paper is that of an air source heat pump (ASHP) supplying hot water and space heating to a typical detached dwelling. A heat pump, whilst being a heat source, is a significant electrical demand. Hence, the exercise here is one of demand shifting. Note, that the approach taken can equally be applied to μ-CHP and so can also be viewed as a means to analyse time-shifting of local despatchable electrical supplies.

Specifically, the consequences of back-shifting the start and stop times of the heat pump are analysed against prevailing space temperatures, hot water supply temperatures and heat pump energy consumption. This is done with a view to determining the 'limits' of flexibility, where those limits are dictated by the increasing thermal discomfort and loss of utility (reduced hot water supply temperatures) engendered by back-shifting the period of heat pump operation.

Four different situations are examined:

1. The heat pump start time is altered in the dwelling 'as-is', making use of the intrinsic thermal capacity of the building fabric and the heating system.
2. The heating set point is increased in order to boost the quantity of heat stored in the fabric thus increasing the potential to time shift the heat pump at the expense of increased energy consumption.
3. Thermal capacity is added to the heating system in the form of a 300L buffer tank placed between the heat pump and distribution system.
4. The buffer tank capacity is increased to 500L

## METHOD

In order to explore the relationship between the heat pump operation and occupant comfort and utility, an 'integrated' simulation approach was adopted, where the performance of the heat pump was simulated along with the environment (i.e. the building) within which it operates. To achieve this end, a dynamic simulation model was developed on the ESP-r building simulation tool (Clarke, 2001). This comprised a detailed representation of the building geometry and fabric materials; a representation of the dwelling's air leakage characteristics plus a detailed, component-based representation of the heating system and its controls. The model also incorporated details of the temporal heat gains from the occupants and equipment and hot water draws by householders.

Within ESP-r, the building model is decomposed into a large number of small 'control volumes' for which energy and mass balance equations are derived. The solution of these equations at discrete time steps with real, time-varying climate data and user-defined control constraints (e.g. set point temperature and heating operating times) yields the dynamic evolution of temperature, energy and fluid flows within the building and plant system. The values of these quantities are calculated at a user-defined frequency over the simulated period. In this case, the state of the building and plant system is simulated using 1-minute time steps over the course of a characteristic winter week.

## MODEL

The integrated model used for this study comprised three main elements; these are:

- the building and fabric;
- the heating system; and
- system control.

### Building and Fabric

The ESP-r building model used in this paper is a thermodynamic representation of a typical UK detached dwelling (Figure 1) the characteristics of which are described in detail elsewhere (Kelly and Beyer, 2008). This particular example was chosen in that it is stereotypical of those dwellings into which ASHP devices could be retrofitted. Further, retrofitting of low-carbon technologies coupled with fabric improvements in existing dwellings (which in the future will still comprise the majority of the housing stock) will be key mechanisms for reducing the carbon intensity of UK housing and the built environment in general.

*Figure 1: a typical detached dwelling*

The model of the building is divided into three main functional zones: living, non-living and loft area. The living and non-living zones differentiate the areas where active and inactive (i.e. sleeping) occupancy occurs. This form of model, whist not an architecturally faithful representation of a real building is capable of replicating its thermodynamic behaviour for the purposes of an energy analysis.

The model's characteristics have been adapted so that they are representative of a dwelling that has had its fabric upgraded in addition to the retrofit of an ASHP (a fabric upgrade is a requirement of qualifying for the UK's renewable heat incentive [RHI, 2011]). The model in this paper has a floor area of $136m^2$, external cavity walls with a U-value of $0.45W/m^2·K$ and double glazed windows $3.304W/m^2·K$. The average outside air infiltration rate is 0.5 air changes per hour. The building is a lightweight construction, with suspended timber floors and ceilings, the internal walls are finished in plasterboard and most floors are carpeted.

The model is set up to include intermittent heat gains from a family of four. The occupants are active between 6:00-9:00hrs and 17:00-23:00hrs and sleeping between 23:00hrs and 6:00hrs. Outside these periods the house is empty.

