Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 158 (2019) 3878–3885

Energy Procedia

www.elsevier.com/locate/procedia

10th International Conference on Applied Energy (ICAE2018), 22-25 August 2018, Hong Kong, China

High Temperature Air Source Heat Pump Coupled with Thermal Energy Storage: Comparative Performances and Retrofit Analysis

Khoa Xuan Le[a,]*, Ming Jun Huang[a], Nikhilkumar Shah[a], Christopher Wilson[a], Paul Mac Artain[b], Raymond Byrne[b], Neil J. Hewitt[a]

[a]*Centre for Sustainable Technologies, University of Ulster, Northern Ireland, United Kingdom*
[b]*School of Engineering, Dundalk Institute of Technology, Dundalk, Ireland*

Abstract

This paper presents the performance analysis and retrofit assessment of a domestic high temperature air source heat pump coupled with thermal energy storage in terms of different system configurations. TRNSYS simulations were used to model the dynamic system, which has been validated against field trial results. The validated models were then simulated with different system configurations under the same boundary conditions to assess its performances. Simulated results showed that continuous coupling between the heat pump and storage to heat the house had the worst performance, while the operation of only the heat pump heating the house obtained the highest efficiency. Although the heat pump cannot compete with the gas boilers in terms of running costs, between 6.6% and 33% of carbon cut can be achieved if the heat pump was retrofitted.

Keywords: high temperature air source heat pump; thermal energy storage; TRNSYS; system performances, retrofit evaluation.

1. Introduction

Fossil-fueled boilers providing heat for space heating and domestic hot water accounted for 78% of domestic energy consumption and 40% of domestic greenhouse gas emissions in the UK [1]. With the target to reducing carbon emissions up to 80% by 2050 [2], heat pumps are a promising alternative technology to replace existing boilers at domestic level. This technology has increasingly attracted the interests of many researchers in the UK. For example, the Energy Saving Trust [3] conducted 83 field trial air-source and ground-source heat pumps with varied housing

* Corresponding author. Tel.: +44(0)2890366122.
E-mail address: le-k@ulster.ac.uk

10.1016/j.egypro.2019.01.857

types and applications across the UK; Kelly and Cockroft [4] investigated the retrofit assessment of air source heat pumps in the existing dwellings in central Scotland by means of measurement and simulations. Most of those studies consisted of standard heat pumps (the maximum outlet water temperature is 55°C) installed with under-floor heating and/or oversized/advanced radiators. However, this kind of heat pump cannot work efficiently with traditional wet radiators which are suggested the flow temperature of 75°C and the return temperature of 65°C [5]. Therefore, existing dwellings equipped with boilers and traditional radiators are unlikely to be retrofitted by standard heat pumps due to the high cost of replacing heat distribution systems. To mitigate this issue, high temperature air source heat pumps (HT-ASHP) with flow temperature up to 80°C, which is similar to the supply of boilers, can provide a potential solution to avoid the displacement cost of heat distribution systems.

There are various studies investigating HT-ASHPs; however, most of them focus on industrial applications (e.g. the extensive review of Arpagaus et al. [6]) or domestic hot water heating context reviewed in [7] rather than space heating utilization. Furthermore, regarding retrofit perspective in the UK, there is lack of in-situ and modelling results to fully evaluate this retrofit technology for the domestic built environment.

Thermal energy storage (TES) integrated with air source heat pumps has considerable advantages for demand-side management that may play a significant role in future energy systems with increased proportions of non-dispatchable renewable energy [8]. Such a combined system can shift the electric demands from peak-load to low-load periods, which can help to balance the grid and reduce electricity bills for homeowners. This system can also provide benefits for building thermal comfort [9].

There is much research available in literature focusing on the coupled system of air source heat pumps and TES as a demand-side management (DSM) tool at the domestic level, but the majority was carried out with standard air source heat pumps. To best of our knowledge, investigation on domestic HT-ASHPs and TES as a DSM tool is still scarce. Therefore, more studies are required to further understand how this system performs.

