FIELD TESTING OF AN

ECONOMISED VAPOUR INJECTION HEAT PUMP

MING JUN HUANG, NEIL J. HEWITT AND NGUYEN MINH
Centre for Sustainable Technologies, University of Ulster
Newtownabbey, Co. Antrim, N. Ireland, BT37 0QB

ABSTRACT

The purpose of this work was to evaluate the performance of an economised vapour injection (EVI) compressor utilised in an air source heat pump. It was to ascertain if such a unit is capable of economically heating a typical UK family home that was originally heated by a high temperature wet radiator system supplied with hot water from an oil boiler. The concerns were based around maintaining a sufficiently high seasonal coefficient of performance when utilising cold air as a heat source and delivering hot water to a heating circuit originally designed for temperatures of 60°C or more.

The EVI compressor has the capability of overcoming some of the difficulties of high temperature lift operation (namely reduced capacity) by allowing liquid subcooling to maintain high evaporator capacity in order to provide adequate heating during cold ambient air periods. The laboratory tests (based on the EN14511 standard) carried out using air temperatures of -12°C to +15°C and water was delivered from 35°C to 65°C were found to have a superior performance to a compressor without EVI. The system was installed in a family home and its performance evaluated.

1. INTRODUCTION

While it is considered that air source heat pumps are energy efficient space heating devices for homes in moderate winter conditions encountered in UK and Ireland, concerns exist around maintaining a sufficiently high seasonal coefficient of performance (COP) when utilising cold air as a heat source and delivering hot water to a heating circuit originally designed for temperatures of 60°C or more [Payne and O'Neal, 1995]. The economised vapour injection (EVI) compressor has the capability of overcoming some of the difficulties of high temperature lift operation (namely reduced capacity) by allowing liquid subcooling to maintain high evaporator capacity in order to provide adequate heating during cold ambient air periods [Hewitt et al., 1991, Ding et al., 2004; Beeton and Pham, 2006; Hewitt and Huang, 2006 (a)]. A research programme was developed to optimise the components and operating regime of such a heat pump and a number of component improvements were developed, particularly in evaporator design and defrost regime. This work outlines how this system was then used to displace a fossil-fuel boiler in a conventional wet radiator system in a residential house. The heat pump was developed around the Copeland ZH13-KVE series of compressors delivering a nominal 11 kW at a typical domestic home design temperature of -3°C [Hewitt and Huang, 2006 (a) (b)].

2. THE ECONOMISED VAPOUR INJECTION (EVI) COMPRESSOR

The high lift compression is accomplished by extracting a portion of the condenser liquid and expanding it through a small thermostatic expansion valve to an intermediate pressure above evaporator pressure. The expanded refrigerant feds a subcooler where it is evaporated by subcooling the remaining condenser exit liquid. The superheated vapour is then injected into an intermediate compressor port and the subcooled liquid, with its decreased evaporator inlet enthalpy, leads to an improved refrigeration effect. Further, a reduced mass flow is compressed from the suction to the discharge pressures thus reducing the power requirement, and yet the condenser capacity is not reduced because the intermediate pressure and suction pressure gases are recombined in the compressor to ensure full mass flow is attained in the condenser.

The original test facility is illustrated in Figure 1. The test apparatus utilising R407c consisted of the test heat pump system, the hot water supply system and the data acquisition system. For the majority of the tests, the environmental parameters – air temperature, air velocity, and air humidity remained constant for each group of tests. Temperature and pressure measurements were made throughout the system as well as compressor power consumption. The refrigerant circuit distribution and hot water supply system were instrumented with Type-T thermocouples to monitor the variation of temperature (accuracy ±0.29°C). Pressure transducers with a 0-30 bar range were utilised to monitor the variation in refrigeration pressure (accuracy ±0.037 bar). Finally a flowmeter was used to monitor the flow rate variation of hot water system. The humidity of the air in the test chamber was measured with a Sontay combined relative humidity and temperature sensor. The accuracy is ±3%. Data were recorded by ΔT logger at 10 second intervals and stored for later analysis.

