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Thesis Title:

An Experimental Characterisation of Ground and Air Source Heat Pump Technologies in the Irish Maritime Climate

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Abstract

As energy derived from hydrocarbons now constitutes 87% of worldwide energy consumption there has been mounting concern about the potential negative environmental, economic and sustainability impacts of such over reliance on fossil fuel. Recent national and international policies reflect these concerns by setting targets for renewable energy supply by 2020 and 2050. Ireland has committed to achieve 12% renewable derived thermal energy consumption by 2020 and heat pump technologies are targeted as a key contributor. However, this project responds to the absence of climate specific design knowledge for the Irish Maritime climate and the need for effective source side management.

This study addressed this knowledge deficit by developing a unique test facility with the capacity to characterise both ground-source and air-source heat pumps. It consisted of an automated weather station; $15kW_{th}$ horizontal collector ground source heat pump ($GSHP_{HC}$); $15kW_{th}$ vertical collector ground source heat pump ($GSHP_{VC}$); $8kW_{th}$ air source heat pump (ASHP), along with 111 sensors and supporting data acquisition system which allowed the characteristics of the climate, collector and heat pumps to be continuously monitored.

This facility was operated for 745 test days between 2007 and 2009, during which time 22 tests were conducted and 168,522kWh (606GJ) delivered. The average Seasonal Performance Factor (SPF) for the $GSHP_{HC}$, $GSHP_{VC}$ and ASHP were 2.90, 2.95 and 3.74 respectively, highlighting the suitability of the Maritime climate for ASHPs. The impact of climate on the ground's upper layer has been characterised in terms of ground temperature and moisture content and its impact on heat pump Coefficient Of Performance (COP) has been quantified. The influence of heat pump duty and collector design parameters such as soil type and ground cover type has also been assessed. A new parameter known as the Collector Performance Indicator (*CPI*) has been established to allow horizontal collector performance to be quantified. A numerical model has been developed to assess the performance of a new climate sensitive split level horizontal collector that delivered an 8% higher SPF than the standard collector design. Potential exists to boost performance by a further 7% using more effective source side management techniques. Suggestions were also made to insulate a portion of the vertical collector return pipe; that would boost performance by 5%. The study concludes with a series of recommendations that would further exploit the potential of the test facility and test data and boost the contribution of heat pumps in a sustainable energy fuelled future.

Declaration

The substance of this thesis is the original work of the author and due reference and acknowledgment has been made, when necessary, to the work of others. No part of this thesis has already been accepted for any degree and is not been concurrently submitted in candidature for any other award.

Signed:________________ Date: 24/9/10

Niall Burke (Candidate)

Signed:________________ Date:__________

Dr. John Lohan (Supervisor)

DEDICATION

To my family, friends and especially to my ever supportive wife…

STATEMENT OF CONFIDENTIALITY

The material contained in this thesis may not be used, sold, assigned or disclosed to any other person, organisation or corporation without the expressed permission of each of the following:

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PUBLISHED WORK

The following publications result directly from this study:

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Dimache, L., Greene, M., Fitzgerald, F., Burke, N. and Lohan, J., (2010), "Optimum Sizing and Operation Control of an Ideal Hybrid Thermal Renewable Energy System Investigated *a Posteriori*", Renewable Energy Beyond 2020, Trondheim, Norway, 7-8 June 2010.

Burke, N., Greene, M. and Lohan, J., (2008), "Experimental Characterisation of Both Air and Ground Source Heat Pump Technologies in the Irish Maritime Climate", The Challenge of Sustainability Conference, Dundalk IT, Ireland, 20 June, 2008.

Lohan, J., Clarke, R., Burke, N., Greene, M. and Fitzgerald, F., (2008), "Development of an Experimental Test Facility to Continuously Monitor the Performance of Sustainable Energy Technologies in Maritime Climates", The Challenge of Sustainability Conference, Dundalk IT, Ireland, 20 June, 2008.

Greene, M., Burke, N., Lohan, J. and Clarke, R., (2008), "Ground Temperature and Moisture Content Surrounding Horizontal Heat Pump Collectors in a Maritime Climate Region", 9th International Energy Agency Heat Pump Conference, Zürich, Switzerland, 20-22 May, 2008.

Burke, N., Greene, M., Lohan, J. and Clarke, R., (2008), "Heat Pump Performance Sensitivity to Horizontal and Vertical Collector Configurations in a Maritime Climate Region", 9th International Energy Agency Heat Pump Conference, Zürich, Switzerland, 20-22 May, 2008.

Burke, N., Greene, M., Lohan, J., Dimache, L., Clarke, R. and Fitzgerald, F., (2008), "Experimental Characterisation of both Air and Ground Source Heat Pumps Operating under the Irish Maritime Climate", AGMET, Energy and the Irish Climate – Harnessing Irish Climate for Energy, Dublin, Ireland, 29 February, 2008.

Dimache, L., Lohan, J., Allen, B., Burke, N., Greene, M. and Fitzgerald, D.F., (2008), "Investigating Energy Solutions for Sustainable Aquaculture", AGMET, Energy and the Irish Climate – Harnessing Irish Climate for Energy, Dublin, Ireland, 29 February, 2008.

Greene, M., Burke, N., Lohan, J. and Clarke, R., (2007), "Ground Temperature Gradients Surrounding Horizontal Heat Pump Collectors in a Maritime Climate Region," First International Conference on Energy and Sustainability (ENERGY 2007), New Forest, UK, 20-22 June, 2007.

Lohan, J., Burke, N., and Greene, M., (2006), "Climate Variables that Influence the Thermal Performance of Horizontal Collector Ground Source Heat Pumps," Proceedings of ESDA 2006: 8th Biennial ASME Conference on Engineering Systems Design and Analysis, Torino, Italy, 4-7 July, 2006.

Lohan, J., Greene, M., and Burke, N., (2006), "The Development of a Ground Source Heat Pump Characterisation Facility for the Irish Maritime Climate," Proceedings of the 2nd Renewable Energy in Maritime Island Climates Conference (REMIC2), Dublin Institute of Technology, Dublin, Ireland, 26-28 April, 2006.

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NOMENCLATURE

1. Abbreviations

Abbreviation	*Name*	*Unit*
ASHP	air source heat pump	(-)
BHE	borehole heat exchanger (vertical collector)	(-)
COP	instantaneous (every minute), coefficient of performance	(-)
COP_{DAY}	daily average coefficient of performance	(-)
COP_{AVG}	average coefficient of performance over stated period	(-)
CPI	collector performance indicator	$(K/(W/m^2))$
DAQ	data acquisition system	(-)
DHW	domestic hot water	(-)
$GSHP_{HC}$	ground source heat pump utilising the horizontal collector	(-)
$GSHP_{VC}$	ground source heat pump utilising the vertical collector	(-)
HP-IRL	abbreviation used to refer to this study	(-)
IiBC	innovation in business centre	(-)
SH	space heating	(-)
SPF	seasonal performance factor	(-)
TAZ	thermally affected zone	(-)
TRT	thermal response test	(-)
WSHP	water source heat pump	(-)

2. Symbol

Symbol	*Name*	*Unit*
Q_{HC}	horizontal collector thermal energy extract rate	(kW)
A_{HC}	horizontal collector area	(m^2)
Q'_{HC}	horizontal collector demand	$(kWh/m^2/hour)$
q''_{HC}	instantaneous horizontal collector thermal demand	(W/m^2)
Q''_{Depth}	vertical ground heat flux at given depth	(W/m^2)
Q_{Depth}	ground volumetric thermal energy content between given depths	(kWh/m^2)
Q_{Diff}	difference in thermal energy content between given depths	(kWh/m^2)
Q_{VC}	vertical collector thermal energy extract rate	(kW)
Q_{AS}	air source thermal energy extract rate	(kW)

Symbol	*Name*	*Unit*
T_F	fluid flow temperature from HP to collector	(°C)
T_R	fluid return temperature to HP from collector	(°C)
$T_{HC,F}$	fluid flow temperature from HP to horizontal collector	(°C)
$T_{HC,R}$	fluid return temperature to HP from horizontal collector	(°C)
$T_{HC,\infty}$	horizontal collector 'farfield' temperature	(°C)
$T_{P1,0.9m}$	control temperature at profile 1 ($T_{P1,0.9m} = T_{HC,\infty}$)	(°C)
$\Delta T_{HC,G}$	collector drawdown, difference between $T_{HC,\infty}$ and $T_{HC,R}$ ($\Delta T_{HC,G} = T_{HC,R} - T_{HC,\infty}$)	(K)
$\Delta T_{HC,B}$	brine, temperature difference between $T_{HC,R}$ and $T_{HC,F}$	(K)
$\Delta T_{HC,R}$	horizontal collector ground temperature recovery difference between $T_{P1,0.9m}$ and $T_{P5,1.0m}$	(K)
$T_{VC,F}$	fluid flow temperature from HP to vertical collector	(°C)
$T_{VC,R}$	fluid return temperature to HP from vertical collector	(°C)
$T_{VC,\infty}$	vertical collector 'farfield' temperature	(°C)
$\Delta T_{VC,G}$	difference between $T_{VC,\infty}$ and $T_{VC,R}$ ($\Delta T_{VC,G} = T_{VC,R} - T_{VC,\infty}$)	(K)
$\Delta T_{VC,B}$	brine, temperature difference between $T_{VC,R}$ and $T_{VC,F}$	(K)
q'_{VC}	instantaneous vertical collector thermal demand	(W/m)
$T_{HP,F}$	heat pump sink flow temperature for the $GSHP_{HC}$, $GSHP_{HC}$ or ASHP	(°C)
$T_{HP,R}$	heat pump sink return temperature for the $GSHP_{HC}$, $GSHP_{HC}$ or ASHP	(°C)
ΔT_{HP}	heat pump temperature lift - difference between source and sink temperatures	(K)
$\Delta T_{HP,B}$	fluid temperature difference between $T_{HP,R}$ and $T_{HP,F}$	(K)
$T_{P\#,Depth}$	Ground temperature at given Profile number and depth	(°C)
$M_{P\#,Depth}$	ground moisture content at given Profile number and depth	(m^3/m^3)
λ_G	ground thermal conductivity	($W/m\cdot K$)
λ_{eff}	ground *effective* or *bulk* thermal conductivity	($W/m\cdot K$)
α	ground thermal diffusivity	(m^2/s)
θv	ground volumetric water content	(m^3/m^3)
C_G	ground volumetric heat capacity	($MJ/m^3\cdot K$)
Ta	ambient air temperature	(°C)
T_P	temperature penalty	(°C)

CHAPTER 1 – INTRODUCTION

While this study delivers a deeper insight of heat pump performance in the Irish Maritime climate, this chapter reviews the wider energy consumption, policy and legislative trends that motivated this study and underpin its relevance in a more sustainability conscious world.

Over the past two decades there has been mounting concern about the potential negative environmental, economic and sustainability impacts of over reliance on fossil fuel based development. The following sections review the issues that face future worldwide energy provision and the international legal frameworks that encourage a shift towards a more energy efficient and sustainable future.

1.1 ENERGY POLICY

Energy derived from hydrocarbons constitutes 87% of the total energy consumed worldwide (IEA, 2007). The continued increase in global energy demand is putting further pressure on energy supply. Coupling a growing demand with potentially diminishing resources generated a dramatic price rise in 2008 which saw crude oil reach an unprecedented $147 per barrel (PRRC, 2008). This also triggered a significant and unexpected negative impact on the supply and cost of food. This dramatic crude oil price rise coincided with a levelling off in production of key oil producing nations (IEA, 2008) sparking concern about the arrival of 'peak oil' (ASPO, 2005; Stevens, 2008). Responding to heightened concerns about diminishing worldwide fuel stock, along with CO_2 driven climate change, worldwide attention has steadily refocused on achieving more sustainable energy solutions (ASPO, 2005; IEA, 2007; IPCC, 2007; IEA, 2008).

The United Nation's *Economic and Social Council* report on sustainable development sets out the climate change policy options and actions required to expedite implementation (ECOSOC, 2006). These options reflect other international reports that recommend urgent action to stave off the worst effects of climate change in the future.

Primarily, international concern has focused on the potential harmful effects of unconfined burning of fossil fuels on the environment and a continuous assessment of the risk is being managed by the Intergovernmental Panel on Climate Change (IPCC). The most recent IPCC report emphasised the need for immediate global action to alleviate the worst effects of climate change (IPCC, 2007).

Explicit action on this issue has come primarily in the form of commitments made under the *Kyoto Protocol* within the United Nations Framework Convention on Climate Change (UNFCCC, 2007). This has helped reduce CO_2 emissions through its 1992 Protocol and to highlight the environmental advantages of sustainable energy technologies. Reflecting 15 years of unprecedented economic growth, Ireland's greenhouse gas emissions have grown from 53.37 Mt of CO_2 per year in 1990 (DOE, 2007) to 69.95 Mt in 2009 (SEI, 2009), running 24.5% above 1990 levels (SEI, 2009). Ireland, under the Kyoto Protocol agreed to limit its increase of CO_2 production levels to just 13% on 1990 levels between 2008 and 2010 and has now to purchase the shortfall of carbon credits on the international market. Besides incurring the economic cost of purchasing carbon credits, there is a depletion of indigenous gas and pat reserves which leaves Ireland in the vulnerable position of importing 89% of its primary energy, up from 68% in 1990 (SEI, 2009).

The United Kingdom's Stern Review on the economics of climate change outlined the need for the world to take economic action now and move to a low-carbon economy in moderate steps to reduce the effects of climate change. It states that "*the benefits of strong, early action on climate change outweigh the costs*" and concludes that a '*business as usual*' stance on the issue will only make the necessary economic changes in the future much harder to take, causing a worldwide recession and possibly being too late to reverse the climate change effects (Stern, 2006).

Climate change, which is understood to be triggered by the unnatural build up of greenhouse gasses in the Earth's atmosphere, such as CO_2, creating an imbalance in the Earth's heating by reducing it's capacity to re-radiate solar energy back into space, is becoming an accepted challenge internationally. It is believed that associated global warming is "*very likely*", with 90% confidence, due to human activity (IPCC, 2007). Although climate scientists agree the global average surface temperature has risen over the last century, a vocal element within the scientific community still question the validity of the modelling techniques and inputs which attribute global warming to anthropological actions (Singer and Avery, 2005; Khilyuk and Chilingar, 2006). However, heating (and cooling) systems must be cognisant of the predicted rise in temperature and designed accordingly.

The European Commission has responded to the international call for action through policy position papers such as *An Energy Policy for Europe* (EC(a), 2007), White Paper for a community strategy (EC(a), 2006), Green Paper *A European Strategy for Sustainability, Competitive and Secure Energy* (EC(b), 2006), *Renewable Energy Roadmap* (EC(b), 2007) along with the ambitious *Action Plan for Energy Efficiency* (EC(c), 2006). These publications all recognise the need to achieve security of energy supply by means of diversification, increased energy utilisation efficiency along with reaffirming key international commitments like the reduction in the amount of greenhouse gas emissions by 30% on 1990 levels by 2020 under the *Kyoto Protocol*. It also outlined a doubling of renewable energy domestic supply to 12% by 2010 and encouraged the utilisation of sustainable energy technologies. The White Paper stated that there will be active promotion of solar geothermal and heat pump heating systems with a tripling of the total installed heat pump capacity from its 1995 level delivering up to 2.5 GW_{th} across Europe. In a government national renewable energy action plan, the Republic of Ireland responded in 2010 by committing to achieve 40% renewable energy derived electricity consumption and 12% renewable derived thermal energy consumption by 2020 (DCENR, 2010).

The implementation of the *Energy Performance in Buildings Directive* (EPBD) in Ireland during 2007 raised the profile of energy use within the built environment (EU, 2002). This European Commission directive enables sustainable energy technologies to benefit from their low emissions credentials by positively affecting the price of buildings through an enhancement of the Building's Energy Rating (BER) and was further reinforced by revision of the Building regulations where a renewable energy thermal supply of $14kWh/m^2$ per annum was made mandatory for new dwellings (DOE, 2007). Heat pumps are one such technology that contributes to a lower rating. The governmental policies now established will have a twofold impact on the use of heat pumps in Ireland. Firstly, it is going to impact on the use of sustainable energy heating systems as fossil fuel based heating systems are rated lower on BER and do not attract financial incentives. Secondly, by increasing the percentage of sustainably generated electricity from approximately 5% in 2007 to 40% by 2020 in will lower the Seasonal Performance Factor (SPF) at which heat pumps can be considered CO_2 neutral. This SPF currently stands at 2.5 with 5% green energy, which means that the portion of the SPF above 2.5 is deemed renewable. This will reduce to 2.0 or below when green energy increases market share to 40% of delivered electricity.

1.2 HEAT PUMP OPERATION, UTILISATION AND CLIMATE SENSITIVE SYSTEM DESIGN

The term "heat pump" encompasses numerous products that offer the capacity to move heat. All heat pumps share two characteristics, the heat "source" and "sink". When a heat pump operates in heating mode the building or application to which the heat is delivered is considered the heat sink, while the heat source describes the place from which the heat energy is taken, for example, the ground. As indicated in Figure 1.1, Ground Source Heat Pumps (GSHP) extract heat from the ground, ground water, lakes or rivers. Whereas Air Source Heat Pumps (ASHP) extract thermal energy from the ambient outdoor air (Kavanaugh, 1991; IGSHPA *et al.*, 1997; ASHRAE, 2003 - 2006; Twidell and Weir, 2006).

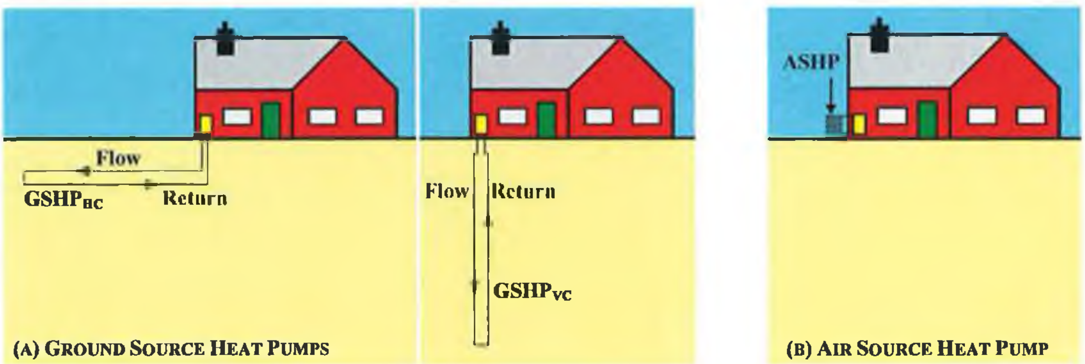


FIGURE 1.1 TYPICAL HEAT PUMP COLLECTOR CONFIGURATIONS.

Instantaneous heat pump efficiency is referred to as the Coefficient of Performance (COP) which is a measure of the useful thermal output (Q_{OUT}) divided by the electrical power consumed by the compressor and circulation pump (W_{IN}). An idealised heat pump thermodynamic cycle is illustrated in Figure 1.2 and from which the theoretical maximum heat pump thermal performance in heating mode is called the *Carnot Coefficient of Performance, COP_{Carnot}*:

$$COP_{Carnot} = Q_{out} / (Q_{out} - Q_{in}) \qquad \textit{Equation 1.1}$$

or

$$COP_{Carnot} = T_C / (T_C - T_E) \qquad \textit{Equation 1.2}$$

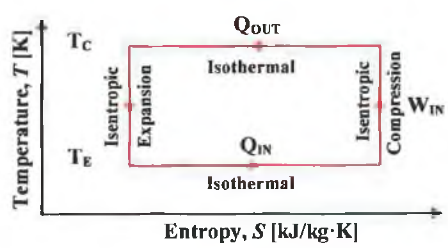


FIGURE 1.2 HEAT PUMP TEMPERATURE-ENTROPY (*T-S*) DIAGRAM.

where T_C is the condenser temperature and T_E is the evaporator temperature in degrees Kelvin. The difference between the source temperature (T_E), and the sink temperature (T_C), is referred to as the temperature lift (T_C - T_E) and as indicated in Figure 1.3(a) temperature lift is inversely proportional to the COP. Equation 1.2 is only theoretical as it assumes that the compression and expansion processes are isentropic and that the energy transfer across the evaporator and the condenser are achieved isothermally. Since this Carnot COP is only theoretically possible, actual heat pump *effectiveness* (ε) can be defined as:

$$\varepsilon = COP_{Actual} / COP_{Carnot} \qquad \textit{Equation 1.3}$$

Actual COPs for heat pumps typically range between 2 and 5 with corresponding effectiveness between 0.3 and 0.5. Examples of heat pump COP and effectiveness are shown in Figure 1.3(a).

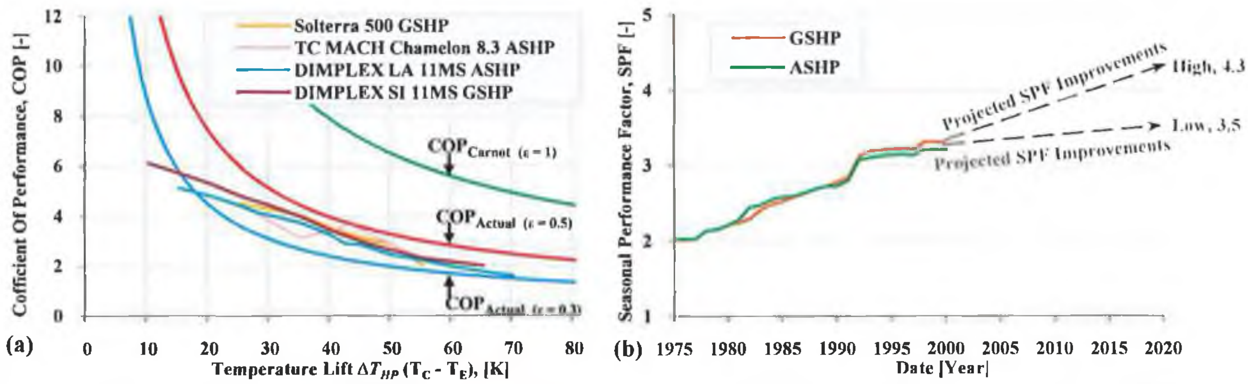


FIGURE 1.3 (A) MANUFACTURER DEFINED EXAMPLES OF HEAT PUMP COP VERSUS TEMPERATURE LIFT FOR GSHP AND ASHPS AND (B) SPF IMPROVEMENTS OVER PAST 25 YEARS (LAZZARIN, 2007), WITH POTENTIAL FUTURE IMPROVEMENT TRENDS.

The efficiency of a heat pump over a heating season is referred to as the Seasonal Performance Factor (SPF), details of which are provided in Appendix A. Figure 1.3(b) highlights that there has been a steady increase in the SPF of both ASHP and GSHP systems over the past 30 years, with recent (2000) values 60% above those in 1976. Projecting past performance trends forward to 2020 indicate that SPFs ranging between 3.5 and 4.3 should be possible.

Domestic heat pumps in Ireland typically range between 10 to 14 kW_{th} and up to 100 kW_{th} or greater for commercial buildings. Heat pumps have generally been used within the built environment for air conditioning (cooling) and refrigeration, but heating is now becoming a mainstream application in Ireland and indeed worldwide.

With around 40% of Europe's primary energy being used for heating and cooling in the built environment (SEI, 2006; EHPA, 2008) the issue of energy efficiency and energy reduction requires a focused and concerted effort in order to reduce fossil fuel dependence, energy consumption and CO_2 emissions. When compared with conventional heating systems heat pumps offer lower CO_2 emissions, higher reliability, 25 year life expectancy, require no boiler or fuel tank, generate no combustion, explosive gases or pollution within the building (Rawlings *et al.*, 2004).

Heat pump system installations have seen a global annual growth in the range of between 20% and 30% (Bose *et al.*, 2002) with over 130 million systems installed worldwide delivering over 1,300 TWh (MEZ, 2006). In Ireland, heat pumps for domestic heating were introduced in the 1990's, reached 1% of heating system market in 2004 (O'Connell, 2004) and 3% of market (2,500 units) in 2007 (O'Connell and Cassidy, 2003; O'Connell and Cassidy, 2004; SEI, 2007b). Figure 1.4 shows heat pump market share in selected EU countries.

FIGURE 1.4 MARKET SHARE OF HEAT PUMPS FOR DOMESTIC HEATING FOR NEW & RENOVATED HOUSES IN SELECTED EUROPEAN COUNTRIES IN 2007 (SEI, 2007B; EHPA, 2009).

While the European heat pump market has seen strong growth, market penetration has been dented in some countries by poor quality heat pumps and installations (AR(b), 2004). Concern therefore exists as to the depth and level of expertise that underpins the design, selection and installation of heat pump technologies as the industry has developed over a relatively short timeframe and was not developed in tandem with standardised installer training.

Figure 1.5 illustrates the interaction and interdependency between the climate, built environment and heat pumps.

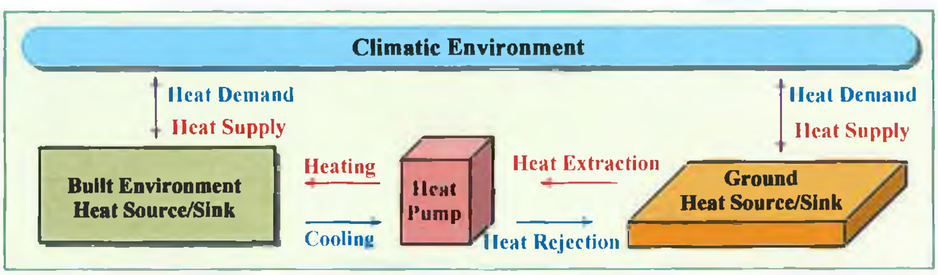


FIGURE 1.5 INTERDEPENDENCE BETWEEN BUILDING DEMAND, CLIMATE AND HEAT SOURCE/SINK.

There is a general lack of information available to cater for the specific needs of heat pump installation and operation in the Maritime Climate. Resources such as the BSRIA Technical

Note on Ground Source Heat Pumps (1999) deployed in the United Kingdom concludes that efficiencies are inherently higher in ground source heat pumps than those of air source heat pumps because of lower air temperatures at times of peak demand (Rawlings, 1999). However, Rawlings relied heavily on data from continental climate regions (CEN, 1994; Kavanaugh and Rafferty, 1997), and Dumont and Frere (2005) cautioned about translating results from one climate region to another. Moreover, no specific research is referenced in Rawlings' report stating that the ASHP is less efficient than that of a GSHP operating in either the continental or Maritime climate.

Arsenal Research, a recognised test laboratory for certifying heat pump performance, were commissioned by Sustainable Energy Ireland in 2003 to assess the status of heat pump utilisation in Ireland. The report emphasised the requirement for heat pump research and development in Ireland, stating that there is a lack of "*neutral, practical information*" as "*there is only information from some manufacturers and importers available*" and this is seen as a severe barrier to the long term viability and growth of heat pumps in Ireland (AR(c), 2004). This report also identifies a need for training courses and training facilities with practical test equipment (AR(a), 2004).

Indeed the lack of understanding of heat pump performance variation with climate has recently been addressed by the *Intelligent Energy Europe* funded SEPEMO-Build Programme (Anon., 2010) which monitors heat pump performance operating in real applications across various European countries. Other recent *Intelligent Energy Europe* funded projects such as the GROUND-REACH, GROUND-HIT and SMART-HEAT have also begun to address the deficit in real application heat pump performance evaluation under various European climates. Indeed there has also been a call from *Sustainable Energy Ireland* (SEI) to investigate the seasonal performance of heat pumps under the Maritime Climate of Ireland (Durkan, 2007).

Figure 1.6 presents 30 years averaged air temperatures, for six widely dispersed locations at similar latitudes that experience continental and Maritime climates. Three striking effects are evident: i) the amplitude of the annual air temperature fluctuation in Maritime climates is less than half that of continental regions that generate greater demand for heating in winter and which turns potentially to cooling demand in the summer; ii) the average winter air temperatures show a 6°C variation between Maritime and continental climates, with Maritime regions recording the highest winter temperature of +5°C and as a result; iii) Maritime regions show a +2°C higher average year-round air temperature. While all above points are positive for the $GSHP_{HC}$, the latter suggests the horizontal collector will, for a given collector depth, operate in the heating mode for longer periods and at a higher temperature and this should lead to a higher SPF in Maritime regions.

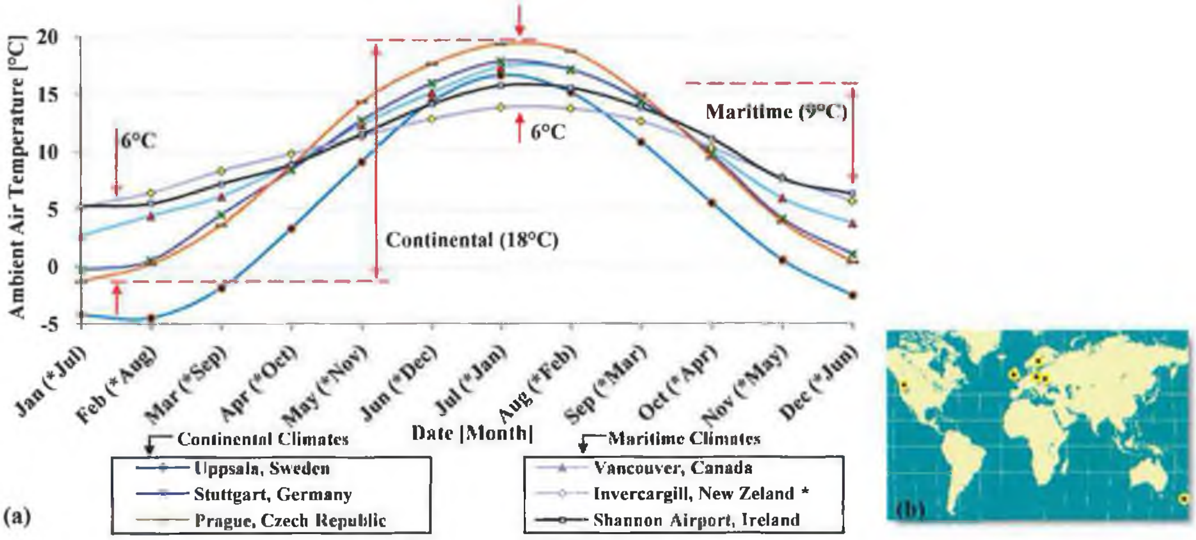


FIGURE 1.6 CONTINENTAL V MARITIME CLIMATES (A) 30 YEAR AVERAGE MONTHLY AIR TEMPERATURES FOR SIX WIDELY DISPERSED LOCATIONS THAT SHARE A SIMILAR LATITUDE (WORLD CLIMATE, 2007) AND (B) GEOGRAPHICAL SPREAD OF SITES IN (A).

The low peak summer temperature within the Maritime climate drives a space heating demand for as much as 10 months of the year, and is clearly illustrated in Figure 1.7 and contrasts with the more extreme continental climate where the peak heating demand can be up to 50% more than that of the Maritime Climate.

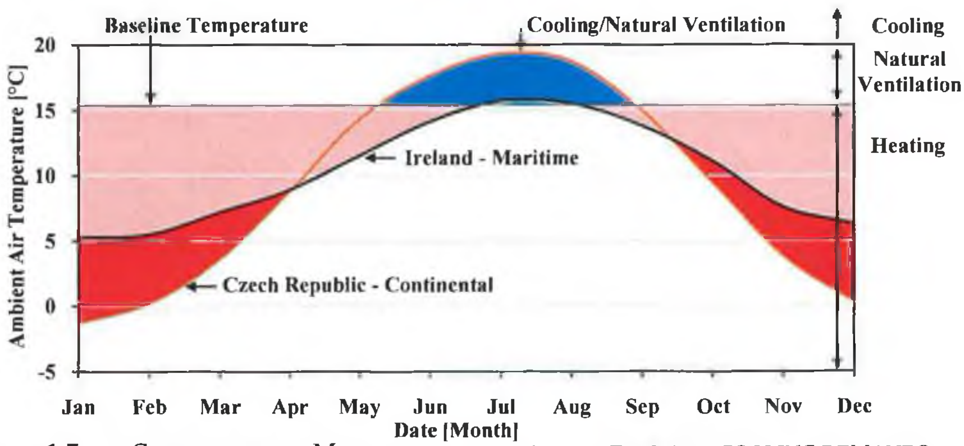


FIGURE 1.7 CONTINENTAL V MARITIME CLIMATES – HEATING AND COOLING DEMANDS.