In this paper, the integrated system of a domestic HT-ASHP and TES is conducted by means of simulations which have been validated against the field trial results. The main objective of this work is to assess how the system works in different scenarios, allowing to provide insight into the retrofit potential of this technology when comparing with the performances of boilers.

Nomenclature

W	electric input power of the heat pump	(kW)
COP	coefficient of performance	(-)
T	temperature	(˚C)
t	time	(minute)
RH	outdoor relative humidity	(%)

Subscript

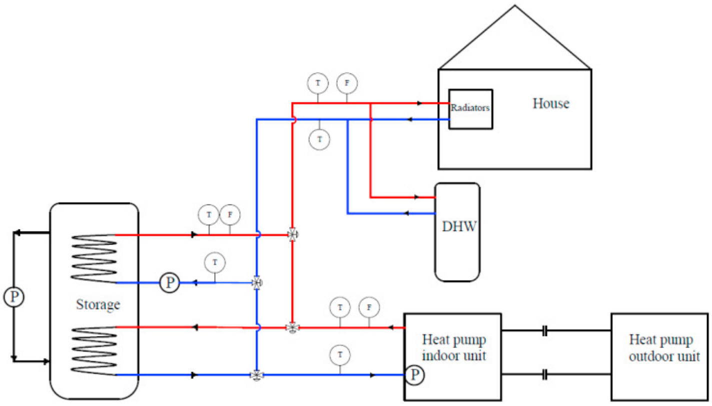
w	water outlet of the heat pump
a	ambient
def	defrost
fr	frost

2. Field trial descriptions

Two mid-terraced “Hard to Heat” houses, which is typical house stock in Northern Ireland, were built at Jordanstown campus of the Ulster University to investigate experiments and research for retrofit technologies in the UK. These dwellings were constructed of solid walls with loft insulation along with timber double-glazed windows and doors. This study concentrated on the house to the left in Fig. 1a which was retrofitted with a variable capacity HT-ASHP integrated with a TES tank, providing heat for space heating and domestic hot water (DHW) of the house. The scheme of the system can be seen in Fig. 1b. The building was occupied by a family of three people. The heating system was controlled based on a programed schedule and a thermostat in the dining room where its temperature was maintained at 21°C.

The retrofit variable capacity HT-ASHP was a cascade unit. This HT-ASHP had a nominal COP of 2.5 with the nominal heating capacity of 11 kW. The TES was a sensible tank that was custom made with copper material, 600-liter capacity, 2m height and 0.6m diameter and 75mm foam thick insulation. There were two coiled tube heat exchangers for charging and discharging the TES as well as seven temperature probes at an equal distance used for control and monitoring purposes. There was also a de-stat pump installed on the storage to mix thermally inside the tank so that the stratification effect was eliminated.

a. Tested houses



b. Schematic of the system

Fig. 1. Field trial houses and schematic of the system.

Data was monitored and stored via the equipment including a wireless radio data logger, 15 transmitters with built-in sensors and a desktop computer. Data was logged in 1-minute interval program. Uncertainties of the sensors were used to calculate the error ranges of the measured COP and heat output which are ± 5.59% and ± 5.17%, respectively. Data collection used for this study was of three periods followed by three control modes, which is described below:

- Direct mode (26/11/2014 – 10/02/2015): The heat pump delivered heat directly to the house. Water outlet temperature of the HT-ASHP was set 75°C, which is the same as the flow temperature of the replaced boiler.
- Storage mode (21/02/2015 – 30/03/2015): The HT-ASHP provided heat to the TES tank, and that heat was then transferred to the house. The heat pump was on to charge the tank if the tank temperature was below 65°C, and it was off when the tank reached 75°C. This mode can be named as a buffering system.
- Combined mode (16/04/2015 – 07/06/2015): The heat pump was switched on during night time (e.g. 1.am – 5.am) to store energy in the storage, bringing the tank temperature to 75°C. When the house required the first heating demand of the day, the stored energy was delivered to the house till its temperature dropped to 55°C. After that, the heat pump took over to provide heat to the house in the rest of the day. The purpose of this mode was to take advantage of the Economy 7 tariff where the electricity price in night time was cheaper than in day time.