## Heating System

A detailed sub-system model of the heating system has been developed and integrated into the building model. The variants with and without buffering are shown in figures 2a and 2b respectively.

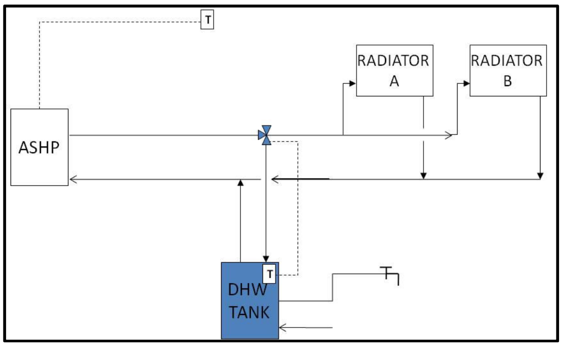

*Figure 2a: the unbuffered heating system configuration*

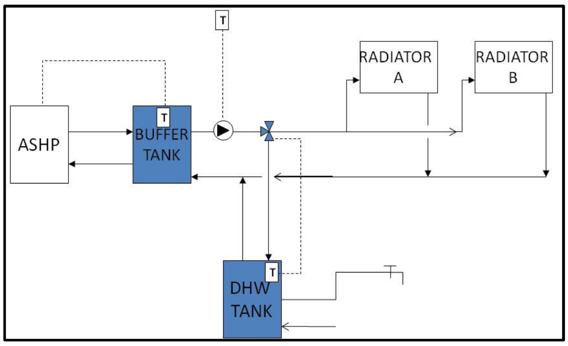

*Figure 2b: the buffered heating system configuration*

This heating system model comprises the ASHP device model, buffer tank (when appropriate) and models representing the balance of plant: radiators, diverting valves, a hot water tank, etc. The hot water tank is subject to time varying hot water draws (totalling 120L/day) consistent with the activities of a typical family of four and with an intermittent occupancy pattern (Knight and Ribberink, 2007).

The buffered heating topology is similar to that deployed in domestic microgeneration field tests undertaken by the Canadian Centre for Housing Technology (CCHT) (Entchev *et al*. 2006). The unbuffered configuration is similar to that employed in UK Carbon Trust's trials of microgeneration (Carbon Trust, 2007).

The ASHP model was calibrated using an experimental data set. The variation of the COP for the heat pump model and the actual device with the heating system return/external air temperature difference are shown in Figure 3. More details regarding the particular model of the ASHP employed in these simulations and its calibration/verification are given by Kelly and Cockroft, (2011).

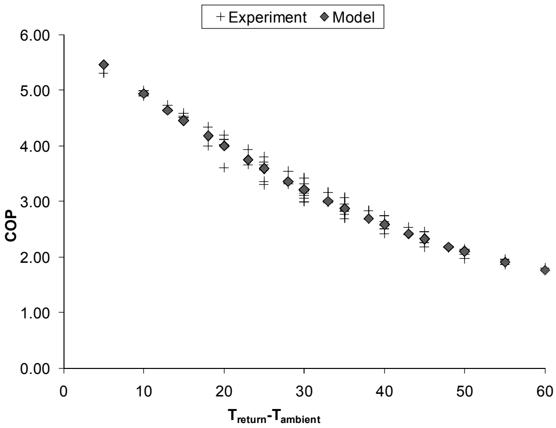

*Figure 3: the variation in the ASHP model's predictions of COP with return water – ambient air temperature difference* (Kelly and Cockroft, 2011)

## System Control

The heating system model incorporates various controllers that dictate its operation. The heat pump is subject to on/off control with a 4°C dead band which attempts to maintain the air temperature in the living room between and 19-23°C; for a second group of simulations, the dead band set points were altered to 21-25°C. There is also a hot water precedence control within the heating system model: if the hot water tank temperature is below its set point of 45°C, heat is diverted to it in preference to the radiators (this is a common configuration for heating systems in the UK). The heat pump also features an internal safety control that switches the device off if return water temperatures breach a user-defined set point of 55°C: in a real unit, this prevents component failures due to excessively high pressures in the heat pump's refrigerant loop.