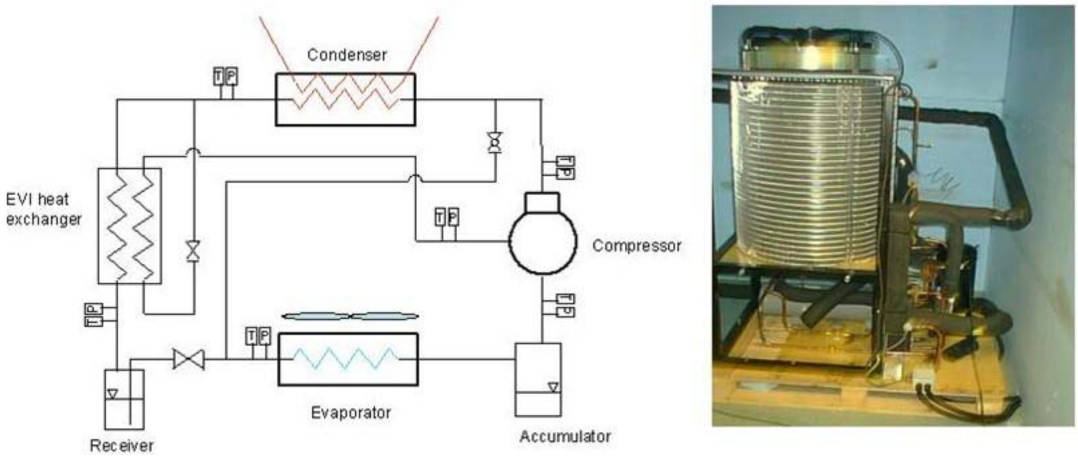

Figure 1. EVI Test Facility

4. TEST RESULTS

The performance of the overall system was tested, based on EN14511 (Anon, 2004), at the following points.

Air-on temperature = -12°C	Water off temperature = 60°C
Air-on temperature = -12°C	Water off temperature = 50°C
Air-on temperature = -7°C	Water off temperature = 60°C
Air-on temperature = -7°C	Water off temperature = 50°C
Air-on temperature = -2°C	Water off temperature = 50°C
Air-on temperature = 2°C	Water off temperature = 35°C
Air-on temperature = 2°C	Water off temperature = 60°C
Air-on temperature = 2°C	Water off temperature = 50°C
Air-on temperature = 7°C	Water off temperature = 60°C
Air-on temperature = 7°C	Water off temperature = 50°C
Air-on temperature = 15°C	Water off temperature = 60°C

A test room capable of maintaining the heat balance across the range of air-on temperatures was equipped to undertake these tests. Tests at temperatures below 0°C where carried out in accordance with EN14511 and their evolution is explained in more detail in Hewitt and Huang (2006). Figure 2 illustrates the performance attained during these tests when compared to that predicted by the compressor manufacturer's software. Is this performance sufficient to economically heat a home?

