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The Experimental Analysis of the Effect of Ambient Factors on the Defrosting of Economised Vapour Injection Compressor Air Source Heat Pump in Marine Climates

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Abstract
Air source heat pumps have numerous advantages in many applications over other heating equipment in marine climates with regard to energy efficiency. The main concerns are based around maintaining a sufficiently high seasonal coefficient of performance (COP) when (a) utilising cold air as a heat source and (b) delivering hot water to a residential heating circuit originally designed for water temperatures of 60°C or more with oil heating. The Economised Vapour Injection (EVI) compressor has the capability of overcoming some of the difficulties of high temperature lift operation during cold ambient conditions. However it is not clear except ambient temperature how the other ambient factors may affect the performance of EVI air-source heat pump in marine climates. This paper evaluates operating performance with the defrost effect for a retrofit residential EVI air-source heat pump in Belfast, UK. The ambient factors which affect the performance of the heat pump defrosting were studied. The investigation was to optimise the operation of an EVI air source heat pump operating under defrost conditions encountered in maritime climates.

Key Words: Testing, Domestic heat pump, Thermal analysis, High temperature, COP, Efficiency.

NOMENCLATURE

ASHP	air source heat pump

COP	Coefficient of Performance	
C_p	Water specific heat	kJ kg^{-1} K^{-1}
EVI	Economised vapour injection	
i	Enthalpy of refrigerant	kJ kg^{-1}
m	Water volume flow rate	m^{3}s^{-1}
$\dot{m}$	Evaporated refrigerant mass flow rate	kgs^{-1}
$Q_{extract}$	Heating extract from the ambient	W
$\dot{Q}_h$	Heating output rate from the condenser	W
T	Temperature	°C
T_{air}	Air Temperature	°C
ΔT	temperature difference between hot water inlet and outlet in the hot water cycling system	°C
W	The electrical power consumption of running the heat pump	W

Subscripts

CondExit	Location at the exit of condenser
Discharge	Location at the compressor discharge
EvapInlet	Location at the evaporator inlet
EVI Inlet	Location at the EVI heat exchanger inlet
EVI Suct	Location at the EVI suction inlet
ExpanIn	Location at the main expansion valve inlet
Suction	Location at the compressor suction
WaterIn	Hot water temperature at the water circuit cycle entrance point
WaterRet	Hot water temperature at the water circuit cycle return point

Greek symbol

ρ	Water Density	kgm^{-3}
η	Efficiency	

1. INTRODUCTION

Air source heat pumps (ASHP) are energy efficient space heating devices. They have a significant impact on carbon dioxide emissions and the potential to save on running costs when compared to oil-fired central heating. The challenge is how to utilise cold air as a heat source and maintain the desired levels of comfort whilst utilising an existing wet radiator oil/ fossil heating circuit system originally designed for water at temperatures of 60°C or more. European maritime island climates with the moderating influences of the sea in winter there are relatively few days below 0°C and, as a consequence, air source units may be viable (Hewitt et al, 2011). The heating capacity of ASHP decreases as outdoor air temperature drops, especially when there is frost formation on the outdoor heat exchanger surfaces in humid climates (Stoecker, 1957; Yasuda et al., 1990; Payne and O'Neal, 1995). Adopting an economised vapour injection (EVI) compressor into the ASHP allows it to overcome the high temperature lift operation (namely reduced capacity) which allows low temperature liquid refrigerant subcooling to be attained while maintaining high evaporator capacity in order to provide adequate heating during cold ambient air periods (Hewitt et al., 1991; O'Neal et al., 1991; Beeton and Pham, 2003; Wang et al., 2009; Hu et al., 2011). During the refrigerant cycle in the EVI ASHP (shown in Figure 1) a small portion of the liquid refrigerant from the condenser (at state 3) passes through the EVI thermostatic expansion valve to an intermediate pressure. This expanded refrigerant exchanges heat with the remaining refrigerant from the condenser in the internal heat exchanger to subcool the main-stream refrigerant to state 5, At the same time the saturated vapour in the intermediate pressure is injected into the intermediate compression chamber through an injection port (at state 4). The subcooling refrigerant is expanded through the main expansion valve to the low pressure (state 6) and enters the evaporator with lower enthalpy. This therefore attains high evaporator capacity. The reduced mass flow rate through the evaporator causes the temperature on evaporator drops which may sensitive to the ambient factor conditions. (**after EVI operating, the pressure there is a very little change compare with the evaporator inlet temperature dropped.)