Early research into the performance of heat pumps operating under the Irish Maritime climate by O'Conner aptly suggested that while the capital cost of these systems was prohibitive it identified the need "*.... for further research to improve domestic heat pump performance to suit Maritime climates*" to avoid over-dependence on oil (O'Connor *et al.*, 1982). Similar calls to investigate the interaction between the horizontal collector, the soil/air interface and the climate has been made by Bose *et al.* (2002). ASHRAE also recently acknowledged the need for a comprehensive evaluation of horizontal collector GSHPs operating in heating dominated climates such as the Maritime climate (ASHRAE, 2003 - 2006). It is therefore envisaged that this study, referred to as '*HP-IRL*', should determine the impact of the Irish Maritime climate on the performance of two heat pump technologies shown in Figure 1.1.

While horizontal collectors are less expensive to install, vertical collectors occupy less surface space. Air source heat pumps are the easiest and cheapest to install but the technology is generally more expensive that its GSHP counterpart due to the larger thermal capacity requirement. Hence, for the purpose of evaluating the viability of heat pump technology in Ireland this *HP-IRL* study focuses on the three most popular heat pump sources; vertical ($GSHP_{VC}$) and horizontal ($GSHP_{HC}$) collector ground source heat pumps and air source heat pumps (ASHP) which collectively accounts for 98% of the Irish market as shown in Figure 1.8.

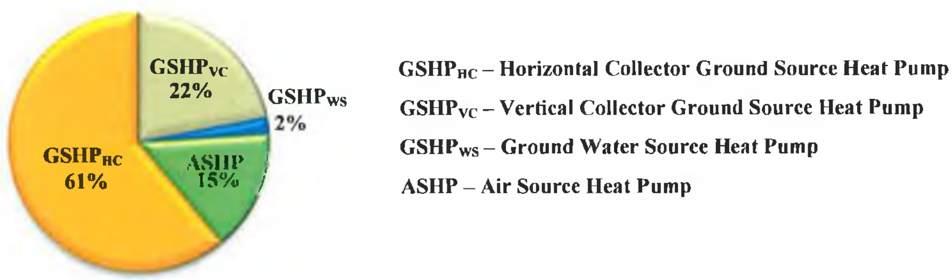


FIGURE 1.8 MARKET SHARE OF DIFFERENT HEAT PUMP TYPES IN IRELAND (SEI, 2007B; JOLLEY, 2007).

The market share for domestic dwelling is still highly biased towards the GSHP due to its perceived superior performance over ASHPs. Within GSHPs horizontal collectors are more popular than vertical collectors due to the reasonably shallow depths required to locate the pipes which reduce installation costs to less than 50% of vertical collectors.

Heat pump collector design has undergone enormous changes over the past decade with the development of better materials (grouts), configurations (Slinky^{TM}, double U-tube) and understanding of the heat transfer dynamics. However, the majority of this work has been directed towards the improvement of vertical collector systems. By comparison to vertical collectors, the analysis of horizontal collector systems is often seen as too small in capacity, less predictable and harder to analyse using short-term analysis methods to warrant thorough investigation. Due to the $GSHP_{HC}$'s efficiency having a close relationship with the climate, as opposed to the $GSHP_{VC}$, the design cannot be easily simplified into general rules of thumb without recourse to the specifics of the climate. This in effect demands a climate sensitivity analysis to be performed on $GSHP_{HC}$ operation that can lead to the development of climate or region specific design rules of thumb.

In summary, the drivers for this *HP-IRL* study can be summarised as follows:

- calls for neutral, practical experimental data on GSHP system performance in the Maritime climate
- lack of information for heat pump performance in Maritime climates
- lack of climate sensitive collector design and operational guidelines
- need to support the Irish heat pump emerging market with design information and training facilities
- need to identify diversifying applications for heat pump technologies so that the utilisation of this sustainable technology can be increased
- improve the overall system performance in tandem with heat pump performance improvements through source side management
- maintain heat pump cost benefits with regard to changing electricity supply, both from an environmental and cost perspective

1.3 AIMS AND OBJECTIVES

This *HP-IRL* study aimed to develop a comprehensive test facility for the functional characterisation of GSHP and ASHP technologies in the Irish Maritime climate to support climate sensitive performance analysis and collector design with source side management. This was to be accomplished by delivering the following objectives:

- Conduct a literature review to examine the following:
    - international heat pump test standards and characterisation facilities
    - climate classifications and factors that affect heat pump performance
    - operational dynamics of heat pumps and collector regions
    - methods of assessing heat pump performance and design of experiments
    - heat pump collector design criteria and best practice
- Design, build and commission a comprehensive test facility that would allow the performance of functioning $GSHP_{HC/VC}$ and ASHP technologies to be established within the Maritime climate of Ireland.
- Perform extensive experimental testing of two heat pump technologies and three collector types to evaluate different interrelated characteristics such as:
    - impact of variable weather, ground and operational conditions on all three heat pump's performance
    - impact of multi-year thermal extraction on source temperatures

  - influence of heat pump duty on ground temperature drawdown, thermally affected zones around the collector pipes and recovery rates, as well as the influence of ground cover
- Generate experimental data for benchmarking numerical models
- Investigate the potential for horizontal collector performance improvements in Maritime climate conditions with variation in collector length, depth, surface covers and configurations
- Development of new climate sensitive design and operational parameters for horizontal collectors
- Examine the economic and environmental aspects of heat pump utilisation in Ireland
- Promotion of project through publications, training material and GMIT's website
- Document study in the form of a PhD thesis

1.4 METHODOLOGY

The methodology employed during *HP-IRL* was influenced by published literature, industrial interaction and experimental evidence.

Literature review

Established current state of the art in heat pump technology by means of a comprehensive literature review, revealing past and present research focus and approach.

Industry contact

Interaction with the following members of the energy industry helped refine the research priorities and objectives; the International Ground Source Heat Pump Association (IGSHPA), the International Geothermal Association (IGA), the European Heat Pump Network (EHPN), the European Heat Pump Association (EHPA), the Geothermal Association of Ireland (GAI), Renewable Energy Skillnet (Heat Pump Installer Group) and Ireland based sustainable energy companies such as Dunstar Ltd. and Energy Master Ltd.

Empirical analysis

Practical research carried out by the installation, testing and monitoring of a comprehensive weather station, two ground source heat pumps (horizontal and vertical collector) along with the installation, testing and monitoring of an air source heat pump as part of the *HP-IRL* study.

Monitoring of the two heat pump technologies and three collector types was carried out through the use of high accuracy temperature sensors (both heat pump temperatures and ground temperatures), ground moisture sensors, flow-metering and electricity consumption. Data retention was provided via a Data Acquisition system (DAQ) which not only recorded data but makes certain data available for display through the World Wide Web.

Finally, an analysis of the capabilities of the various heat pump sources carried out through a thorough evaluation of the recorded data after systematic testing. The analysis of the heat pumps was benchmarked against other heating systems for both cost and environmental impact. Furthermore, the sources of energy and the utilisation of heating demand were evaluated against that of differing climates.

1.5 THESIS STRUCTURE

Chapter 1 – Introduction. This chapter identifies the drivers and scope of this *HP-IRL* study.

Chapter 2 – Literature Review. This chapter presents the relevant literature including the characteristics of world climates, the potential impact upon heat pump performance and international studies that have informed the design and implementation of the *HP-IRL* test facility.

Chapter 3 – Experimental Investigation. This chapter details the *HP-IRL* test facility, including the weather station, vertical, horizontal ground and air source heat pumps, the 111 instruments and sensors used to monitor heat pump performance along with the data acquisition system. This chapter also presents the experimental procedures and rationale for the 22 number (747 days) heat pump test programme.

Chapter 4 – Horizontal Collector: Experimental Analysis. This chapter presents the results of nine test periods conducted over three years, a comprehensive analysis of the test data and examines the behaviour of the thermal interaction between the Maritime climate, ground and collector.

Chapter 5 – Horizontal Collector: Modelling and Design. This chapter develops a model for horizontal collector design and deploys if to design a split level horizontal collector system, optimising system performance from a Maritime climate perspective.

Chapter 6 – $GSHP_{VC}$ and ASHP Performance Evaluation. This chapter evaluates the performance of the vertical collector ground source heat pump over three years, examining the effect of the Maritime climate, ground type, geothermal gradients, of multiple year operation on system performance. This chapter also evaluates the performance of an air source heat pump over a six month test period, examining the relationship between heat pump system performance and climate.

Chapter 7 – Techno-Economic Evaluation. This chapter provides a comparison of the three collector types in terms of thermal provision, environment aspects of heat pump utilisation in Ireland, along with an economic evaluation of heat pump systems using a market survey.

Chapter 8 – Conclusions and Recommendations. In this chapter conclusions of the research are presented, along with recommendations for future work.

Chapter 2 – Literature Review

The following literature review was undertaken to reveal the impact of climate on heat pump and collector performance, the extent of previous heat pump studies in Maritime climate regions and to help define test methodologies for such investigations. It draws on over 50 previous studies and not only presents key findings but also the experimental methodologies employed, many of which influenced the design and implementation of the *HP-IRL* study.

Since many of the cited studies were undertaken in different climatic regions the first sub-section identifies the most recent global climate classification to gain appreciation for the range of climates investigated and to identify which studies were conducted in a similar climate to that of Ireland.

2.1 Climate classification

Climate classification is used to identify geographical locations that experience similar climate characteristics. Hence, all locations within a specific climate region should experience weather patterns that generate similar heat pump demands and collector thermal charging and discharging capacities. In doing so, this allows one to identify different regions with similar climates, which removes the climate factor or vice versa.

Climate is a mean of the weather elements and is represented universally to be the mean over a thirty year period. Dry Bulb Temperature (DBT) and to a lesser degree rainfall are the most common parameters used to succinctly describe a climate. The *Köppen* system of climate classification shown in Figure 2.1 was developed in 1884 and is the most commonly used climate classification today. It is largely quantitative and is based primarily on the level of precipitation and the ambient air temperature, both annually and seasonally (Koeppe and De Long, 1958).

However, this classification lacks resolution. For instance, Ireland's climate falls into the category *Cfb* which identifies Ireland as having a temperate climate with cool summers (averages between +10°C and +22°C for the hottest months), mild winters (averages between -3°C and +18°C for the coldest months), with all months moist. This climate, as indicated in Figure 2.1, covers the majority of central Europe. Clearly, heating demands vary considerably across Europe and the 21K range of winter time ambient outside air temperatures does not offer enough accuracy for the current purpose.

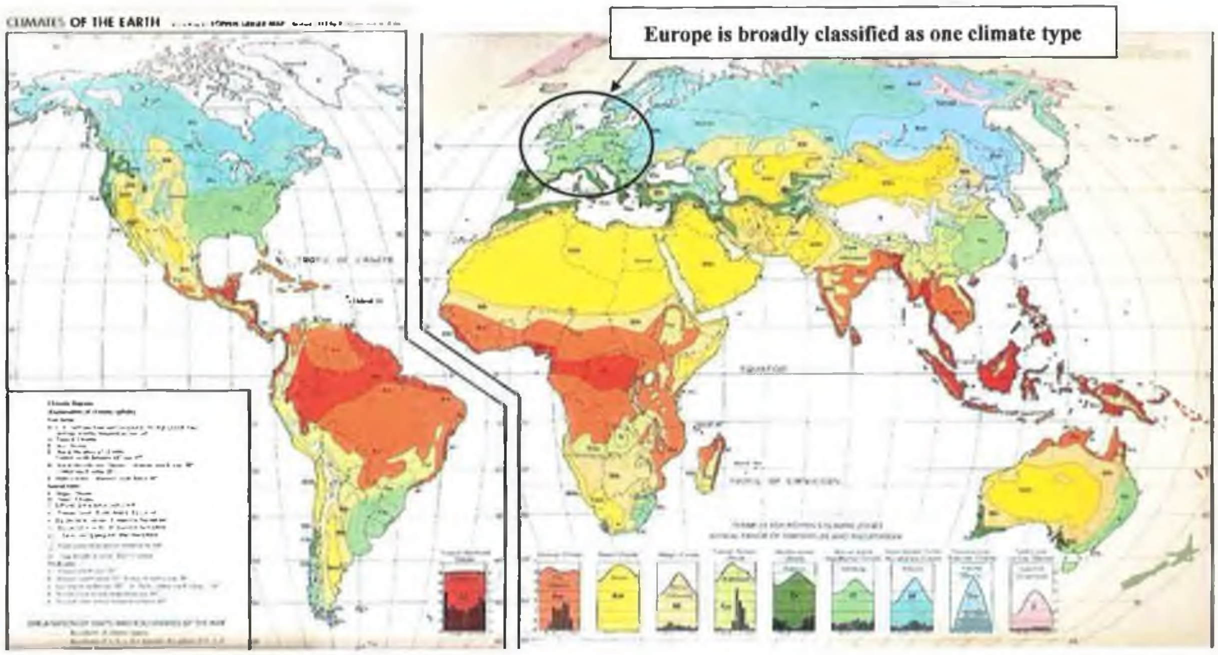


FIGURE 2.1 KOPPEN CLIMATE CLASSIFICATION OF THE WORLD IN 1884 (KOEPPE AND DE LONG, 1958).

A more recent climatic evaluation published by *Koeppe* in 1958 and shown in Figure 2.2 offers greater geographical resolution of climates (Koeppe and De Long, 1958). Ireland is identified as having a '*Cool Marine*' climate consisting of relatively mild, almost perennially moist winter conditions with cool, cloudy summers, with average ambient outside air temperature ranging between -1°C and +7°C for the coldest months.

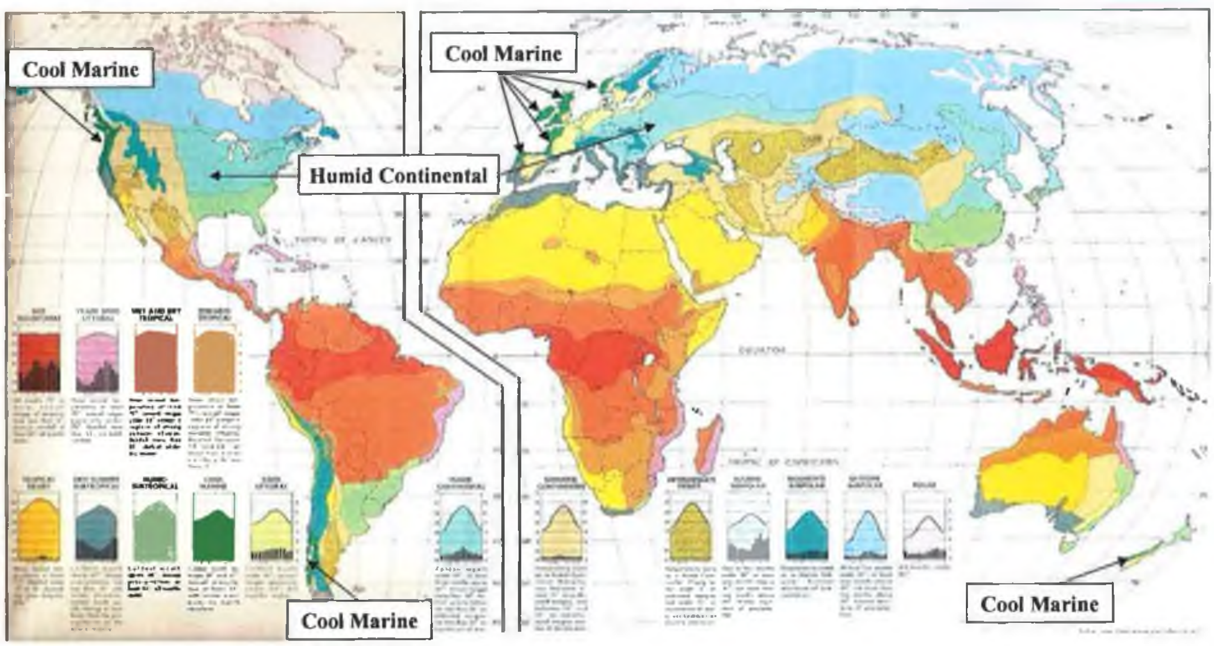


FIGURE 2.2 KOEPPE CLIMATE CLASSIFICATION OF THE WORLD IN 1958 (KOEPPE AND DE LONG, 1958) – LARGER VERSION AVAILABLE IN APPENDIX B.

This climate classification also includes the western seaboards of the United Kingdom (UK), France, Portugal, Canada, the United States of America (USA), Norway, Chile and the southern island of New Zealand and it is estimated that over 100 million people experience this climate worldwide. The climatic parameters of each climate class shown in Figure 2.2 are outlined in Appendix B.

While Ireland's climate is defined as '*Cool Marine*' it is typically referred to as 'Maritime' and this tern is used throughout this thesis.

The warm North Atlantic Drift sea current, which passes Ireland's west coast, has the effect of elevating the average ambient air temperature in Ireland during the winter by approximately 5K and conversely reduces the summer temperature by approximately 5K. It is therefore an influential stabilising mechanism on the Irish climate, moderating ambient air temperature fluctuation, which contrasts with more pronounced temperature fluctuations of the continental climate (Koeppe and De Long, 1958; McElwain, 2004). As a result it is difficult to compare heating demand, heat pump collector thermal dynamics and Seasonal Performance Factor (SPF) between these two geographically close regions. The climate distinctions recognised in Figure 2.2 are an important first indicator of these distinctions.

Heat pump operational efficiency can vary from climate to climate and will be a major determining factor in the type and sizing of system used (Fairey *et al.*, 2004). Indeed previous studies that attempted to evaluate building heat loss have highlighted how the accuracy of many numerical methods and models, applied homogenously across various climates, have been questioned, and generally led to not only climatic but also regional specific evaluation methods being adopted (van Hoof and Hensen, 2007). Hence *HP-IRL* sought to establish if similar adjustments were required for heat pump performance in the Maritime climate.

2.2 CLIMATE SPECIFIC HEAT PUMP STUDIES

This section identifies research undertaken over the previous three decades to evaluate and compare the performance characteristics of ground and air source heat pumps within specific climates. It is confined to studies that focused on heat pump collector performance indicated by COP and SPF of those whose collector was exposed to the climate, as opposed to laboratory tests that focus on specific aspects of heat pump design such as refrigerants, heat exchanger and compressor design or control algorithms.

In keeping with the climate classifications in Figure 2.2, Figure 2.3 illustrates the geographical spread of the twenty-one climate specific experimental heat pump studies reviewed.

FIGURE 2.3 GEOGRAPHICAL SPREAD OF TWENTY-ONE EXPERIMENTAL HEAT PUMP STUDIES, WITH DETAILS PRESENTED IN TABLE 2.1.

While eight climate types are represented the majority of studies report data recorded over periods of less than one year. Nine studies present findings based on data recorded over more than one year and of those four were undertaken in the *Humid Continental* climate, with one each in *Dry Summer Subtropical, Semi-arid Tropical, Cool Littoral, Moderate Sub-polar* and *Cool Marine* climates.

Comparison is further complicated by the range of technologies studied, differing project aims, operating conditions, study length and diverse climatic influences. However, in an effort to facilitate comparison a profile of each study is presented in Table 2.1 to define the year of study, heat pump technology analysed and key findings. Table 2.1 also details five laboratory based studies, five numerical and two techno-economic studies in addition to the twenty-one climate specific experimental studies in Figure 2.3.

Table 2.1 shows that the concentration of work has focused on establishing heat pump performance in continental and subtropical climate regions with nine each. While nine (27%) of the thirty-three studies shown in Table 2.1 are based in the *Cool Marine* (Maritime) climate, they are limited in terms of length of experimental investigation and do not evaluate the interaction between the climate, ground and heat pump performance.

TABLE 2.1 SUMMARY TABLE OF KEY HEAT PUMP STUDIES CONDUCTED BETWEEN 1981 AND 2009

No.	Researcher (Year)	Climate Type (Country)	ASHP	$GSHP_{HC}$	$GSHP_{VC}$	WSHP	Approach/Study Parameters
1	Metz, P.D. (1981)	Humid Continental (USA)		E			Study conducted over 1 year. Horizontal collector submerged at 2m depth. Only heat pump SPF analysis performed.
2	O'Conner and McGovern (1982)	**Cool Marine** (Ireland)	E				Trials conducted between 1975 and 1979 revealed COP's in the range of 1.2 to 1.6 for an air-to-air heat pump and between 2.26 and 2.47 for an air-to-water heat pump.
3	Parker and White (1982)	Humid Continental (USA)	E	E			Study conducted over 2 years. Horizontal collector submerged at 2m depth. Study did not provide detail on performance evaluation.
4	Rosell *et al.* (1983)	**Cool Marine** (Ireland)	E				Study conducted over six months, recording a SPF of 2.5 with no other parameter results.
5	Reistad, G.M. (1984)	**Cool Marine** (USA)	E			E	Study conducted over 3 months. Climate similar to Ireland's where testing concluded that an ASHP has the potential to be a cost effective system for mild climate regions.
6	Mei, V.C. (1986)	Humid Continental (USA)		N			Study of the effects of soil freezing around a horizontal collector. Study highlighted the need for horizontal collectors to be designed to operate on and below freezing point in continental climates.
7	Morehouse *et al.* (1992)	Humid Subtropical (USA)	N	N			Study focused on ground thermal recovery after heat extraction.
8	Piechowski, M. (1996)	Humid Subtropical (Australia)		E N			Study conducted over 2 months. Horizontal collector submerged at 1.8m depth. Study evaluated the thermal storage capacity of the ground in cooling mode.
9	Mihalakakou *et al.* (1996, 1997)	**Cool Marine** (Ireland)		E			Study conducted over six months. No heat pump performance testing. Identified the characteristics of heat absorption through various ground covers. This experiment showed a source temperature increased under a bare surface (asphalt) being up to +1°C higher than that under short grass.
10	Kent, E.F. (1997)	Dry Summer Subtropical (Turkey)	E				Evaluation of an ASHP system over 8 months of the heating season. System performance was highly dependent on air source temperature.
11	Leong *et al.* (1998)	Humid Continental (USA)		N			Study into the effect of soil moisture content on heat pump performance.
12	Eugster and Rybach (2000)	Moderate Subpolar (Switzerland)			E N		Study into the effect multiple years of thermal extraction on borehole source temperature.
13	BRECSU (2000)	**Cool Marine** (UK)		E			Study conducted over 1 year. Horizontal collector submerged at 1m depth. No details on exact location of study within the UK. Only heat pump SPF analysis presented.
14	De Swardt and Meyer (2001)	Semi-arid Tropical (South Africa)	E N			E N	Study conducted over 1 year. Measurements showed that air source temperature varies greatly. Historical climate data utilised in simulations was not detailed enough to show effect of variation on COP.
15	Ito *et al.* (2001)	Humid Continental (Japan)	E N		E N		Laboratory based experimental study that relied on historical climate data to simulate the source temperatures and duty cycles. Study combined a vertical collector and ambient air collector. Numerical study assumed a fixed drawdown of the ground source temperature.
16	Popiel *et al.* (2001)	Humid Continental (Poland)		E			Study conducted over 1 year, and evaluated (shallow) ground temperatures versus climate. No heat pump performance was presented.
17	Li *et al.* (2003)	Humid Subtropical (China)	T	T	T		Techno-economic analysis. Determined that the $GSHP_{VC}$ system best suited this climate.
18	Lam and Chan (2003)	Trade Wind Littoral (China)	E			E	Study conducted over 6 months. Very detailed account of the experimental design and followed international standards. Heat pump SPF analysis only.
19	Hepbasli *et al.* (2003)	Dry Summer Subtropical (Turkey)			E		Study conducted over 4 months. Heat pump system generated low COPs due to collector sizing error.

TABLE 2.1 CONTINUED.

No.	Researcher (Year)	Climate Type (Country)	ASHP	$GSHP_{BC}$	$GSHP_{VC}$	WSHP	Approach/Study Parameters
20	Huang and Murphy (2003)	Humid Continental (USA)			N		Numerical evaluation of heat pump performance with variable duty cycles. Highlighted need to carefully balance vertical collector thermal extraction/injection if conduction is the only form of heat transfer in the ground (i.e. no groundwater movement)
21	Guoyuan *et al.* (2003)	Humid Subtropical (China)	E				Experimental (laboratory) investigation of an ASHP with a two-stage compressor and simulated outside ambient air temperatures. Study focused mainly on heat pump performance evaluation with minimal climate evaluation.
22	Al-Huthaili, S. (2004)	**Cool Marine** (UK)			E		Study conducted over 2 months. Only heat pump COP analysis presented.
23	Inalli and Esen (2004)	Semi-arid Continental (Turkey)		E			Study conducted over 6 heating season months. SPF was recorded for collector depths of 1m and 2m.
24	Marcic, M. (2004)	Humid Continental (Slovenia)	E				Study conducted over 9 years. (1989 – 1998). The ASHP worked as part of a bivalent heating system, with a condensing oil boiler.
25	Romero *et al.* (2005)	Dry Summer Subtropical (Spain)	E		E		Study conducted over 1 year. Comparative study revealing a higher efficiency performance by the $GSHP_{VC}$ over the ASHP. Study evaluated heat pump COP and climatic conditions.
26	Lindholm *et al.* (2005)	Cool Littoral (Sweden)	N T		N T		Study revealed a novel design utilising both the air and ground as heat pump dual sources. No experimental heat pump performance data.
27	Blanco-Castro *et al.* (2005)	Dry Summer Subtropical (Spain)	E				Experimental (laboratory) investigation of ASHP operating in a *Dry Summer Subtropical* Mediterranean Climate. Laboratory experiment in a climate controlled chamber using historical climate data (30 year average).
28	Dumont and Frere (2005)	Cool Littoral (Belgium)		E			Study conducted over nine months. Horizontal collector submerged at 0.6m depth. COP and ground temperature within the collector area was recorded.
29	Karlsson and Fahlen (2007)	Cool Littoral (Sweden)		E N			Laboratory based experimental investigation. Study looked at demand side capacity controlled ground source heat pump performance. Unclear what ground and collector fluid temperatures were used in the study.
30	Hewitt and Huang (2008)	**Cool Marine** (Northern Ireland)	E				Study evaluating the performance of a novel ASHP evaporator. Showed the potential for ASHP system to operate effectively in the Maritime climate.
31	Mustafa Omer, A. (2008)	**Cool Marine** (UK)	T	T			UK based techno-economic study. Indicates the superior qualities of ground source heat pumps over alternative systems. Air source heat pumps were recommended for mild climate regions where winter temperatures remain above 30°F (-1°C), as in Maritime climates. However, this conclusion is not supported by literature or empirical evidence.
32	Verhelst *et al.* (2008)	Cool Littoral (Belgium)	E N				Study conducted over 1 year. Study monitored the operation (COP & SPF) of an ASHP with regard to climatic influences.
33	Koyun *et al.* (2009)	Dry Summer Subtropical (Turkey)		E			Non seasonal study looking at the potential for improved collector pipe design. Horizontal collector submerged at 1.8m depth.

E - Experimental evaluation
N - Numerical evaluation
T - Techno-economic evaluation

While Tables 2.2, 2.6 and 2.8 provide more detailed characteristic and performance data by technology type, the following observations were also made from Table 2.1:

- Despite 26 of the 33 studies (79%) reporting experimental results, no study presented results for two heat pump technologies and three collector types ($GSHP_{HC}$, $GSHP_{VC}$ and ASHP). *HP-IRL* is the first such study to report data from the same location, climate type and test methodology.
- The longest continuous study providing validated experimental results ran for just 12 months (BRECSU, 2000; De Swardt and Meyer, 2001; Romero *et al.*, 2005; Verhelst *et al.*, 2008). *HP-IRL* benefits from a 36 month long test program.
- The key heat pump performance metrics used by 30% of researchers was COP and 21% was SPF, with no $GSHP_{VC}$ study generating a SPF and few studies defining heat pump duty, climate conditions during testing or collector characteristics such as; ground temperature and moisture content, ground cover type, collector return temperature, collector pipe temperatures or ground temperature drawdown and recovery. *HP-IRL* details all these parameters.
- ASHPs are the most climate sensitive technology (Mustafa Omer, 2008) whereas the vertical collector GSHP is the least sensitive (Romero *et al.*, 2005).
- No previous experimental data exists to describe source-side characteristics of climate sensitive GSHP collector design and generally limited to collector depth and area.

The following three sections focus on those experimental studies listed in Table 2.1 with one section devoted to each collector type.

2.3 HORIZONTAL COLLECTORS

Horizontal collectors are the most popular collector type installed, with 61% of installations in Ireland (Jolley, 2007), yet are poorly documented in the context of the performance parameters identified in the previous section. Table 2.2 details thirteen horizontal collector focused studies performed between 1981 and 2010 and highlights the climate type, length of study, key results and notes on the collector design. For instance, researchers have utilised collectors buried at depths ranging from 0.6m to 2m and recorded SPFs ranging between 2.2 and 3.2.

TABLE 2.2 KEY $GSHP_{HC}$ STUDIES UNDERTAKEN BETWEEN 1981 AND 2010

No.	Researcher (Year)	Climate Type (Country)	Winter Farfield Ground Temp.*	Heat Pump Capacity & Performance	Methodology / Collector Design / Key Findings
1	Metz, P.D. (1981)	Humid Continental (USA)	0 to +4°C (Winter)	3.4 kW (Heating) ----- 2.2 (SPF)	Experimental study conducted over 1 year. No specific collector design methodology was employed. Horizontal collector submerged at 2m depth. Winter ground temperatures at collector depth dropped to -7°C. Ground temperature drawdown or recovery not determined (ΔT_G).
2	Parker and White (1982)	Humid Continental (USA)	-	2kW (Heating) ----- SPF Not Given	Experimental study conducted over 2 years. Horizontal collector submerged at 2m depth. No ground temperatures recorded. Due to climatic demands, the $GSHP_{HC}$ offered a more stable heat source/sink than the ASHP. $GSHP_{HC}$ found to be a more cost effective system.
3	Mei, V.C. (1986)	Humid Continental (USA)	NA	NA ----- NA	Numerical study. Study of the effects of soil freezing around a horizontal collector. Established design requirements for horizontal collectors under prolonged freezing conditions. Highlighted ground thermal properties and moisture content as important factors. Thermal interference between collector pipes is exacerbated by high thermal extraction rates. Study highlighted the need for horizontal collectors to be designed to operate on or below freezing point in continental climates.
4	Morehouse *et al.* (1992)	Humid Subtropical (USA)	NA	NA ----- NA	Numerical study. Simulated ground thermal extraction and recovery time.
5	Piechowski, M. (1996)	Humid Subtropical (Australia)	-	NA ----- NA	Experimental study conducted over 2 summer months. No heat pump employed, thermal energy provided by electrical heating element. Study evaluated the thermal storage capacity of the ground in cooling mode. Horizontal collector submerged at 1.8m depth. No design methodology employed. Study indicates that this climate dictates that the cooling load is far superior to heating load in collector design.
6	Leong *et al.* (1998)	NA	NA	NA ----- NA	Numerical study. Study concluded that heat pump performance is strongly linked to soil moisture content: Heat pump performance is severely affected when the moisture content is below 12.5%, somewhat affected between 12.5% & 25%, slightly affected between 25% & 50% and not affected between 50% & 100%. (COP decline of 1.5%)
7	BRECSU (2000)	**Cool Marine** (UK)	+3 to +7°C (Winter)	3.9 kW (Heating) ----- 3.2 (SPF)	Experimental study monitored heat pump COP over one year (March 1998 to February 1999). Test site location not defined, test methodology not defined. Horizontal collector submerged at 1m depth. No ground temperatures recorded. Collector return temperature ranged from +0.2°C to +4.3°C in winter.
8	Popiel *et al.* (2001)	Humid Continental (Poland)	-1 to +6°C (Winter)	NA ----- NA	No collector design methodology employed. No heat pump or collector utilised. Experimental study on shallow ground temperatures only. Study highlighted the need for careful evaluation of the ground surface type as the ground-climate interface plays an important role in ground thermal rejuvenation. Summer ground temperature at 1m below car park surface was +4°C higher than below the short grass surface. Winter conditions showed no significant temperature variation. Recommended horizontal collector depth of 1.5 to 2m to suit climate. Study highlights how surface cover impacts on ground temperatures.
9	Li *et al.* (2003)	Humid Subtropical (China)	NA	NA ----- NA	Techno-economic analysis with no experimental results. The simulated horizontal collector was designed based ASHRAE (1996). Study determined that the $GSHP_{VC}$ was a more economic system than a $GSHP_{HC}$ operating in this particular climate. Notable that the collector performance simulation assumed a fixed drawdown (ΔT_G) on the ground source temperature when heat pump in operation.
10	Inalli and Esen (2004)	Semiarid Continental (Turkey)	+8 to +10°C (Winter)	2.5 kW (Heating) ----- 2.4 - 2.9 @ 1m 2.6 - 3.2 @ 2m (SPF)	Developed the following collector design criteria that were specific to Turkey: Optimum horizontal collector depth is between 1m and 2m; ground thermal properties should be precisely measured before designing the GSHP system and; showed how an optimised collector flowrate is important factor in collector design & performance (increase in flowrate increases electricity use but decreases the temperature lift across the evaporator which affects the COP). Ground temperature drawdown not reported (ΔT_G).

TABLE 2.2 CONTINUED.