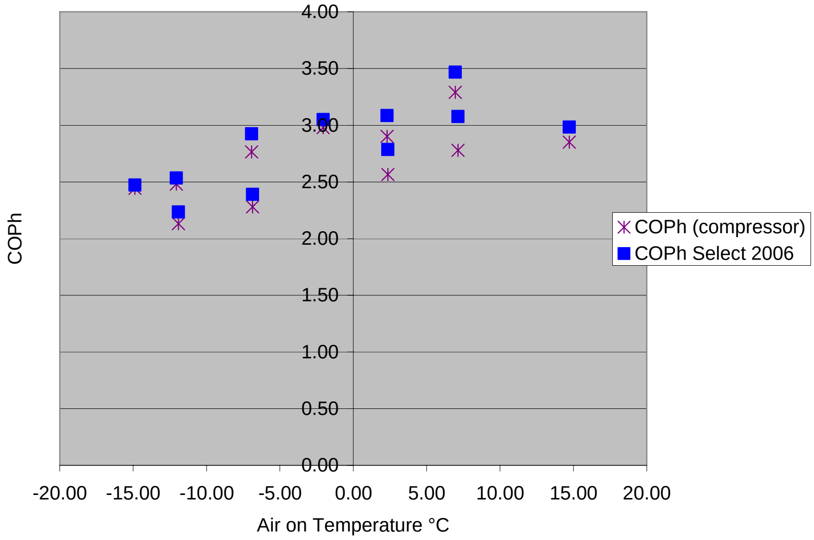

Figure 2. Comparison with Copeland Select Software of Laboratory EVI Heat Pump

3. FIELD TRIAL DESCRIPTION

The field unit is based in the enhanced tube circular evaporator coil and is built in a quasi-compact manner such that it is capable of being transported through a typical UK domestic doorway (Figure 3).

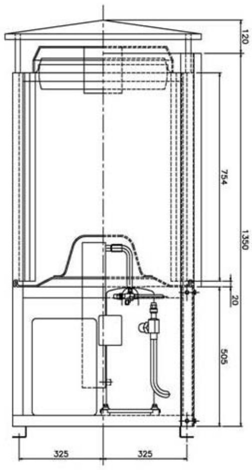

Figure 3. Field Trial Unit

The field unit was installed in a $105m^2$ semi-detached 3 bed-roomed family house in Carrickfergus, Northern Ireland. The home was heated with a conventional UK wet radiator system designed for 82°C-72°C flow and return temperatures. The oil fired boiler was housed in the garage to the rear of the property and therefore it was a relatively simple installation to connect to this system. A 32 amp single phase electricity connection had to be brought from the house to the rear of the garage. The heat required by the home for a range of ambient conditions is shown in Figure 4. The heat pump is oversized by a minimum of 30% of all applications.

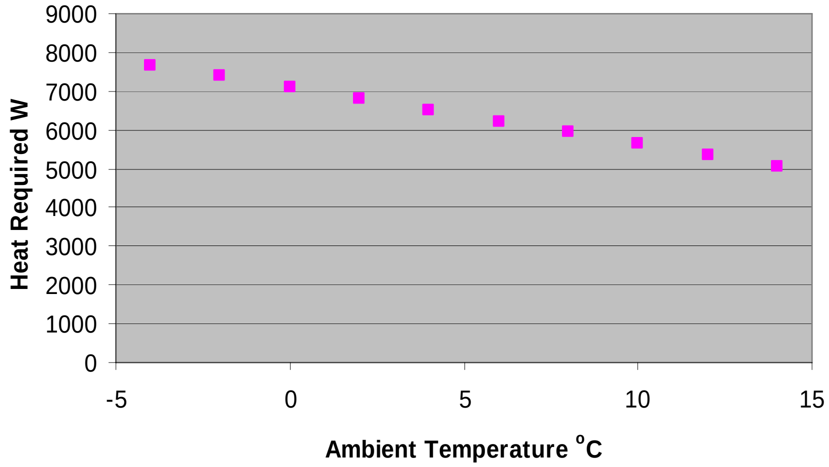

Figure 4. Carrickfergus Test Site Heating Demand (inc. 2000W for hot water).

4. FIELD TRIAL EXPERIMENTS AND RESULTS

The unit was installed in February 2006 and has been operation since then with no major difficulties except that of a minor refrigerant leakage. The increased trends of COP with increasing ambient temperatures are consistent for each of the different supplied hot water temperatures (55 to 63°C) (Figure 5).