A research programme was developed to optimise the components and operating regime of such a heat pump (HP) and a number of component improvements were developed, particularly in the evaporator design for residential application (Hewitt et al., 2006 and Hewitt et al., 2011). The developed ASHP with EVI compressor has been tested in the laboratory environmental chamber with EN14511 standards and thereafter a field trial unit was developed and installed. It was found that the EVI ASHP unit has a superior performance to an ASHP without EVI under test conditions (Huang et al., 2009). In order to achieve the best performance of the EVI ASHP, it is essential to understand the effect from the multiple ambient conditions. Frost on the outdoor heat exchanger coil reduces its ability to absorb heat from the outdoor heat exchanger coil and thus degrades the thermal performance of the ASHP. The layer of frost reduces the airflow area needed for heat exchange as a result of the blockage and in addition the frost that builds up over the coils acts as an insulating layer. The hot gas bypass defrost method can defrost the evaporator coil without utilising valuable space and hot water heating. However the study of the performance of hot gas bypass for defrosting in the residential ASHP system is limited. Stoecker (1957) analysed the size of pipes carrying hot gas to defrost evaporators. Using a photo-coupler for detecting frost formation in an air source heat pump and also for determining the initiation point of the defrost cycle has been experimentally studied (Byun et al, 2006; 2008). The performance of the hot-gas bypass defrosting test with a circular evaporator coil for EVI ASHP was tested in a laboratory simulated environment by Hewitt and Huang (2008). The conditions include the defrost initiation condition, defrost operating time and interval between defrosts. However the performance evaluation under realistic conditions for this type of HP is essential before any optimised hot gas defrosting suggestions can be achieved. In this work a detailed performance analysis for EVI ASHP in a field trial with repeatable ambient conditions cases in Belfast, UK is presented. This paper presents the capacity of a high temperature ASHP with/without defrosting conditions to replace a fossil-fuel boiler in a conventional wet radiator system in a residential house in the coldest period of the year in Belfast. The main work of this paper is not only to study the EVI system with hot gas bypass defrosting, but also the ambient factors that affect the EVI system defrosting under freezing conditions.

2. FIELD TRIAL UNIT DESCRIPTION AND EXPERIMENTAL PROCEDURE

A semi-detached 3 bed-roomed family house with $105m^2$ in Carrickfergus, Northern Ireland was heated by an ASHP. The EVI ASHP was pretested in the laboratory before being fixed into the residential building to supply heat for hot water and space heating for the field trial. The system was operated and monitored in Belfast since 2007, N. Ireland. In this paper only the operations under repeatable freezing conditions have been selected and classified as seven cases for analysing to find the effect of ambient conditions on the system thermal performance. The field trial unit is developed around the ZH13-KVE series of economized vapour injection compressors (Copeland, 2009) delivering a nominal 11 kW heat to meet the heat requirement of the house with an outdoor winter design temperature of -3°C (Figure 2). The refrigerant in the system is 407C. The heat supplied by the ASHP was delivered to the residential rooms by a conventional radiator system. The temperatures at different positions in the EVI ASHP cycle system, along with the ambient temperature and hot water heating system water inlet/outlet temperature, were monitored using T-type thermal couples (Figure 1). The pressures at different positions in the cycle system were monitored by ALCO PT4-30M pressure transducer as listed in Figure 1. A pulse power measurement ME4zrt with accuracy of ±0.05kWh was used to monitor the compressor power consuming, *W*. The measurement accuracy for the flow rate is ±2 percent. Data were recorded by an ΔT logger at 30 second intervals for analysis.

The ASHP unit was on-off controlled with the supplied hot water temperature at 63/50 °C. When the ambient temperature was below zero in Feb to March, which is the coldest months in N. Ireland, hot gas bypass was used for defrosting. To commence hot gas defrosting, a solenoid valve in the hot-gas bypass line was energized, connecting the compressor discharge with the evaporator inlet downstream of the expansion valve and upstream of the evaporator distributor, thus supplying the heat for defrosting. The evaporator fan shuts off during defrosting. At the termination of the two minutes defrost cycle, the ASHP and the evaporator axial fan are switched back to heating mode and resumes normal operation. Two defrosting strategies with an interval cycles at half hour and 45 minutes and lasting for a 2 minutes operation time were tested on the ASHP. In order to avoid becoming frozen during cold evenings the ASHP was operated at 3:00am for 10 minutes. The hot gas bypass defrost performance of the ASHP was analysed on those days. In the residential house the ASHP operated twice from 6:35am for 2 hours 10 minutes and from 18:10pm for 3 and half hours.

The integrated cyclic Coefficient of Performance (*COP*) was used to evaluate the defrosting test which is defined as the ratio of the integrated heating output over the integrated energy consumption. The integration was carried out over some complete frost-defrost cycles under stable conditions. Assuming there is no heat loss from the condenser to heat up the water, the heating output $\dot{Q}_h$ (a) was calculated using the water flow rate *m* and temperature difference ΔT between water inlet and water outlet in the hot water cycling system as shown in Eqn. (1)

and (b) checked against condenser refrigerant mass flow and enthalpy balance. C_p is water specific heat, ρ is water density.

$$\dot{Q}_h = m\rho c_P \Delta T \qquad (1)$$

COP was calculated as:

$$COP = \frac{\dot{Q}_h}{W} \qquad (2)$$

The heat extraction $\Delta Q_{extract}$ from the ambient to the evaporator can be described as:

$$Q_{extract} = \dot{m}(i_1 - i_6) \qquad (3)$$

Where 1 and 6 represent the positions at entrance and exit the evaporator (see Figure 2).

3. FIELD TRIAL RESULTS AND ANALYSIS

3.1. The Performance of the EVI Compressor Air Source Heat Pump Unit at Freezing Ambient Conditions

The performance of the EVI ASHP with defrosting at freezing ambient conditions was studied. A typical morning heating operation from 6:35 am to 8:45 am period of 2 hours and 10 minutes in a typical freezing ambient conditions Case II (2nd March) was selected for analysis. The supplied hot water temperature setting for on/off cycle was 50/63°C. The ASHP was energized until the hot water supply temperature rose to 63°C and then shut off until the temperature dropped to 50°C. The EVI was opened after 10 minutes system operating. The hot gas bypass defrosting was operated every each half hour and lasted for 2 minutes.