In the case of the buffered heat pump system, the operation of the heat pump is controlled based on the buffer tank temperature, which is maintained between 45-50°C by an on/off controller with a 5°C dead band. A circulating pump draws heat from the buffer tank to maintain the living space air temperature between 19-23°C (on/off control with a 4°C dead band). The controller settings are drawn from practical experience from field trials (Kelly and Cockroft, 2011)

The active period of the controllers mirrors the house occupancy, with the heat pump initially scheduled to turn on 1-hour prior to active occupancy and turn off at the end of occupied periods.

## SIMULATIONS

With reference to the four different thermal mass cases described in the introduction, the following groups of simulations were run.

1. The basic heating system (Figure 2a) was used in an initial base case simulation with the controllers operating as described previously. In five subsequent simulations within this group, the heat pump start and stop times were back-shifted in half-hourly increments.
2. As case 1. However, the heating on/off set point temperatures were altered to 21-25$^o$C.
3. As case 1. However, a 300L buffer tank was introduced (in addition to the hot water storage tank – see Figure 2b) and maintained between 45-50$^o$C. Heating set point temperatures are the same as in case 1.
4. Case 3 repeated with a 500L buffer tank.

In each group, six simulations were run in total, with the start-stop times for the ASHP back-shifted incrementally by up to 3-hours.

All simulations used a temperate climate data set, characteristic of the West of Scotland. The simulations were conducted for a characteristic winter week, 9-15 January using a simulation time step of 1-minute. Such a fine time resolution (for thermal systems simulations) was required to capture those phenomena that affect the ASHP energy and environmental performance such as on/off cycling and the defrost action.

## RESULTS

The large and detailed datasets emerging from each simulation were processed to extract the key quantities pertinent to this analysis, namely internal temperatures in the building, hot water tank temperatures and heat pump energy consumption. These are shown in the tables (2a-2d) at the end of the paper. Figure 4 shows an example of the type of raw data obtained from the simulations showing the variation in living space temperatures in response to heat input from the heating system along with the ASHP power consumption.

Figure 5 shows the typical temporal variation in living space temperatures over the course of a simulated morning. The temperature plots include the effects of excitation from the heating system, internal gains, infiltration of outside air and solar radiation in addition to heat exchanges with the building fabric. The diagram combines several simulation runs and clearly illustrates the effect of back-shifting the heat pump start and stop times.

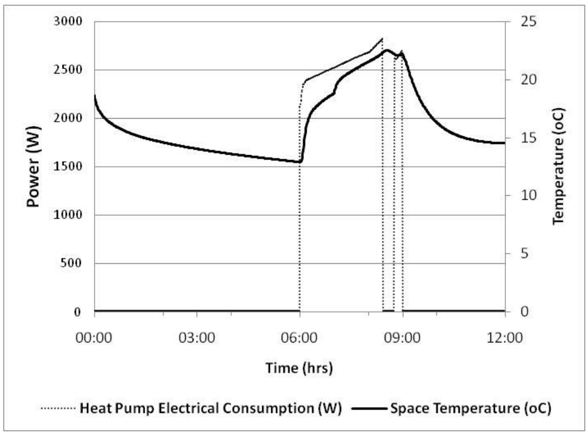

*Figure 4: typical simulation time series data (heat pump power consumption and the response of the living space temperature to the resulting heat input).*

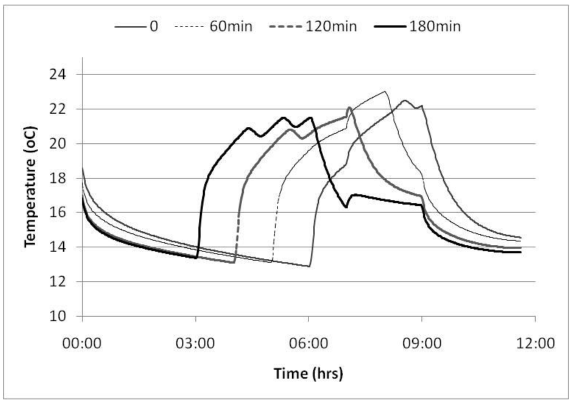

*Figure 5: the effect of back-shifting heat pump operation time on living space temperatures*

In the analysis that follows, the two metrics used to gauge comfort and utility (i.e. the ability of the system to supply hot water) are a living space temperature of 18$^o$C and a hot water temperature of 40$^o$C, both of which are analysed during periods of *active* occupancy. The space temperature reported here is the *operative* temperature – the average of the living space air and mean radiant temperature, though in practice for a well-insulated lightweight construction there is usually little difference between these two values.