No.	Researcher (Year)	Climate Type (Country)	Winter Farfield Ground Temp.*	Heat Pump Capacity & Performance	Methodology / Collector Design / Key Findings
11	Dumont and Frere (2005)	Cool Littoral (Belgium)	+20°C (Summer) +6°C (Winter)	9 kW (Heating) ----- 2.7 – 3.0 (SPF)	Study conducted over nine months. No specific design methodology employed. Horizontal collector submerged at 0.6m depth. Ground temperature drawdown not reported (ΔT_G). COP and ground temperature within the collector area were recorded. Collector operates in permafrost conditions for 3 months of the year. Concluded that the collector design and weather conditions can (ambient air temperature, rainfall & solar radiation) have a significant effect on heat pump performance. Study recommended climate specific collector design.
12	Karlsson and Fahlen (2007)	Cool Littoral (Sweden)	-	NA ----- NA	Laboratory based experimental investigation, performed in accordance with EN-255 test standard. The mean annual ambient air temperature was set at +6°C. Study looked at demand side capacity controlled ground source heat pump performance. Study unclear about ground temperature and collector fluid temperature defined. COP improved with demand side capacity control against fixed speed by between 7 – 15%. Study illustrated the need for heat pump demand side management, but also suggested that a source side management could offer similar efficiency improvements.
13	Koyun *et al.* (2009)	Dry Summer Subtropical (Turkey)	NA	4 kW (Heating) 2.7 kW (Cooling) ----- NA	Non seasonal study investigating the potential for higher performance collector piping. Collector buried at a depth of 1.8m. Better designed collectors can be up to 26% more effective at gathering heat.

* Stated ground temperature at collector depth over the winter heating season.

2.3.1 CLIMATE EFFECTS ON HORIZONTAL COLLECTOR PERFORMANCE

This section reviews research undertaken to determine specific climate effects on horizontal collector performance.

In New York, USA, Metz (1981) examined the utilisation of a $GSHP_{HC}$ over one year. This study shared the *HP-IRL* objective to analyse heat pump performance under real climatic conditions. The horizontal collector was submerged at 2m. The ground collector was designed to operate at a fluid temperature as low as -7°C and a SPF of 2.2 was achieved. While the severity of New York's *Humid Continental* winter climate contrasts greatly with that of Ireland's milder Maritime climate this study offers little in terms of performance comparison since near and farfield ground temperatures, drawdown/recovery rates or operational characteristics (duty) were not published.

Piechowski (1996) experimentally and numerically studied the performance of a horizontal collector in the hot and *Humid Subtropical* climate of Australia over two summer months. The horizontal collector was buried at a depth of 1.8m. Ground temperatures between collector pipes, collector fluid temperature and collector fluid flow rates were recorded. All temperature sensors (thermocouples) were calibrated to within ±0.1°C. However, ground farfield temperature was not recorded. The study outlined the optimal collector design for summer cooling in hot climates and indicates the potential need for supplementary water injection to maintain high soil moisture content, since it plays a crucial role in the effective storage and diffusion of thermal energy. The accompanying numerical study showed that, due to the large amount of variables in designing a collector system, optimum collector pipe spacing could not be defined. This study highlights the significance of soil moisture content in collector design, especially in hot and dry climates.

In a numerical investigation Leong *et al.* (1998) characterised the specific effect moisture content has on a horizontal collector when extracting thermal energy and, in particular, on the ability of the collector to maintain as high a temperature as possible. Again, this study details the relationship between ground moisture content and collector performance. High moisture content facilitates higher collector return temperatures that translate into higher heat pump COPs for a given output temperature set point. However, ground moisture content was only shown to be significant when it fell below 50%. Therefore, ground moisture content plays a significant role in climates that fail to deliver consistent rainfall.

In 2000, the UK based Building Research Energy Conservation Support Unit (BRECSU) reported on the operation of a $GSHP_{HC}$ system over one year (BRECSU, 2000). The study recorded heat pump fluid flow and return temperatures through the condenser and evaporator, along with fluid flow rates and electricity consumption. No calibration results were presented. COP ranged between 2.5 and 3.7 over the year and the SPF was 3.2. While no ground temperatures were recorded, this report highlighted the performance gains that accrue in the *Cool Marine* (Maritime) climate, as the collector return temperature never dropped below 0°C. As indicated by Mei (1986) in Section 2.3.3.1, the avoidance of freezing conditions plays an important role in increasing heat pump performance.

In a study of heat pump control concepts by Bianchi *et al.* (2005) it was suggested that real weather and duty patterns should be used so that a more realistic simulated indicator of heat pump efficiency could be established. While no test of simulation results were reported this approach forms the basis for this *HP-IRL* study.

In summary, this section presented a sample of those studies that concentrated on revealing the impact of climate influences, such as in ambient air temperature and rainfall, on heat pump performance.

The findings reinforce the recognised design principle to position the horizontal collector deep enough to avoid the negative impact of extremely low ambient air temperatures while retaining high moisture content about the collector. Experiences articulated in these studies helped to shape the *HP-IRL* performance evaluation by closely investigating ground temperature and moisture content with depth, imposing a range of practical duty cycles and exposing the collector to the climate.

2.3.2 HORIZONTAL COLLECTOR THERMAL EXTRACTION & RECOVERY

This section reviews those studies that analysed the ground's response to thermal extraction by monitoring parameters such as duty cycle, ground surface effects and collector depth.

Considering the ground response to thermal extraction, using a $GSHP_{HC}$ in a numerical study Morehouse (1992) characterised the ground's ability to recover from thermal extraction in a *Humid Subtropical* region of the USA. A notable finding identified the need to control the thermal extraction rate from the ground in order to enable efficient thermal recovery. It was concluded that in order to avoid permafrost and an associated reduction in collector performance, as much ground thermal recovery time should be allowed as possible before thermal extraction is resumed. The critical extraction-recovery time ratio, or duty cycle, depends on collector size and thermal demand, where the system is ideally designed for a maximum 50/50 ratio of extraction versus recovery time under the coldest winter conditions. As no further guidelines were presented, this aspect was investigated in the *HP-IRL* study and is reported in Chapter 4 as the duty cycle.

Dumont and Frere (2005) monitored the performance of a $GSHP_{HC}$ adjacent to a domestic dwelling for a period of nine months in Belgium's *Cool Littoral* climate. The collector was buried at a depth of 0.6m. As indicated in Figure 2.4 the ground temperature at the collector was monitored at one location, although the precise location of temperature sensor was not given along with that of the evaporator temperature. While no farfield ground temperature was recorded its notable from Figure 2.4 that heat pump operation induced freezing of the collector ground, causing permafrost for three months between mid-December and mid-March. The ground temperature within the collector region indicates the impact of heat pump operation, but the lack of a reference ground temperature outside of the collector area, made it impossible to discern the impact of heat pump operation on ground temperature within the collector. However, conclusions drawn from the study indicate that the collector design and climate can both have a significant effect on heat pump performance, especially for such shallow collectors, since COP ranged between 2.69 to 2.96.

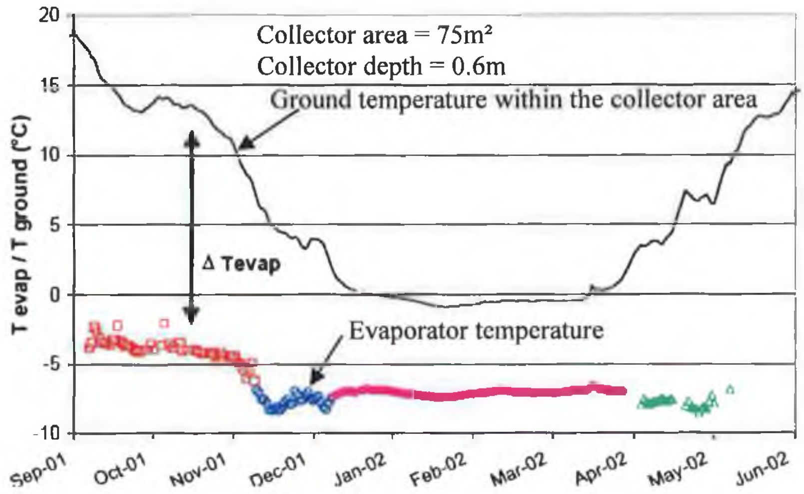


FIGURE 2.4 HEAT PUMP EVAPORATOR AND THE GROUND TEMPERATURE WITHIN THE HORIZONTAL COLLECTOR (DUMONT AND FRÈRE, 2005).

This study highlighted the variation of the heat pump efficiency, with the above ground climate, collector ground temperature and heat pump operation. The authors also recognised that manufacturer's stated range of COPs, measured to internationally recognised standards, did not give any indication of how a heat pump performs under real conditions, with varying climates and duty cycles since true indication of a source temperature is climate, collector depth and heat pump duty specific. Indeed manufacturers do not include the electrical consumption of the collector pump in the COP calculation. This study helped inform the experimental design of the *HP-IRL* study, which not only targeted similar ground temperature data but also the impact of the above ground climate on the collector and farfield ground temperatures.

Inalli and Esen (2004) experimentally tested the performance of a horizontal collector in the Turkish, semi-arid continental climate over six months. They used two horizontal collector loops of 50m length, installed at 1m and 2m depths. The study recorded the heat pump fluid flow and return temperatures on the heating and collector side (condenser and evaporator), collector fluid flow rates, electricity consumption and ground temperatures in the collector region. Test results were recorded every half hour. The SPF for the 1m and 2m deep collector loops were 2.66 and 2.81 respectively with the temperature at 2m showing a 3°C higher mean annual temperature. This 6% improvement in SPF with depth is a key finding and was attributed to the 4°C higher minimum winter ground temperature at 2m. Minimum winter ground temperatures at 1m and 2m was +8°C and +12°C respectively. This climate type generates a substantial swing in ambient air temperatures throughout the year of ±15K, permitting the horizontal collector the potential to buffer the heat pump from the worst of these temperature swings, offering a stable and high temperature source. However, as the heating season is short and cooling requirements dominate, it was concluded that the horizontal collector may not justify the initial high investment cost, unless the heat pump can be reversed to provide summer cooling.

Addressing the impact of ground cover on collector performance, Mihalakakou *et al.* (1996; 1996; 1997) established the potential of pre-heating and pre-cooling air through the ground for ventilation and identified the characteristics of heat absorption through various ground covers in the Maritime climate of Ireland. This experimental study showed that the annual ground temperature was up to 1°C higher (at a depth on 1m) under a bare asphalt surface than under short grass. This showed that the ground cover is a significant and controllable factor that influences ground temperature and therefore collector performance.

In a Polish study into the effect of ground surface cover on underlying ground temperatures, Popiel *et al.* (2001) experimentally investigated vertically aligned temperature profiles of the ground under both a car-park (tarmac) and short grass. Measurements during summer showed that the temperature 1m below the car park surface was 4°C higher than that of 1m below the short grass surface. However, in winter there was no significant temperature variation. It was recommended that a horizontal collector should be installed at a depth of between 1.5 and 2m for this continental climate. Furthermore, due to the demands of a *Humid Continental* climate, cooling is required in the summer and as the ground is up to 4°C cooler under short grass rather than under a tarmac surface, short grass was the preferable surface condition for enhanced collector performance. Florides and Kalogirou (2007) also identified ground surface cover as an important parameter whose influence on the near surface ground temperature (0-2m) was underestimated. These studies highlight the impact of surface cover on the collector region temperature at 1m depths and possible knock-on effects on heat pump performance. As a result, this aspect of collector design was also considered in the *HP-IRL* study.

Karlsson and Fahlen (2007) raised the topic of source side management in a laboratory based evaluation of a capacity controlled ground source heat pump that operated on a typical domestic house duty cycle in Goteborg, Sweden. This study focused on 'demand side management' of heat pump operation, where the flow temperature to the hydronic (radiator) heating system was controlled and kept as low as possible. This modification increased the COP by between 7 and 15% against that of a fixed speed system. However, the study concluded that the SPF may not improve as a result of inefficiencies in the variable speed compressor along with increased cost associated with increased run time of the fixed speed circulation pumps on the heating side and the collector. Indeed, it identified that control of the collector side is an equally important factor in improving the overall system efficiency. This recent study's acknowledgment of the potential performance improvements that could accrue from heat pump source side management is significant and it also highlights the lack of awareness in this area. In developing the *HP-IRL* study, this recent study provides further evidence that temperature management on either or both the source or demand sides can significantly improve heat pump performance. *HP-IRL* complements the Karlsson and Fahlen (2007) study by quantifying the performance improvements that result from better source side management.

2.3.3 Collector performance and design

This section identifies those early studies undertaken to explore specific aspects of collector design which has informed the collector design protocols/methods described in Section 2.3.3.2.

2.3.3.1 Collector performance

Oak Ridge National Laboratory (ORNL) in the United States has led heat pump research on ground source heat pumps operating under *Humid Continental* climate conditions since the 1980's (ORNL, 1980; Mei, 1986; ORNL, 1997; Bose *et al.*, 2002). This work led to the formation of the International Ground Source Heat Pump Association (IGSHPA) which released independent standards for ground source heat pump installation (IGSHPA *et al.*, 1989; IGSHPA *et al.*, 1997). A template for this *HP-IRL* study was performed over two years by Parker and White (1982) of the ORNL, comparing the use of an ASHP and $GSHP_{HC}$ from both a techno-economic and collector design perspectives. This investigation exposed both heat pump types to the same operating and climatic conditions (*Humid Continental*). The external ambient air temperature, heat pump thermal performance and electrical consumption were recorded every 15 minutes. It concluded that, due to the large difference between the average winter (-1°C) and summer (+38°C) air temperatures, the ground source provided a more stable and cost effective thermal source than the air source. However, heat pump technology has since moved on significantly and as the study was conducted in the highly variable continental climate of Oklahoma, USA, it makes it hard to compare with the *HP-IRL* study where the Maritime climate generates just a 15K temperature difference between the summer and winter ambient air temperatures. While acknowledging the difficulty in translating the findings from such studies into the modern setting that uses different refrigerants and collector design, this study provided insight of how earlier researchers approached such experimental investigations.

In a similar study to Parker and White (1982), Li *et al.* (2003) carried out a study comparing the techno-economic operation of an ASHP and GSHP which was conducted in the *Humid Subtropical* climate conditions of Hangzhou, China. This study revealed that the performance of the GSHP was both technically and economically superior to that of the ASHP, with the vertical collector performing better than the horizontal collector with surprisingly short payback periods of 1.6 and 4.2 years respectively. Source temperatures were also recorded and are shown in Figure 2.5.

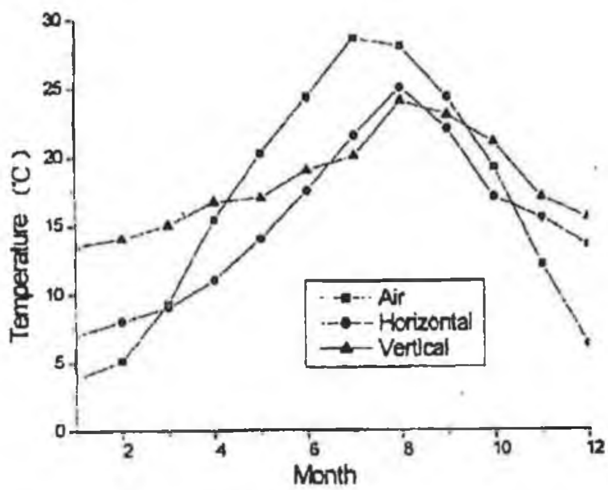


FIGURE 2.5 MONTHLY AVERAGE AIR AND GROUND TEMPERATURES IN HANGZHOU, CHINA (LI *ET AL.*, 2003).

Despite the substantial climate differences, this research is particularly relevant to the *HP-IRL* study as it mirrors the technology range. Although this study by closely resembles the aims of the *HP-IRL* study, it lacked experimental results and relied on a numerical analysis of the three sources using ambient air and ground temperatures as boundary conditions to determine the year round performance of the ASHP and GSHP systems. Notably it also assumed a fixed drawdown (ΔT_G) temperature difference between the ground and the collector fluid, regardless of the heat pump duty cycle. The vertical collector design was based on recommendations from Ingersoll *et al.* (1948, 1954) and the horizontal collector was designed based on recommendations from ASHRAE (1996).

Mei (1986) carried out a numerical analysis on the effect of freezing around the collector pipe to enhance horizontal collector design for the *Humid Continental* climate region of the USA. The study emphasised that while it is important to avoid collector freezing, if freezing does occur the system can still function, but with reduced performance. The study concluded that in order for the system to operate effectively in freezing conditions the ground must be sufficiently moist and the collector be submerged at an optimal depth for the chosen climate, although "optimal depth" was not defined. The study also recognised the dilemma associated with placing the collector as deep as possible to avoid the harshest winter conditions and the increased likelihood that the collector may remain in permafrost caused by thermal extraction at these depths for longer due to reduced prospects of thermal recovery induced by increased spring/summer ambient air temperatures. This study illustrated the severe nature of continental climates and the need for horizontal collectors to perform under permafrost conditions. In such situations heat pumps must deliver high temperature lifts (ΔT_{HP}) that depress COP. It also suggests the need for adequate collector sizing, to avoid sub-zero Celsius collector temperatures for as long as possible and source side feedback.

Collins (1998), in a limited two week long experimental test program in May 1998, examined the performance of a horizontal and a vertical collector in the Maritime climate of Ireland. In this study, the heat pump collector flow and return temperatures, electrical power consumption and flow rates was monitored. While this study endeavoured to show a comparison between a $GSHP_{HC}$ and a $GSHP_{VC}$, its short test duration did not enable a full performance evaluation and weather parameters and ground temperature profiles were not monitored. However, this relatively recent study called for a dedicated heat pump test facility to comprehensively assess heat pump systems over a full year, with additional monitoring of ground temperature distributions, rates of discharge and the thermally affected zone. *HP-IRL* responds to this call by monitoring all the identified parameters.

Working on collector pipe design, Koyun *et al.* (2009) developed a new design of finned aluminium horizontal collector pipe that performed up to 26% better than conventional HDPE piping. However, while this offered the potential to reduce collector overall area or a given fixed area, deliver a higher year-round source temperature, the study showed that higher heat transfer rates were only achieved during the initial stages (hours) of heat pump operation. This study also recommends the high performance collector designs depends heavily on the thermal properties of the ground in close proximity to the collector pipe and use of thermally enhanced backfill was recommended in conjunction with high moisture content (no percentage reported) ground around the collector pipe.

Heat pump monitoring extending over short periods of months have been carried out under the *Cool Marine* climate and this has delivered useful operational characteristics of heat pump systems (BRECSU, 2000; Al-Huthaili, 2004) that complement economic and environmental viability studies by Mustafa Omer, 2008. However, these omit key performance metrics outlined in Section 2.2 and any reference to climate sensitive collector design. Indeed, the underlining premise guiding collector design is how best to avoid the climate or the negative aspects associated with the coldest periods. *HP-IRL* targets collector design guidelines that would allow the collector benefit from the mild aspects of the *Cool Marine* climate and be protected from the cold periods.

Many of the studies identified above have influenced horizontal collector design standards and these are reviewed in the next section.

2.3.3.2 INDUSTRY STANDARD COLLECTOR DESIGN GUIDES

This section reviews the key design guides that have emerged for horizontal collector design. These guidelines offer generic guidance on collector characteristics such as depth, area, pipe length and spacing as well as operational parameters such as extract rate and ground temperature drawdown.

TABLE 2.3 INDUSTRY STANDARD HORIZONTAL COLLECTOR DESIGN GUIDES AND PARAMETERS RECOMMENDED BY LEADING HEAT PUMP REPRESENTATIVE BODIES AND RESEARCHERS

Author/Institute (Year)	Recommended Minimum Fluid Temperature ($T_{HC,R}$)	Ground Temperature Drawdown range ($\Delta T_{HC,G}$)	Extract rate	Collector Thermal Characteristics	Collector depth (m)	Pipe spacing (m)	Trench size (m)	Trench spacing (m)
Kavanaugh & Rafferty (1997) – USA	-	-	-	-	1.5m (5') Minimum	0.3 – 0.6m	-	-
IGSHPA *et al.* (1997) – USA, for cooling dominated climates.	40°F above coldest *Ta*, or 25°F (-4°C), whichever greatest	-	-	-	1.5m	≥ 0.6m	-	-
VDI 4640 (2001) – Germany	-	Not to Exceed: ±12 K base load ±18 K peak load	50 – 70 kWh/m²/a	8 – 40 W/m² (selected on soil type)	1.2 – 1.5m (not less than 1.2m and not to exceed 1.5m)	0.3 – 0.8m	-	-
***Hepbasli*, A. (2004) – Turkey**	-	-	-	Local experience required	1.2 – 3.0m	0.3m	0.9 – 1.8m	3.7 – 4.6m
BRE (2004) – UK	-	-	-	-	1 – 1.5m	-	-	-
SEI (2005) – Ireland	-	-	-	-	1.2 – 1.5m	0.3 – 0.8m	-	-
EHPA (2005) – Europe	-	-	-	-	0.9 – 1.5m	-	-	-
RETScreen (2005) – Canada	-6°C	-	-	35 – 55m of collector pipe per kW of capacity	1.2 – 2m	-	0.15 – 0.6m	1.5 – 4m
ASHRAE (2003 - 2006) – USA	-	-	-	Minimum of 20 W/m²	≥ 1.2m	≥ 0.6m	-	-
EN 15450 (2007) – Europe	-	Typical Central European value of -12 K	50 – 70 kWh/m²/a	8 – 40 W/m² (selected on soil type)	-	-	-	-
Rawlings *et al.* (2007) – UK	-5°C	-	-	-	1 – 2m	0.3m	-	3m
Brown, R. (2009) - BSRIA UK	-	-	-	8 – 40 W/m² (selected on soil type)	≥ 1.5m	≥ 0.8m	-	-

Table 2.3 reveals that most common design parameters are collector depth and pipe spacing. However, VDI, RETScreen and ASHRAE introduce other operational parameters such as annual extract rates and the extent of temperature drawdown (ΔT_G) within the collector region.

While there are clearly many general design guidelines they are confined to advising on collector geometry without addressing operational characteristics or source side management to reflect sensitivity to climate and/or duty cycle. Indeed this was clearly highlighted by ASHRAE who stated that "*even though many thousands of these systems* ($GSHP_{HC}$) *were installed in heating climates, no comparable analysis has been performed to determine proper design guidelines*" (ASHRAE, 2003 - 2006).

In acknowledging the substantial body of work relating to the design of $GSHP_{VC}$ systems Bose *et al.* (2002) highlighted the absence of similar studies that characterise the behaviour of horizontal collectors, where interaction with the above ground environment is important.

In terms of collector design methods employed, Le Feuvre (2007) analysed how engineers in the UK designed collectors and concluded that the approaches were evenly split between software tools and general 'rules of thumb'.

In monitoring the performance of a $GSHP_{HC}$ system in Carouge, Switzerland (*Humid Continental*), Hollmuller and Lachal (2001) concluded that simulation tools are not sufficiently capable of recreating the multiple climatic effects that influence both heat pump duty and performance, and recommended that simplified rules of thumb are more practically useful for specific locations.

In a study of the technical status of $GSHP_{HC}$ design softwares, Eugster and Sanner (2007) recommend that, due to the rapid changes in heat pump technologies, modern rules of thumb should be constantly developed and recognised the challenge to bring these new software techniques to engineers and planners. This *HP-IRL* study responds to this call.

In Germany and Switzerland, the VDI guidelines issued by the "*Association of German Engineers*" (VDI 4640, 2001) are used to design vertical and horizontal collectors for residential houses. These are the most comprehensive collector design guidelines and the preferred design guidelines used by installers in Ireland, particularly for vertical collector installations (Goodman and Pasquali, 2009). Indeed, this is the only design guideline that makes reference to acceptable limits for the important collector ground temperature drawdown (ΔT_G) parameter. The collector ground temperature drawdown is the difference in temperature between the collector return temperature (temperature of collector fluid going into the heat pump) and the ground's farfield temperature at the depth of the collector. Since heat pump efficiency increases as source-sink temperature difference decreases, it is an important design goal to correctly size the horizontal collector to minimise the ground temperature drawdown (ΔT_G). It also reflects the degree of energy depletion within the collector which varies depending on collector design, duty cycle, thermal extraction rates and duration. In reference to the temperature drawdown on a horizontal collector VDI 4640 states:

> *"The temperature of the heat carrier fluid returning to the horizontal ground heat exchanger should not exceed the limiting range of ±12K temperature change in base load operation (weekly average) compared to the undisturbed ground temperature, at peak loads the value is ± 18K"* (VDI 4640-Part 2, 2001)

Under the weather conditions of Ireland this would allows the collector fluid temperature to range between -7°C, under base load, and -13°C under peak loads, assuming a minimum ground temperature of +5°C. This estimate would therefore allow the collector to operate in permafrost conditions for large portions of the winter heating season in Ireland. This ΔT_G operating parameter was recorded within *HP-IRL* to ascertain if the VDI guidelines were satisfied by the *HP-IRL* horizontal collector design under the imposed duty cycle and Maritime climate.

Another relevant and prominent guide is the European standard EN 15450 (EN 15450, 2007), entitled "*Heating systems in buildings – design of heat pump heating systems*". Using the ΔT_G definition from the VDI 4640-Part 2 (2001) this recommended that during continuous operation ΔT_G should not exceed 12K in continental Europe. It is notable that heat pump testing to EN 14511 (EN 14511, 2004), only tests the fluid return temperature to a low of -5°C, which is equivalent to a minimum ground temperature in continental Europe of +7°C based on the nominal ΔT_G of 12K. Since Chapter 4 provides evidence that minimum ground temperatures continental climate regions are much lower than +7°C this indicates that performance testing at -5°C may not be sufficiently low enough. EN 14511 does not define typical drawdown in other climates. A list of all relevant heat pump standards is provided in Appendix C.

A highly referenced guide for the installation of GSHP systems is Kavanaugh and Rafferty's "*Ground Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings*" (1997). It gives a detailed account of the parameters that relate mainly to Water-to-Air systems utilising $GSHP_{VC}$ technology in the USA (Kavanaugh and Rafferty, 1997). However, collector duty variations and ground temperature drawdown are not covered for GSHP horizontal collector systems.

The soil type is factor that impacts on ΔT_G since a collector's ability to gather thermal energy is governed by the soil's thermal conductivity and specific heat capacity. Based on the soil type, a rule-of-thumb for collector sizing based on specific extraction output shown in Table 2.4 is commonly used to quantify collector thermal extraction in watts per meter squared of collector area. Different soil classifications are defined and Table 2.4 presents a sample of specific extraction outputs defined by two guidelines. Note that the output increases with moisture content.

TABLE 2.4 SOIL THERMAL EXTRACTION CAPACITY EXAMPLES

VDI 4640, 2001	
Soil Type	**Specific extraction output**
Dry, non-cohesive soils	8 W/m²
Cohesive soils, damp	16 - 24 W/m²
Water Saturated sand/gravel	32 W/m²

Viessman Heat Pump Sizing Manual	
Soil Type	**Specific extraction output**
Dry, sandy soil	10 - 15 W/m²
Damp, sandy soil	15 - 20 W/m²
Dry, loamy soil	20 - 25 W/m²
Damp, loamy soil	25 - 30 W/m²
Ground with groundwater	30 - 35 W/m²

While the specific extraction output is commonly used by heat pump designers to determine collector size, once soil type and moisture content are known, it still does not give any indication of the ground temperature drawdown. Indeed it also does not indicate a recommended collector length to complement the specified collector area. Another popular method for sizing the horizontal collector area uses a nomogram (SIA-Documentation D0136, 1996) shown in Figure 2.6.

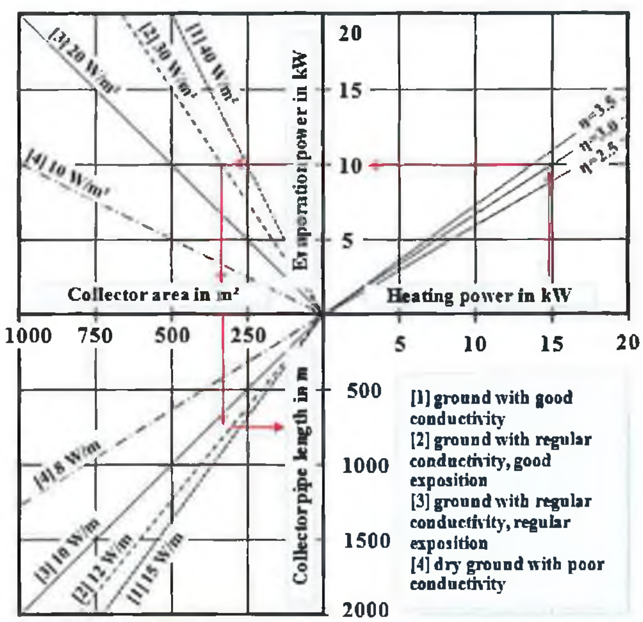


FIGURE 2.6 NOMOGRAM FOR SIZING HORIZONTAL GROUND LOOPS (SIA-DOCUMENTATION D0136, 1996).

The nomogram shown in Figure 2.6 is a reproduction of the original German nomogram (SIA-Documentation D0136, 1996) used in the VDI 4640 standard. As an operational example the red arrows shown in Figure 2.6 represent the 15kW $GSHP_{HC}$ in the *HP-IRL* study. Based on an SPF (η) of ~3.0 the thermal extraction from the ground is 10kW. If the ground is assumed to have good thermal properties (see ground condition [2] in Figure 2.6) the projected collector area should be 320m² and utilise 750m of collector pipe. Chapter 3 outlines that the *HP-IRL* collector area is generously sized at 430m² with 1500m of collector pipe.

A more recent collector design guide by Brown (2009), published by the UK based Building Service Research and Information Association (BSRIA), primarily drew its recommendations from the VDI 4640 and EN 15450 standards (EN 15450, 2007). It stated that for the UK climate the horizontal collector should be buried at least 1.5m deep.

Canada has been at the forefront of heat pump research and development over the past twenty years with the development of numerous guides for heat pumps along with simulation tools such as *RETScreen® International* and *GS2000* (Hosatte and Sunyé, 2005). A number of other heat pump evaluation software have been developed including those listed in Table 2.5; GchpCalc (USA), $GLHE_{PRO}$ (Sweden/USA), GLDesign, GeoStar (China), EnergyPlus (USA), Earth Coupled Analysis (USA) and CLGS (USA).

TABLE 2.5 HEAT PUMP COLLECTOR DESIGN SOFTWARE

Software	Cost	$GSHP_{VC}$	$GSHP_{HC}$	Calculation Method	Country of Origin
RETScreen®	Free	✓	✓	IGSHPA *et al.* (1997)	USA & Canada
GS2000™	Free	✓	×	Not available	USA & Canada
GchpCalc	$300 (€210)	✓	×	Kavanaugh and Rafferty (1997)	USA
$GLHE_{PRO}$	$525 (€370)	✓	×	Eskilson (2000)	USA/Sweden
GLDesign	Free	✓	✓	Kavanaugh and Rafferty (1997) & Eskilson (2000)	USA
GeoStar	NA	NA	NA	Not available	China
EnergyPlus	Free	✓	×	Not available	USA
ECA	$395 (€280)	✓	✓	ASHRAE (2003 - 2006)	USA
CLGS	$500 (€350)	✓	✓	IGSHPA *et al.* (1997)	USA

However, these software tend to be only applicable to the country of origin, due to the availability of climate data, utilising over-simplified climatological aspects such as air temperature only, can be limited to one collector type and/or be cost prohibitive. They tend to employ the design guidelines described in Table 2.3.

In determining the efficiency of the overall system it is important to be able to estimate the average winter heating season source temperature supplied to the heat pump. If both the source and sink temperatures can be estimated then the SPF can be estimated from the heat pump manufacturer's operating efficiency typically measured to the EN 14511 standard. However, in calculating heat pump efficiency EN 14511 excludes the pumping power required to circulate fluid through the horizontal collector which therefore overestimates COP.

In summary, the horizontal collector design guides make little reference to the following important collector characteristics or operational parameters:

- Drawdown rates and extent of collector ground temperature drawdown (ΔT_G)
- Extract rate and duration
- Duty cycles
- Collector pumping power impact on performance
- Recommended recovery periods
- Need to monitor ground source temperature to support source side management

All these aspects are investigated in *HP-IRL*, along with the detailed monitoring of the climate and ground cover.

2.3.4 HORIZONTAL GROUND SOURCE

Ground temperature varies in response to changes in the radiant, thermal and latent energy exchange processes that take place primarily through the ground surface (Hillel, 1980). The ground absorbs up to 46% of the incident solar radiation (Peuser *et al.*, 2002) and also exchanges thermal energy with both the incident rainfall and air movement. Year-round vertical temperature profiles of the ground show that the ground serves as a diurnal store to a depth of approximately 0.6m, a monthly store to approximately 5m and a seasonal store with temperature variations occurring between 5m and 15m. All these ground temperature variations can be described as weather dependant, but temperature variations below 15m are largely climate and geothermal gradient dependant. Climatic influences on ground temperature conditions are summarised in Figure 2.7.