The average ambient temperature was around -1.2°C within the operation (Figure 3). Although defrosting was operated at half hour intervals, the frost still accumulated during the morning operation (photo 1). Ice build-up on the evaporator was observed and resulted in reduction of the airflow area with time. During the 2 hours 10 minutes operation not enough heat was able to be moved from the surrounding air; thus the maximum temperature of supplied hot water was below 55°C.

The supplied hot water temperatures in the inlet and outlet, refrigerant temperatures along the system cycle, super-heat/subcooling and the heat supply and power consumption along with *COP* are shown respectively in Figure 3(a), (b), (c) and (d). From Figure 3(a) it can be seen that the hot water could quickly go to above 50°C, but was enable to reach the required 63°C. The accumulated frost on the surface of the evaporator reduces the heat extraction $\Delta Q_{extract}$ from the ambient to the evaporator to such an extent that the refrigerant cannot be fully evaporated (see equation 3). This reduces the refrigerant mass flow rate $\dot{m}$ in the cycle and can contribute to expansion valve hunting which can explain the pressure fluctuation and therefore the temperature fluctuation along the ASHP system cycle (Figure. 3(b)). The continuing pressure decrease reflects the frosting on the evaporator, hence the temperature for evaporating is reduced. With the frost accumulating the discharge temperature is increasing thus leading to the compressor working under hard conditions. The super-heat and subcooling are obtained from REFPROP of NIST (2008) using the R407C and pressure as reference. The super-heating and subcooling have severe variations due to the lack of mass fluid in the cycle (Figure. 3(c)). If this condition was allowed to continue, it would result in compressor failure. The reduced heat supply with frost accumulation can be seen from Figure 3(d). The ice on the evaporator outside coil can't be fully defrosted, hence the supplied heat is not only failing to provide the required heat but is also reducing with time progression in each defrosting cycle. The average COP is 2.67 which is calculated based on the European standard tests EN14511-2 (Anon, 2004) for heat pumps.

3.2. The Effect of Different Ambient Factors with Freezing Conditions on the Performance of the ASHP System with EVI Compressor

The field trial test has proved that the ambient temperature significantly affects the ASHP performance. However there are other factors that can affect the performance of the ASHP rather than just the ambient dry-bulb temperature. The morning operation records for 7 consecutive days in March with variable humidity, wind speed and average operation dry-bulb temperature between -1.2 to +1.9°C (except one day up to +6.8°C) were selected for analysis. The operation was started at 7:30am and lasted for two hours and 10 minutes. The ASHP was under 50/63°C on-off operation. The ASHP defrosting intervals were set at 30 minutes and 45 minutes separately for comparison.

Table 1 lists the detailed comparison for the system *COP*s with average and initial ambient temperatures, humidity, wind speed and the time taken to reach the maximum hot water supply temperature. The comparison of the effect of the ambient factors on the ASHP performance is presented in Figure 4. From the test results the positive effect of ambient temperature on the performance of the ASHP can be seen, but often the supplied hot water cannot reach the required 63°C under freezing conditions, even when the average ambient temperature is above 0°C. From I to III (1st to 3rd March) with similar average and initial ambient temperatures and the same defrosting interval in freezing conditions the system performance is different. On the Case I (1st March) the system can provide the hot water to the required level with *COP* of 3.13, while the water supply temperature could not reach the required 63°C on both Case II and Case III (2nd and 3rd March) with *COP*s of 2.67 and 2.79 respectively. With the solar radiation effect being negligible during the test winter morning sessions; the humidity and the wind speed are the main factors contributing to the ASHP's performance difference. On the Case I (1st March) the high wind chill (3.47m/s) with high relative humidity (97%) can improve the defrosting performance for the 30 minutes defrost interval with similar ambient conditions on Case II and III (2nd and 3rd March). The ambient temperatures along with the water inlet/outlet temperatures for the Case I and II (1st and 2nd March) are compared in Figure 5 respective to wind speed of 3.47 and 1.29 m/s. The high relative humidity of air in the maritime climate contains more latent heat which can moderate the frosting when ambient temperature is around -1°C. The comparison of the system performance under similar ambient temperature shows the wind speed has more effect on the ASHP performance than the relative humidity. The ASHP operated under higher wind speed can provide higher hot water temperature, although it is still under the required temperature (Case III and Case II; Case III and Case V) (3rd and 2nd days; 3rd and 5th days).

The performance of the EVI ASHP can be judged by COP and also the time to get to the required hot water temperature. For the 45 minute defrost interval, when the average ambient temperatures are higher than 1.4°C (Case IV, VI and VII in Table 1) (4th, 6th and 7th March in Table 1), the wind chill can improve the ASHP performance with *COP* higher than 3.12 and the time for the supplied hot water to 63°C can be shortened from 90 to 45 minutes with wind speeds respectively of 0.9 m/s on Case IV (4th) and 2.44 m/s on Case VI (6th March). When the surrounding air temperature is around freezing point (Case V (5th March) with initial air temperature at 0°C) the highest possible hot water temperature is 54°C under defrosting operation with wind chill of 1.8 m/s therefore it has low *COP* 2.63. Comparing the HP performance with *COP* and the time to get the required hot water temperature, when the ambient temperature is above 1.4°C the 45 minutes defrost interval can meet the requirement.

Thus when the average ambient temperature is around to the freezing point during the ASHP operation, the 30 minutes defrost interval will be suitable for the climate similar to N. Ireland ambient conditions.