An operative temperature threshold of 18$^o$C can be regarded as towards the lower end of acceptable thermal comfort as defined by Fanger[1] (Fanger, 1970), whist water supplied

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[1] Note that a temperature of this magnitude does not guarantee comfort; this is dependent upon many other factors including clothing and activity, hence this is a rather approximate metric. Whilst acceptable

below $40^{o}$C[2] would begin to feel lukewarm to the occupant.

*Case 1- as-is*

Table 2a shows the effect on internal operative temperature, hot water supply temperature and heat pump energy consumption as the intermittent start/stop times are back-shifted in 30-minute increments. If the start/stop is back-shifted by a full hour, there is a negligible impact upon environmental conditions and system utility, with temperatures being below $18^{o}$C for only 1.8% of the actively occupied hours and hot water being supplied at a temperature of over $40^{o}$C for the whole simulation period. The mean living space is $21^{o}$C. However, shifting the start/stop time back by a further 30-minutes results in a dramatic deterioration in environmental conditions, with living space temperatures below $18^{o}$C for over 12% of occupied hours. Hot water is still supplied above $40^{o}$C for all actively occupied hours.

*Case 2- increased set point*

Table 2b shows the same performance metrics but with the space heating set point temperatures raised by approximately $2^{o}$C in order to increase the quantity of heat stored within the building fabric. In this case a 1-hour back-shift in start/stop time results in temperatures being below $18^{o}$C for only 0.3% of active occupied hours, with a mean living space temperature of $22.4^{o}$C. An additional 30 minutes shift resulted in temperatures below $18^{o}$C for just over 5% of the active occupied periods and a mean space temperature of $21.8^{o}$C. In both cases the system consistently supplied hot water above $40^{o}$C.

A 1.5-hour back-shift results in a dramatic deterioration in environmental conditions with the number of hours that the living space is below $18^{o}$C rising to almost 14% of the active occupied hours.

*Case 3 –additional thermal buffering (300L)*

The results for the case where the heat pump is buffered from the rest of the heating system by a 300L tank (see Figure 1b) are shown in Table 2c. In this case, a back-shift of up to 1.5 hours has no effect on comfort conditions, with living space temperatures above $18^{o}$C and hot water supply temperatures above $40^{o}$C for all active occupied hours Mean living space temperatures are $21.3^{o}$C. A back-shift of more than 1.5-hours results in a significant deterioration in environmental conditions, with living space temperatures being below $18^{o}$C for over 10% of active occupied hours and hot water temperatures being below $40^{o}$C for slightly over 4% of active occupied hours.

*Case 4 –additional thermal buffering (500L)*

The results from the simulations where the buffer tank capacity was increased to 500L are shown in table 2d. In this case, a back-shift of 2-hours is possible without significant effect on either environmental conditions or hot water temperatures: with this amount of back-shift space temperatures are below $18^{o}$C for only 0.3% of the time. Mean living space temperatures are $21.2^{o}$C. Hot water temperatures fall below $40^{o}$C for 4.4% of active occupied hours.

## DISCUSSION

For all four sets of simulations, including the case of the building ‘as-is’, it appears that the effect of back-shifting the heat pump start/stop times by up to 1-hour has little or no effect on the comfort and utility of the building occupants during active occupied hours.

Making use of additional thermal capacity, by increasing the set-point temperature or increasing the size of the buffer tank has a limited effect (in this case) on the amount of back-shift that can be achieved without a deterioration in space and hot water temperatures.