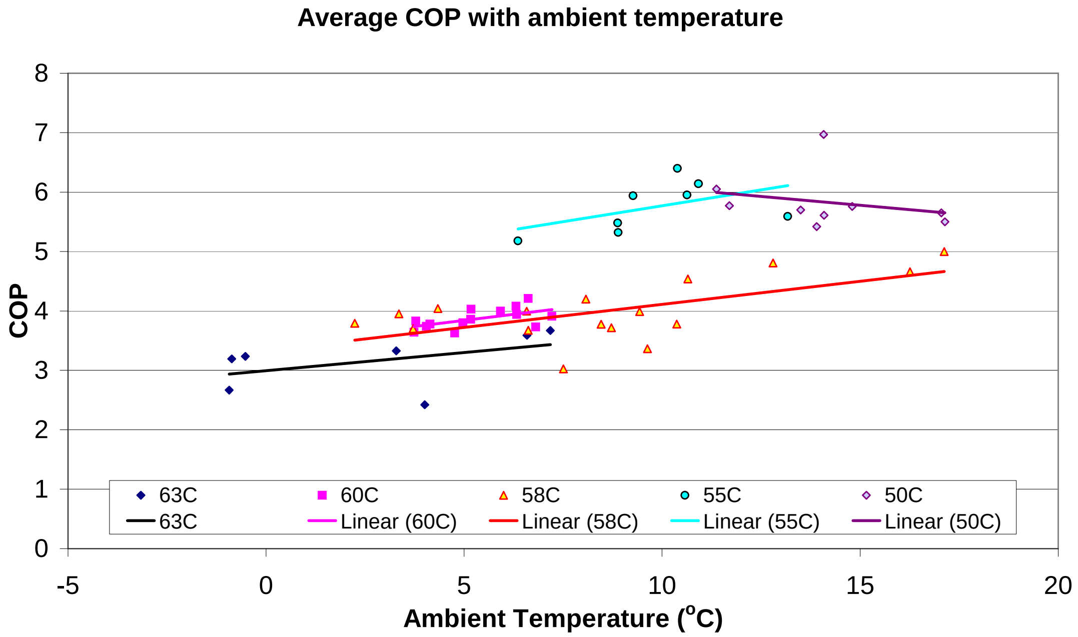

Figure 5. Average COP for a range of conditions

The data is general is very satisfactory with improvements in performance noted as both ambient air temperature rose and water supply temperature was reduced in accordance with a simple weather compensated control strategy. This was devised in accordance with meeting the heat demand of the home through a series of fixed area high temperature radiators whose performance would deteriorate as lower temperature water was circulated through them, as illustrated in Figure 6.