3.3. The Effect of Sharp Rising Ambient Temperature on the ASHP Performance with Defrosting Operated

The EVI ASHP performance is affected by the ambient temperature. During the early spring season there is a significant rise in the ambient temperature from freezing to a warm on a sunny morning. The defrosting is needed when the ambient temperature is below 2°C from all of the recorded trials. In the warm climate the ASHP will be operated using the heat pump on-off control and not the defrosting dominant control. The effect of the variable ambient on the ASHP performance has been studied with the field trial data. A typical morning operation with initial temperature at -1.1°C and reaching 6°C by the end of the 2 hours and half hours was studied on Case IV (4th March) (Figure 6 and 7). The defrost interval was 45 minutes. After the initial defrosting the hot water supply temperature can reach 63°C in 1.5 hours and this is then followed by the on-off cycle control. This is due to the increased ambient temperature which increases the ASHP capacity and therefore switches the system to the 53/63°C on-off control. The ASHP is energized until the hot water supply temperature rises to 63°C and then shuts off until the temperature dropped to 53°C in this case. The interval time for the on/off 53/63°C heating cycle was decreasing when the ambient temperature raised, the period of on/off cycle was reduced from 20 minutes to 15 minutes. Another reason which caused the cycle reduction is the frost formed on the evaporator which reduced the refrigerant mass flow rate in the refrigeration cycle. The performance of the ASHP with temperatures along the cycle line is presented on Figure 7 which reflects the hot water supply pattern. The accumulated frost on the surface of evaporator can be seen as the evaporating temperature is dropping with time progression.

3.4. Field Trial Unit Performance with Ambient Temperature above 5°C

With ambient temperature increasing the required heating load is declined, so the setting temperature changed to 60/50°C on/off cycle control. The requirement for defrosting when the ambient temperature is above 2°C depends on the operation time and other ambient conditions. This section is to study the EVI ASHP performance with long operation hours when the ambient temperature is above 5°C (Figure 8). Figure 9 shows the ASHP performance for ambient temperature around 7°C at the beginning of operation and around 5°C at the end. The average *COP* for ramp up is 4.55, but beyond the full heat supply. After the full heat supply stage, the average *COP* can reach up to 4.06. With 3.5 hrs of operation, the heat supply reduces and therefore the *COP* decreases from 4.06 to 3.66. One reason for this is the drop in ambient temperature from 7.5°C to 5.2°C. However this appears to be an insufficient explanation. With time progression although the ambient temperature is 5°C at the end, the surface temperature on the outside coil is below zero hence the frost is accumulated with continuing operation. The regular on-off cycle can regulate the frost accumulation, but for the unit operating for more than 2 two hours the performance will be adversely affected until beyond the acceptable range.

4. CONCLUSIONS

The field trial performance of defrost for an economised vapour injection (EVI) compressor utilised in an ASHP was ascertained and evaluated. It was found that the ambient temperature significantly affected the performance of the ASHP unit with defrosting. The frost that formed on the outdoor coil is not just dependant upon the ambient temperature but a combination of different factors: solar intensity, humidity, wind speed and operation strategies along with defrosting control. By operating a 30 minutes defrost interval and with increased air flow around the evaporator the EVI ASHP can satisfy the requirements for daily heating demand under freezing conditions in the similar maritime island climates. Therefore an air source heat pump is capable of direct retrofit into an existing home originally heated by an oil boiler with added ventilation around the evaporator even in when the ambient temperature was below -1°C.

the conditions need defrosting.

For the extremely cold day in March, , the ASHP with the half hour interval and 2 minutes hot gas bypass defrosting cycle cannot provide enough heat for residential space heating with low wind speed in N. Ireland, UK.

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Figure captions and table:

Figure 1. A schematic of the system with thermocouple locations

Figure 2. Air source heat pump for field trial in Carrickfergus, N. Ireland UK

Figure 3. Performance of heat pump with wind speed of 1.29 m/s and freezing conditions

Figure 4. Comparison of the effect of ambient temperature, wind speed and humidity on EVI ASHP performance with 30 and 45 minutes' defrosting interval

Figure 5. Hot water supply temperature comparison for Case I and Case II (1st and 2nd March) respective to wind speed 3.47 and 1.29 m/s

Figure 6. Ambient temperature and water supply temperature variation for system with 45 minutes' defrost interval (Case IV (4th March))

Figure 7. Temperature variation along the heat pump cycle with 45 minutes' defrosting interval (Case IV (4th March))

Figure 8. EVI ASHP hot water supply performance with ambient temperature above 5°C

Figure 9. EVI ASHP energy performance with COP for ambient temperature above 5°C

Photo 1. HP evaporator with frost on a morning session operation

Table 1. Effect comparison of ambient factors on ASHP performance

Table 1. Effect comparison of ambient factors on EVI ASHP performance

Defrost interval	**time**	**Humidity**	**Amb temp (°C)**		**Wind**	**Peak state**		**COP**
		%	AV	Initial	(m s^{-1})	Temp (°C)	Time (mins)	
30 mins	1st (I)	97	-1.1	-1.3	3.47	63	100	3.13
	2nd (II)	94	-1.2	-1.5	1.29	54	120	2.67
	3rd (III)	88	-1.1	-2.0	2.32	60	120	2.79
45 mins	4th (IV)	93	1.4	-1.1	0.90	63	90	3.12
	5th (V)	89	0.8	0	1.80	54	45	2.63
	6th (VI)	98	1.9	0.9	2.44	63	45	3.25
	7th (VII)	97	6.8	6.4	3.60	63	30	3.73