Adding a 300L buffer tank or raising the heating set point by $2^{o}$C enables only another 30-minutes back-shift before environmental conditions deteriorate and loss of utility occurs.

Additionally, increasing the heating on/off set points from 19-23 to 21-$25^{o}$C resulted in an energy use penalty of approximately 10%. Increasing the buffer tank capacity to 500L permitted a back shift of up to 2-hours.

It should also be noted however, that a heavily insulated 500L buffer tank (in addition to a 300L water tank) would require a significant amount of space in a dwelling – a typical 500L tank is 1.9m high with a diameter of 0.7m (Ariston, 2010).

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within the context of this paper, a more serious comfort analysis would require a more sophisticated approach. Also note that Fanger’s concept of static comfort (Fanger, 1970) criteria has been challenged by the more recent concept of adaptive comfort (e.g. deDear and Brager, 1998).

[2] Note that in many boiler-based hot water systems, hot water is stored at over $60^{o}$C for among other reasons to prevent growth of Legionella. However, this is an inefficient practice as the Legionella threat can be removed by occasionally raising water storage tank temperatures above $60^{o}$C. Additionally, hot water at $60^{o}$C must be mixed with cold water prior to use to prevent the risk of scalding. Hence, a $40^{o}$C supply temperature is both safer and more energy efficient.

For both case 1 and case 2 there is a slight energy penalty associated with back-shifting the start/stop times as shown in Table 1 above (for case 2 this is in addition to the energy penalty resulting from increasing the internal air temperature set point).

For all four cases, the maximum back-shifting time is marked by a very rapid deterioration in environmental conditions and utility thereafter.

*Table 1: Back shift time before significant discomfort and change in energy use.*

| | Back-shift time (hrs) | Energy penalty % |
|---|---|---|
| Case 1 | 1.0 | 5.6 |
| Case 2 | 1.5 | 5.2 |
| Case 3 | 1.5 | -3.8 |
| Case 4 | 2.0 | -7.0 |

For the cases with a 300 and 500L thermal buffer (in addition to the hot water tank), the ASHP energy consumption falls slightly without a marked deterioration in performance for 1.5 and 2-hour back shifts, respectively. Thereafter energy consumption continues to fall, but environmental conditions and utility deteriorate rapidly.

It is interesting to note than in all four cases the energy consumption associated with back-shifting the start-stop times initially increases until about 1.0-1.5 hours back-shift. After this point, energy consumption falls again. The reason for this phenomenon is that with more than 1.5 hours back-shift there is very little overlap between the heat pump controller ‘on’ period and the time at which space heating and energy demands occur (particularly in the morning). The ASHP device controller therefore cannot react to changes in space and hot water temperatures and so activate the unit when these fall below the set point. Hence, accompanying the fall in energy demand after 1.5 hours back-shift is a dramatic rise in the time that living space temperatures are below 18$^{o}$C and water temperatures below 40$^{o}$C; effectively this level of time-shifting results in a lower energy consumption *but* the heating system does not adequately meet the needs of the occupants.

From the perspective of the operation of electricity distribution networks, the ability to shift the heat pump operating time only within a time window of 60-to 120-minutes (for a lightweight dwelling) is, at first sight, not particularly inspiring. From a radical demand shifting perspective it would not, (for example) permit sufficient flexibility to flatten the characteristic peaks for electricity seen in housing during the morning and evening periods. It would however, permit a local network operator to stagger the operation of the heat pumps within their area without any discernible affect on the occupants. This would be a useful facility to prevent concurrent ASHP morning or evening start-ups, perhaps negating the need for investment in local low voltage network reinforcement in areas with significant penetration of heat pumps.