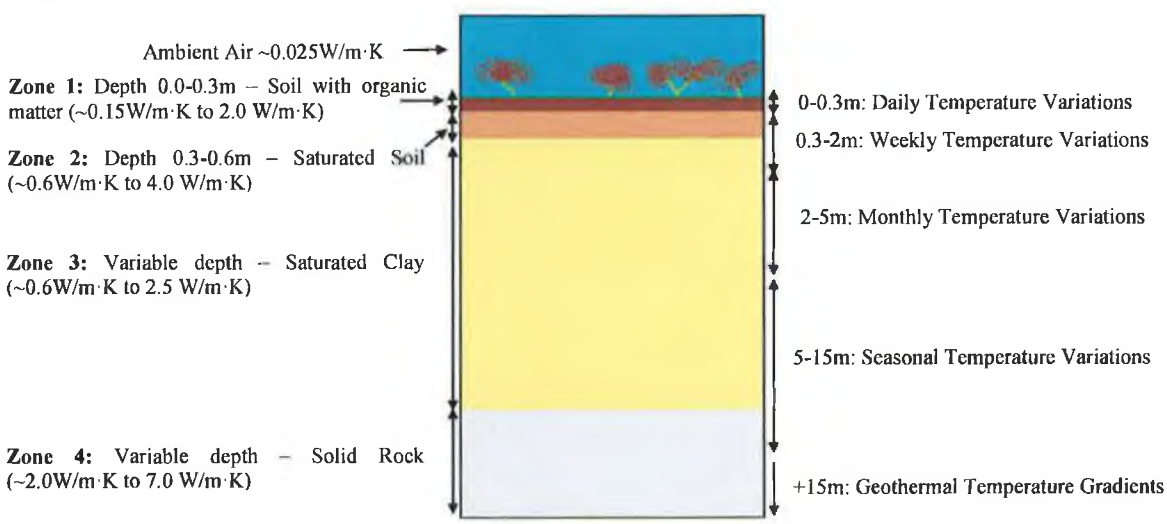


FIGURE 2.7 GROUND TEMPERATURE AND CONDUCTIVITY VARIATIONS WITH DEPTH.

The following inter-related factors govern the rate at which thermal energy is extracted from the ground using typical water filled piping collectors:

- Ground thermal conductivity and diffusivity
- Moisture migration, as a result of both rainfall and induced temperature gradients imposed on the ground through heat extraction
- Possibility of ice formation around and along the collector coil, generating a step change in the ground's thermal characteristics
- Seasonal induced ground temperature variations
- Collector pipe material and pipe-soil contact thermal resistance
- Collector area, pipe diameter, length and internal fluid flow rates

The following sub-sections outline the specific characteristics required of a horizontal collector thermal source.

2.3.4.1 GROUND THERMAL PROPERTIES

Ground volumetric heat capacity, C_G, is the ability of a given volume of ground to store energy while it is subjected to sensible temperature changes. It is a determinant of the average specific heat capacity of the constituent materials multiplied by the average bulk density. The volumetric heat capacity increases linearly with volumetric water content, which impacts on both the thermal conductivity and diffusivity as indicated in Figure 2.8.

The thermal conductivity of the ground can be defined as the amount of heat (W) that passes per 1°C temperature gradient per metre depth of ground induced by a temperature gradient (W/m·K) applied in the direction of heat flow (Farouki, 1986).

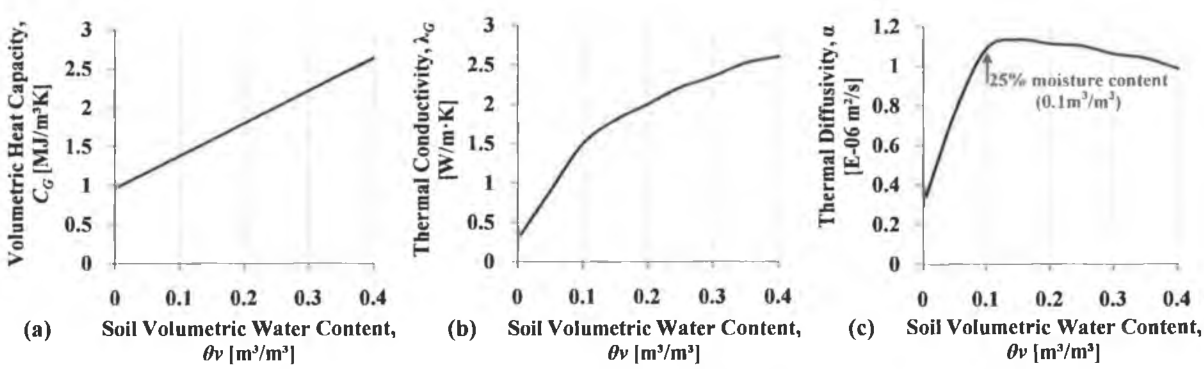


FIGURE 2.8 (A) VARIATION IN VOLUMETRIC HEAT CAPACITY WITH MOISTURE CONTENT FOR IRISH SANDY/ SILTY LOAM SOIL WITH AN AVERAGE BULK DENSITY OF 1200KG/M^3 (B) VARIATION OF SOIL THERMAL CONDUCTIVITY, AND (C) VARIATION OF THERMAL DIFFUSIVITY (WITH INCREASED IN SOIL MOISTURE CONTENT FOR IRISH SANDY LOAM SOIL WITH AN AVERAGE BULK DENSITY OF 1200KG/M^3) (LOHAN *ET AL.*, 2006).

Conduction is the dominant mode of heat transfer in the ground although heat transfer is also facilitated by advection, convection, radiation and evaporation-condensation. The ground's formation is made up of various layers of different composite materials shown in Figure 2.7, each layer having differing thermal characteristics.

The soil's thermal conductivity has been shown in many reports to be the most important factor in utilising the ground as a thermal source and soil moisture plays a significant role (Piechowski, 1996; Leong *et al.*, 1998). Soil with a volumetric water content (θv) of $0.4 m^3/m^3$ is generally considered to be fully saturated and therefore has a moisture content of 100% (Piechowski, 1996). What is noticeable from Figure 2.8(c) is the improvement in the ground's diffusivity above soil moisture content of 25% ($0.1 m^3/m^3$). As thermal diffusivity is an important factor in horizontal collector performance, soil moisture therefore plays an important role. Further analysis is presented in Lohan *et al.* (2006).

2.3.4.2 FLUCTUATIONS IN GROUND TEMPERATURE GRADIENTS

The undisturbed ground temperature at any depth z in the ground and at any time t, $T_u (z,t)$, can be predicted using Equation 2.1 (Carslaw and Jaeger, 1959):

$$T_U(z,t) = T_m - \left(A_S \cdot e^{\left(-z\left(\frac{\pi}{365\alpha}\right)^{0.5}\right)}\right) \cdot \left(cos\left(\left(\frac{2\pi}{365}\right) \cdot \left(t - t_o - z/2\left(\frac{365}{\pi\alpha}\right)^{0.5}\right)\right)\right) \qquad \textit{Equation 2.1}$$

where: α is the ground's thermal diffusivity (m^2/s); A_s is the amplitude of the ambient air temperature fluctuation at the ground's surface; T_m is the mean ambient air temperature at the ground's surface and; t_o is the phase lag.

The phase lag increases proportionally with depth. Therefore, the amplitude of temperature change in the ground reduces with depth. As indicated in Figure 2.9, the transient variation in ground temperature with depth goes from being similar to that of the ambient air, at the ground air interface, to being stable at a temperature similar to that of the yearly mean ambient air temperature below 10m ($T_m = \bar{T}_a$).

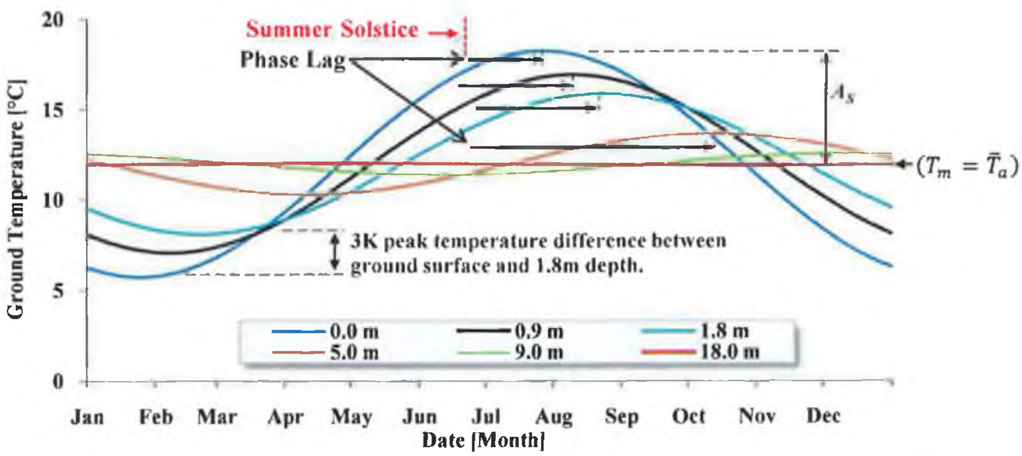


FIGURE 2.9 PREDICTED GROUND TEMPERATURE VARIATIONS WITH DEPTH FOR THE GROUND CONDITIONS IN THE *HP-IRL* STUDY.

The ground temperature fluctuates in response to seasonal swings to depths of between 5m and 15m. Below this level lies a '*neutral zone*', which remains at a constant temperature throughout the year. However, geothermal gradients can have an effect on the ground's temperature within the neutral zone and the amount is influenced by geological conditions. Geothermal gradients are a factor in determining the temperature at various depths in the ground around the collector, where these gradients are denoted by the rise in temperatures with depth. Temperature gradients vary globally depending on factors such as continental plate thickness, proximity to fault lines, ground material and thermal conductivity. Geothermal gradients generally range between 0.5-3K per 100m (Gehlin, 2002).

It is evident from Table 2.3 that most guidelines recommend that horizontal collectors be installed at a depth of between 0.3m and 0.6m below frosting depth (BRE, 2004; Mustafa Omer, 2008). In Austria, this requires a collector depth of between 0.8 and 1.2m (Halozan, 2008), whereas the VDI 4640 standard recommends between 1.2m and 1.5m depth (VDI 4640, 2001). However, due to Ireland's mild Maritime climate prolonged periods of frost are rare, particularly in costal regions and it may be acceptable to install horizontal collectors at shallower depths. This option is explored in this *HP-IRL* study, where ground temperature and soil moisture measurements are recorded to 0.6m above and below the 0.9m deep collector.

2.3.5 SUMMARY

While most studies agree that the climate plays a significant role in the overall performance of $GSHP_{HC}$ systems (Parker and White, 1982; Piechowski, 1996; Popiel *et al.*, 2001; Dumont and Frère, 2005) it is noticeable that there has not been any substantial experimental study evaluating both the positive and negative relationships between the climate, collector condition and heat pump performance, especially in the *Cool Marine* (Maritime) climate. Recent studies have also called for manufacturer specifications for horizontal collector operation and installation based on laboratory tests (Dumont and Frère, 2005) along with the impact of ground cover (Mihalakakou *et al.*, 1996; Popiel *et al.*, 2001; Florides and Kalogirou, 2007), source side temperature management (Karlsson and Fahlen, 2007) and the performance of horizontal collector systems in heating dominated climates (ASHRAE, 2003 - 2006) to be investigated. In response, this *HP-IRL* study has been designed to facilitate a thorough investigation of these aspects under Ireland's *Cool Marine* climate.

2.4 VERTICAL COLLECTORS

This section reviews relevant literature associated with the performance characteristics of vertical collectors.

2.4.1 STANDARD SITE ASSESSMENT

Over the past decade, vertical collector, or borehole heat exchanger (BHE) site assessment has become an accepted part of the design process with numerous studies reporting results from borehole Thermal Response Tests (TRT) (Ekelof and Gehlin, 1996; Spiker, 1998; Gehlin, 2002; Sanner, 2007; Sanner *et al.*, 2008; Esen and Inalli, 2009).

This relatively short TRT enables the likely performance of a vertical collector to be predicted by evaluating the thermal conductivity of the surrounding ground. A TRT takes approximately fifty hours and can be conducted using the mobile apparatus shown in Figure 2.10 (Gehlin, 2002). It involves monitoring both the temperature change of the collector fluid over time and the heating or cooling load applied until steady-state conditions arise, typically after 50 hours.

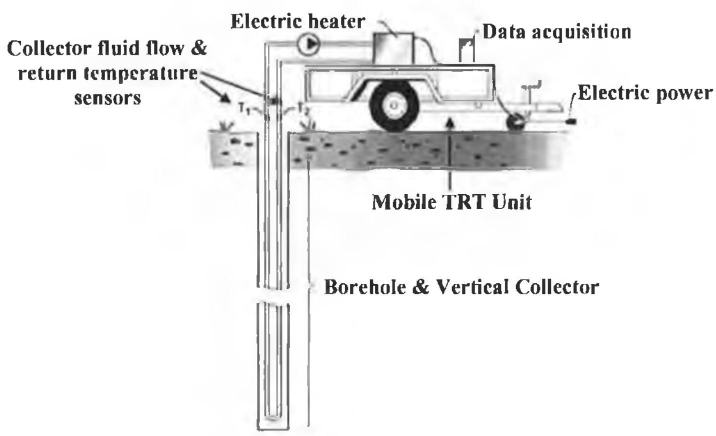


FIGURE 2.10 APPARATUS USED TO CONDUCT A THERMAL RESPONSE TEST (GEHLIN, 2002).

The TRT was first presented by Mogensen (1983) where the thermal conductivity of a borehole was determined *in situ* by chilling the ground. A number of mobile thermal response test rigs similar to that shown in Figure 2.10 were developed in the 1990's, which enabled the test to be carried out at any given location (Ekelof and Gehlin, 1996; Austin, 1998). More recently the typical utilisation of the TRT method injects heat into the vertical collector rather than chilling it. However, the TRT cannot be used where groundwater flow exists, and since it is conducted over just fifty hours it does not provide information on the likely year-on-year drawdown of the source temperature (Eugster and Rybach, 2000; Huang and Murphy, 2003; Acuña *et al.*, 2008). This latter aspect of vertical collector performance requires continuous monitoring over successive years of heat pump operation and such studies are documented in the next section.

2.4.2 VERTICAL COLLECTOR PERFORMANCE

Table 2.6 summarises the key studies conducted on vertical collectors over the past decade and identifies collector design, test duration, ground temperatures and climate specific performance results.

TABLE 2.6 SUMMARY OF KEY $GSHP_{VC}$ STUDIES

No.	Researcher (Year)	Climate Type (Country)	Average Ground Temperature	Heat Pump Capacity & Performance	Key Findings
1	Eugster and Rybach (2000)	Moderate Subpolar (Switzerland)	+9 to +12°C (Winter)	NA	Numerical study on the effects of long-term heat extraction on ground temperature. Single, coaxial, 105m long BHE. Study indicates that "dry" boreholes show significant drop in ground temperature in first year of operation, stabilising in subsequent years.
2	Ito *et al.* (2001)	Humid Continental (Japan)	+8°C (Winter)	0.25kW (Heating) ----- 2.7 – 3.2 (COP)	Laboratory based experimental study using historical climate data to simulate source temperatures and duty cycles. 11m long BHE. Vertical collector was supplemented with thermal energy from ambient air when heat flux was positive. Assumed a fixed drawdown (ΔT_G).
3	Hepbasli *et al.* (2003)	Dry Summer Subtropical (Turkey)	+5.5 to +13.2°C (Winter)	3.8kW (Heating) ----- 1.3 - 1.6 (COP)	Experimental study over 4 months. Single U-tube, 50m long BHE. Heat pump system generated low COPs due to collector sizing error. Soil moisture is a key element in maximising heat transfer (Turkish climate offers extreme variation in rainwater supply).
4	Huang *et al.* (2003)	Humid Continental (USA)	+12 to +15°C (Winter)	Not Presented	Numerical evaluation of heat pump performance with variable duty cycles. Highlighted need to carefully balance vertical collector thermal balance if conduction is the only form of heat movement in the ground (i.e. no groundwater movement). Uses a fixed collector drawdown, but reduces the ground temperature with increased duty.
5	Al-Huthaili, S. (2004)	Cool Marine (UK)	+3 to +6°C (Winter)	2.98 – 3.32 (COP)	Experimental study conducted over 2 months. No specific collector design methodology was utilised. Ground temperature drawdown not determined (ΔT_G), heat pump COP analysis only.
6	Romero *et al.* (2005)	Summer Subtropical (Spain)	+18°C (Year average)	Not Presented	Experimental study conducted over 1 year. Study evaluated $GSHP_{VC}$ & ASHP COP and climatic conditions. Revealing a higher efficiency performance by the $GSHP_{VC}$ over the ASHP, where vertical collector offered superior source temperature stability, particularly where a similar amount of heating and cooling is required. Study highlighted impact of climate on heat pump performance. Study concluded that, at this geographic location, the $GSHP_{VC}$ system has efficiency up to 32-36% better than the ASHP over the heating season and 50-60% more efficient over the cooling season.
7	Lindholm *et al.* (2005)	Cool Littoral (Sweden)	0 to +6°C (Winter)	3.0 (Simulated COP)	Numerical study revealed a novel design utilising both the air and ground as heat pump dual sources. No experimental heat pump performance data and therefore revealed little insight into the impact of heat pump duty on drawdown (ΔT_G).

Eugster and Rybach (2000) monitored the thermal extraction of a borehole heat exchanger in operation for over thirty years in Zurich, Switzerland, where a Moderate Subpolar Climate exists. Using the experimental results of borehole thermal extraction a simulation tool was developed to show the potential impact of excessive extraction, which reduces collector return temperatures and thus reduces year-on-year heat pump performance. This study however was limited in its scope and only examined the effects of prolonged thermal extraction from "dry" boreholes with no groundwater movement. It did however highlight the need to examine borehole performance over a number of years to generate reliable performance data and this approach was taken in the *HP-IRL* study.

Ito *et al.* (2001) carried out a laboratory based study in the Japanese *Humid Continental* climate which utilised an ASHP and $GSHP_{VC}$ in series. The ambient air was used to 'boost' the temperature of the ground as a dual source whenever the ambient air temperature exceeded the fluid temperature returning from the ground by more than 2°C. Heat pump testing was conducted under laboratory conditions and the experiment was designed to highlight COP sensitivity over a range of input temperatures regulated using a temperature controlled bath. The authors concluded that there was merit in using the ambient air source when its temperature rose above that of the ground. The system was determined to be effective in increasing the overall COP although no specific level of COP improvement was defined. However, it was stated that this approach was highly dependent on suitable climatic conditions, requiring both heating and cooling to be economically feasible.

In an experimental evaluation of a vertical collector GSHP in the Turkish *Dry Summer Subtropical* climate, Hepbasli *et al.* (2003) reported on a four month long, winter study. The rated capacity of the heat pump was 3.8kW with a 50m single U-tube BHE. The COP ranged between 1.4 and 1.65, significantly lower than the manufacturers rated COP of 4. The large discrepancy was attributed to under-sizing of the evaporative heat exchanger, highlighting the need for careful collector design. This experimental evaluation showed the need to not just monitor the heat pump efficiency but equally the overall system, including the borehole. This research reinforced the need to evaluate the heat pump system during operation in order to fully understand both the heat pump and collector performance. The need for climate specific design was illustrated by the need to design the collector to satisfy the cooling and heating needs equally.

Huang and Murphy (2003) numerically investigated the effects of building operation on $GSHP_{VC}$ performance in a school exposed to the *Humid Continental* climate of Kentucky, USA. The study recognised the importance of balancing the net thermal energy injected and extracted energy from the borehole so that long-term temperature stability was achieved. The climate, along with a large amount of heat gain from electrical equipment and students, generated a larger requirement for cooling that heating. However, since the school was closed in summer, the seasonal thermal imbalance was not a problem. While this numerical model only applied to conditions where no ground water movement was presented, this study highlighted the need to carefully control demand in order to maintain long-term temperature stability.

In a UK (*Cool Marine*) based study, Al-Huthaili (2004) developed a novel collector that utilised run off rainwater as a heat pump thermal reservoir, supplemented with vertical collector heat pipes. The study evaluated the vertical collector system for two months of winter operation. While this study was limited in its scope and duration it recognised that optimal heat pump performance was dependant on suitable ground conditions and highlighted the reduced potential of dry boreholes due to sub-optimal ground moisture content, even in Maritime climates. As a result, this study called for a more thorough investigation into collector designs in Maritime climates.

Research conducted by Romero *et al.* (2005) and Urchueguia *et al.* (2006) compared a $GSHP_{VC}$ with an ASHP in the Spanish Subtropical (*Mediterranean*) Climate. The experimental investigation was conducted over one year, with large seasonal variations in the ambient air temperature requiring both heating and cooling. In this climatic region satisfactory year-round indoor thermal comfort relies equally on heating and cooling. A key finding suggested that the ground offered superior energy saving potential over air source due to its more stable temperature, which reduced the temperature lift (ΔT_{HP}) in both the heating and cooling modes. As a result, the $GSHP_{VC}$ technology outperformed the ASHP system by 32-36% in the heating season and by 50-60% during the cooling season. However, such increased returns were likely to be diminished by high vertical collector installation cost. This research highlighted:

- The significant impact of climate in driving heat pump demand and hence COP
- The importance of establishing system capability within a climate
- The need to justify or accommodate the initial system capital installation cost

However, while the ability of the $GSHP_{VC}$ to outperform the ASHP in cooling mode was emphasised, the applicability of these findings may not translate to Ireland, where cooling demand is lower and heating demand is moderate and prolonged.

Lindholm *et al.* (2005) carried out a numerical investigation into the combined use of the ambient outside air and the ground as alternate sources for a single heat pump operated in the *Cool Littoral* climate of Gothenburg, Sweden. The system drew its energy from the air while its temperature exceeded +3°C, a condition which existed for 77% of the time, and then reverted to the borehole for the remaining 23% (2000 hours) of the heating season. The overall SPF was 3, with the individual $GSHP_{VC}$ and ASHP systems being 3.1 and 2.7 respectively. While the single source $GSHP_{VC}$ system had a higher SPF (3.1) than the combined source system (SPF 3.0), the combined system can be cost effective in terms of reduced ground space and initial capital cost. This hybrid design may have applicability in Ireland where, based on 2007 data, the air temperature drops below +3°C for just 2.5% of the time. As a potential application within the milder Irish climate (below +3°C 2.5% of year in Ireland versus 23% in Sweden) the ASHP system would provide a substantially greater proportion of heat than the vertical collector, enabling the vertical collector size to be considerably reduced and utilised only in severely cold conditions.

In Summary, experimental $GSHP_{VC}$ studies over the past decade have primarily concentrated on borehole thermal evaluation by means of the Thermal Response Test (TRT). While this testing may return a bulk thermal conductivity value for the ground surrounding the collector, it does not offer any indication of the variation in source temperature with time and the effects of groundwater movement on the thermal energy provision. Indeed, transient responses to long term thermal extraction from boreholes have been somewhat overlooked. The recent studies outlined in Table 2.6 illustrate ways in which $GSHP_{VC}$ system performance is evaluated with regard to time, climate and compared to the performance of other heat pump (collector) technologies. However, it is noticeable from the Table that the studies are generally limited to short-term evaluation of borehole thermal characteristics (fifty hours up to one year). The *HP-IRL* study has drawn on the findings of these key studies to inform the $GSHP_{VC}$ characterisation methods and performance evaluation criteria. These were deployed over three heating seasons with a view to establishing the influence of duty on drawdown, long-term operation on thermal fatigue and possible impact of climate on heat pump performance.

2.4.3 VERTICAL GROUND SOURCE

Vertical collectors are used where ground space is limited and while the drilling costs are high, it offers a potentially higher source temperature and heat transfer coefficient than horizontal collectors (Hepbasli and Kalinci, 2009).

A general rule of thumb for vertical collector sizing is that 1m of vertical borehole should be provided for each 50 to 80W of heat pump capacity (ASHRAE, 2003 - 2006). Alternatively it can also be expressed as 100 to 150kWh/(m·annum) (VDI 4640 / Part 2, 2001). This is subject to suitable pipe diameter and number of collector pipes per borehole. The vertical collector in this *HP-IRL* study was designed based on 50 W/m of heat pump capacity.

As there is a substantial distance between the vertical collector's thermal energy source and the collector piping, it may take a number of years of heat pump operation to establish steady-state thermal conditions (Ekelof and Gehlin, 1996; Halozan, 2008). This is also dependent on the net balance of thermal energy collected/available/injected, where a system that is used for heating purposes (thermal extraction only) will take time to reach thermal equilibrium (Huang and Murphy, 2003). However, a system that provides both heating and cooling has a much smaller net balance of thermal energy, where possibly as much thermal energy is rejected as collected and the initial thermal equilibrium is maintained. The typical trend in heating only systems start with high collector extraction temperature during the first year of operation and this drifts lower over the subsequent three to five years before stabilising (Halozan, 2008).

Figure 2.11 shows the change in collector temperature of heating only $GSHP_{VC}$ system (single, co-axial 105m vertical collector) in Switzerland from an initial high in the year 1986 and stabilising over the preceding three years (Eugster and Rybach, 2000). It shows a drawdown from the farfield surroundings (indicated as the red line) of approximately 1°C in temperature over the first year of operation and slows down over the following number of years monitoring. In this example significant geothermal heating occurs, generating a temperature rise of 2.4°C per 100m depth.

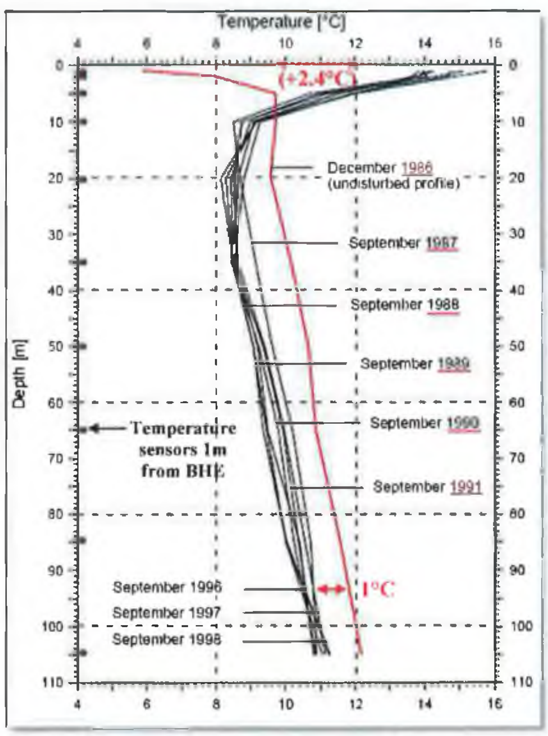


FIGURE 2.11 TEMPERATURE DRAWDOWN OF A VERTICAL COLLECTOR MONITORED OVER A FIVE YEARS OPERATION IN SWITZERLAND (EUGSTER AND RYBACH, 2000).

In Figure 2.11 temperature sensors were installed at 0.5 and 1.0m horizontal distance from the borehole and at depths of 1, 2, 5, 10, 20, 35, 50, 65, 85 and 105m. Temperatures were recorded at 30 minute intervals. Recording was conducted over an initial 5 years from 1986 to 1991 and then a further 3 years from 1996 to 1998.

Figure 2.12 shows the modelled radial temperature effects of short and long term thermal extraction from the same vertical collector in Switzerland over a thirty year period.

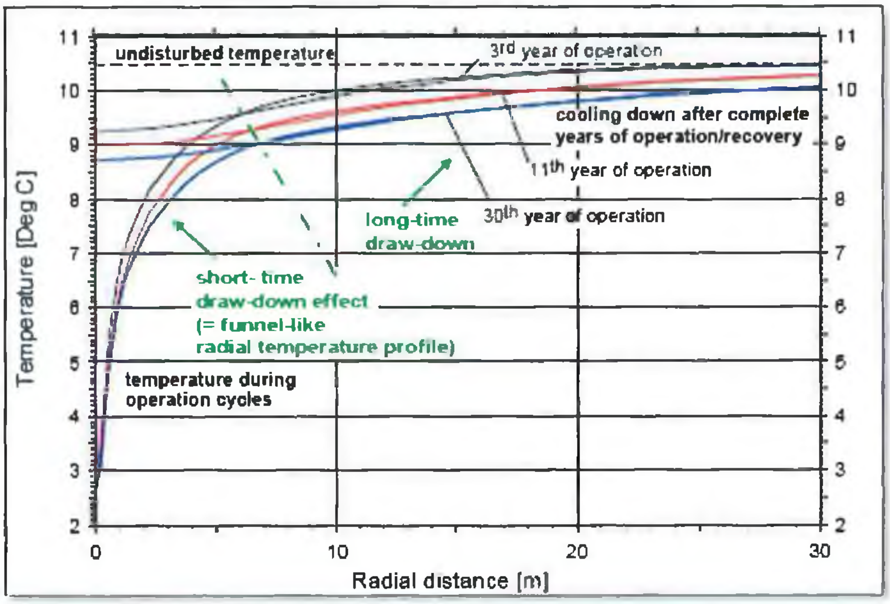


FIGURE 2.12 POTENTIAL TEMPERATURE DRAWDOWN AT VARIOUS RADIAL DISTANCES FROM THE VERTICAL COLLECTOR OVER THIRTY YEARS OF OPERATION IN SWITZERLAND (EUGSTER AND RYBACH, 2000).

It is noticeable from Figure 2.12 that the undisturbed farfield temperature is +10.5°C and the collector fluid temperature is around +3°C, indicating a substantial drawdown of 7.5K on the source temperature. It also highlights the potential for thermal interaction between boreholes is most acute within a radial distance of 10m.

Borehole thermal resistance (R_b) is an important factor in the successful operation of a vertical collector (Eugster and Rybach, 2000). The borehole region that envelops the vertical collector plays a critical role in facilitating thermal transfer and must be designed correctly in order to supply the highest temperatures possible (Eugster and Rybach, 2000). This is dependant upon the thermal resistance between the brine circulating fluid in the collector (T_f) and the temperature of the borehole wall (T_b), under a specific heat transfer rate $Q_{VC}^{''}$ (W/m) (Gehlin, 2002). This can be represented by Equation 2.2:

$$T_f - T_b = R_b \cdot Q_{VC}^{''} \qquad \textit{Equation 2.2}$$

where T_b will vary over the initial period of time (hours) when delivering a constant heat input, and thereafter the temperature difference ($\Delta T = T_f - T_b$) will remain constant (steady state conditions). Assuming that the borehole thermal resistance remains constant, the ΔT becomes proportional to the heat extraction rate, Q''_{VC}. If designed correctly, using thermally enhanced grout material, the thermal resistance of the borehole should remain constant over the heating season (Gehlin, 2002).

The thermal performance of the overall borehole system, subjected to specific heating or cooling loads, depends not only on the borehole thermal resistance but also on the transient thermal resistance of the surrounding ground (Gehlin, 2002). Furthermore, the influence of boreholes on one another is important as closely spaced boreholes can interact thermally (Gehlin, 2002). The impact of such interference is determined by the duration and intensity of the interaction (Gehlin, 2002).

A temperature penalty, T_p, relates to the adverse effects that can arise as a result of long term heat pump operation with vertical collectors. In terms of its maximum deliverable heating capacity over an annual basis, energy in via solar and geothermal gradients will equal energy out via the vertical collector (Gehlin, 2002). T_p is the temperature reduction of the collector fluid return temperature over time due to slow depletion of the ground's thermal capacity along with possible interference from adjacent vertical collectors. Many investigations have been carried out into the long term effects of vertical collector interaction and a multitude of design guidelines have been developed defining minimum spacing between adjacent vertical collectors, from 4.5m to 15m (Kavanaugh and Rafferty, 1997; VDI 4640 / Part 2, 2001; Reub and Sanner, 2001; Kohl *et al.*, 2002; Le Feuvre, 2007; Michopoulos and Kyriakis, 2009). However, these investigations have generally considered the implications of thermal interference with a balanced, or near balanced system where heat extraction and injection work in series over the year (Kavanaugh and Rafferty, 1997). In heating only applications, the general design guideline is to compensate for these T_p penalties by increasing the vertical collector length and increasing the distance between adjacent boreholes (Gehlin, 2002). This approach has the effect of increasing the initial cost of installation and reducing the overall capacity of the installation per unit site area. Indeed, potential future use of vertical collector heat pump systems in high density populations need to be cognisant of the potential for neighbouring vertical collector boreholes/systems effecting each other.