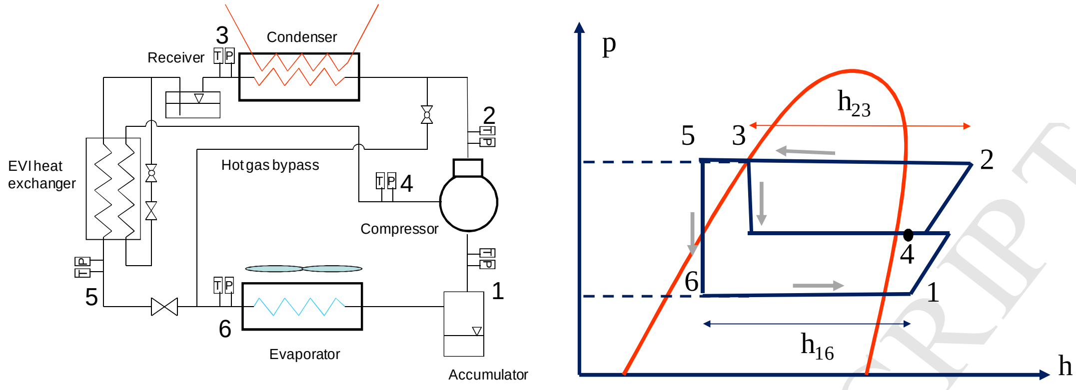


Figure 1. A schematic of the system with thermocouple locations

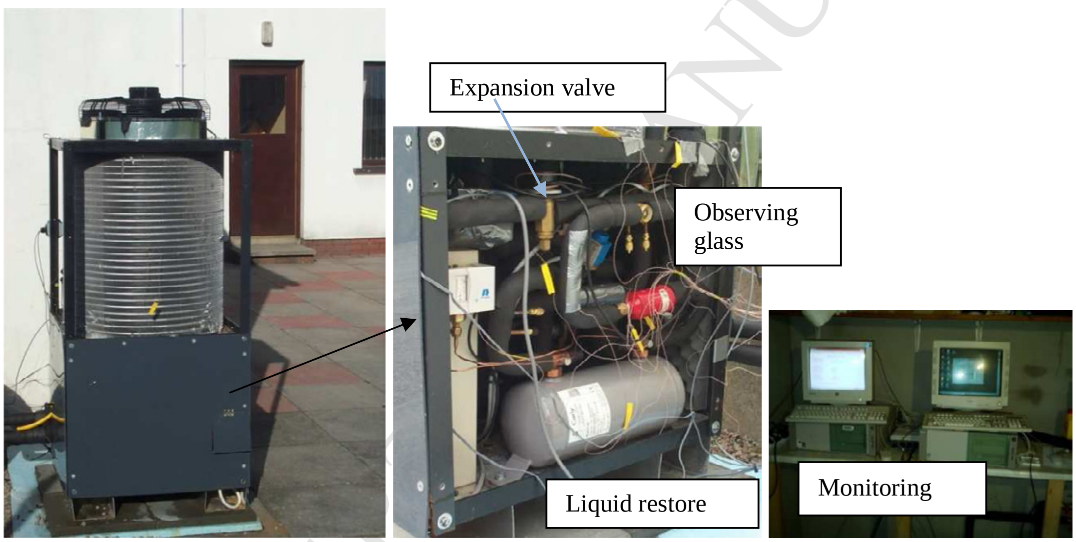


Figure 2. Air source heat pump for field trial in Carrickfergus, N. Ireland UK

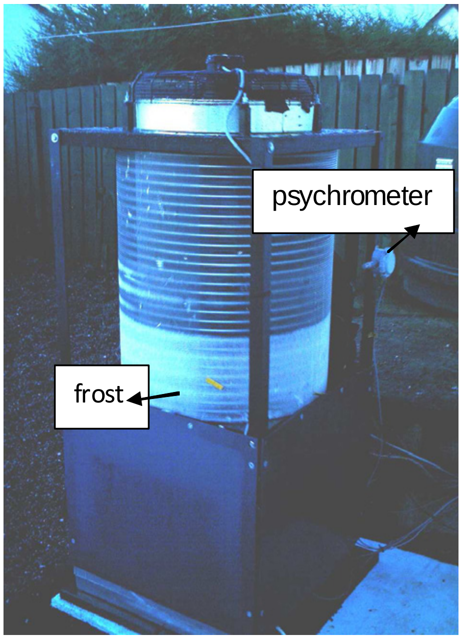
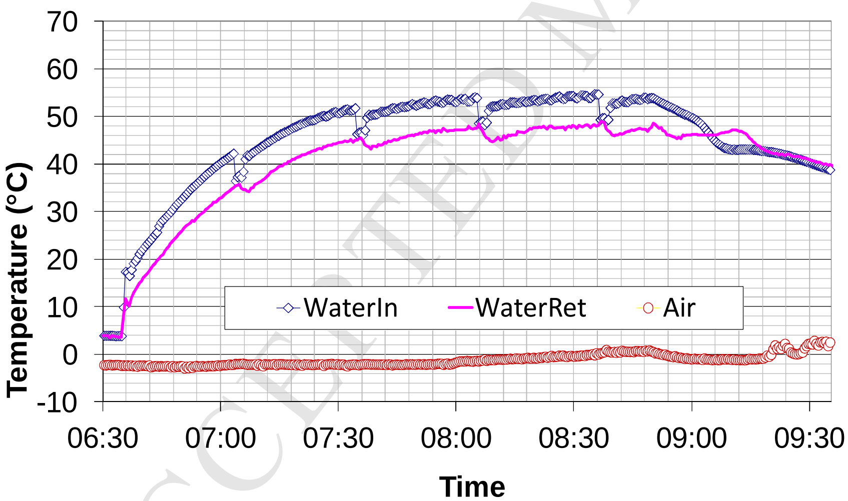