To facilitate significant shifting of heat pump (or μ-CHP) operation beyond 2 hours in this case would require significant extra buffering: either in the fabric of the building or with a larger buffer tank. Both of these options are problematic. Concerning increasing the fabric storage, it has already been shown that one means of achieving this, increasing internal temperatures, produces a limited benefit at a significant energy penalty. Additionally, raising temperatures further would lead to thermal discomfort from over-heating. An alternative is to increase the dwelling’s effective thermal mass by (for example) installing an underfloor heating system. Whilst this is achievable in a new build dwelling, it would be difficult and costly to implement by retrofitting to an existing dwelling. Increasing the size of the buffer tank would also be problematic as the majority of UK urban houses are space-constrained with limited room to install large hot water tanks. Further, many houses in the UK have installed combination boilers and so have had any hot water storage space that was available converted to other uses (e.g. shower rooms). One possible solution to this problem could be to utilise phase change heat storage rather than sensible heat storage in water; this could reduce buffering space requirements. However, domestic phase change heat storage products are not widely available. Hence, the scope for significant extra thermal buffering is limited at present.

Given the problems with making further use of thermal mass, an alternative approach to augment the time-shifting potential of microgeneration would be to further improve the fabric of the building in order to reduce the decay rate of internal temperatures after the heat pump is switched off. The efficacy of this approach will be explored in future studies.

## CONCLUSIONS

In this paper, the use of thermal storage to facilitate the time-shifting of microgeneration operation has been examined using a detailed simulation-based approach.

The specific case examined is that of a typical (thermally lightweight) detached UK dwelling with slightly above-average levels of insulation; this is representative of the large number of such dwellings that could be retrofitted with an ASHP (installed along with a fabric upgrade).

Without any additional thermal buffering, the operation of the heat pump could be set-back by up to 1-hour without any significant effect on the occupants.

Increasing the space heating set-point temperature by approximately 2°C enabled a back-shift of up to 1.5 hours. However, this resulted in the ASHP energy consumption increasing by approximately 10%.

With 300L of thermal buffering between the ASHP and the heat distribution system (in addition to the hot water tank) the maximum back-shift was 1.5 hours. Energy consumption was slightly reduced and this case back-shifting of start/stop times caused minimum deterioration in environmental conditions and hot water supply temperatures to the occupant.

Increasing the thermal buffering to 500L increased the possible back-shift to 2-hours. Again this resulted in slightly reduced heat pump energy consumption. It was also noted that a fitting a 500L buffer tank *and* a 300L water tank into a space constrained dwelling may be problematic.

From the perspective of the interaction of the heat pump devices and the local LV network, the magnitude of time-shifting shown for the case of the lightweight building analysed in this paper is not sufficient to facilitate radical demand-side manipulation such as flattening of the demand profile for a population of heat pumps. However, the magnitude of time shifts shown in this paper are sufficient to facilitate staggering of start times to prevent concurrent device start-ups.

## FURTHER WORK

A follow-on to this paper will feature expanded results from an additional 3 dwelling types featuring different levels of insulation, thermal buffering and internal thermal mass. Buildings with a higher thermal mass (typical of may older UK dwellings) are of particular interest as these could offer greater scope for greater temporal shifts on demand..

## ACKNOWLEDGEMENTS

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## REFERENCES

Ariston (2010), Unvented Cylinders 100 to 500 litres. Product Brochure. Available at: http://www.ariston.co.uk/products/unvented-cylinder/direct/high-capacity/documents/0000000029/

Clarke J A, Energy Simulation in Building Design, 2nd Ed., Butterworth-Heinemann, 2001.

Carbon Trust, Micro-CHP Accelerator Interim Report (2007), Carbon Trust Report CTC726. Available at: http://www.carbontrust.co.uk/publications/publicationdetail.htm?productid=CTC726

DECC, Department for Energy and Climate Change (2009), The UK's National Strategy for Climate Change and Energy, HMSO, London.

deDear R. and Brager G. (1998), Developing and adaptive model of thermal comfort and preference. ASHRAE Transactions 104 1 pp. 145–167.

Entchev E, Gusdorf J, Swinton M, Bell M, Szadkowski F, Kalbfleisch W and Marchand R (2004), Micro-generation technology assessment for housing technology, Energy and Buildings, 36(9), September, pp925-931.

Fanger P. O., (1970) *Thermal Comfort*, McGraw-Hill, New York,

Kelly N J, Cockroft J (2011), 'Analysis of retrofit air source heat pumps performance: results from detailed simulations and comparison to field trial data', Energy and Buildings (43) pp 239-245.DOI: 10.1016/j.enbuild.2010.09.018 (IN PRESS).