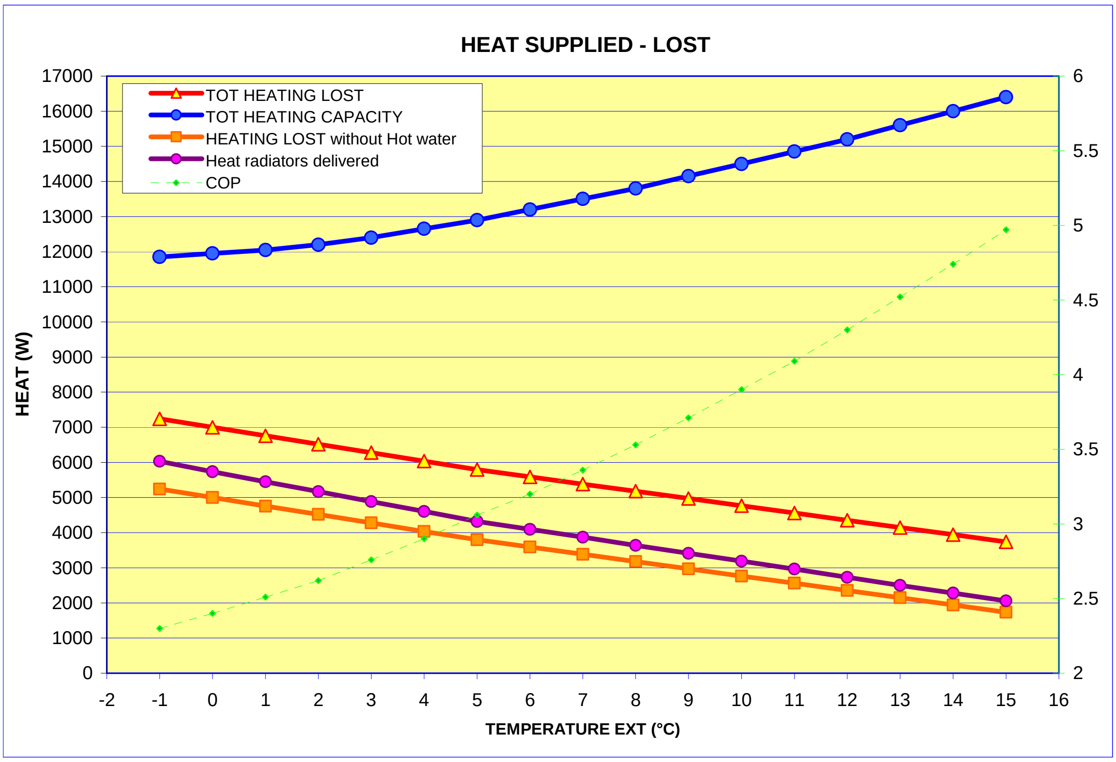

Figure 6. Relationship between Heat Pump, House Heating Systems, Heat Demands and Ambient Temperature

It was noted that at higher ambient temperatures, the performance was not increasing as predicted. This was due to the main thermostatic expansion valve controlling evaporator conditions having reached its maximum capacity.

The improved performance can be in the most part attributed to the effects of improved average COP caused by ramp-up and on-off cycling. Ramp-up takes the water in the heating system from typically about 16 °C to a maximum of 60 °C leading to a high "average" COP (Figure 7).

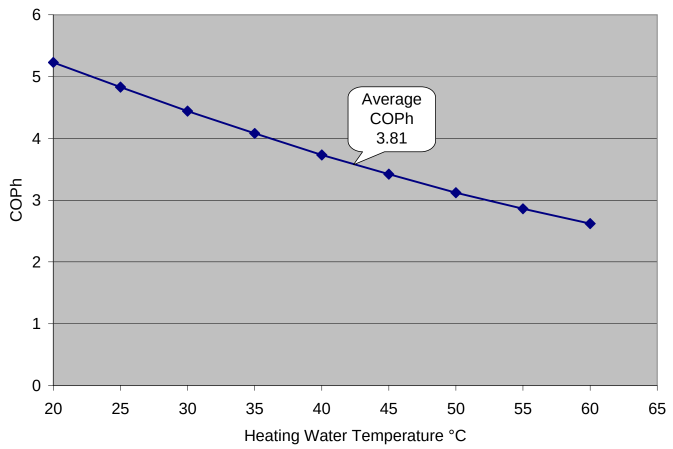

Figure 7. Average COPh of Ramp-Up

However this appears to be an insufficient explanation. The data from the Carrickfergus test site reveals a lower ramp-up COP and a higher average COP for water at 60°C and an ambient temperature of 5.5 °C (Figure 8). The average COP during the ramp period is lower than the cycling period (Figure 8). This is because of during the on-off cycling stage, when the compressor off, the circulating hot water is still able to transfer heat to the house. Thus heat is being supplied for zero compressor power. Further, the role of the 144 litre domestic hot water tank acting as an additional heat supply during start up and a heat sink once the heat pump supply water temperature exceeds the tank temperature is not yet fully understood.