As rock conditions can vary greatly, so too can the thermal conductivity. Ireland's ground normally consists of metamorphic and crystalline rock formations, with some deposits of igneous rock (GSI, 2004). The most sizable rock formation is that of carboniferous limestone covering a large portion of the midlands (GSI, 2004). The vertical collectors monitored in this *HP-IRL* study were installed in carboniferous limestone rock, with a conductivity value of approximately 4 W/m·K. Figure 2.13 gives the conductivity range for some common rock and soil types. Further detail on rock types in Ireland is presented in Appendix D.

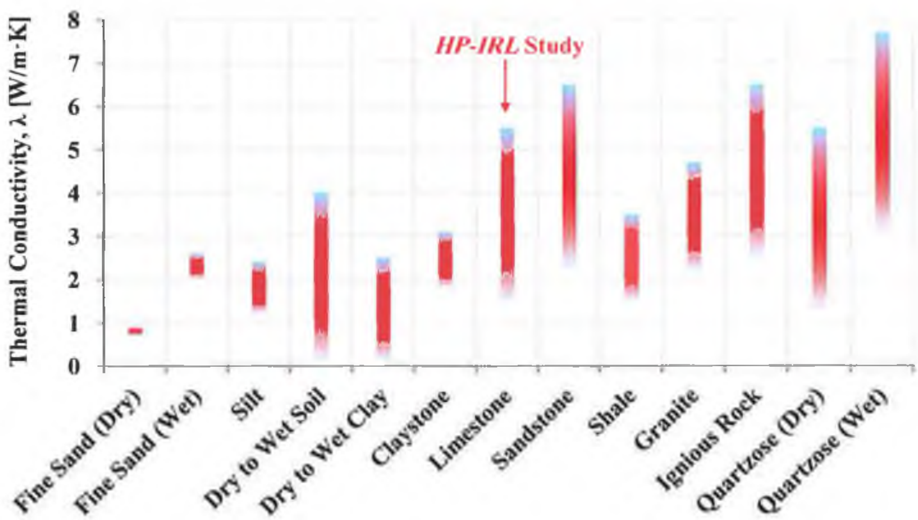


FIGURE 2.13 THERMAL CONDUCTIVITY OF TYPICAL SOIL AND ROCK TYPES (IGSHPA *ET AL.*, 1989).

From a geothermal perspective, O'Brien (1987) carried out an analysis of the potential offered by high temperature springs in Ireland and revealed that Ireland possessed just a modest amount of hot springs, with temperatures in the region of +20°C to +30°C, but these resources are sparse throughout the country.

In Ireland, Allen *et al.* (2003) identified and quantified the potential for elevated temperatures exhibited by certain features of the ground, such as that generated by the insulation effect of the surrounding buildings and increased albedo on roads and pathways, which is described as the "*heat island*" effect. It points out that due to the heat island effect there is rich potential for GSHP use of gravel filled buried valleys. Due to the gravels porosity they are filled with groundwater and have in some cases an elevated temperature above that of Ireland's mean ground temperature by between 3-4K (Allen and Milenic, 2003). However, this study did not extend to heat pump testing or modelling.

O'Connell *et al.* (2005) assessed the potential of deep bore geothermal resources in Ireland and in agreement with O'Brien (1987), concluded that Ireland has only modest opportunities to benefit from high temperature (+150°C), direct use, heat from the ground. At best Ireland has just 20 to 30 sites that deliver low grade heat of around +20°C.

For the application of deep borehole geothermal systems, CSA Group was contracted by Sustainable Energy Ireland (SEI) in 2006 to generate a temperature gradient map of Ireland. The map relied on the data from 80 sites and yielded ground temperature gradients at depths of 100m, 500m, 1000m, 2500m and 5000m. as illustrated in Table 2.7 the temperatures at 500m varied from +18°C in the south to +26°C in the north and at 2,500m temperatures varied from +28°C to +45°C in the south to +64°C to +97°C in the north (SEI, 2004).

TABLE 2.7 ASSESSING IRELAND'S DEEP BOREHOLE THERMAL RESOURCE TO DEPTHS OF 2,500M (SEI, 2004)

Location	Depth: 500m	Thermal Gradient	Depth: 2,500m	Thermal Gradient
Southern Ireland	+18°C	1.6°C per 100m	+28°C to +45°C	0.7°C to 1.4°C per 100m
Northern Ireland	+26°C	3.2°C per 100m	+64°C to +97°C	2.2°C to 3.5°C per 100m

2.4.4 SUMMARY

Vertical collectors are used where ground space is limited and offers a potentially higher source temperature and heat transfer coefficient than horizontal collectors (Hepbasli and Kalinci, 2009). However, care must be taken in designing BHEs that minimise borehole thermal resistance and consideration is given to the potential for a year-on-year reduction in source temperature. Indeed, while the short-term TRT can give an indication as to the thermal conductivity of the ground surrounding the BHE, it will not give any indication of the year-on-year source temperature reduction and may not be a reliable indicator in areas with large groundwater movement. Vertical collectors source temperature show the least sensitivity to climate effects (Lindholm *et al.*, 2005). While geothermal gradients offer a certain improvement in the BHE source temperature, Ireland is not endowed with a large geothermal resource (O'Brien, 1987; O'Connell *et al.*, 2005) and thus heat pump are employed to boost the low temperature thermal resource to usable temperature levels.

2.5 AIR SOURCE HEAT PUMPS (ASHPS)

Since ASHPs draw thermal energy from the air, they display a greater sensitivity to climate and this section reviews the studies that characterise the performance of ASHPs in different climatic regions. It also highlights the effect of weather parameters and system design improvement on ASHP performance.

2.5.1 CLIMATE SENSITIVE HEAT PUMP STUDIES

Table 2.8 details the thirteen climate sensitive studies that have been performed in six different climates and operational performance achieved.

TABLE 2.8 SUMMARY OF KEY ASHP STUDIES UNDERTAKEN TO IDENTIFY THE IMPACT OF CLIMATE ON ASHP PERFORMANCE

No.	Researcher (Year)	Climate Type (Country)	Ambient Air Temperature Range	Heat Pump Performance	Key Findings
1	Parker and White (1982)	Humid Continental (USA)	-	-	Two year comparative study between an ASHP and a $GSHP_{HC}$. ASHP system proved inadequate as a monovalent system, requiring additional resistance heating if ambient air temperature dropped below 32°F (0°C).
2	O'Conner and McGovern (1982)	**Cool Marine** (Ireland)	-	A/A – 1.2 to 1.6 (COP) A/W – 2.26 and 2.47 (COP)	Trials conducted between 1975 and 1979. Recommended that further research required to improve domestic heat pump performance to suit Maritime climates.
3	Rosell *et al.* (1983)	**Cool Marine** (Ireland)	-	2.5 (SPF)	Study conducted over six months. Recorded SPF with no other parameter results.
4	Reistad *et al.* (1984)	**Cool Marine** (USA)	-	1.7 – 2.1 (COP)	Study conducted over 3 months. Climate similar to Ireland's where testing concluded that an ASHP has the potential to be a cost effective system for mild climate regions.
5	Kent, E.F. (1997)	Dry Summer Subtropical (Turkey)	+5.1°C (Min) +23.3°C (Max) +13.8°C (Avg)	2.0 – 3.2 (COP Range)	Evaluation of an ASHP system over 8 months of the heating season. System performance highly dependent on source temperature (1°C ambient air temperature change shows 12% drop in COP).
6	De Swardt *et al.* (2001)	Semi-arid Tropical (South Africa)	+11°C (Min) +23°C (Max) +18°C (Avg)	3.1 – 3.9 (COP_{COOL}) 3.1 – 3.3 (COP_{HEAT})	Experimental evaluation of ASHP and WSHP systems over 1 year. ASHP performance is economical in this climate, but is predominantly required for cooling purposes. Historical climate data utilised in simulations was not detailed enough to show effect of variation on COP.
7	Lam and Chan (2003)	Trade Wind Littoral (China)	-	1.5 – 2.4 (Heating COP)	Experimental study over 6 months. Very detailed account of the experimental design and followed international standards. Heat pump SPF analysis only. Lacked sufficient depth in analysis of the climatic (weather) effects on heat pump performance.
8	Guoyuan *et al.* (2003)	Humid Subtropical (China)	-	NA	Experimental (laboratory) investigation of an ASHP with a two-stage compressor and simulated external ambient air temperatures. System shown to be potentially successful in extremely cold climates where minimum winter ambient air temperatures of below -15°C are experienced. The analysis the ASHP compared favourably in these conditions with that of other mainstream heating systems. Study focused mainly on heat pump performance evaluation with minimal climate evaluation.
9	Marcic, M. (2004)	Humid Continental Climate (Slovenia)	+5.1°C (Winter Average)	3.1 – 3.2 (Heating season COP)	Study conducted over 9 years. (1989 – 1998). The ASHP worked as part of a bivalent heating system, with a condensing oil boiler. The ASHP provided 80% of the thermal energy requirements of the house (74% of days), and over the test period achieved an average COP of 3.16. It contributes to the design parameters for the installation of ASHPs in that, for certain climatic regions, it may need to be integrated into a bivalent system in order to maintain an acceptable efficiency, enable reliability and remain cost effective.
10	Blanco-Castro *et al.* (2005)	Dry Summer Subtropical (Spain)	+11.5°C (Jan. Average)	3.1 – 3.3 (SPF_{HEAT}) 3.1 – 3.4 (SPF_{COOL})	Experimental (laboratory) investigation of ASHP operating in a *Dry Summer Subtropical* Mediterranean Climate. Laboratory experiment in a climate controlled chamber using historical climate data (30 year average). Performed in accordance to test standards EN 255-1 and EN 12055. A conclusion drawn from this study is that the use of historical climate data to predict heat pump performance is not a simple application as it can lack sufficient resolution, and historical ambient air temperatures alone may not reflect actual thermal energy usage patterns.

TABLE 2.8 CONTINUED.

No.	Researcher (Year)	Climate type (Country)	Ambient Air Temperature Range	Heat Pump Performance	Key Findings
11	Hewitt and Huang (2008)	**Cool Marine** (Northern Ireland)	NA	NA	Study evaluating the performance of a novel ASHP evaporator. Laboratory testing carried out in accordance with EN 14511 standard. Study showed the potential for ASHP systems to operate as a standalone heating system in the favourable Maritime climate, provide an adequate defrost strategy is employed.
12	Verhelst *et al.* (2008)	Cool Littoral (Belgium)	+5.7°C (Winter Average) +15.7°C (Summer Average)	2.4 – 4.1 (COP Range) 3.0 (Annual SPF) 3.5 (Heating season SPF)	Experimental study conducted over 1 year. Study monitored the operation (COP & SPF) of an ASHP with regard to climatic influences. Backup heating is required when ambient air temperature dropped below 0°C. Study highlights the need for intelligent control in order to maintain economic performance under cold conditions. Study clearly indicates the performance capability of an ASHP even under cold winter temperatures.

As outlined in Section 2.3.3.1, Parker and White (1982) contrasted the characteristics of a $GSHP_{HC}$ and an ASHP, which showed that in *Humid Continental* climates under "cold" conditions (32°F, 0°C), the $GSHP_{HC}$ was capable of furnishing adequate heat to the building, but the ASHP capacity would drop and supplementary resistance heating was required. This study concluded that the ASHP may not be a particularly economic solution under harsh winter conditions (*Humid Continental*). However, as this study was conducted over 25 years ago its findings may not be replicated, especially as this *HP-IRL* study is being conducted under the mild *Cool Marine* climate of Ireland, utilising modern ASHP technology.

O'Connor and McGovern (1979), in a study based in Ireland, conducted experimental testing of the use of air source heat pumps for domestic heating and identified that Ireland offered an ideal climate for such systems. In a follow-on study the same research group carried out field trials into the application of ASHPs to establish COP's in the range of 1.2 to 1.6 for an air-to-air heat pump and between 2.26 and 2.47 for an air-to-water heat pump (O'Connor *et al.*, 1982). The trials were conducted between 1975 and 1979. Aptly, while this research concluded that the capital cost of these systems was prohibitive it identified the need "*....for further research to improve domestic heat pump performance to suit* Maritime *climates*" to avoid over-dependence on oil.

Rosell *et al.* (1983) developed, installed and monitored the use of an ASHP in a family home over one winter period in northern Ireland (*Cool Marine*). Their results show the ASHPs performance (COP) to be highly sensitive to fluctuations in external ambient air temperature and, although the climate delivered relatively high and stable ambient air temperatures, the heat pump offered a SPF of 2.5. Such a low SPF reflects early stage technology, along with the constantly high output temperature demand of up to +50°C. This research does however, by it own admission, lack the resolution needed in order to evaluate the system under specific weather conditions as it only monitored the daily electrical consumption and matched it to the average daily ambient air temperature to generate a daily COP. Indeed, it was assumed that the relative humidity level in the external ambient air was unimportant.

The experimental investigation performed over three months by Reistad *et al.* (1984) combined two thermal sources, utilising the best thermal source at any given time. The study was conducted in the *Cool Marine* climate of Oregon, USA, similar to that of Ireland. The concept evaluated the use of ambient air as the heat pump thermal source until the ambient air temperature reduced to a changeover temperature of +3°C where the water source was then utilised. Although the system utilised mains water as the water source the concept gave a detailed account of the contrasting nature of both sources of thermal energy, detailing how ambient air can be a better thermal source than the ground at certain times of the day and/or year. The combined air and water source solution fared better than the ASHP, but worse than the water source heat pump (WSHP). The additional cost of the combined system proved to be negligible and with better control along with an intelligent design for the changeover temperature the dual source system could be feasible. This investigation concluded that an ASHP has the potential to be cost effective for mild climate regions.

Kent (1997) evaluated the performance of a compact air-to-air heat pump system over an eight month heating season in Istanbul, Turkey (*Dry Summer Subtropical* climate). The heat pump performance characteristics show high sensitivity to ambient air temperature, where a 1°C change in ambient air temperature generated a 12% change in COP, as shown in Figure 2.14. The study did not define the SPF and did not indicate whether the system was cost effective.

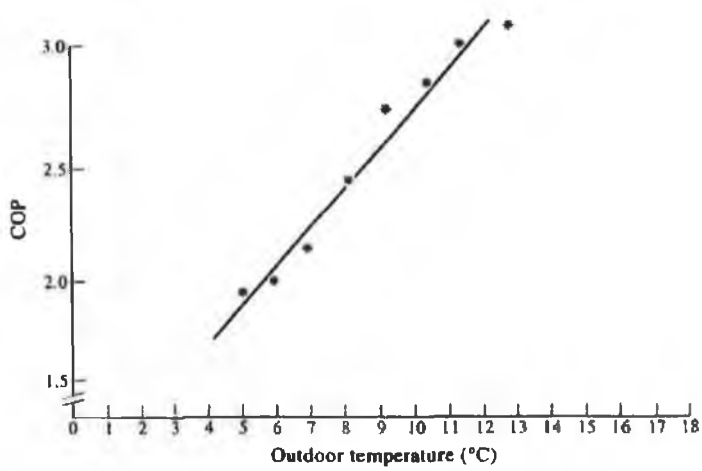


FIGURE 2.14 VARIATION IN ASHP COP WITH AMBIENT AIR TEMPERATURE (KENT, 1997).

It is noticeable that the system performance shown in Figure 2.14 would not be sufficient to operate economically in the Irish climate, where the average winter temperature is approximately +5°C (+8°C for Istanbul), returning a COP of just 2.

An experimental performance comparison between an ASHP and a WSHP was carried out over one year by De Swardt and Meyer (2001) in Johannesburg, South Africa (*Semiarid Tropical* climate). Measurements included; fluid temperature and flowrate, ambient air temperature (wet and dry bulb) and flowrates for both heat pumps. To characterise the impact of heat pump's operation on the source, ambient air and ground temperatures (to a depth of 3m) were measured and recorded. Relative humidity data was also obtained from a local climatological database and typical data is shown in Figure 2.15. Using the recorded data a numerical model was developed to simulate the heat pump operation using the climatological data.

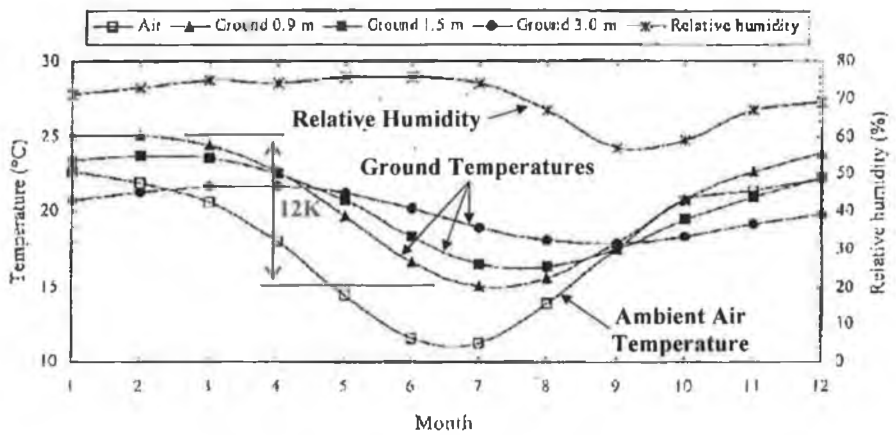


FIGURE 2.15 SOUTH AFRICAN MONTHLY AMBIENT AIR AND GROUND TEMPERATURES (DE SWARDT AND MEYER, 2001).

Ground temperatures shown in Figure 2.15 were recorded outside the horizontal collector region and did not capture the impact of heat pump operation.

Figure 2.16 presents predicted results which show that the GSHP COP was not greatly affected by variation in ground temperature. Possibly reflecting the lack of sensitivity between the ground temperature and COP shown in Figure 2.16, the authors also concluded that using daily averaged ambient air temperature and relative humidity data may not offer sufficient resolution to predict the performance of a ground source heat pump collector.

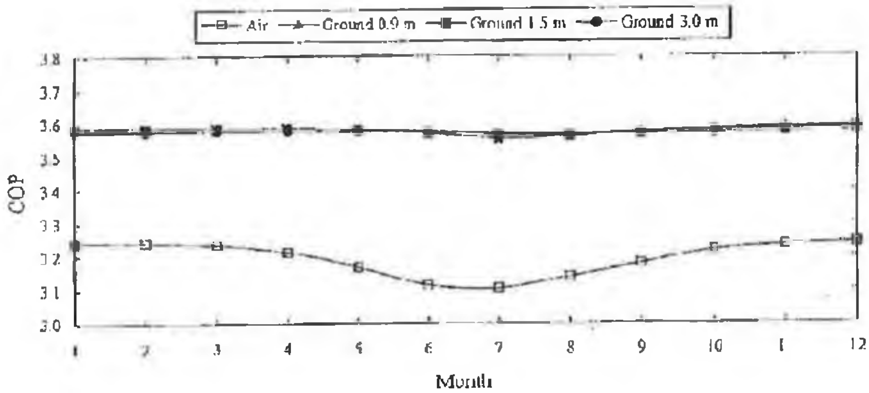


FIGURE 2.16 PREDICTED MONTHLY HEATING COPS FOR AIR AND GROUND SOURCE HEAT PUMP SYSTEMS (DE SWARDT AND MEYER, 2001).

In the context of heat pump performance variation with changing temperature lift (ΔT_{HP}) it is difficult to explain such invariable numerical COP predictions in Figure 2.16 given that there is a 12K variation in ground temperature annually shown in Figure 2.15. However, despite these results the study has once again raised the issue of the impact of climate on heat pump performance from a number of perspectives, and *HP-IRL* sought to remove such uncertainties through experimental measurement.

Lam and Chan (2003) compared the performance of an ASHP and a WSHP (open loop well water) within the *Trade Wind Littoral* climate of Hong Kong, heating two hotel swimming pools. The experimental analysis was conducted over one heating season, spanning six months. Evaporator and condenser temperatures were recorded every 15 minutes, along with flow rates and electrical consumption. Comparable with a gas boiler system, the ASHP's COP varied between 1.5 and 2.4 over the heating season, and delivered a payback of two years, whereas the WSHP's COP remained stable at 1.7, and also delivered a two-year payback. The relatively short payback is due to the high and consistent demand for thermal energy by the swimming pool, along with moderate installation costs. While this study did give a detailed account of the experimental design and followed international standards, the study lacked sufficient detail in its analysis of the climatic effects, ground source temperatures, drawdown on the source temperature and frost effects on the ASHP performance.

Guoyuan *et al.* (2003) showed in a laboratory investigation that an ASHP can be deployed successfully even in extremely cold climates, where winter ambient air temperatures of below -15°C are experienced, such as in Beijing, China (*Humid Subtropical*). However, such systems need to be designed and controlled correctly to perform efficiently. From their analysis, the ASHP compared favourably with that of other mainstream heating system types.

Marcic (2004) analysed the performance of an ASHP in the cold Slovenian climate (*Humid Continental*) over a period of 9 years (1989 – 1998). This study describes the climate of Maribor, Slovenia, as a "Central European Climate". The ASHP worked as part of a bivalent heating system, with a condensing oil boiler. The oil boiler provided for heating needs when the ambient air temperature dropped below 0°C. The ASHP provided 80% of the thermal energy requirements of the house (74% of days) and achieved an average COP of 3.16. Considering that the external ambient air temperature can be as low as -18°C, the condensing oil boiler provided 42% of the heating for the house with external ambient air temperatures of between 0°C and -18°C. It contributes to the design parameters for the installation of ASHPs in that, for certain climatic regions, it may need to be integrated into a bivalent system in order to maintain an acceptable efficiency, enable reliability and remain cost effective. It is also notable that the average winter temperature averages at +5°C for Slovenia, which is similar to that of Ireland. However, the Slovenian winter temperatures can drop below -1°C for periods of a month or more, which does not occur in Ireland.

Blanco Castro *et al.* (2005) carried out laboratory testing in a $50m^3$ climate controlled chamber to determine the SPF for an ASHP in Spain using historical climate data (*Dry Summer Subtropical Mediterranean* climate). They used historical thirty year average air temperature data to predict heat pump seasonal performance. However, they amended this input data to only include the average temperatures from 2001 to 2003, which they felt were more representative of current climatic conditions. Again and in agreement with De Swart and Meyer (2001) this study concluded that the use of a single climatic parameter such as ambient air temperature may not be enough to indicate true thermal energy usage patterns, or building heat loss. Laboratory testing using multiple real weather data inputs can also be ineffective in the evaluation of heat pumps seasonal performance if the weather data lacks sufficient detail and resolution, which was the case in this study. This study does however extend the performance of a heat pump to include the climate data which informed optimum evaporator design. Such results influenced the *HP-IRL* study to monitor all aspects of the local climate, as well as simultaneously monitoring the ground condition and heat pump performance.

In a Belgian *Cool Littoral* climate context, Verhelst *et al.* (2008) evaluated an air-to-water heat pump system and monitored its performance characteristics over a one year period. The ASHP delivered space heating and domestic hot water (DHW) for a low energy house in Belgium. The ASHP system was supplemented by an immersion heater when the ambient air temperature dropped below 0°C, delivering 6% of annual heat demand. The year-round SPF was 3.0, with a winter heating season SPF of 3.5. Figure 2.17 shows the recorded COP as a function of ambient air temperature over the winter heating season of 2006/2007, where "PF" is the weekly performance factor (COP) and "PF DHW" is the weekly performance factor for domestic hot water production at +55°C.

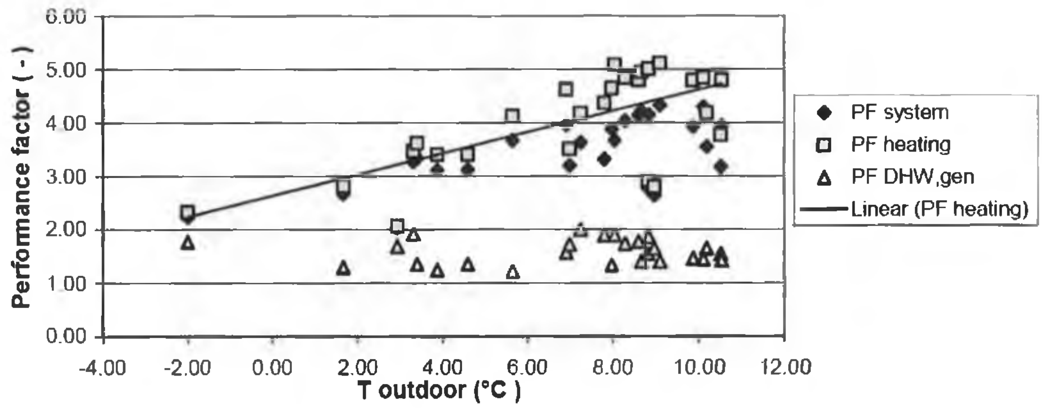


FIGURE 2.17 ASHP PERFORMANCE DATA AS A FUNCTION OF EXTERNAL AMBIENT AIR TEMPERATURE IN 2006/2007 (VERHELST *ET AL.*, 2008).

The summer-time heat demand was primarily DHW and as it required a high delivery temperature (+55°C) it had a negative effect on the heat pump performance. This study highlighted the importance of heat pump evaluation for climate specific operation and further illustrated that ASHP systems can be cost effective as a primary heating system. However, the evaluation did not detail heat pump performance versus climate, where for example, no relative humidity was monitored and no timeframe was given for when the supplementary immersion heating was required, only indicating the percentage of total heat delivered. This study has however provided an important insight into the need to develop climate specific ASHP integration within the built environment and the key role that climate sensitive heat pump control could play in optimising performance.

Hewitt and Huang (2008) in an northern Ireland based study (*Cool Marine*), tested a novel heat pump circular evaporator, where its performance was monitored while operating with a number of different defrost strategies. The study details the heat pump operation, performance characteristics (both capacity and COP) and defrost dynamics, noting the need for careful selection of the defrost strategy, particularly for performance in a Maritime (*Cool Marine*) climate with high humidity levels. Testing was carried out in a temperature and humidity controlled test chamber and in accordance with test standard EN 14511-2. This study showed the potential for ASHP systems to operate as a standalone heating system in the favourable Maritime climate, provided a suitable defrost strategy is utilised.

large or expensive collector space and can be used to heat both air and water directly. Indeed, most ASHPs have the capacity to work effectively to ambient air temperatures as low as -20°C (Guoyuana *et al.*, 2003). While the risk of frosting and the associated need for defrosting increases as air temperature drops below +5°C this risk is lower in Maritime climates that typically offer winter ambient air temperatures between +5°C and +15°C. Ambient air is the most accessible and cheapest energy source for heat pumps. Despite this, air source heat pumps deliver their poorest performance (COP) when heating is most needed, as the cold ambient external air that drives the building heating demand will also require the heat pump to work harder to compensate for the higher temperature lift (Hewitt and Huang, 2008). Therefore, running costs are generally higher due to COP's that can be as much as 30% lower than GSHPs (De Swardt and Meyer, 2001; Fairey *et al.*, 2004). Hence this technology was added to the heat pump technology portfolio evaluated under this *HP-IRL* study.

The air source system requires air to be pulled across an evaporator by means of an electric fan in order to facilitate the heat transfer from the air to the primary refrigerant. This process can be improved with higher heat transfer characteristics such as increased natural wind speeds and higher humidity and some systems have looked at the use of rainwater for additional energy (Hino, 1995). Moisture in the air increases the specific heat capacity and thermal conductivity. However, the downside of this is an increased tendency for the evaporator to frost up, constrict air flow and ultimately stop heat exchange.

A major technical problem with ASHPs occurs when the ambient air temperature falls to a point in which airborne moisture condenses and freezes on the evaporator. Prolonged ice build up on the evaporator diminishes heat transfer rates and an energy consuming defrost capability is also required to remove the unwanted ice.

2.5.3 AIR SOURCE HEAT PUMP OPERATION

ASHPs display a number of advantages and disadvantages when compared with GSHPs. The main advantages include; more compact collectors, ease of installation and lower installation costs and these are balanced against disadvantages such as COP variation with fluctuating external ambient air temperature, performance drops generated by parasitic energy losses due to defrosting of ice build-up on the evaporator and fan induced noise.

Defrosting plays an important role in the successful operation and more importantly the efficiency of an ASHP (Donnellan, 2007; Karlsson and Fahlen, 2007). Evaporator frosting reduces both heat transfer and air flow across the evaporator, which decreases thermal performance. Frost formation on the evaporator occurs when the source air goes through two phase changes, from gas-to-liquid and then from liquid-to-solid. For the majority of the time the air that passes through the evaporator will to some degree experience the initial phase change from gas to liquid, generating a positive impact on COP, as a disproportionate amount of energy is released for the temperature drop. There are no negative repercussions on COP associated with this phase change, with the possible exception of condensation water build-up which may cause corrosion in costal regions. Once the evaporator drops below 0°C a second phase change occurs, transforming the liquid to solid (ice). This second phase change across the evaporator will also release a disproportionately high amount of thermal energy for the temperature drop in the ambient air. However, this gas-to-liquid change generates an ice build-up on the evaporator which tends to increase both the pressure drop which reduces air flow, and thermal resistance which reduces heat transfer. The heat pump must then trigger defrost mode, which has negative impact on COP because a similar amount of energy is required to achieve defrost and this energy is typically lost to the environment, reducing COP.

From an efficiency perspective, the greatest risk of frost formation takes place with ambient temperatures between +5°C and +1°C, where lower air temperatures have low specific humidity levels and do not deposit appreciable amounts of ice on the evaporator (Lazzarin, 2007).

There are a number of defrost methods available including compressor shutdown defrosting, electric heat defrosting, reverse-cycle defrosting and hot-gas defrosting (Hewitt and Huang, 2006; Hewitt and Huang, 2008). The reverse cycle defrost is the most common defrost technique used by ASHPs (Hewitt and Huang, 2008) where the refrigerant that normally travels through the evaporator and picks up thermal energy from the air is reversed and heats the evaporator and thus melts the frost build-up, essentially switching the heat pump from the heating mode to the cooling mode. The disadvantage of this method is that the heat required to defrost the evaporator is taken from within the building interrupting the supply during defrost (Hewitt and Huang, 2008). Within reverse cycle defrost strategies the most commonly used method in industry is the time/temperature method (Donnellan, 2007). The most rudimentary and inefficient defrost cycle is executed using a timer only where the defrost cycle is initiated at predetermined time intervals regardless of need. More intelligent systems are capable of interpreting the temperature data to define a more appropriate defrost start time and duration. This is especially true in Maritime climates where the relative humidity throughout the heating season is consistently above 70% (Rohan, 1986; Garcia-Suarez *et al.*, 2002; Hickey *et al.*, 2003). For the time/temperature method the temperature sensor measures the conditions in which frost formation can occur and the timer measures the amount of time that has elapsed in this condition before a defrost cycle is initiated, and a defrost routine will continue for a specific amount of time, whether required or not. This simple technique has been improved upon with the introduction of more dynamic strategies such as multi-stage defrost, where various timed off and on reverse cycles are allowed to elapse in a controlled defrost, demand defrost cycles and multi-evaporator defrost cycles.

A demand defrost cycle senses the evaporator temperature and only initiates a defrost cycle if it detects the presence of ice and stops the defrost cycle when all ice is removed. A typical defrost duration is between two and ten minutes (Lazzarin, 2007). Further improvements on COP can be made by reducing the energy required for defrost or by recovering the heat emitted from the heat pump during the defrost cycle.

The measurement of an ASHP COP is carried out in accordance with the European standard EN 14511, where a typical set of source (dry-bulb air) temperatures and sink temperatures are evaluated, which are shown in Figure 2.18.

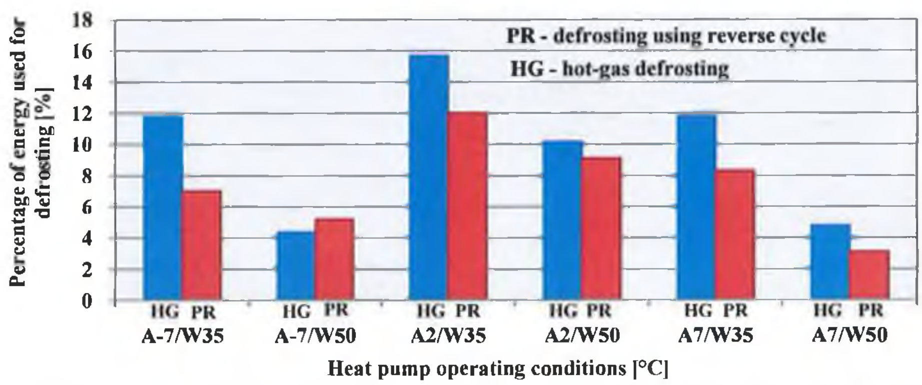


FIGURE 2.18 PERCENTAGE OF TOTAL HEAT PUMP ENERGY CONSUMPTION THAT IS PARASITIC FOR DEFROSTING PURPOSES (ZOGG, 2003).

Humidity for the testing must be in the range of 72.5% to 86.8%. Three characteristics are required in order to establish the operational capability: defrost time, interval between defrosts and the resultant COP. The test must be run for a number of cycles to reach a steady state before the test is validated.