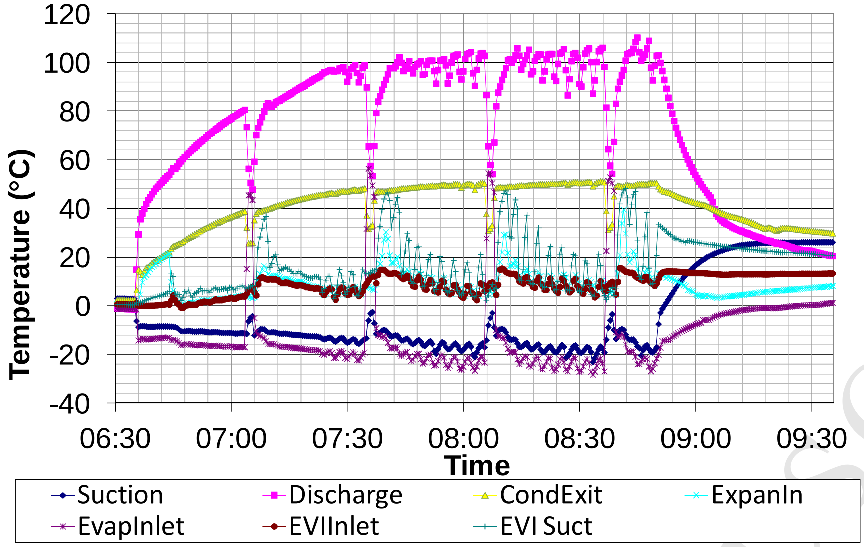


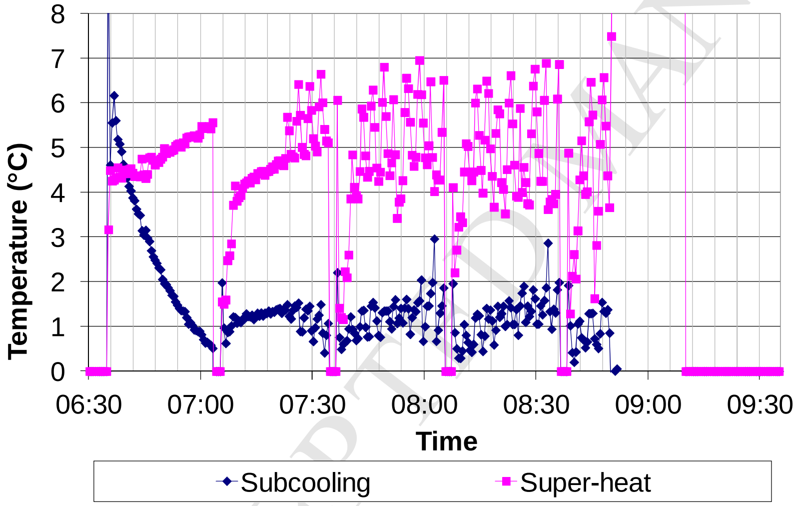
Photo 1. HP evaporator with frost on a morning session operation



(a)


Temperature (°C)
120
100
80
60
40
20
0
-20
-40
06:30
07:00
07:30
08:00
08:30
09:00
09:30
Time
Suction
Discharge
CondExit
ExpanIn
EvapInlet
EVIInlet
EVI Suct

(b)