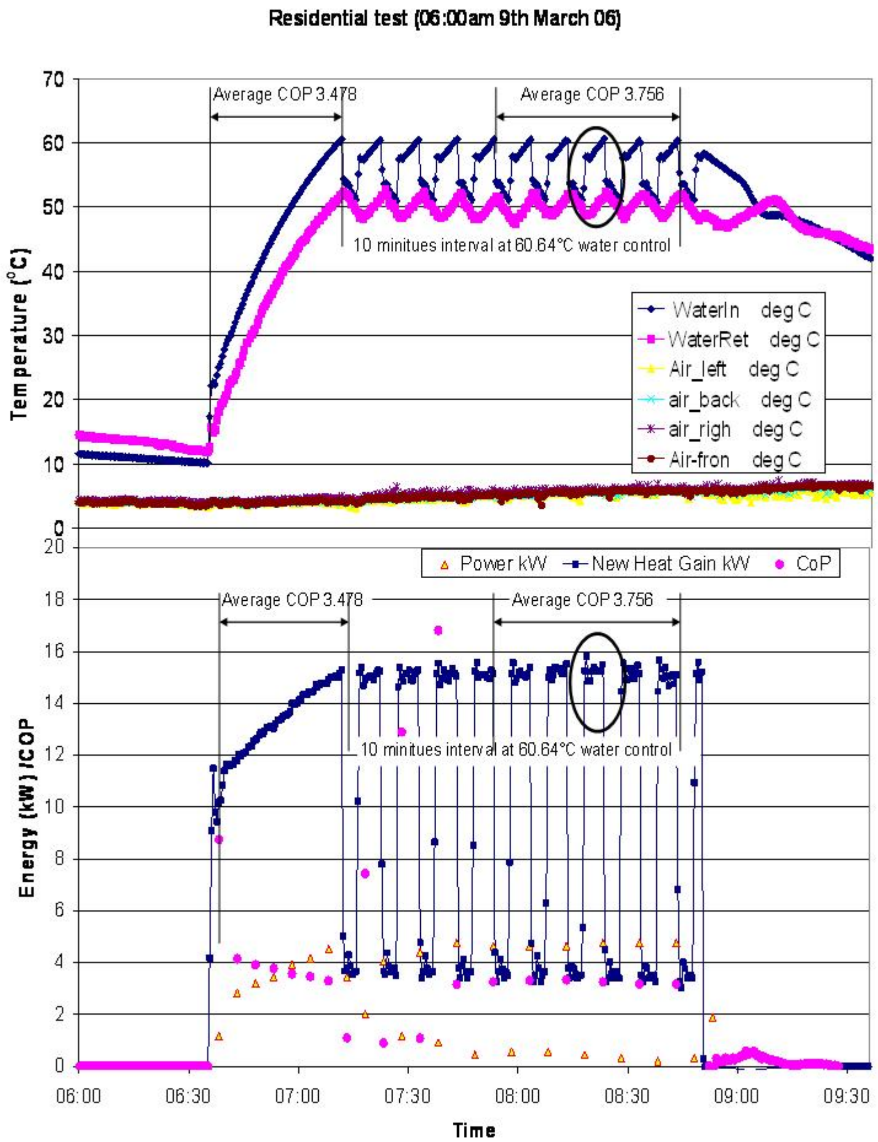

Figure 8. Ramp-up and average COP from Carrickfergus test site

Summer operation has evidence of improved performance but much of this operation occurred outside the current compressor envelopment. Figure 9 illustrates typical summer conditions with the high ambient air temperatures coupled with short run times (as the heat demand is only that for domestic hot water) raising concerns over system longevity and requiring some form of capacity control or an acceptance that another form of water heating is required during such conditions.

5. OPERATING COSTS

Running costs for the unit are as follows. Typical winter operation (Space Heating & Hot Water) at an ambient temperature of -1°C to 6°C with hot water being delivered between 60°C and 65°C is shown in Table 1:

Table 1. Winter Operation Running Costs

Period	Heating Method	Total Domestic Energy Usage	Average Cost/wk	Notes
9 Sept 05 - 1 Feb 06 (22 Weeks)	Oil Boiler	Oil: 900 Litres + Electricity for Appliances	£16.40 £ 5 Total £21.40	Oil @ 40p/l (2006 price)
1 Feb 06 - 30 Mar 06 (8 Weeks)	ASHP	Electricity for ASHP and Appliances	£20	Electricity @ 10.21p/kWh (not off peak tariff)

Summer operation (Hot Water Only) requires the water to be delivered at 65°C (Table 2) from an air source temperature >15°C. Considerable savings can be achieved with the air-source heat pump in summer operating conditions.