The amount of energy required to defrost an ASHP ranges between 3% and 15% with an average of 8% (Lazzarin, 2007). Note that it exceeds 10% for air temperatures between +2°C and +7°C for all typical outputs shown. The greatest frost formation occurs when the entering air temperature is between +5°C and +1°C, below which the air's specific humidity is small and it does not produce significant ice formation (Lazzarin, 2007).

Differing levels of air humidity have an effect on the rate and density of the frost build-up, and also an increase in air velocity with less volumetric flow but an increased velocity leads to a reduction of thermal resistance and an increase in frost thickness and frost density (Lee *et al.*, 1996). As noted in Section 2.1, Ireland has a particularly humid Maritime climate. However, an increase in the flow speed across the evaporator will deliver a smaller temperature drop in the ambient air temperature than a slower flow rate, where for example if the ambient air temperature is above +5°C and by having a fast flow rate the exhaust air can be kept above 0°C, minimising the risk of frost formation.

Therefore, the frosting of the ASHP evaporator and the subsequent defrost cycle can have a substantial bearing on the system's seasonal efficiency. This is the biggest difference between the ASHP and the GSHP, since the latter never requires defrosting. However, many advances have been made to improve defrosting efficiency in recent years and the SPFs have increased by 15% over the past 20 years (Lazzarin, 2007) and can rival those of GSHP systems. One such ASHP system has been installed as part of the *HP-IRL* research project which has a novel defrost strategy that minimises parasitic heat losses by utilising two evaporators, details of which are provided in Section 3.5.

Good defrost strategies require a high standard of control and programming technology and cost can be a substantial factor. Since GSHP systems do not require defrost control the unit costs are generally cheaper. A more in-depth economic analysis of heat pump systems is presented in Section 7.2.

Historically, the coast of Ireland experiences on average ten frost days per year, whereas this increases to sixty per year inland (EPA, 2002). Drawdown of source temperature is not usually a problem for the ambient air source as both buoyancy forces and Ireland's generally windy climate ensure that ambient air can be seen as an infinitely large and constantly recharged source. Wind and buoyancy induced air flow negates the need for a large collector and additional pumping. Therefore, with frost build-up likely to generate the greatest problems, this *HP-IRL* study sought to select a heat pump and test program that would evaluate this aspect under Ireland's Maritime climate.

One of the biggest problems utilising ASHPs as a monovalent heating system is to enable the heat pump to satisfy the heating demand of the building under the climate's most extreme circumstances. This can lead to a significant oversizing of the system for a building's nominal heating requirements. Where oversizing is a problem, the use of variable speed compressors would help alleviate the problems that may arise in terms of lower than optimal evaporating temperatures, rapid frost formation, high amounts of system cycling and ultimately a reduced SPF (Duprez *et al.*, 2008). This oversizing problem is not as pronounced under the Irish Maritime climate where the ambient air temperature fluctuation during the heating season is low. However, the use of variable speed compressors is still advisable to improve SPFs (Duprez *et al.*, 2008). As part of the *HP-IRL* economic analysis of heat pump systems presented in Chapter 7, companies specified ASHPs for a specific dwelling with a design heat loss of 10kW that were 20% larger than GSHP systems.

ASHPs can generate greater noise emissions than GSHPs with 90% of noise emissions originating from the fans used for the circulation of ambient air across the evaporator (Zogg, 2003). Lowering the fan circulation speed, improving the airflow guidance into the evaporator, using low pressure drop evaporators and sound absorbing baffles can all help reduce noise emissions (Zogg, 2003).

2.5.4 SUMMARY

Cited experimental ASHP studies revealed how performance is highly dependant on favourable climatic conditions, in particular ambient air temperature and relative humidity (Rosell *et al.*, 1983; Reistad, 1984; Guoyuana *et al.*, 2003; Marcic, 2004). Depending on the technology, the Maritime climate of Ireland is therefore conducive to ASHP performance that is potentially comparable to that of GSHPs. Indeed While they reveal a general need for bivalent ASHP systems to overcome interruption in supply in continental climate regions during severely cold periods (Marcic, 2004; Verhelst *et al.*, 2008) there is sufficient evidence to suggest that ASHP systems can operate successfully as a stand-alone heating system with the right design and control within mild climates such as the Irish *Cool Marine* climate (Guoyuana *et al.*, 2003; Hewitt and Huang, 2008; Mustafa Omer, 2008).

Guided by these findings the *HP-IRL* study sought to conduct an experimental performance characterisation of a commercial ASHP, by simultaneously monitoring:

- all aspects of the local climate
- temperature delivered by the ASHP
- hourly COPs and ASHP system seasonal performance (SPF)

2.6 Literature review summary

This literature review sought to reveal the impact of climate on heat pump performance, identify the extent of previous studies in Maritime climate regions and to help define test methodologies for such investigations. The review was informed by over 50 previous studies of three heat pump collector types, along with a review of the climate types.

Complementing the emergence of generic collector design guides ((Kavanaugh and Rafferty, 1997; BRECSU, 2000; VDI 4640 / Part 2, 2001) further research has been called for in areas such as collector source side management (Karlsson and Fahlen, 2007) and climate exposed collector studies (ASHRAE, 2003 - 2006; Dumont and Frère, 2005; Florides and Kalogirou, 2007). While the understanding of climate sensitive heat pump performance continues to grow, the review shows that few studies have:

- focused on climate sensitive collector performance in any climate
- evaluated three heat pump collector types ($GSHP_{HC}$, $GSHP_{VC}$ ASHP) in one study
- ran for more than 12 months
- reported collector characteristics such as heat pump duty, collector return temperatures, collector pipe temperature, recovery rates or horizontal collector ground temperature drawdown
- provided details of the input temperature for their respective sources and tended to give the overall source temperature (i.e. the ground but not the input fluid temperature, or visa versa) and the sink temperature (i.e. +50°C)
- evaluated the link between the ground temperature drawdown, heat pump duty cycle and recovery periods
- developed the potential benefits of horizontal collector design that takes advantage of the mild aspects of the *Cool Marine* climate and be protected from the cold periods
- recognised the need for, or potential of, climate sensitive collector design coupled with improved source side management techniques

The *HP-IRL* study has been designed to facilitate a thorough investigation of these aspects under Ireland's *Cool Marine* climate, addressing each of these in a methodical way, using best practice approaches from the literature.

The following Chapter presents the experimental facility developed to allow this study to be undertaken.

CHAPTER 3 – EXPERIMENTAL INVESTIGATION

Based on the deficits identified by the *Literature Review* this *HP-IRL* study sought to establish a comprehensive test facility that would allow the performance of two heat pump technologies and three collector types to be established within the Irish Maritime climate.

The principle objective was to capture the performance of the two GSHPs and one ASHP while placing attention on the performance improvements that could be achieved through source side management (Karlsson and Fahlen, 2007). Combining these project objectives with established measurement techniques from the literature, gave rise to the test facility described in this chapter. Reflecting the almost exclusive need for heating it was decided that this facility would focus solely on heating mode.

3.1 EXPERIMENTAL FACILITY

This section provides details of five key aspects of the test facility; an automated weather station, $15kW_{th}$ horizontal collector heat pump, $15kW_{th}$ vertical collector heat pump, $8kW_{th}$ air source heat pump and 111 sensors and supporting data acquisition system which allowed the performance of the heat pumps to be continuously monitored.

The experimental design was influenced by previous Irish (O'Connor *et al.*, 1982; O'Brien, 1987; Mihalakakou *et al.*, 1996; Collins, 1998; D'Arcy, 2004; O'Connell and Cassidy, 2004) and international researchers (Eskilson, 1987; Piechowski, 1996; Kavanaugh and Rafferty, 1997; Austin, 1998; Zogou and Stamatelos, 1998; Chiasson *et al.*, 2000; Gehlin, 2002; Guoyuana *et al.*, 2003; Hepbasli *et al.*, 2003; Al-Huthaili, 2004; Blanco Castro *et al.*, 2005); established design guidelines (IGSHPA *et al.*, 1997; VDI 4640 / Part 2, 2001; ASHRAE, 2003 - 2006; Rawlings *et al.*, 2004) and calls made for further study in the literature.

The guiding principles followed during the design of the test facility were:

- replicate international best practice in experimental design and test methodologies
- develop a flexible test facility which complements existing studies, addresses recognised deficits and enables new insights and designs to be evaluated
- heat pumps were integrated within live applications
- different types of heat pumps could be simultaneously studied in the one climate, which allows performance to be compared

- develop tools that would allow heat pump performance to be enhanced and the contribution of heat pumps to sustainability to be advanced
- generate accurate test results that have a consistency, longevity and depth of detail to be used for later numerical studies

3.1.1 Innovation in Business Centre (IiBC)

The *HP-IRL* experimental investigation was conducted on GMIT's Dublin road campus between September 2006 and January 2010. As indicated in Figure 3.1 both GSHPs serviced a 1200m² nearby office building known as the Innovation in Business Centre (IiBC), at latitude 53°16'39" N and longitude 9°00'43" W. The heat pumps were an integrated part of this building's space heating system.

Figure 3.1 (a) Location of Galway in Ireland, (b) the GMIT campus and (c) the IiBC building (west side).

The IiBC is naturally ventilated, and has a maximum static fabric heat loss of 40 kW_{th} with a maximum static air infiltration heat loss of 20 kW_{th} (Stephens, 2003). This design heat loss is based on an external temperature of -2°C and an internal temperature of +21°C.

3.1.2 IiBC building's heating requirements and system

The IiBC accommodates approximately 70 people. The average casual heat dissipation per person in an office environment is 100 W and based on an 8 hour working day, 5 day week, contributes an average of 14,000 kWh/annum heat gain. Passive solar gain has been estimated at 9,000 kWh/annum (Stephens, 2003). However, the installed Building Energy Management System (BEMS) highlights the dominant heat gains from lighting and other electrical equipment as running at approximately 260,000 kWh/annum, corresponding to a constant 30 kW_e demand, varying between 20 kW_e at night and 40 kW_e during the day. The design heat loss and heat gain is shown in Figure 3.2.

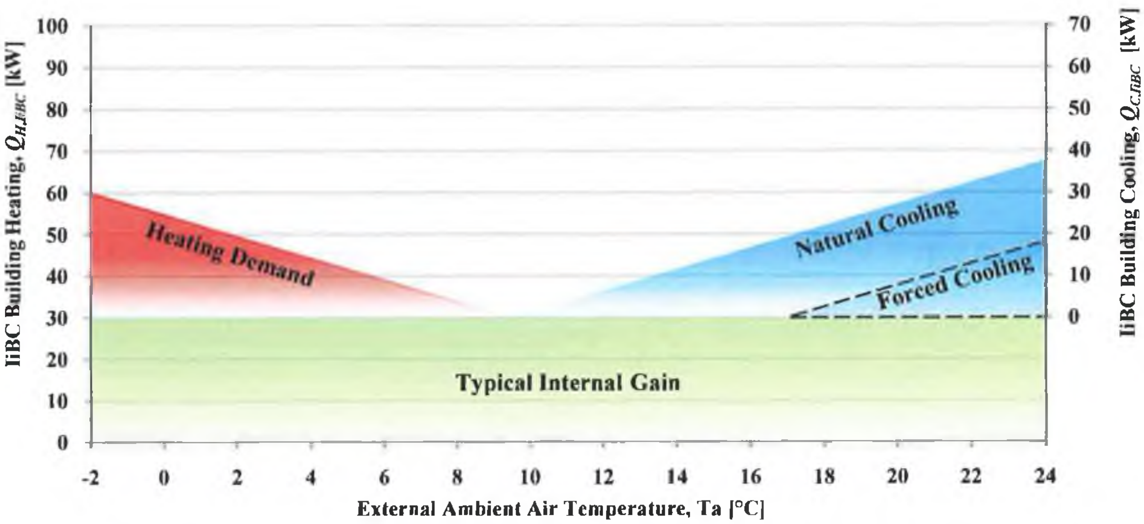


FIGURE 3.2 IIBC BUILDING DESIGN HEAT LOSS AND GAINS ANALYSIS VERSUS EXTERNAL AMBIENT AIR TEMPERATURE.

The heating system consists of two 60 kW_{th} liquified petroleum gas (LPG) condensing boilers acting as the primary source, with a maximum output temperature of +80°C and two secondary 15 kW_{th} GSHPs delivering a maximum output temperature of +50°C. Heat is distributed through hydronic radiators. This heat distribution system was chosen at a stage when the gas condensing boilers were the sole supply (monovalent), but it was also considered suitable, although not ideal for use with GSHPs, which were added after the primary gas heating system was designed. The GSHP heating system can therefore be described as a retrofit. The IiBC's building's heating system schematic is shown in Figure 3.3.

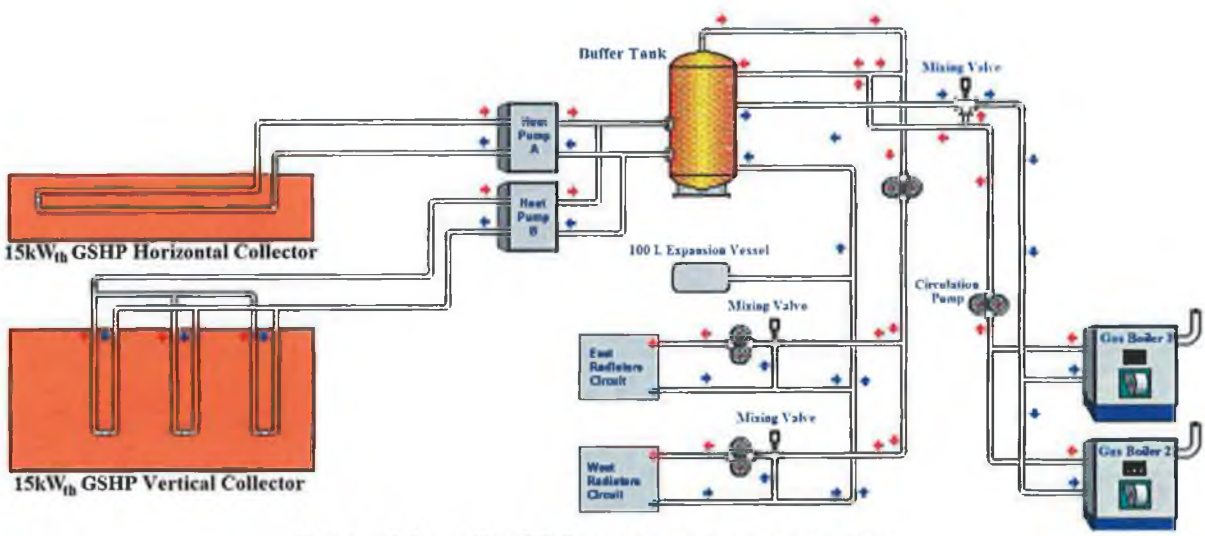


FIGURE 3.3 THE IIBC HEATING SYSTEM SCHEMATIC.

The radiators are designed for a typical flow temperature of +70°C. As the heat pumps were a late inclusion the output capacity of the radiators diminishes the usefulness of the heat pumps as they can deliver a combined maximum of 30 kW_{th} of energy at +50°C, yet the radiators are limited to a dissipation rate of 15 kW_{th} at the low external ambient temperature of -2°C. Due to this constraint, the heat pumps can be limited to operating between external air temperatures of +3°C and +16°C, although this can fluctuate with wind speed, rainfall level, solar intensity and/or internal gains. Indeed, as internal gains can consistently exceed 40kW during business hours due to high computer usage, the heat pumps offer sufficient heat to cater for the building's thermal needs to an external ambient air temperature as low as -2°C.

3.1.3 IIBC SPACE HEATING DEMAND

Figure 3.4 illustrates the monthly heating requirement of the IiBC building and the actual contribution of each of the three heating systems to meet demand.

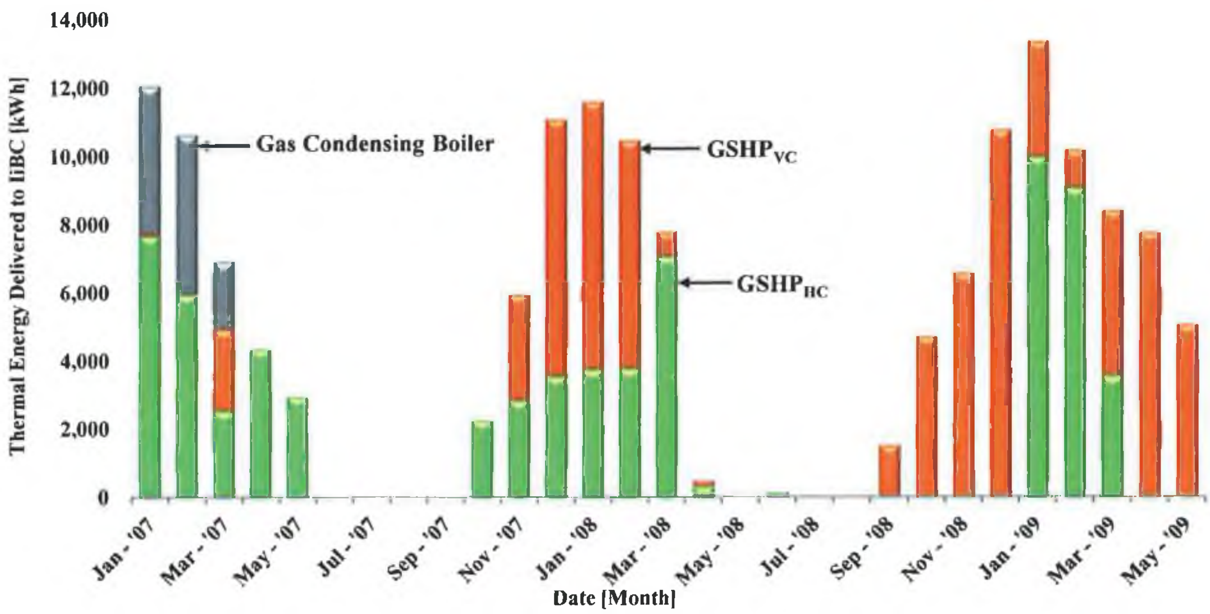


FIGURE 3.4 RESPECTIVE CONTRIBUTIONS OF THE $GSHP_{HC}$, $GSHP_{VC}$ & GAS HEATING SYSTEMS TO THE IIBC HEAT DEMAND PER MONTH BETWEEN JANUARY, 2007 AND JUNE, 2009.

Table 3.1 illustrates the respective contribution of the heat pumps to the IiBC heating over the three heating seasons, including the hours of operation and degree-days.

TABLE 3.1 HEAT PUMP CONTRIBUTION TO IIBC HEATING DEMAND OVER THREE HEATING SEASONS

	$GSHP_{HC}$ thermal supply in kWh (Hours of operation)	$GSHP_{VC}$ thermal supply in kWh (Hours of operation)
2006/2007	23,348 kWh (2,515 hours)	2,492 (109 hours)
2007/2008	23,623 kWh (2,766 hours)	26,010 (2,458 hours)
2008/2009	22,543 kWh (1,581 hours)	45,509 (5,582 hours)

Figure 3.4 and Table 3.1 shows that the Gas condensing boiler was only utilised in the initial winter heating season of 2006/2007, the $GSHP_{HC}$ was utilised with a steady year-on-year thermal supply of approximately 23,000kWh over the 3 heating seasons. The $GSHP_{VC}$ was the first season initially tested for only 109 hours, in the subsequent year it was increased to

match the GSHPHC supply and in the final heating season it was extensively utilised to characterise its performance under intensive duty.

Degree days are calculated based on the temperature difference between the hourly average external ambient air temperature (*Ta*) and the standard design heating base temperature of +15.5°C for a commercial building with large internal gains. For a domestic dwelling the design heating base temperature is generally +18°C (CIBSE, 2006). To generate a value for degree-days over the heating season each day's hourly average temperature differences are added together.

The IiBC's internal temperature is controlled between 8am and 9pm, Monday to Saturday by an automated Building Energy Management System (BEMS). The BEMS controls the gas boilers and the heat pumps. The gas boilers and the heat pumps cannot work in tandem as the optimum operating temperatures of the gas condensing boilers is a flow temperature of +70°C and a return temperature of +55°C which is above the +50°C maximum operational temperature of the heat pumps. The facilities manager must then select either the gas boilers or the heat pumps for heating duty and this decision is based on the external ambient air temperature, with gas activated if a prolonged cold period of below +0°C is predicted.

The BEMS system also provides frost protection when required, where the heating system is activated, regardless of time, when the external ambient air temperature falls below 0°C.

Drawing on measured data over the three heating seasons ('06/'07, '07/'08, '08/'09) Figure 3.5 illustrates the increased heat demand with decreasing external ambient air temperature.

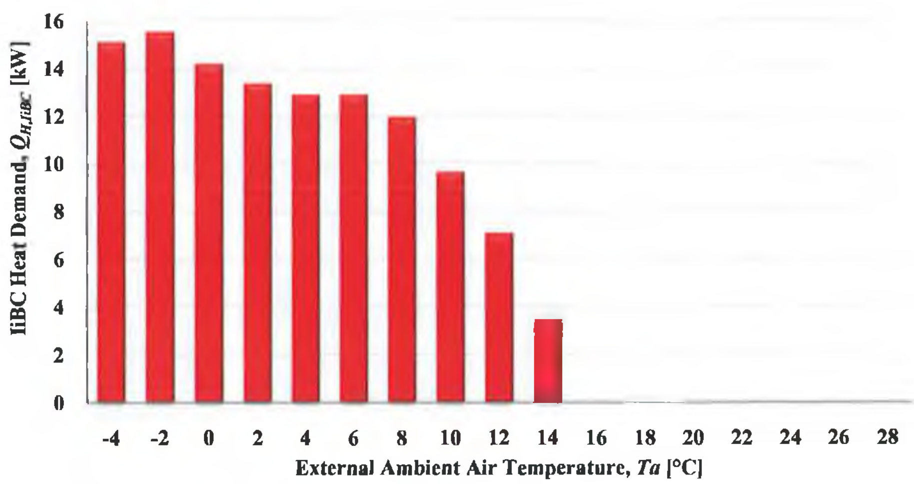


FIGURE 3.5 TOTAL HEAT DEMAND OF THE IIBC FROM THE HEAT PUMPS AS A FUNCTION OF EXTERNAL AMBIENT AIR REQUIREMENTS.

Results presented in Figure 3.5 deviates greatly from the design heat load, reflecting the unexpected heat gain from the electrical equipment, and limitations on the part of the radiators to dissipate the lower temperature supply. It also explains why the heat pumps had the capacity to fully satisfy the IiBC's heating demand during the '07/'08 and '08/'09 heating seasons shown in Figure 3.4.

Figure 3.6 shows the percentage run time per day for both GSHPs as a function of external ambient air temperature. This shows that the heat pump secondary heating system was used over almost their full range from 5% to 75% of total output, with individual heat pumps contributing 100% output for extended periods.

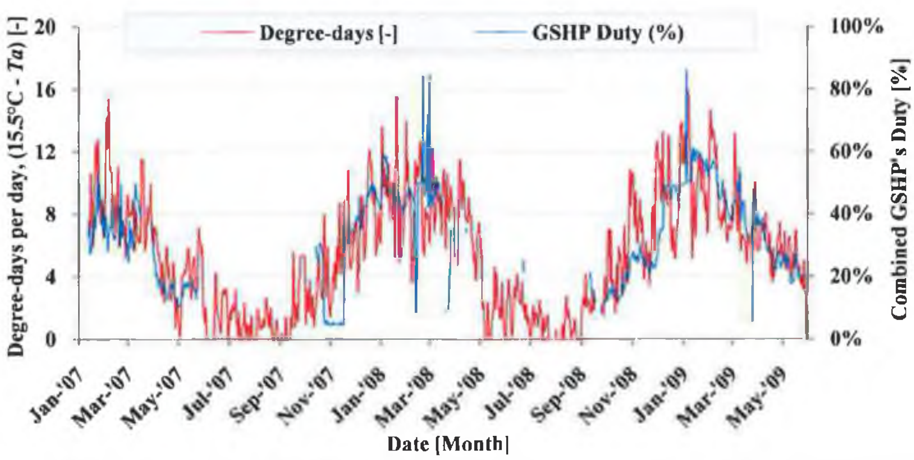


FIGURE 3.6 IIBC DAILY AVERAGED HEAT DEMAND VERSUS HEAT PUMP DUTY BETWEEN 2007 AND 2009.

Figure 3.7 illustrates the consistently even monthly averaged IiBC heating demand over the three years of recorded data.

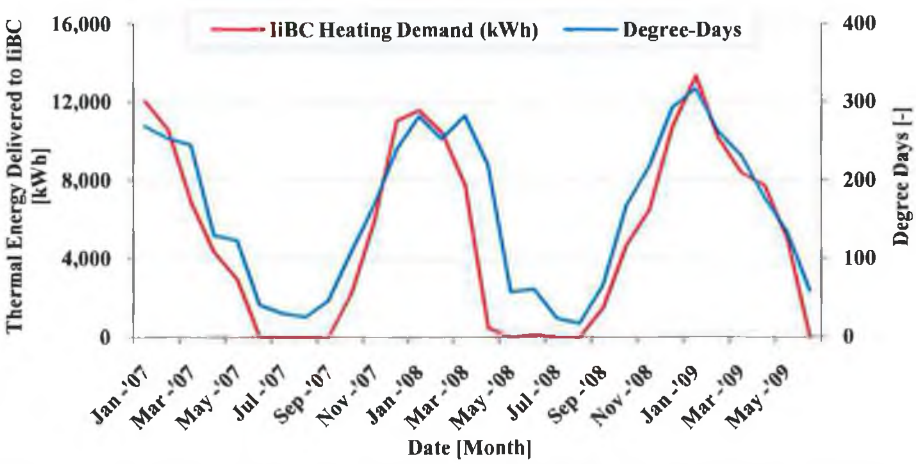


FIGURE 3.7 IIBC MONTHLY HEAT DEMAND VERSUS DEGREE-DAY DEMAND OVER 3 YEAR PERIOD.

From the data shown in Table 3.2 the IiBC's average annual thermal consumption is 55 kWh/m²/annum.

TABLE 3.2 IIBC HEATING SYSTEM THERMAL DEMAND

Winter Period of Operation	Degree-Days (based on +15.5°C)	IiBC Thermal Demand (kWh/m²/annum)
2006/2007	1,774	57
2007/2008	1,781	46
2008/2009	1,975	63

As a distribution of recorded daily average ambient air temperatures throughout 2007, it is notable in Figure 3.8 that the majority (78%) of the temperature distribution is within +5°C to +15°C.

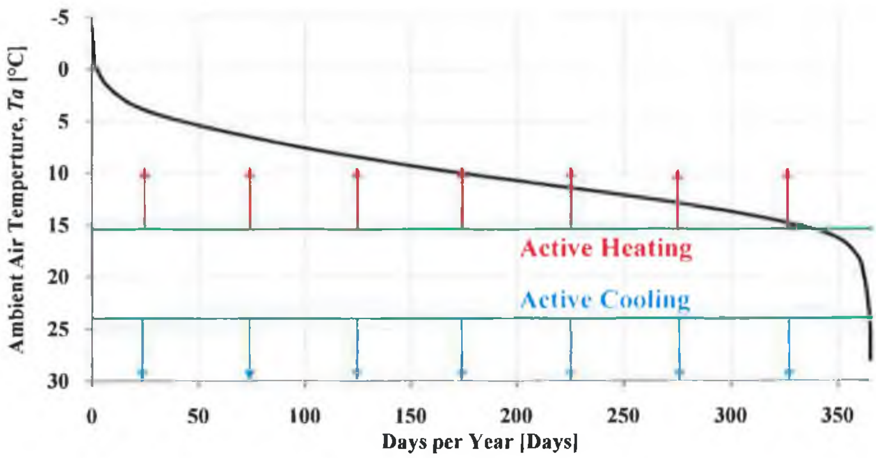


FIGURE 3.8 MEASURED DAILY AVERAGE EXTERNAL AIR TEMPERATURE DISTRIBUTION OVER 2007.

Figure 3.8 shows that the daily averaged ambient air temperature (*Ta*) was below +5°C for 45 days (12% of time), between +5°C and +15°C for 285 days (78% of time) and above +15°C for 35 days (10% of time). Using the standard +15.5°C as the base heating demand temperature, the heating season represents 90% of the year 2007, with active cooling required for 2 days or 0.5% of the year. This justifies the focus placed on the heating mode in this *HP-IRL* study and the potential for air source heat pumps.

3.2 WEATHER STATION

Since the weather drives both energy demand (duty) and supply and since it plays a key role in understanding the role of the Maritime climate in heat pump performance and delivering improved source side management it was considered necessary to develop the automated weather station shown in Figure 3.9. The weather station is within 300m of all three heat pumps and mounted on the roof of the Innovation in Business Centre (IiBC) building. The nine aspects of the local weather shown in Table 3.3 were continuously monitored to the accuracies shown from 2006 onwards. Each climate variable was continuously monitored and recorded every 5 minutes.

FIGURE 3.9 IIBC'S ROOF MOUNTED, AUTOMATED WEATHER STATION INSTALLED IN 2006.

Reflecting the important role of the climate in understanding heat pump performance and source side management the weather station is a key component of this *HP-IRL* study.

TABLE 3.3 *HP-IRL* WEATHER STATION SENSORS AND MEASUREMENT ACCURACY (2006 – 2010)

No.	Climate Variable	Measured Accuracy
1	Rainfall Levels	± 2% up to 25mm/hr
2	Rainfall Temperature	± 0.35°C @ 0°C
3	Wind Speed	± 0.3 m/s
4	Wind Direction	± 0.3°
5	External Air Temperature (Dry Bulb)	± 0.35°C @ 0°C
6	Air Pressure	± 0.5mbar (50Pa) @ 20°C
7	Relative Humidity	± 2.5% @ 10-100% RH
8	Net Solar Radiation	± 10% of daily totals
9	Net Infrared Radiation	± 10% of daily totals

By way of introduction to each climate sensor, key variables that impact on the status of the horizontal collector are presented in Figure 3.10.

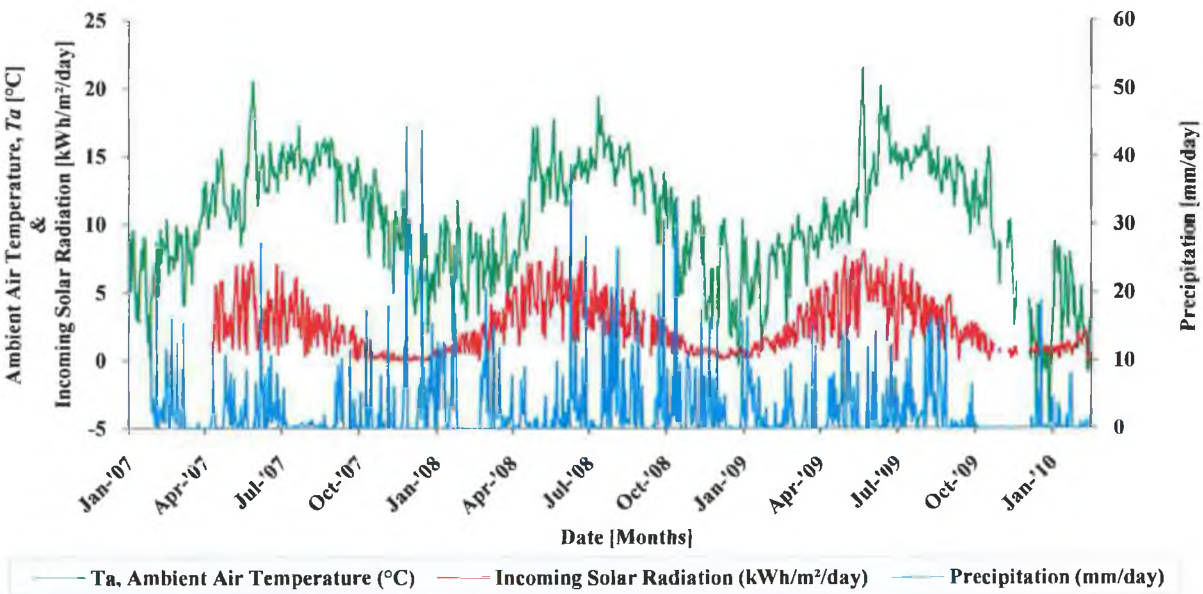


FIGURE 3.10 WEATHER VARIABLES THAT IMPACT ON GSHP HORIZONTAL COLLECTOR OPERATION RECORDED BY THE *HP-IRL* WEATHER STATION (2007-2010).

Ambient air temperature and incident solar radiation show typical annual cycles, whereas rainfall is distributed relatively evenly through-out the year. Above all climate variables the ambient air temperature has the greatest effect on a building's heating requirements. A *Vaisala* (HMP 45C) temperature and relative humidity probe, complete with a radiation shield, was used to measure these two variables. Historically recorded climate data for Galway reveals that the 30-year average external ambient air temperature was +10.2°C (1961 to 1990). Meanwhile, the *HP-IRL* weather station recorded an average ambient air temperature of +10.6°C between 2007 and 2009.