Temperature (°C)
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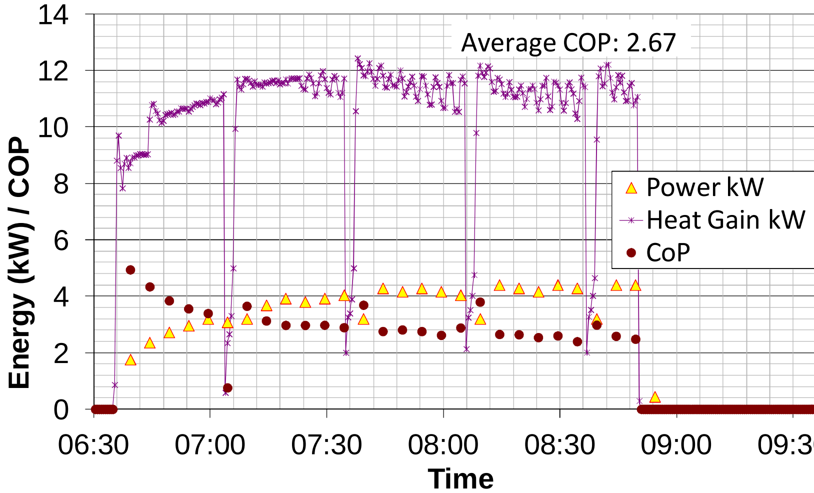
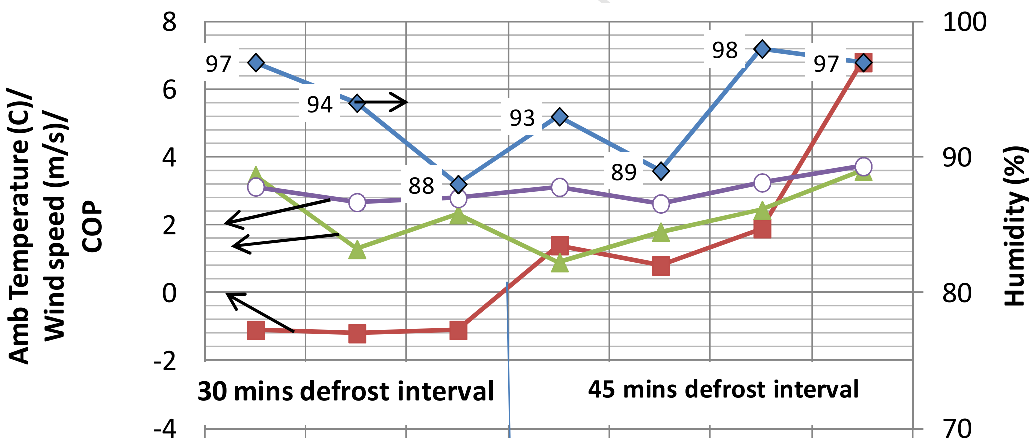
7
6
5
4
3
2
1
0
06:30
07:00
07:30
08:00
08:30
09:00
09:30
Time
Subcooling
Super-heat

(c)



(d)

Figure 3. Performance of ASHP with wind speed of 1.29 m/s and freezing conditions (Case II, 2nd March)



	1st	2nd	3rd	4th	5th	6th	7th
Temp	-1.1	-1.2	-1.1	1.4	0.8	1.9	6.8
Wind	3.47	1.29	2.32	0.9	1.8	2.44	3.6
COP	3.13	2.67	2.79	3.12	2.63	3.25	3.73
Humidity	97	94	88	93	89	98	97

Figure 4. Comparison of the effect of ambient temperature, wind speed and humidity on EVI ASHP performance with 30 and 45 minutes' defrosting interval

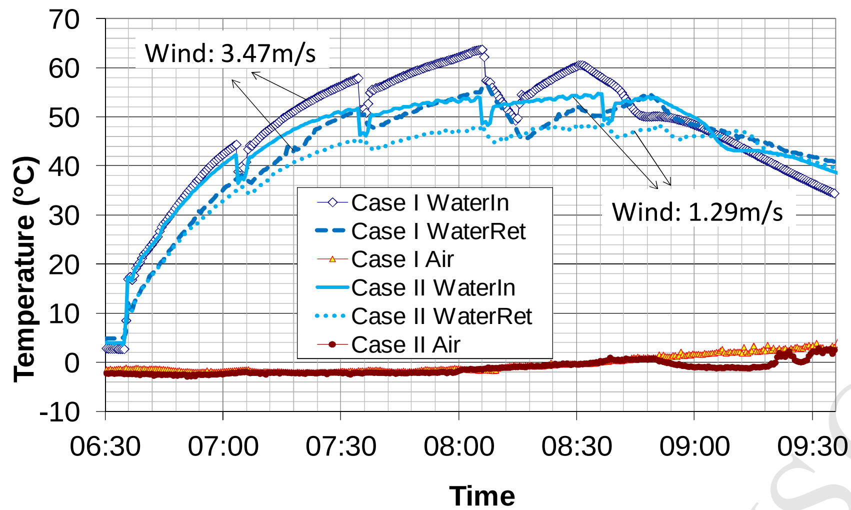


Figure 5 Hot water supply temperature comparison for Case I and II (1st and 2nd March) respective to wind speed 3.47 and 1.29 m/s

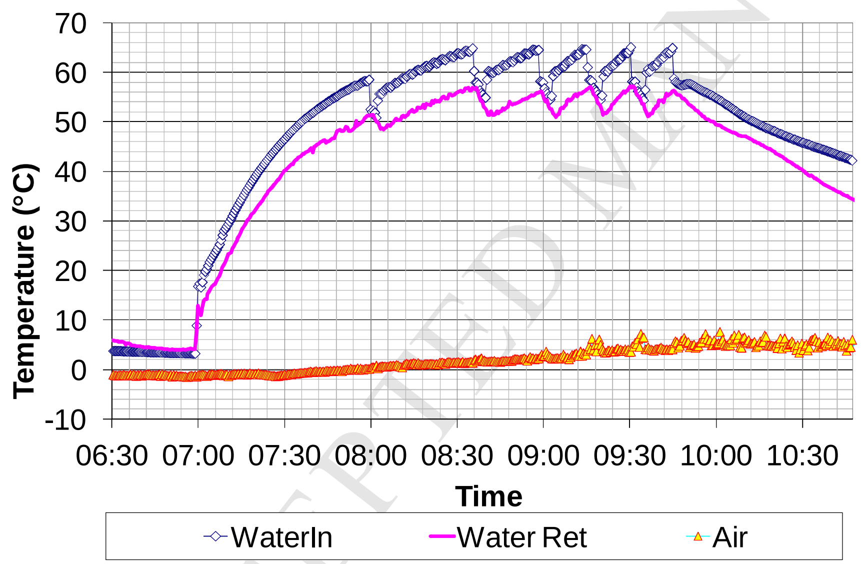


Figure 6. Ambient temperature and water supply temperature variation for system with 45 minutes' defrost interval (Case IV) (4th March)