3. Methodology

3.1. Modelling

In this work, TRNSYS 17 [10] was used to model the investigated system of a variable capacity HT-ASHP and a TES tank coupled in a residential dwelling. The following subsections explain in detail the developed models.

3.1.1. Heat pump model

TRNSYS Type 1217 was utilized to model the steady state performance of the selected variable capacity HT-ASHP. This model mainly relied on the performance curves combining full load and part load. In our case, the catalog data from the manufacturer just contained nominal values, which is limited and different in practice. Therefore, the field trial data were used to develop the curves alternatively.

3.1.1.1. Full load

The full load curves of the heat pump are the functions of the outdoor dry-bulb temperatures and the water outlet temperatures, as expressed by the following empirical relationships:

$$W = -1.01 + 0.7T_w + 0.24T_a - 0.01T_w^2 + 0.000051T_w^3 - 0.0105T_wT_a + 0.000075T_w^2T_a \quad (1)$$

$$COP = -1.797 + 2.133T_w - 0.107T_a - 0.0034T_w^2 + 0.000016T_w^3 + 0.0046T_wT_a - 0.000035T_w^2.T_a \quad (2)$$

3.1.1.2. Part load

The retrofit HT-ASHP's compressor was variable speed, meaning its heat capacity can be modulated to maintain the desired water outlet temperatures. From the collected data, the part load curve is depicted in Fig. 2.

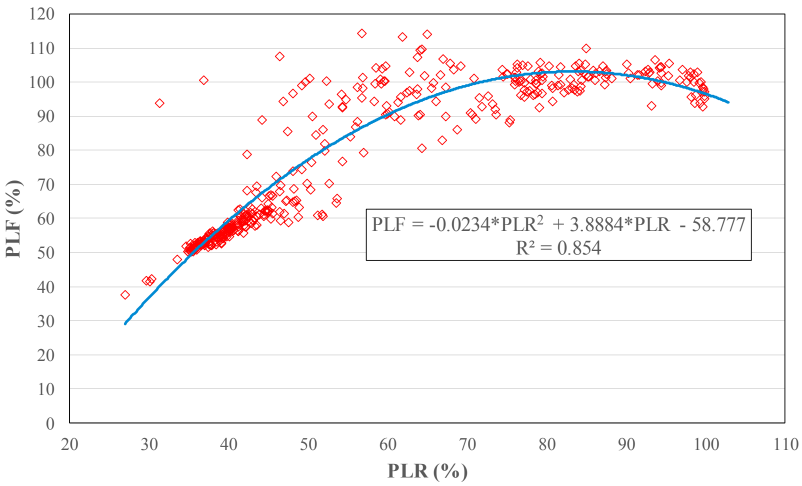


Fig. 2. Part load curve of the heat pump.

3.1.1.3. Defrost cycles

The performance curves above were derived from the collected data excluding the periods of defrost cycles, so a defrost model was developed and coupled outside the heat pump model Type 1271. When the outdoor temperatures dropped below 7°C and the relative humidity rose above 65% for a long period, the heat pump switched to defrost mode. It was observed from the field trial data that: 1) the average cooling power and electric input power of a defrost cycle equalled 2.17kW and 1.75kW, respectively; 2) the time of a defrost cycle was from 1 minute to 10 minutes depending on the frosting time t_{fr} and weather conditions, all of which are given by the following equations:

$$t_{fr} = 39 - 1.06T_a + 0.33RH + 0.13{T_a}^2 - 0.0093RH^2 - 0.018{T_a}^3 - 0.00006RH^3 \quad (3)$$

$$t_{def} = 56.2 - 0.34T_a - 3.56t_{fr} - 0.047{T_a}^2 + 0.079t_{fr}^2 + 0.0096{T_a}^3 - 0.00057t_{fr}^3 \quad (4)$$

3.1.2. Building and DHW models

Heating demand of the house included space heating and DHW, so both of them were developed and integrated into the heat pump model above. Building geometry was firstly drawn in Sketchup software and then imported into TRNSYS Type 56. Boundary profiles obtained from the data collection containing temperature profiles of the adjacent dwelling (house to the right in Fig. 1a) was assigned as inputs for the house model Type 56. U-values in the model were set with the known building envelop elements. Heat gains of occupants and equipment were set identically for weekday and weekend based on the surveys with the inhabitants. Real weather data of on-site station was used to run the models for validation purpose.