Rainfall levels are recorded by means of a *RM Young* rain gauge (Tipping Bucket Rain Gauge/Heated – Model No. 52202) shown in Figure 3.9. The rain gauge has a rated accuracy of ±2% up to 25mm/hr and ±3% up to a precipitation rate of 50mm/hr. The rainfall levels recorded during 2007 and 2008 were 773mm and 1296mm respectively. Measurements showed that there was a rainfall event for 14% of the hours during 2007 and 19% of all hours during 2008. The rainfall temperature was recorded using a custom designed apparatus, the results of which indicated rainfall temperature corresponds to that of the ambient air wet bulb temperature.

A set of two *Kipp & Zonan* pyranometer (CMP3) and two pyrgeometer (CGR3) sensors recorded both incoming and outgoing long and short wave radiation with factory calibrated accuracies of ±10% of daily totals, with calibration result presented in Appendix E. Solar radiation, particularly direct incoming solar radiation, plays an important role in the thermal recharge of the near surface ground layer, since up to 50% of this incoming energy can be absorbed by the ground (Peuser *et al.*, 2002). Figure 3.11 shows the weekly-averaged solar radiation during 2008 and two photos showing the sun's elevation above the IiBC at solar noon on both the summer and winter equinoxes.

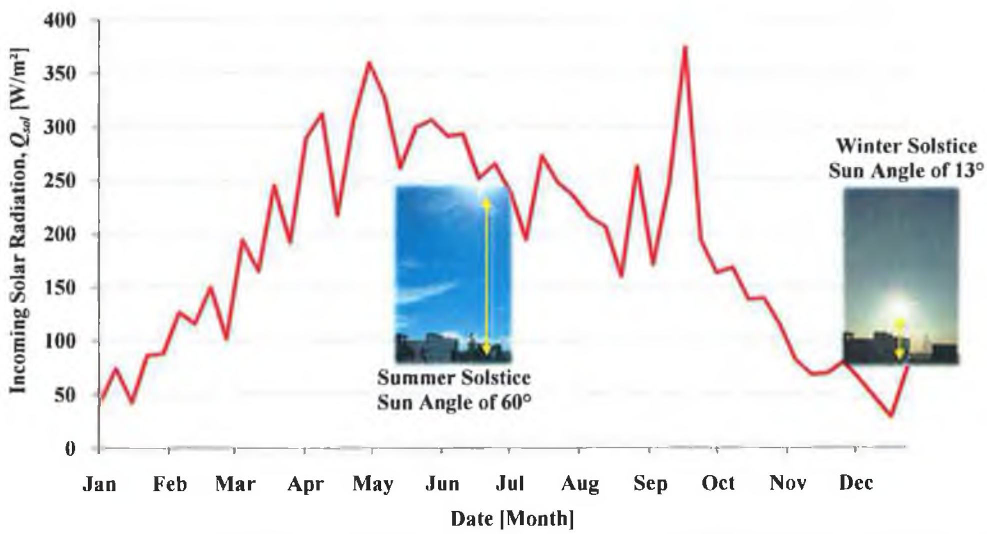


FIGURE 3.11 INCOMING WEEKLY AVERAGED GLOBAL SOLAR RADIATION RECORDED BY *HP-IRL* WEATHER STATION DURING 2008.

Solar radiation plays a key role in the seasonal recovery of the near surface ground temperature. However, while direct solar radiation plays an important role in the grounds thermal balance during summer its impact diminishes in winter. However, this can vary between ground cover types that absorb different amounts of incoming radiation and this aspect of the ground's thermal energy rejuvenation is examined in Chapter 4.

The range of ASHP relevant weather variables are shown in Figure 3.12.

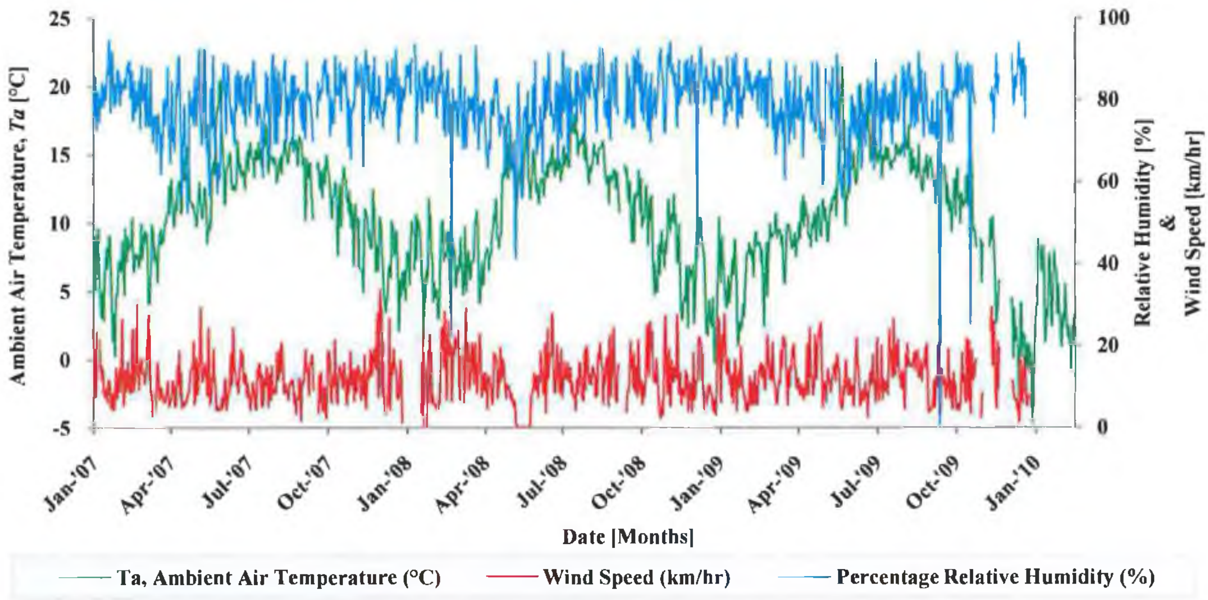


FIGURE 3.12 WEATHER VARIABLES THAT IMPACT ON ASHP OPERATION RECORDED BY THE *HP-IRL* WEATHER STATION (2007-2010).

Ireland has a typically high, year-round relative humidity of 70% and above, which is a common feature of Maritime climates.

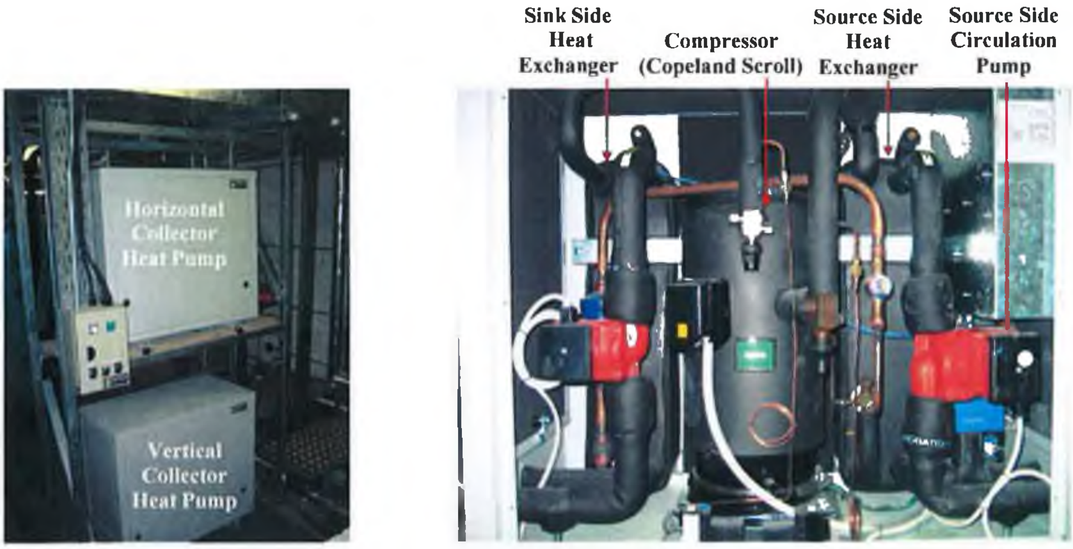
Relative humidity plays a significant role in the operation of ASHP systems. High relative humidity increases the heat capacity of the air, thereby increasing heat exchanger efficiency, but airborne moisture can also condense and freeze on the evaporator below +7°C, reducing heat transfer capacity or triggering a defrost cycle.

Wind speed and direction are recorded using an *RM Young* anemometer. The prevailing wind direction is south-westerly.

It is considered that the inclusion of a weather station provides a novel dimension to the heat pump literature, since it not only helps explain variations in heat pump performance, it also highlights inter-dependencies between collector type and climate. Combining these significantly increase our understanding of how to approach and implement source side management.

3.3 HORIZONTAL COLLECTOR GROUND SOURCE HEAT PUMP

Two identical, Irish made ground source heat pumps each with a capacity of $15kW_{th}$ were utilised in this study. They were manufactured by *Geostar Engineering Ltd.* as part of the *Solterra*™ heat pump range (*Solterra 500*) and are shown in Figure 3.13.



(a) Heat Pumps and controls **(b)** Heat pump components

FIGURE 3.13 SOLTERRA 500 HEAT PUMP INSTALLATION COMPONENTS.

The *Solterra 500* heat pump utilises R-407C primary refrigerant. R-407C is a mass blend of HFC-32 (23%), HFC-125 (25%) and HFC-134a (52%). R-407C displays equivalent capacities to the phased out HCFC refrigerants but heat transfer efficiencies are 5% lower on average, since they tend to fragment or change composition during evaporation and condensation in vapour compression refrigeration applications developing a 5°C temperature glide across the heat exchangers (ORNL, 1997). This departure from isothermal phase change behaviour makes the refrigerant less commercially attractive, but still viable. This refrigerant has low toxicity and no flame spread and is classified so by ASHRAE. Both heat pumps are identical and use a *Copeland* scroll compressor (ZB 42KCE-PFJ-551).

The *Solterra 500* heat pump was evaluated by Arsenal Research's independent test centre in Austria to obtain the heating power and COP at the rated conditions defined in the EN-14511 standard. The results of which are shown in Table 3.4 and these provide a useful benchmark for the COPs measured in this *HP-IRL* study.

TABLE 3.4 SOLTERRA 500 TEST RESULTS AS PER EN 14511 STANDARD

Condition	Average Heating Capacity [kW]	Average Power Input [kW]	Coefficient Of Performance [COP]	Uncertainty of Heating Capacity [±kW]
B5/W35	17.365	3.871	4.5	0.147
B0/W35	15.232	3.844	4.0	0.139
B-5/W35	13.317	3.808	3.5	0.133
B5/W50	16.584	5.317	3.1	0.144
B0/W50	14.576	5.273	2.8	0.136
B-5/W50	12.778	5.240	2.4	0.130

As source side management is central to this *HP-IRL* study much attention is focused on monitoring the collector response to both the climate and heat pump operation. This was made possible using the range of instruments presented in Figure 3.14.

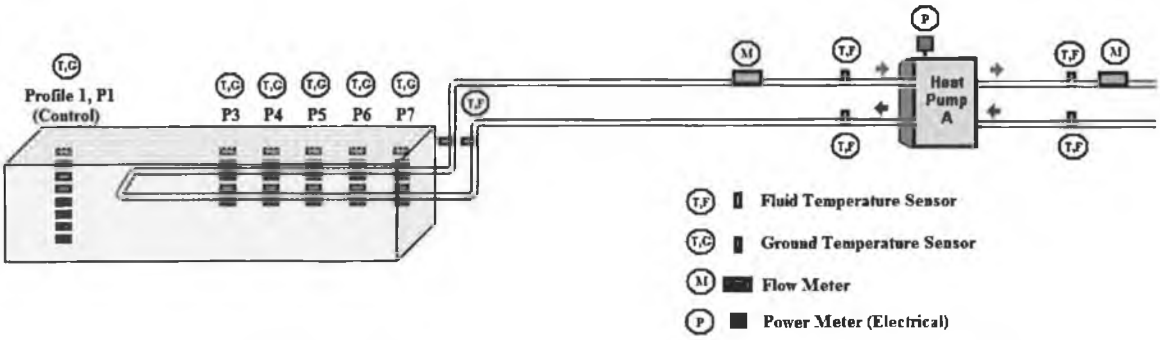


FIGURE 3.14 PIPING AND INSTRUMENTATION SCHEMATIC FOR *HP-IRL'S* HORIZONTAL COLLECTOR GROUND SOURCE HEAT PUMP.

Both heat pumps have their source and sink side flow rates recorded using four *Burkett* paddle wheel flowmeters (8030 HT) with a calibrated accuracy of ±2 % at 1 m/s. The total electrical power consumed is the sum of compressor power usage and the collector's circulating-pump power demand. These values are monitored using *Vydas* power meters (UPC) accurate to ±0.5%.

Both heat pumps have their fluid flow and return temperatures recorded by eight *Omega* PT100 CLASS B 1/10 DIN elements with an accuracy of ±0.03°C at 0°C and validated on-site by both a *Jofra* (D55SE) temperature calibrator and an ice slurry calibration, details of which are detailed in Appendix F. The temperature sensors are immersed centrally in the collector fluid flow and return pipes, allowing the probe tip to be situated in the centre of the pipe fluid flow. In order to prevent external temperature interference the exposed stem of the sensor probes are thermally insulated as shown in Figure 3.15.

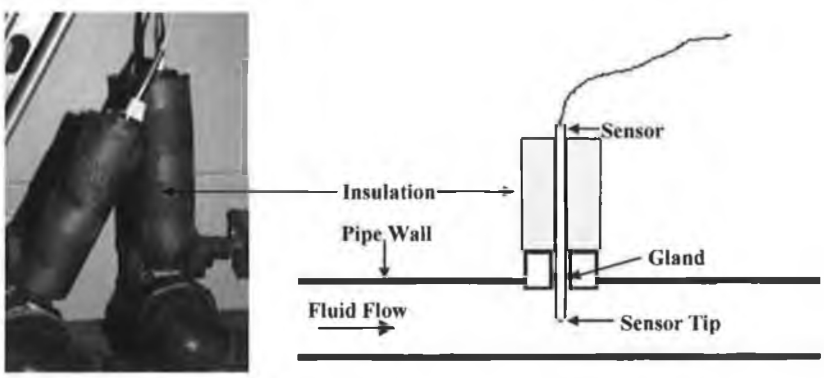


FIGURE 3.15 TEMPERATURE SENSOR *IN SITU.*

These sensors combine to enable the COP of both heat pumps to be evaluated to an accuracy of ±3.3%, details of which are presented in Appendix F.

3.3.1 Horizontal collector dimensions

Figure 3.16 presents a plan view of the 430m^2 collector, relative to the IiBC building. The collector consists of ten high density polyethylene pipes (PE100 SDR-11 HDPE) each of 150m length, with an outer diameter (O.D.) of 32mm and a 3 mm wall thickness. These ten collector loops are connected to a manifold that is supplied by a HDPE flow and return pipe to the heat pump that is 60m long, with an outer diameter of 63mm and a 5.8mm wall thickness. Details of the HDPE pipes are presented in Appendix G. Despite having low thermal conductivity the HPDE piping is typically selected for its tough, versatile and flexible qualities, and is generally the type recommended (IGSHPA *et al.*, 1997).

For long term extraction, it is recommended that the collector's specific annual extraction rate should not exceed 50–70kWh/m^2 (VDI 4640, 2001). Therefore, if the building/application has an annual space heating demand of 100 kWh/m^2 the collector region should be between 1.5 to 2 times the heated space area (Reuss and Sanner, 2001). Following this metric and based on the thermal energy demand of ~30kW, the IiBC building required a horizontal collector area of 1,800 – 2,400m^2, but just 430m^2 was available. Hence the need arose for the additional vertical collector.

The collector pipes are situated at an average depth of 1m below the ground surface, encased at the centre of a ~200mm layer of sand. This layer of sand serves to both protect the collector during installation and to allow ground borne moisture to pass over the coil and avoid stagnation. Characteristically, about 50% of the extracted heat is drawn from within five pipe diameters of the collector pipe, which emphasises the importance of the thermo-physical characteristics of the material surrounding the pipe (Claesson and Dunand, 1983). The importance of this is emphasised by the numerous examples of increased heat transfer in vertical boreholes that utilise thermally enhanced grouts (Spiker, 1998; Khan and Spitler, 2004; Florides and Kalogirou, 2007; Phetteplace, 2007; Florides and Kalogirou, 2008; Esen and Inalli, 2009; Sharqawya *et al.*, 2009). However, similar design practice is not employed for horizontal collectors since most studies show negligible contact resistance (O'Connell and Cassidy, 2004).

As indicated in Figure 3.16 the collector region is divided into these two regions, A and B. The collector consists of ten 150m long circuits with Region A containing eight circuits in the in-line configuration and Region B containing two circuits in the spiral (or Slinky^{TM}) configuration. Thus the spiral configuration represents 20% of the overall collector length, 18% of the collector area. Approximately 352m^2 is devoted to Region A, 78m^2 to Region B.

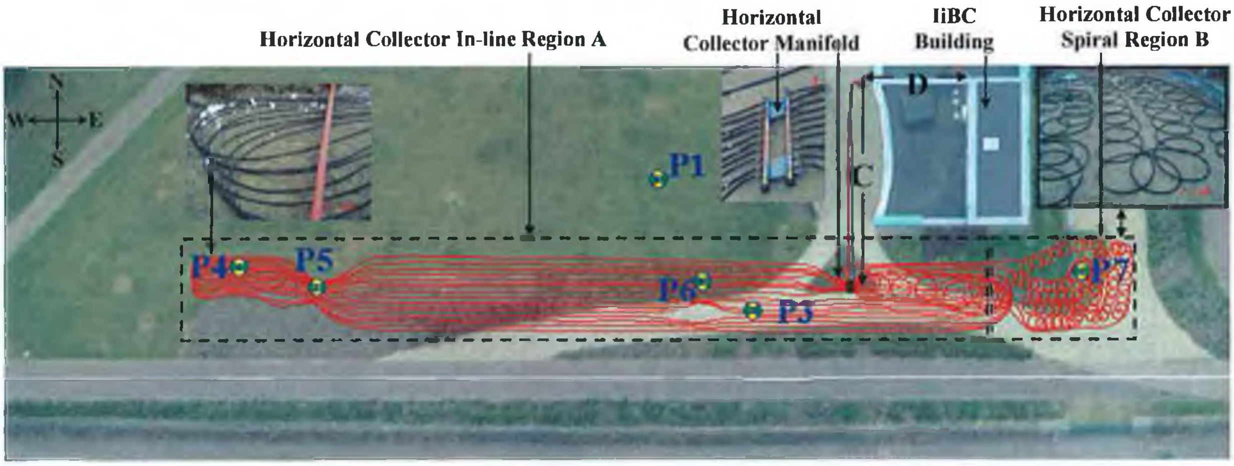


FIGURE 3.16 AERIAL PHOTOGRAPH OF HORIZONTAL COLLECTOR REGION, UPON WHICH THE ACTUAL COLLECTOR CONFIGURATION HAS BEEN SUPERIMPOSED FROM PHOTOS TAKEN DURING INSTALLATION.

While the literature highlights the potential for uneven flow in parallel piping configurations, this was not addressed in this *HP-IRL* study.

As the horizontal collector is located at an average depth of 1.0m, in the saturated clay region outlined in Figure 2.7, the collection of thermal energy at this depth is affected by the variations in weekly weather patterns and these interdependencies are discussed further in Chapter 4.

Table 3.5 outlines the key characteristics of the horizontal collector installed and these will be used in 4.3 to evaluate the heat extraction capacity of the collector.

TABLE 3.5 SUMMARY OF THE *HP-IRL* HORIZONTAL COLLECTOR CHARACTERISTICS

Horizontal Collector Characteristics	Value
Maximum horizontal collector thermal demand (at a sink temperature of +50°C)	11kW
Horizontal collector pipe material	HDPE
Horizontal collector pipe material conductivity	0.46 W/(m·K)
Specific heat capacity of brine	3.73 kJ/(kg·K)
Thermal conductivity of brine	0.48 W/(m·K)
Collector flowrate (per loop)	0.41 m^3/h (0.22 m/s)
Overall horizontal collector brine volume	0.921m^3 (921 litres)
Horizontal collector volume of ethylene glycol	276 litres (30% by volume)
Horizontal collector volume of water	645 litres (70% by volume)
Length of flow & return pipe to manifold, outside IiBC, "C" in Figure 3.16 (63mm O.D.)	36m
Length of flow & return pipe to manifold, inside IiBC, "D" in Figure 3.16 (63mm O.D.)	24m
Overall length of flow & return pipe to collector manifold (63mm O.D.)	60m
No. of horizontal collector loops from manifold (32mm O.D.)	10
Individual horizontal collector loop length	150m
Overall horizontal collector length	1500m
Total horizontal collector area	430m^2
In-line horizontal collector area (Region "A")	352m^2
Spiral horizontal collector area (Region "B")	78m^2
Average horizontal collector depth	1.0m
Maximum horizontal collector extraction per unit collector area	25.5 W/m^2
Maximum horizontal collector extraction per unit collector length	7.35 W/m
Ground conditions	Saturated Clay/gravel
Ground thermal conductivity (λ_G)	2.3 – 2.5 W/(m·K)
Ground thermal diffusivity (α)	1.0 – 1.1 (E-06 m^2/s)

It must be noted that both the horizontal and vertical collector brine solution used in this project has a freezing point temperature of -15°C (30% ethylene glycol, by volume). As the glycol has a lower specific heat capacity than water it is preferable from a heat transfer perspective to keep the proportion of glycol as low as possible to maximise efficiency.

As indicted in Chapter 2, the range of thermal extraction rates range between 8 and 40W/m² and the *HP-IRL* horizontal collector is the upper end of the this range with a maximum thermal extraction of 25.5W/m².

3.3.2 HORIZONTAL COLLECTOR GROUND COVER

As a result of Ireland's lack of geothermal gradients the ground temperature varies in response to fluctuating radiant, thermal and latent heat exchange processes that initially take place at the climate-ground surface interface. This indicates that the ground's surface cover may play an important role in absorbing incident solar radiation, regulating moisture infiltration and responding to ambient air temperature fluctuations. Indeed there is evidence to suggest that the selection of ground cover type can influence the performance of a horizontal collector (Oliver *et al.*, 1987; Mihalakakou *et al.*, 1996; Popiel *et al.*, 2001) and the *HP-IRL* study sought to access the impact of this design parameter.

To investigate the impact of different ground surface covers and to replicate ground covers employed in typical applications, the range of four different covers shown in Figure 3.17 were distributed as shown in Figure 3.18.

Above ground vegetation is separated into two types, *canopy* and *noncanopy*. Grass and bark are classified as noncanopy vegetation While the shrubbery, because of its large volume presence over the ground surface, is referred to as a canopy (Cline *et al.*, 1993). Noncanopy material such as bark is commonly referred to as *mulch.*

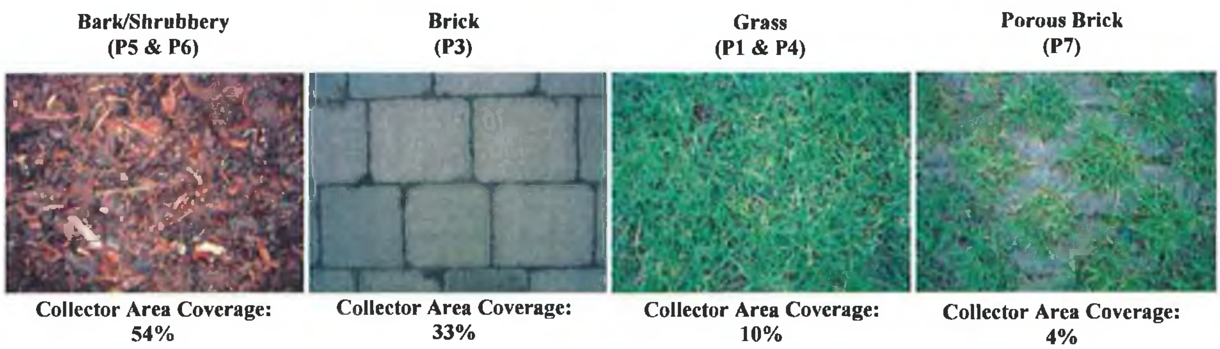


FIGURE 3.17 HORIZONTAL COLLECTOR GROUND SURFACE COVERS.

Locations identified as P1 through P7 in Figure 3.18 refer to vertical measurement profiles at which the variation in the ground temperature and moisture content was recorded to a depth of 1.8m.

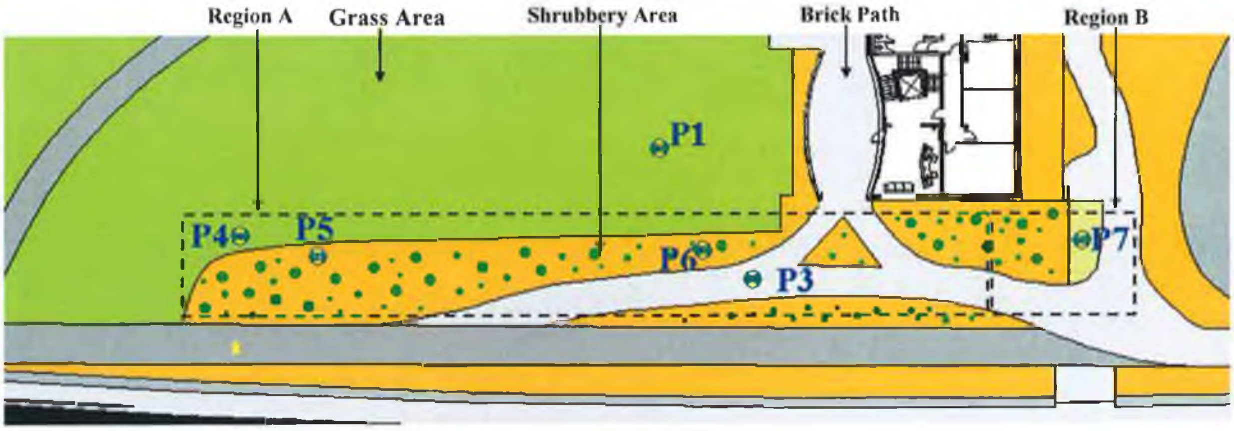


FIGURE 3.18 GROUND SURFACE COVER ABOVE THE HORIZONTAL COLLECTOR.

The thermal and permeability characteristics of the four ground cover types are tabulated in Table 3.6.

TABLE 3.6 THERMOPHYSICAL PROPERTIES OF THE GROUND COVERS INSTALLED ABOVE THE *HP-IRL* HORIZONTAL COLLECTOR

Ground Cover	Coverage of Collector	Physical distance from surface (mm)	Albedo (a)	Emissivity (ε)	Thermal Conductivity (λ)	Permeability (κ, milidarcy)
Bark/Shrubbery	54%	400 above	0.1	0.97 - 0.98	0.12 - 0.04	10^{+8}
Brick	33%	70 below	0.12	0.75	0.8	0.001
Grass	10%	75 above	0.23	0.97 - 0.98	0.12 - 0.04	10
***Porous Brick**	4%	70 below	0.18	0.86	0.4 - 0.8	10^{+5}

* Surface area of porous brick section is 50% brick and 50% grass

3.3.3 SOIL/GROUND ANALYSIS

Soil is the dominant aggregate below the surface cover within the *HP-IRL* horizontal collector region and the following analysis was conducted to estimate the ground's physical characteristics, thermal conductivity and thermal diffusivity. Photographs of the ground composition at four profiles defined in Figure 3.18 and above the horizontal collector are shown in Figure 3.19.

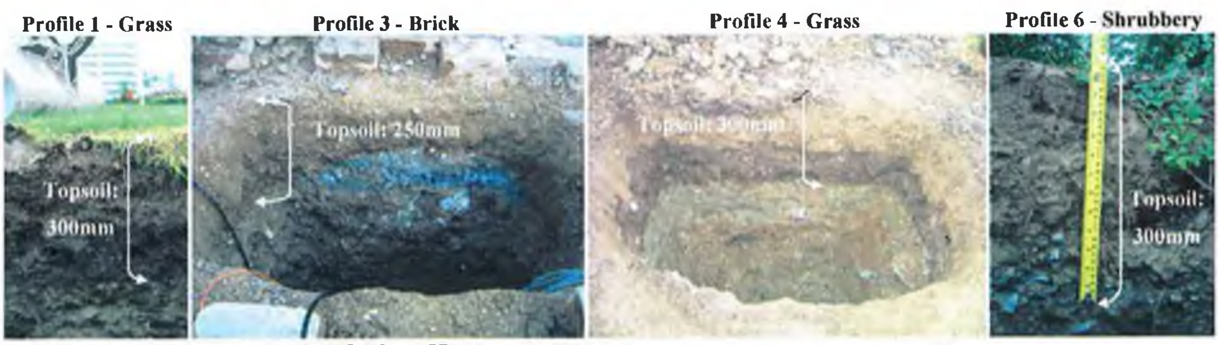


FIGURE 3.19 HORIZONTAL COLLECTOR GROUND COMPOSITION.

Soil thermal analysis is complicated by varying degrees of soil porosity and it is continuously changing and developing. However, this can be overcome by following the standard soil analysis test, breaking down of the soil's structure into it's constituents of minerals and organic matter of various shapes and sizes.

This is then followed by establishing the pore space in the soil and what occupies these pores. These constituents control the thermal characteristics of a soil. The thermal characteristics of each constituent can be examined individually and then brought together to establish the overall thermal conductivity (λ_G). The pores however play a major role in the overall thermal conductivity as it can be filled with air (low thermal conductivity) or water (high thermal conductivity). The size and shape of the particles within the soil determines the volume of pores in the soil and most soil classifications are based on the size of the soil particles. Soils can be classified into categories of gravel, sand, silt or clay.

The properties of soil can differ greatly through the succession of unique layers that constitute the soil profile (Marchall *et al.*, 1996). Organic material tends to be concentrated in the upper regions of the soil, creating a distinct layer that is sometimes referred to in gardening as 'topsoil' and ranges in depth from 50mm to 300mm, under which is called the 'subsoil'. Figure 3.19 shows that the topsoil element ranges from 250 to 300mm for the *HP-IRL* horizontal collector. Horizontal collectors are typically positioned within the subsoil layer and are therefore surrounded by some combination of gravel, sand, silt and clay of varying size particles.

The thermal conductivity of the ground surrounding the *HP-IRL* $GSHP_{HC}$ was determined using a long term transient investigation of the ground's thermal conductivity, developed by Farouki (1986), where the attenuation and the lag of the annual temperature wave of the ground (Farouki, 1986) is recorded.