Table 2. Summer Operating Conditions

Period	Heating Method	Total Domestic Energy Usage	Average Cost/wk	Notes
May onwards when Ambient Air Temperature > 15°C	Oil Boiler	Oil for DHW only + Electricity for Appliances	£13.10 £ 5 Total £18.10	Based on DHW load 364 kWh/w Oil @ 40p/l *
	ASHP	Electricity for ASHP (DHW) and Appliances**	£11.41	Electricity @ 11.21p/kWh (not off peak tariff)

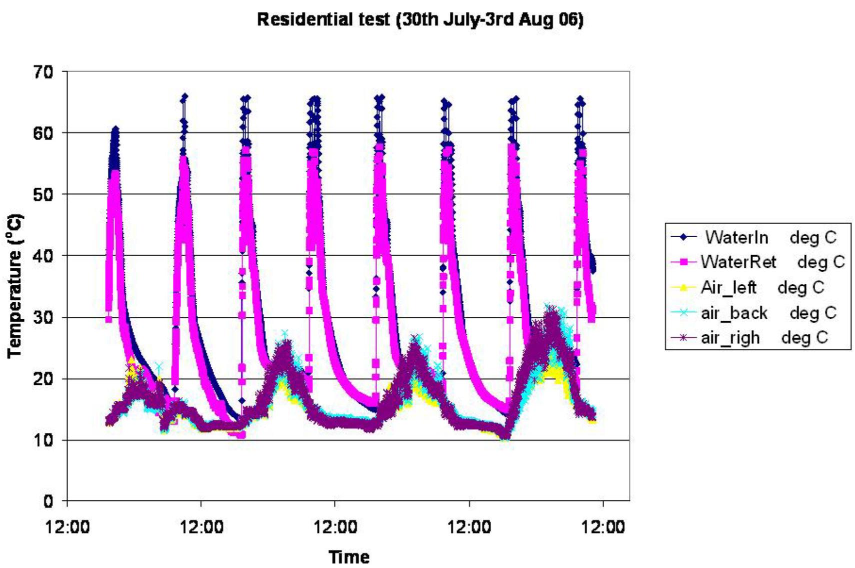

Figure 9. Typical Summer Operation

6. CONCLUSIONS

The performance of an economised vapour injection (EVI) compressor utilised in an air source heat pump is ascertained and evaluated. It is found that such a unit is capable of economically heating a typical UK family home. The performance of the residential air source heat pump for UK conditions, was encouraging, as was the social acceptance of the unit, regarding noise for example. However, most importantly, the running costs were actually lower than heating with oil and secondly, the heat pump met all the domestic requirements placed upon it. An air source heat pump is capable of direct retrofit into an existing home originally heated by a high temperature wet radiator system driven by an oil boiler. Running costs are comparable with an oil fired boiler but there are significant savings in carbon dioxide emissions. Running costs can be improved by operated weather compensated control in warmer months.

REFERENCES

V. Payne and D. L. O'Neal, (1995), Defrost cycle performance for an air-source heat pump with a scroll and a reciprocating compressor, *Int. J. Refrigeration*, 18, pp. 107-112.

NJ Hewitt; JT McMullan and NE Murphy (1991). Development of an Alternative Refrigeration Cycle. *International Journal of Energy Research*. Wileys, Chichester, Vol 15, pp731-745.

NJ Hewitt and MJ Huang (2006). Defrost Cycle Performance for an Air Source Heat Pump. *Proc. The 2nd International Conference of Renewable Energy in Maritime Island Climates*, 26-28 April 2006, Dublin, Ireland

W.L. Beeton and HM Pham, (2003), Vapour Injected Scroll Compressors, *ASHRAE Journal*, April,

Carbon Trust Expert Info – Conversion Factors. Accessed 22nd March 2006. http://www.thecarbontrust.co.uk/energy/pages/page_64.asp, (2006).

Y. Ding, Q. Chai, G. Ma and Y. Jiang, (2004), Experimental study of an improved air source heat pump, *Energy Conversion & Management*, pp. 2393-2403.

ACKNOWEDGEMENTS

The authors would like to acknowledge the support of Copeland Ltd and InvestNI who funded this project. The authors would also like to thank BL Refrigeration Ltd for test bed construction and Mr Philip Dalzell for adding to and maintain these systems.