TRNSYS Type 534 was utilized to model the DHW. Heat storage capacity of the DHW model was 3.78 kWh with 162 liters and heat loss of 2.74 kWh/24 hours, the same as the real tank. The collected data of DHW consumption was used as an input profile for the DHW drawing patterns in the model.

3.1.3. Storage tank model

TRNSYS Type 534 was obtained to model the TES. The tank was set up with seven nodes and two coiled tube heat exchangers. The heat exchanger for charging the tank occupied in three nodes placed at the tank's bottom, while another for discharging was in the other four nodes. The average tank heat loss coefficient found from calibration was about 9.9 KJ/hr.m^2.K. There was a de-stat pump forcing water convection inside the tank to prevent stratification effect so that this pump model (Type 3d) was also implemented besides the storage tank model.

3.2. Simulations

The above developed models were integrated into a whole system model and then simulated following two main steps. Firstly, the model predictions were validated against the field trial results, with some parameters being varied where necessary. Secondly, three control modes including direct mode, storage mode and combined mode were simulated under the same period, three winter months (January to March) of Belfast weather data available in TRNSYS libraries, and the same other boundary conditions to compare the system performances. These results were also compared with the performances of gas boilers to assess its retrofit capability.

4. Results and discussions

4.1. Model validation

In Fig. 3, daily simulated COPs are mainly in the permitted range of the experimental COPs except for the days of electricity fault. Validation details of the seasonal performance factor can be observed in Table 1. SPF is seasonal performance factor of the heat pump, while "Overall SPF" is seasonal system performance factor in which heat loss of the TES is included. It should be noted that the results in the table are for validation purpose only.

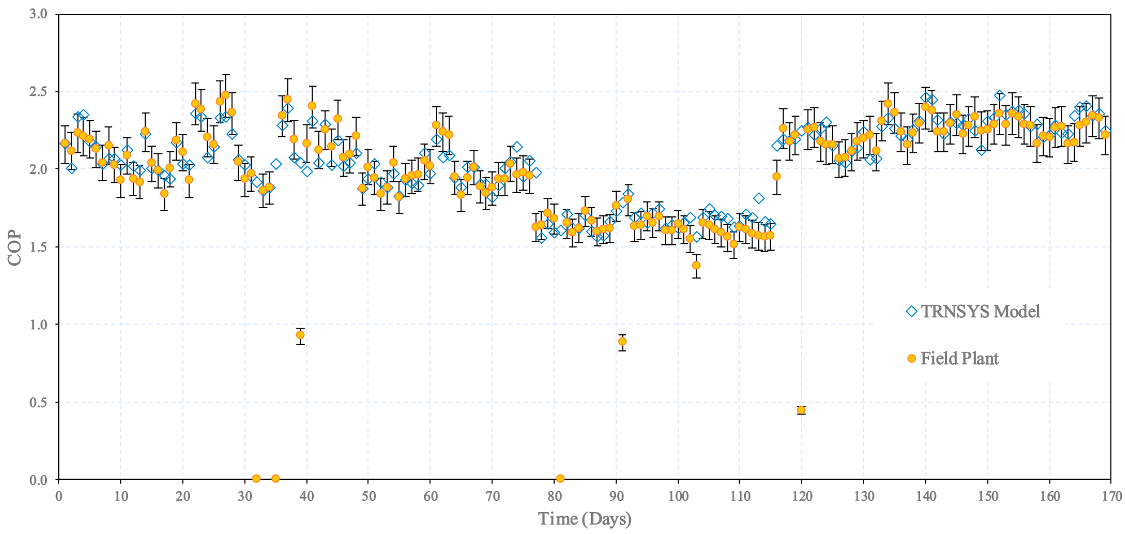


Fig. 3. Comparison between daily predicted COP and daily measured COP in three modes