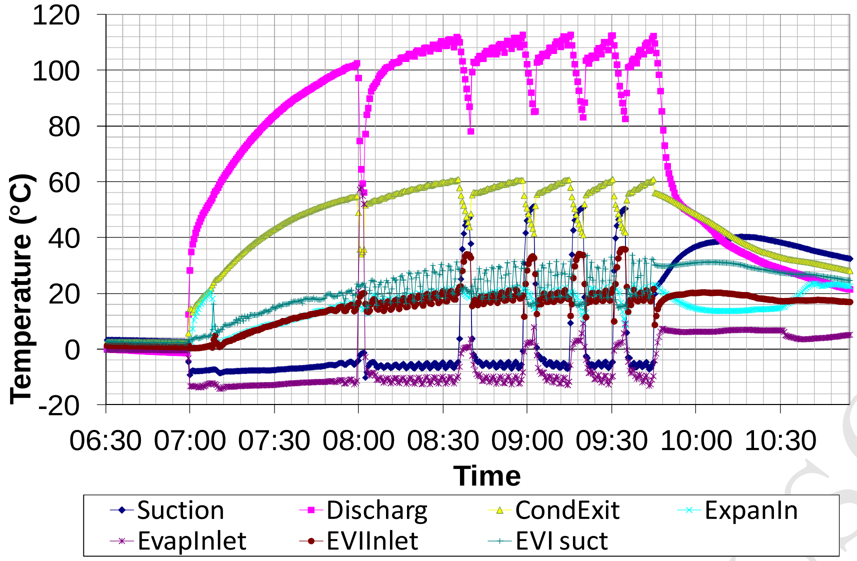
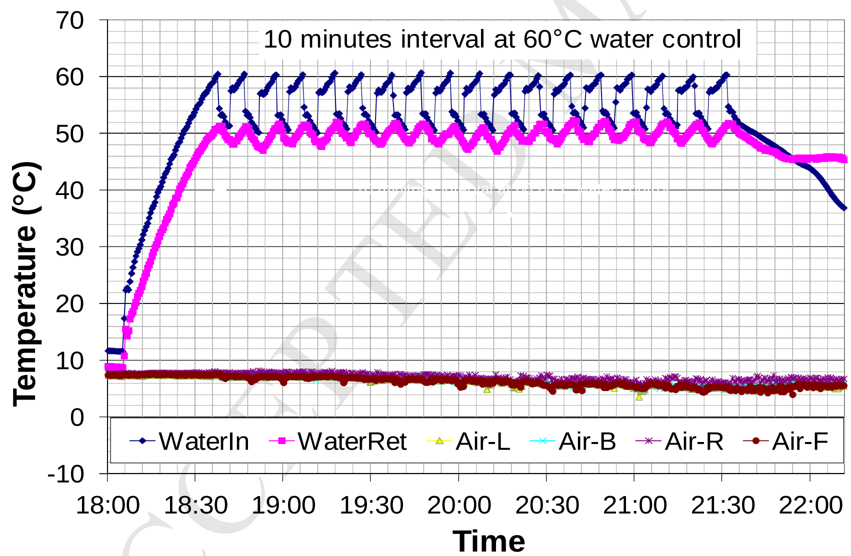


Figure 7. Temperature variation along the heat pump cycle with 45 minutes' defrosting interval (Case IV) (4th March)



On/off temperature control

Figure 8. EVI ASHP hot water supply performance with ambient temperature above 5°C

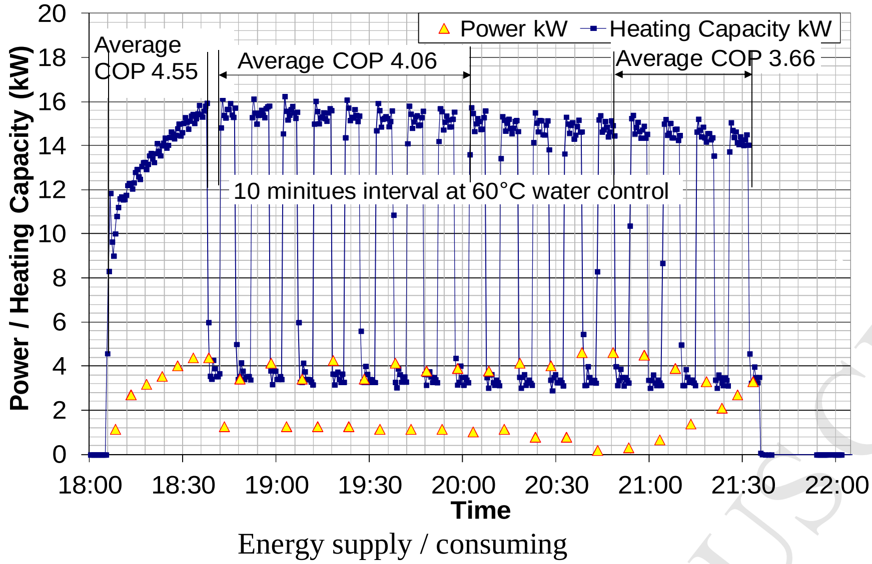


Figure 9. EVI ASHP energy performance with COP for ambient temperature above 5°C