Knight I, Ribberink H (2007), European and Canadian non-HVAC Electric and DHW Load Profiles for Use in Simulating the Performance of Residential Cogeneration Systems, IEA ECBCS Annex 42 publication, ISBN 978-0-662-46221-7.

OPSI, Office of Public Sector Information (2008). `Climate Change Act 2008', The Stationary Office, London.

RHI, UK Renewable Heat Incentive (2011), http://www.rhincentive.co.uk/. Accessed 12/01/11.

*Table 2a results from case with building ‘as-is’**

| Start/stop back-shift (hrs) | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3 |
|---|---|---|---|---|---|---|---|
| Mean Space Temperature @ living (${}^{o}$C) | 21.33 | 21.43 | 21.06 | 20.51 | 19.75 | 19.14 | 18.71 |
| ASHP Energy consumption (kWh) | 85.50 | 87.97 | 90.29 | 91.78 | 88.42 | 89.06 | 87.75 |
| Percentage of hours below 18°C (living) | 0.09% | 0.0% | 1.8% | 12.6% | 28.8% | 44.0% | 50.5% |
| Percentage of hours below 40°C (HW tank | 0.0% | 0.0% | 0.0% | 0.0% | 4.8% | 7.0% | 8.3% |

*Table 2b results from case with increased internal set point temperatures**

| Start/stop back-shift (hrs) | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3 |
|---|---|---|---|---|---|---|---|
| Mean Space Temperature @ living (${}^{o}$C) | 22.31 | 22.58 | 22.36 | 21.78 | 21.07 | 20.28 | 19.68 |
| ASHP Energy consumption (kWh) | 94.16 | 96.80 | 99.89 | 99.12 | 97.34 | 94.75 | 93.33 |
| Percentage of hours below 18°C (living) | 0.06% | 0.0% | 0.3% | 5.5% | 13.9% | 26.6% | 38.7% |
| Percentage of hours below 40°C (HW tank) | 0.0% | 0.0% | 0.0% | 0.0% | 4.7% | 7.1% | 8.3% |

*Table 2c results with 300l thermal buffer between ASHP and heating distribution system**

| Start/stop back-shift (hrs) | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3 |
|---|---|---|---|---|---|---|---|
| Mean Space Temperature @ living (${}^{o}$C) | 21.5 | 21.56 | 21.51 | 21.28 | 20.59 | 19.45 | 18.77 |
| ASHP Energy consumption (kWh) | 85.7 | 86.6 | 86.8 | 82.4 | 73.5 | 66.2 | 61.0 |
| Percentage of hours below 18${}^{o}$C (living) | 0.06% | 0.0% | 0.0% | 0.0% | 10.1% | 29.1% | 33.2% |
| Percentage of hours below 40°C (HW tank) | 0.0% | 0.0% | 0.0% | 0.0% | 4.1% | 10.3% | 12.1% |

Note: living zone setting point: 19 - 23°C– 23${}^{o}$C

*Table 2d results with 500l thermal buffer between ASHP and heating distribution system**

| Start/stop back-shift (hrs) | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3 |
|---|---|---|---|---|---|---|---|
| Mean Space Temperature @ living (${}^{o}$C) | 21.46 | 21.49 | 21.53 | 21.41 | 21.17 | 20.54 | 20.06 |
| ASHP Energy consumption (kWh) | 86.9 | 87.0 | 88.0 | 85.0 | 80.8 | 76.5 | 70.7 |
| Percentage of hours below 18°C (living) | 0.0% | 0.0% | 0.0% | 0.0% | 0.3% | 14.2% | 24.8% |
| Percentage of hours below 40°C (HW tank) | 0.0% | 0.0% | 0.0% | 0.7% | 4.4% | 7.6% | 11.2% |

Note: living zone setting point: 19 – 23${}^{o}$C

*Results above summarise conditions for *active* occupied hours 6:00-9:00, 17:00-23:00 over the simulation week.