Table 1. Models' predictions compared with field trial data

	SPF		Overall SPF	
	Field trial	TRNSYS	Field trial	TRNSYS
Direct mode	2.06	2.05	-	-
Storage mode	1.67	1.63	1.49	1.48
Combined mode	2.26	2.24	1.83	1.85

4.2. Three mode comparison

The simulated results of three modes are reported in Table 2. The heat pump SPF in direct mode was better than the ones in the other modes. This is because, in storage mode and combined mode, the heat pump delivered higher water outlet temperature (above 80˚C) to charge the tank compared to that of the direct mode (75˚C). Comparing storage mode and combined mode, the SPF in the buffering system (1.63) was lower than the one in combined mode (2.01) because the periods of the heat pump charging storage in combined mode was at night only, while the heat pump charged the TES all the time in storage mode. It was also found that due to the set point temperature of the TES in storage mode (mentioned in section 2), the heat pump tended to operate at lower load conditions, leading to the lower SPF compared with the one in combined mode.

Table 2. Summary of simulated results of three modes for three months of winter

	Direct mode	Storage mode	Combined mode
SPF [-]	2.03	1.61	2.01
Total heat output [kWh]	9797	10732	10531 (Night: 1655, Day: 8876)
Total electric use [kWh]	4818	6685	5237 (Night: 871, Day: 4365)
Running cost [£]	714	991	753
Carbon emissions [kg]	1852	2570	2013

** Standard electricity rate: 14.83p/kWh; Economy 7 electricity rate: Night 7.5p/kWh, Day 15.7p/kWh; Gas price: 4.3p/kWh

**Carbon intensity on the grid: 0.3844kg/kWh; carbon factor of gas: 0.243kg/kWh

The heat pump in direct mode consumed the least energy (4818kWh), while the energy use in storage mode was highest (6685kWh). This is because the heat pumps were less efficient in storage mode and combined mode than in direct mode, and furthermore they needed to compensate the heat loss of the storage, with the highest compensation percentage being of storage mode due to the continuous coupling system.

The running costs are also calculated in Table 2 based on the energy use. It is clear in the table that direct mode can help the homeowners to pay less money compared to the other modes. While combined mode took advantage of Economy 7 tariff, it was not beneficial for cost savings. The reason for this is because of the low efficiency of the tank, and future work will concern this issue.

Regarding carbon emissions, again due to the lowest energy utilization in direct mode, the heat pump in this mode released the lowest carbon figure (1852kg) compared to the other modes (Table 2). The highest carbon emissions (2570kg) were accounted for the heat pump in storage mode.

4.3. Retrofit assessment

To assess the retrofit performance of the selected HT-ASHP, the results of gas boilers with 60% efficiency (old gas heavy weight boilers) and 90% efficiency (new condensing boilers), which are popular in the UK housing stock, are shown in Fig. 4. It can be seen in the figure that whatever the configuration was, the selected HT-ASHP cannot beat the gas boilers in terms of operating costs. Therefore, this would be a retrofit challenge for this kind of heat pumps.

As for carbon emissions, however, the heat pump in direct mode and combined mode can attain the relative carbon cut (from 6.6% to 33%) compared to the figures of gas boilers (see Fig. 4). The future UK grid with more proportions of renewable energy sources would make these retrofit HT-ASHPs more competitive with gas boilers to help the UK to achieve the binding target of carbon emissions reduction up to 80% by 2050.

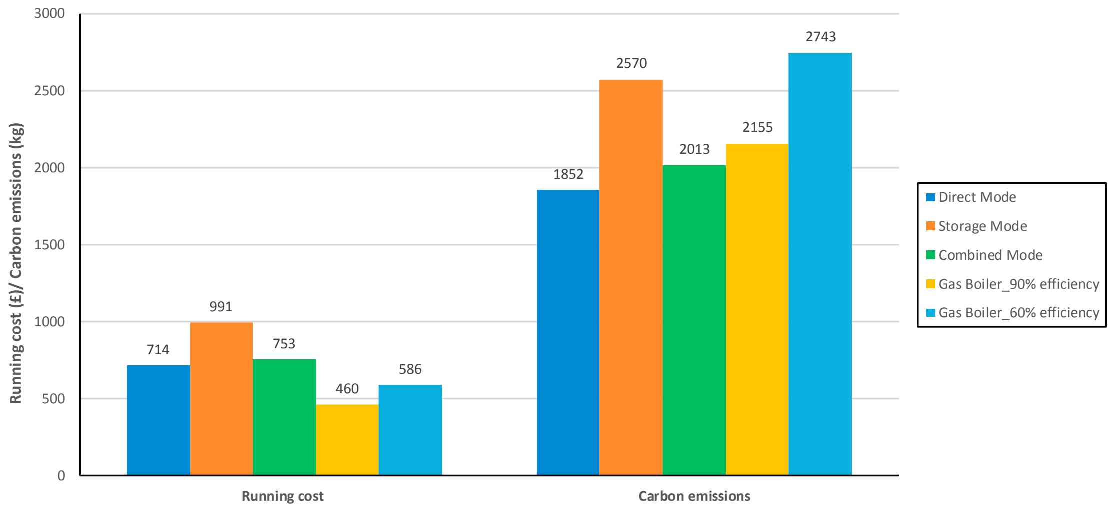


Fig. 4: Running costs and carbon emissions of the retrofit HT-ASHP and gas boilers for three winter months.

5. Conclusions

The retrofit performances of a domestic variable capacity HT-ASHP coupled with TES in three different control modes were compared in this study by means of the developed TRNSYS models which have been validated against the field trial data. The simulated results of three control modes indicate that:

- The buffering system yielded the worst performance including low SPF (1.61) and high running cost (£991).
- Direct mode can allow the heat pump to obtain higher SPF (2.03) compared to the other modes.
- Although combined mode considered Economy 7, its running cost was not lower than the cost of direct mode.
- The retrofit HT-ASHP cannot attain cost savings compared to the gas boilers, but the retrofit heat pump can allow the UK to cut the present carbon emissions.

Future work will carry out the improvement of the efficiency of the storage to enhance the system performance in combined mode. Based on the results of the different system configurations in this study, a demand-side management will be developed to benefit the grid demand and cost savings, which can beat gas boilers' figures.

Acknowledgement

The authors would like to acknowledge the support funding from Research Council UK (RCUK) under i-STUTE project; and from the European Union's INTERREG VA under SPIRE 2 project; and from the European Union's Horizon 2020 under the INPATH-TES project.

References

[1] DECC. Digest of United Kingdom Energy Statistics 2015. 2015.
[2] Parliament of the United Kingdom. Climate Change Act 2008. HM Gov 2008:1–103.
[3] Energy Saving Trust, Department of Energy and Climate Change. Detailed analysis from the first phase of the Energy Saving Trust's heat pump field trial 2012:111.
[4] Kelly NJ, Cockroft J. Analysis of retrofit air source heat pump performance: Results from detailed simulations and comparison to field

trial data. Energy Build 2011;43:239–45.
[5] BS EN 442-1:2014 Radiators and convectors. Technical specifications and requirements. 2014.
[6] Arpagaus C, Bless F, Uhlmann M, Schiffmann J, Bertsch SS. High temperature heat pumps : Market overview , state of the art , research status , refrigerants , and application potentials. Energy 2018;152.
[7] Willem H, Lin Y, Lekov A. Review of energy efficiency and system performance of residential heat pump water heaters. Energy Build 2017;143:191–201.
[8] Hewitt NJ. Heat pumps and energy storage - The challenges of implementation. Applied Energy 2012;89:37–44.
[9] Shah N, Hewitt N. High temperature heat pump operational experience as a retrofit technology in domestic sector. IEEE International Conference Engineering and Technology Innovation/ Internationl Technology Management Conference 2015:1–7.
[10] Klein, S. A., et al. (2014). TRNSYS 17, a TRaNsient SYstem Simulation and program - vol.1. Getting started.