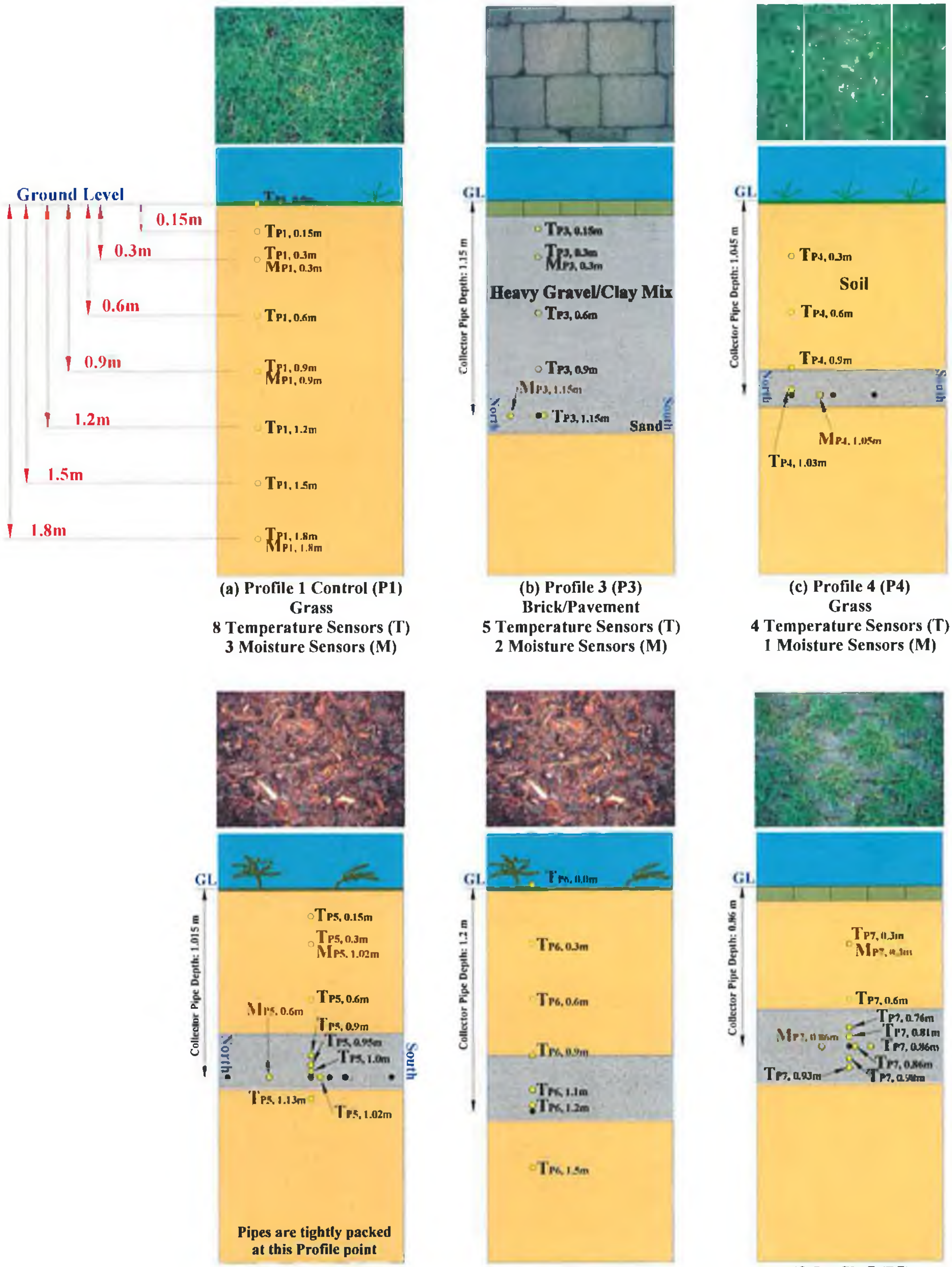
Applying Equation 2.1, a predicted wave is introduced and the ground diffusivity value is changed until the graphs coincide as shown in Figures 2.9 and 4.2. This is achieved where the bulk density (oven dry: 1337 kg/m^3), ground moisture content and specific heat capacity are known. The ground thermal diffusivity is evaluated using Equation 3.1.

$$\alpha = \frac{\lambda_G}{c_G} \qquad \textit{Equation 3.1}$$

From this evaluation the ground thermal conductivity was found to be between 2.3 and 2.5 W/m^2·K.

3.3.4 GROUND TEMPERATURE AND MOISTURE MEASUREMENT PROFILES

The detail of each horizontal collector profile in terms of surface cover, sensor position and soil types are presented in Figure 3.20.



(a) Profile 1 Control (P1)
Grass
8 Temperature Sensors (T)
3 Moisture Sensors (M)

(b) Profile 3 (P3)
Brick/Pavement
5 Temperature Sensors (T)
2 Moisture Sensors (M)

(c) Profile 4 (P4)
Grass
4 Temperature Sensors (T)
1 Moisture Sensors (M)

(d) Profile 5 (P5)
Shrubbery/Wood Chip
8 Temperature Sensors (T)
2 Moisture Sensors (M)

(e) Profile 6 (P6)
Shrubbery/Wood Chip
7 Temperature Sensors (T)

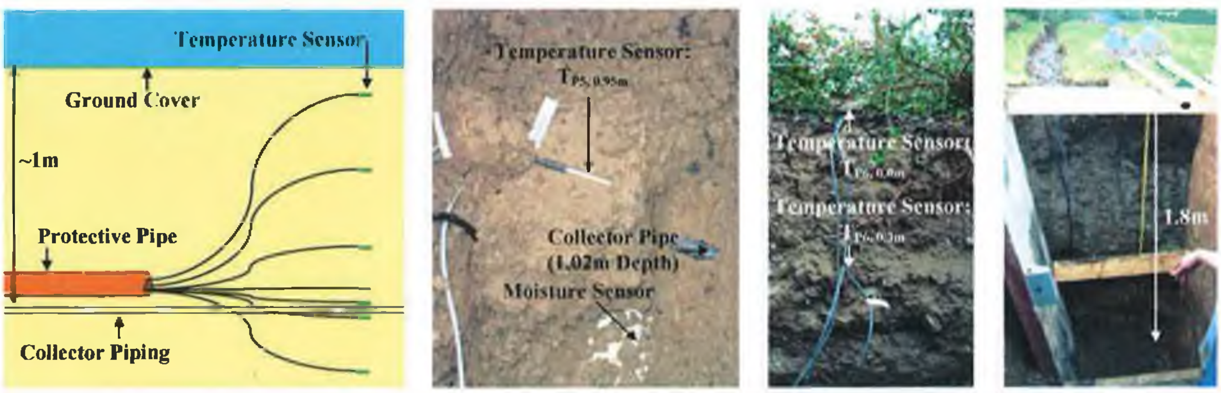
(f) Profile 7 (P7)
Porous Pavement
8 Temperature Sensors (T)
2 Moisture Sensors (M)

FIGURE 3.20 HORIZONTAL COLLECTOR GROUND PROFILES (LOHAN *ET AL.*, 2006).

Influenced by the aim of establishing the link between the climate and horizontal collector performance and recognising that the collector was laid beneath four different ground covers, led to the inclusion of vertical measurement profiles shown in Figure 3.20 that allowed both ground temperature and moisture content to be continuously monitored above and below the collector at the six locations identified in Figures 3.16 and 3.18. Note that each profile is identified using the symbol "P#" and the relative locations of these profiles relative to each other and the collector is presented in Figures 3.16 and 3.18.

One additional profile, P1, was added as a reference profile outside the collector's Thermally Affected Zone (TAZ). This represents a novel and comprehensive approach that enables many aspects of horizontal collector source side management to be rigorously examined.

The ground temperature sensors used were *Sontay* 4-wire RTD PT100 Class A sensors with a factory calibrated accuracy of ±0.3°C which was validated on-site using both a *Jonta* temperature calibrator and a 0°C de-ionised ice-slurry. The temperature sensors were installed after the collector over a three month period from August to September 2006 and Figure 3.21 shows some temperature sensors during installation.



(a) Sensor Installation **(b) P5 Sensor Installation** **(c) P6 Sensor Installation** **(d) P1 Sensor Installation**

FIGURE 3.21 INSTALLATION OF HORIZONTAL COLLECTOR TEMPERATURE AND MOISTURE MEASUREMENT PROFILES.

As can be seen from P6 in Figure 3.21, circular pockets with the same diameter as the temperature sensors were drilled into the side wall and the sensors were placed in them. Precautions were taken during sensor installation to minimise disturbance to surrounding ground, accurately record each sensor's reference number, depth and location, avoid sensor movement or cable damage during backfilling, unusual factors introduced and that the Data Acquisition (DAQ) measurement accuracy was preserved.

The ground moisture content was measures using *Campbell Scientific* electrical frequency water content reflectometer (CS616-L) sensors. The sensors were calibrated on site using structureless massive soil samples generating an accuracy of ±2.5% volumetric moisture content.

3.3.5 HORIZONTAL COLLECTOR PUMPING POWER

In order to facilitate the transfer of thermal energy from the ground to the building, pumping power is required to circulate fluid through the collector circuit. This energy is generally referred to as collector pumping power or parasitic power which is quantified in this section.

A diagram of the head pressure loses for *HP-IRL's* horizontal collector is shown in Figure 3.22.

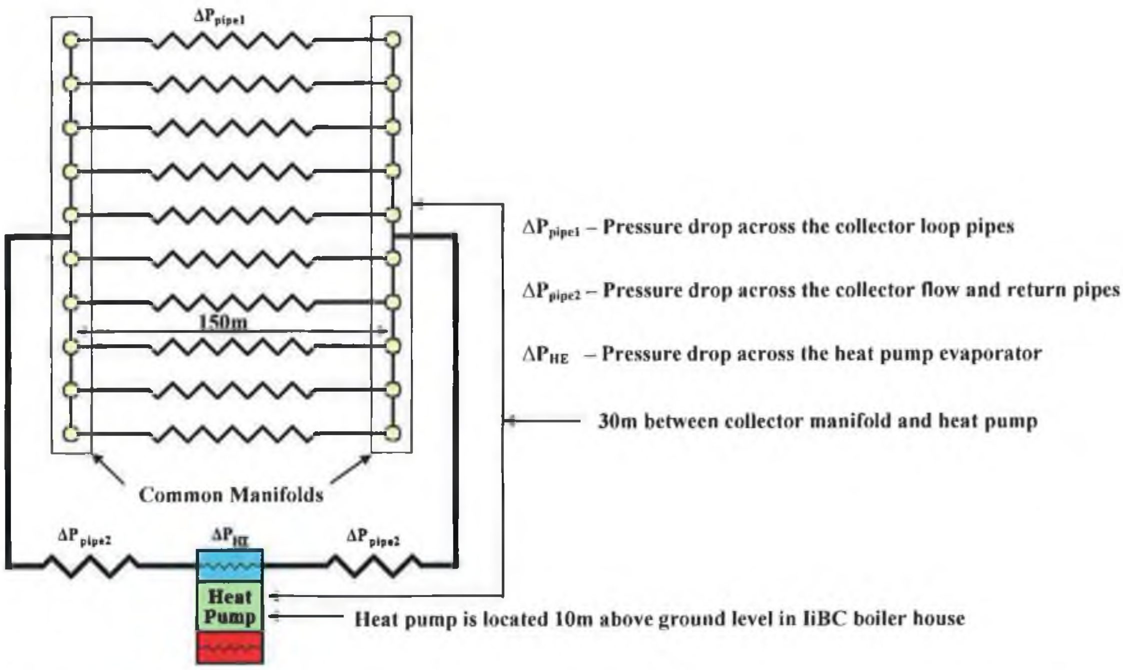


FIGURE 3.22 PLAN VIEW SCHEMATIC REPRESENTATION OF THE HEAT PUMP-TO-COLLECTOR PIPING CIRCUIT.

The circulation of collector fluid is essential to efficient heat extraction from the ground and must therefore be sized to minimise additional or parasitic power required. This can be achieved by reducing the distance between the collector and heat pump and/or avoiding turbulent flow, which can be achieved by reducing the flow speed and/or increasing the pipe diameter. A summary of the pumping power required for the $GSHP_{HC}$ is presented in Table 3.7 and calculation details are presented in Appendix H.

TABLE 3.7 SUMMARY OF PRESSURE LOSSES ACROSS THE $GSHP_{HC}$ CONFIGURATION SHOWN IN FIGURE 3.22

Pressure Parameter	Value
Heat Pump Evaporative Heat Exchanger, ΔP_{HE}	33 kPa
Flow & Return to Collector Manifold, ΔP_{pipe2}	3 kPa
Spiral Loops, ΔP_{pipe1}	25 kPa
In-Line Loops, ΔP_{pipe1}	173 kPa
Total, ΔP_{Total}	**234 kPa**

3.3.6 HORIZONTAL COLLECTOR EXPERIMENTAL TEST PROGRAM

The experimental test program was devised to establish the operational envelope of the heat pump and the conditions imposed on the collector by the variable climatic and associated thermal demands of the IiBC building. Since the weather and to a lesser extent the building demand varies constantly, reliable data could only be established by continuously monitoring all test parameters, testing over extended periods of time and repeating tests where possible. Testing was performed over the period from January 2007 to June 2009. Experimental testing was conducted to continuously monitor COP and other indicators of performance under the following conditions:

- heat pump duty, spanning low, moderate and intensive operation: low duty represents test periods where the heat pump was operational between 20% and 40% of the time, moderate duty between 40% and 70% and intensive duty between 70% and 100%
- test duration, spanning short, medium and long term operation: short-term operation represents distinct test period of no longer than 7 days duration, medium-term represents test period of between 7 and 30 days duration and long-term represents test periods that exceed 30 days continuous operation
- response of ground conditions to climate, including temperature, ground moisture content and thermal energy content

The range of experimental test conditions were imposed over the course of nine test periods identified as "HC#" in Table 3.8 and the results will be discussed in Chapters 4 and 5.

TABLE 3.8 HORIZONTAL COLLECTOR TEST PROGRAM CONDUCTED BETWEEN 2007 AND 2009

Test #	Demand	Duration	Description
HC1	Moderate	Long	First IiBC heating season observational period with moderate thermal extraction rates.
HC2	Low	Long	Prolonged steady state low level thermal extraction, indicative of autumn/spring time domestic dwelling utilisation.
HC3	Moderate	Medium	Fixed daily extract and recovery periods, indicative of domestic dwelling utilisation.
HC4	Low	Long	Comparative heat pump operation period with the $GSHP_{HC}$ and $GSHP_{VC}$ in simultaneous operation.
HC5	Intensive	Short	Steady-state thermal extraction and subsequent recovery period, indicative of extreme utilisation.
HC6	Intensive	Medium	Steady-state thermal extraction and subsequent recovery period, indicative of extreme utilisation.
HC7	Low	Short	Recording localised collector profile thermal extraction and recovery temperature gradients.
HC8	Moderate	Short	Recording localised collector profile thermal extraction and recovery temperature gradients.
HC9	Intensive	Long	Prolonged steady-state intensive thermal extraction, indicative of peak winter utilisation (commercial application).

The impact of heat pump operation was established by comparing the ground condition within the collector region with that of the reference/control Profile 1 as shown in Figure 3.16 and 3.21(a). Observations from the imposed test program relative to the reference profile (P1) also include:

- ground temperature drawdown with various rates of thermal extraction
- the extent of the TAZ surrounding the collector pipes due to thermal extraction
- ground thermal recovery rate
- surface cover effects on ground temperature

This experimental test program was successfully executed while also satisfying the heating demand of the IiBC building using a combination of the three heating systems shown in Figure 3.4.

3.4 Vertical collector ground source heat pump

The vertical collector consists of three 100m deep boreholes spaced 15m apart. The pipes are made of 6mm thick HDPE and have an outside diameter of 63mm. The boreholes are encased in limestone from a depth of 2m. A cavernous aquifer was discovered during drilling at a depth of 95m. A detailed map of the aquifer prone regions of Ireland is shown in Appendix I.

Figure 3.23 shows an aerial photograph of the three vertical boreholes numbered 1, 2, and 3 relative to both the IiBC building and the horizontal collector in Figure 3.16.

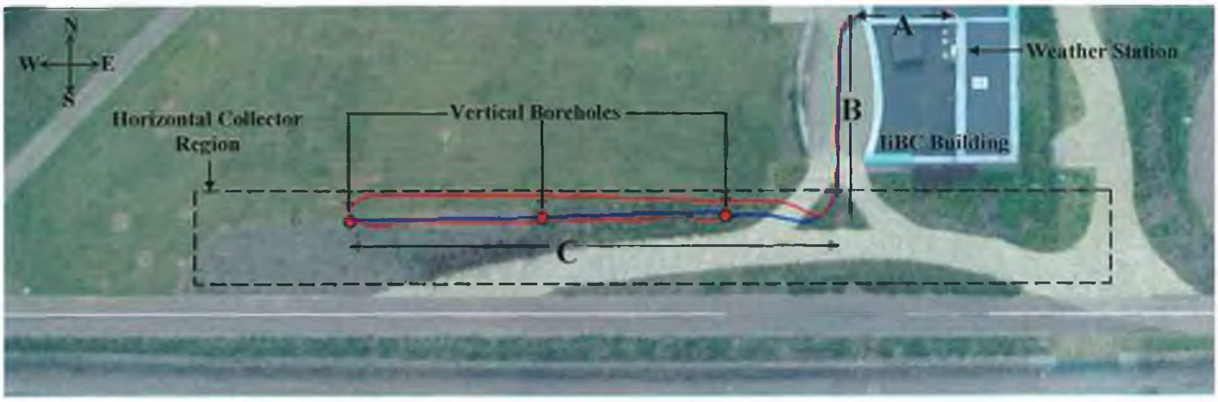


Figure 3.23 Aerial photograph showing locations of the three vertical collector boreholes relative to each other, the horizontal collector and the IiBC building.

A detailed map of the rock formation in Ireland is shown in Appendix D.

3.4.1 Vertical collector instrumentation and thermal characteristics

Due to the focus on source side management, a detailed monitoring of collector response to both the climate and heat pump operation was employed. This was made possible using the range of instruments presented in Figure 3.24.

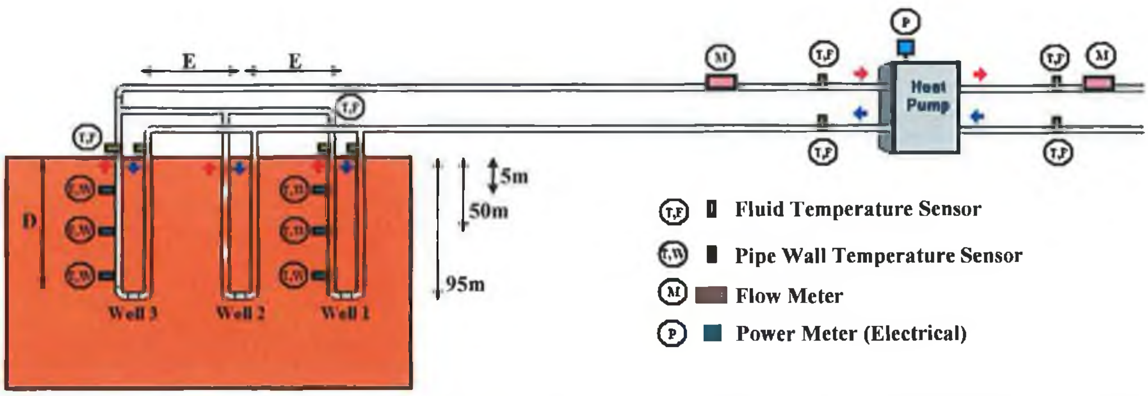


Figure 3.24 Piping and instrumentation schematic for the *HP-IRL's* vertical collector ground source heat pump.

These sensors combine to enable the heat pump COP to be evaluated to an accuracy of ±3.3% and details are presented in Appendix F.

As can be seen from Figure 3.24 temperature sensors that measured the outer wall temperature were installed in Wells 1 and 3 at depths of 5m, 50m and 95m. These temperature sensors were installed to enable an evaluation of the ground temperature variation with depth during heat pump operation, but also to determine the recovery rates after the heat pump ceased operation. Thus the borehole wall temperature sensors formed part of the overall monitoring of the $GSHP_{VC}$ charging and discharging dynamics, over both short-term and long-term periods. These sensors were further employed to determine the undisturbed temperature, when the heat pump was off for over three months, enabling a geothermal gradient evaluation. This aspect represents additional information to complement the thermal response testing. Table 3.9 outlines the key characteristics of the vertical collector.

TABLE 3.9 SUMMARY OF THE *HP-IRL* VERTICAL COLLECTOR CHARACTERISTICS

Vertical Collector Characteristics	Value
Maximum vertical collector thermal demand (at a sink temperature of +50°C)	11kW
Vertical collector pipe material	HDPE
Vertical collector pipe material conductivity	0.46 W/(m·K)
Specific heat capacity of brine	3.73 kJ/(kg·K)
Thermal conductivity of brine	0.48 W/(m·K)
Vertical boreholes flowrate	1.4 m^3/h (0.1 m/s)
Overall vertical collector brine volume	0.851m^3 (851 litres)
Vertical collector volume of ethylene glycol	255 litres (30% by volume)
Vertical collector volume of water	596 litres (70% by volume)
Length of collector pipe within IiBC building ("A" in Figure 3.23)	24m (3%)
Length of collector pipe at 1m depth to the boreholes ("B" and "C" in Figure 3.23)	150m (20%)
Length of pipe between 1m and 15m depth	84m (11%)
Length of pipe between 15m and 100m depth	510m (66%)
Overall vertical collector pipe length (flow & return)	768m
No. of boreholes (63mm O.D.)	3
Borehole spacing ("E" in Figure 3.24)	15m
Borehole depth ("D" in Figure 3.24)	100m
Vertical collector extraction per unit borehole length (at maximum extract rate)	36.7 W/m
Ground conditions – 0 to 2m	Saturated Sandy/Gravel
Ground conditions – 2m to 100m	Limestone: 2 – 4 W/(m·K)
Overall ground thermal conductivity (λ_{eff})	2 – 4 W/(m·K)
Overall ground thermal diffusivity (α)	1.0 – 1.7 (E-06 m^2/s)

From the literature the range of thermal extraction rates are between 20 and 100W/m (VDI 4640, 2001; EN 15450, 2007) depending on the thermal characteristics of the material surrounding the borehole. The *HP-IRL* vertical collector is mid-range with a maximum thermal extraction of 36.7W/m, which is superior to the recommended thermal extraction rate of between 45 and 60W/m for boreholes encased in limestone (VDI 4640, 2001).

Eskilson (1987) showed that seasonal variations in the ground temperature is not of great concern to the overall performance of a vertical collector, by testing a 100m deep vertical collector in granite (λ=3.5 W/m·K) with a 5m thick soil layer on top (λ=2.3 W/m·K) whose thermal performance varies by less than 2% against that of a full 100m deep granite vertical collector. Eskilson concluded that a mean temperature for the entire vertical collector can suffice where all the ground is treated as a homogenous source (Eskilson, 1987). However, due to the large amount of vertical collector piping from the IiBC to the boreholes (see "A", "B" and "C" in Figure 3.23), it may cause a certain variation in the farfield temperature of the *HP-IRL* study from one season to the next as only ~70% of the ground in this study can be said to be located at a depth that maintains a relatively stable temperature (below 15m depth). This aspect is explored in more detail in Chapter 6.

3.4.2 VERTICAL COLLECTOR EXPERIMENTAL TEST PROGRAM

The experimental test program was devised to establish the operational envelope of the heat pump and the conditions imposed on the collector by the variable climatic and associated thermal demands of the IiBC building. Since the weather and to a lesser extent the building demand varies constantly, reliable data could only be established by continuously monitoring all test parameters, testing over extended periods of time and repeating tests where possible. Testing was performed over the period from January 2007 to June 2009.

Using a similar approach to that utilised for the horizontal collector test program in Section 3.3.6 the COP and other indicators of performance under the following conditions:

- Heat pump duty, spanning low, moderate and intensive operation as defined in Section 3.3.6
- Test duration, spanning short, medium and long term operation as previously defined
- response of ground conditions to climate

The range of experimental test conditions were imposed over the course of eight test periods identified as "VC#" in Table 3.10 and the results will be discussed in Chapter 6.

TABLE 3.10 VERTICAL COLLECTOR TEST PROGRAM CONDUCTED BETWEEN 2007 AND 2009

Test #	Demand	Duration	Description
VC1	Intensive	Short	First IiBC heating season observational period with intensive thermal extraction rates.
VC2	Moderate	Long	Comparative heat pump operation period with the $GSHP_{HC}$ and $GSHP_{VC}$ in simultaneous operation
VC3	Intensive	Short	Evaluating the BHE thermal behaviour under a intensive thermal extraction over 3 days. Indicative of a typical winter period of reduced ambient air temperature, inducing a high domestic building thermal demand.
VC4	Low	Short	Evaluating the BHE thermal behaviour under a low intensity, one day thermal extraction period.
VC5	Low	Long	Prolonged steady state low level thermal extraction, indicative of autumn/spring time domestic dwelling utilisation.
VC6	Moderate	Long	Steady state moderate level thermal extraction, indicative of extreme winter time domestic dwelling utilisation.
VC7	Intensive	Medium	Steady state intensive level thermal extraction, indicative of commercial building duty cycle.
VC8	Moderate	Long	Prolonged steady-state moderate thermal extraction, indicative of commercial utilisation.

Observations from the imposed test program also include a characterisation of:

- ground temperature gradients
- seasonal effects on ground temperature around vertical borehole
- ground temperature drawdown with various rates of thermal extraction
- ground thermal recovery

The most common method of establishing a vertical borehole '*farfield*' temperature is to circulate the fluid in the vertical collector without heat extraction, until a steady state temperature is achieved. This established the baseline extract temperature, a benchmark for collector efficiency. Care must be taken to establish the steady state temperature as quickly as possible, as the circulating pump will add heat to the fluid over time, distorting the true steady-state temperature. This test was performed and repeated each year to determine year-on-year effects on the farfield temperature.

This experimental test program was successfully executed while also satisfying the heating demand of the IiBC building using a combination of the three heating systems shown in Figure 3.4.

3.5 *HP-IRL* AIR SOURCE HEAT PUMP

The air source heat pump unit was the *TC-MACH Chamelon 8.3* outdoor unit shown in Figure 3.25. The ASHP has a nominal output capacity of 8.3kW$_{th}$ and was manufactured by a company called *TC MACH Ltd.* based in the Czech Republic.

(a) Front view (Exhaust) of the ASHP **(b)** Rear view (Intake) of the ASHP

FIGURE 3.25 *HP-IRL'S* TC MACH *CHAMELON* AIR SOURCE HEAT PUMP.

The systems uses the refrigerant R407C (16.4kg), a *Copeland* scroll compressor (ZH 21 PFJ) and has dual air/refrigerant heat exchangers. This allows the ASHP to function continuously, where low-grade heat can be delivered from the functioning evaporator to facilitate defrost on the frosted evaporator. Once defrost is complete the roles are reversed if required. A schematic of the TC-MACH system is presented in Appendix J. For this system the manufactures indicate a full defrost power of up to 24% but an overall defrost energy requirement (parasitic) of around 5% in the 0°C to +5°C ambient air temperature range. This energy loss is just 50% of that generated by traditional systems in Figure 2.18.

Each heat exchanger is made of copper pipes and its surface is extended by aluminium lamellas and the flow of air is secured by an axial fan. This arrangement enables reliable and continuous defrosting using the residual heat in the refrigerant after passing the refrigerant/water heat exchanger similar to the alternative hot gas bypass method and was patented in 2005 under the European patent 1 577 624 A2. This defrosting principle can be deployed without the need to interrupt the heating mode, is inexpensive and does not require a complex control system.

The heat pump can operate in heating mode while the external ambient air temperature remains above -15°C and cooling mode can be activated when the external ambient air temperature exceeds +20°C.

The main unit of the heat pump system is located outdoors and it pumps heat into a buffer tank located inside the nearby, Figure 3.26. Heat stored in the buffer tank is used to heat the research facility through fan coil units and water to water heat exchangers.

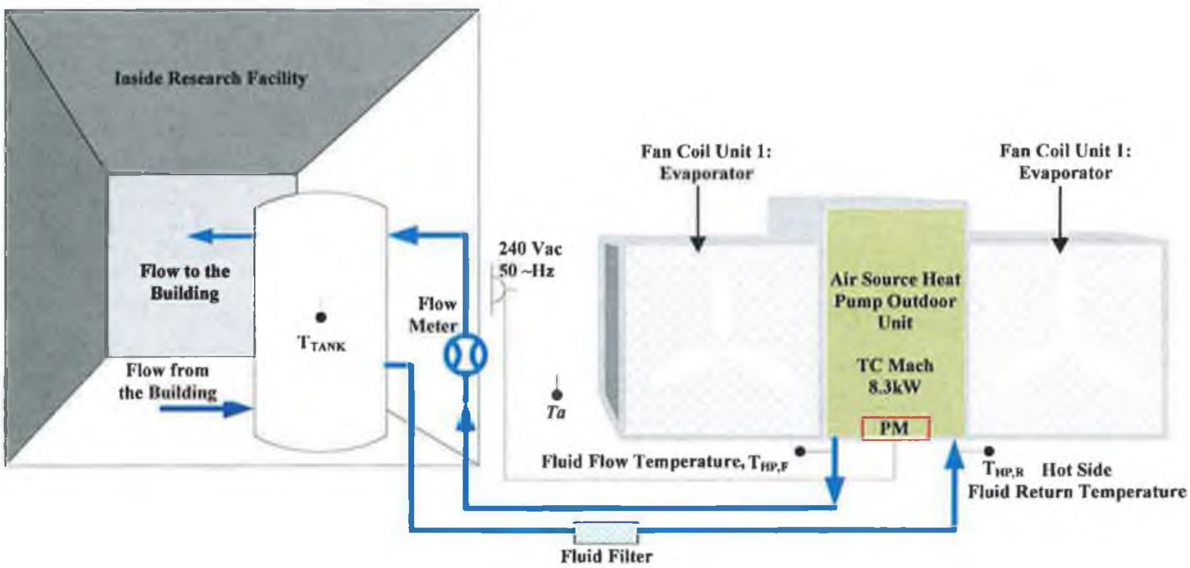


FIGURE 3.26 CONFIGURATION OF THE ASHP AND MEASUREMENT INSTRUMENTATION.

Figure 3.26 also presents a circuit diagram of the air source heat pump system detailing the fluid temperature sensors, flow meter and electrical power monitor's location. These instruments are used to monitor and record COP to an accuracy of ±3%. Both flow and return temperatures were recorded by *Omega* PT100 – 4 wire - PR-11-3-100-M30-75-E-20m DIN 1/10 with ±0.055°C accuracy at +50°C which was validated onsite with an ice-slurry calibration. The temperature sensors were immersed centrally in the flow and return pipes allowing the probe tip (where the sensor is located) to be situated in the centre of the pipe fluid flow. The flow rate is recorded by means of a *Signet* 2551 Magmeter flow meter with an accuracy of ±1%. The flow meter was calibrated on site using both volume and rate calibration methods. The electrical power consumed by the compressor and the main circulation pump was monitored using a *Saia Burgess* AAE1 power meter with a ±1% accuracy. There is also other manufacturer installed instrumentation used to monitor low and high pressure, outlet and inlet heat exchanger temperatures and set point temperatures.

Table 3.11 details the manufacturer's defined ASHP performance over a range of operating temperatures.

TABLE 3.11 TC-MACH CHAMELEON 8.3 TEST RESULTS AS PER EN 14511 STANDARD

Condition	Average Heating Capacity [kW]	Average Power Input [kW]	Coefficient Of Performance [COP]
A7/W35	8.84	2.20	4.0
A0/W35	6.90	2.20	3.1
A-20/W35	4.10	1.90	2.2
A7/W50	9.80	2.75	3.6
A0/W50	7.90	2.85	2.8
A-15/W50	5.20	2.20	2.4

3.5.1 AIR SOURCE HEAT PUMP EXPERIMENTAL PROCEDURE

The ASHP experimental test program was devised to establish the operational performance and the impact of the climatic conditions imposed. Testing was performed over the period from November 2008 to April 2009.

Experimental testing was conducted to continuously monitor COP under the following conditions:

- Heat pump duty, spanning moderate and intensive operation
- Test duration, spanning short, medium and long term operation
- Climatic parameters such as ambient air temperature, relative humidity and wind speed

These test goals were achieved by executing a test program that consisted of the four test periods identified in Table 3.12.

TABLE 3.12 AIR SOURCE HEAT PUMP TEST PROGRAM CONDUCTED BETWEEN 2008 AND 2009

Test #	Demand	Duration	Description
AS1	Moderate	Long	Initial observational period with moderate system demand.
AS2	Intensive	Long	ASHP performances evaluation under a long term intensive duty demand, evaluating start-up and steady-state operational dynamics.
AS3	Moderate	Medium	ASHP performances evaluation under a medium term moderate duty demand, evaluating steady-state operational dynamics.
AS4	Moderate	Long	ASHP performances evaluation under a long term moderate duty demand, evaluating steady-state and intermittent operational dynamics.

The results obtained are presented and discussed in Chapter 6.

3.6 DATA ACQUISITION SYSTEM

The *HP-IRL* project employed 111 sensors to continuously monitor the five systems defined in Table 3.13. While it was the role of another M.Sc. project to commission and test the DAQ system a summary of all the measurement points and sensors used are defined in this section since the responsibility for sensor and DAQ system specification resided with this *HP-IRL* project.

TABLE 3.13 DISTRIBUTION OF SENSORS ACROSS THE *HP-IRL* PROJECT

No.	System Aspect	No. of Sensors	Functionality
1	IiBC Building	6	Recording internal air temperature within the building (2 rooms) for thermal comfort
			Recording flow and return fluid temperatures to the east and west radiator circuits
2	Weather Station	9	Record weather variables listed in Table 3.4
3	$GSHP_{HC}$	9	Recording heat pump performance (COP - source and sink fluid temperatures and flow rates, electrical consumption)
		56	Recording horizontal collector ground temperatures (both inside and outside of collector area)
		10	Recording horizontal collector ground moisture content
4	$GSHP_{VC}$	7	Recording heat pump performance (COP - source and sink fluid temperatures and flow rates, electrical consumption)
		5	Recording vertical collector pipe wall temperatures
		4	Recording individual vertical collector fluid flow and return temperatures
5	ASHP	4	Recording heat pump performance (COP - sink fluid temperatures and flow rate, electrical consumption)
		1	Recording local external ambient air temperature
	Total:	111	

The DAQ system was based around *National Instruments* (NI) FieldPoint modules shown in Figure 3.27. The DAQ system hardware consists of six different types of FieldPoint modules, which were required to interface the 111 sensors with the DAQ software and data storage.

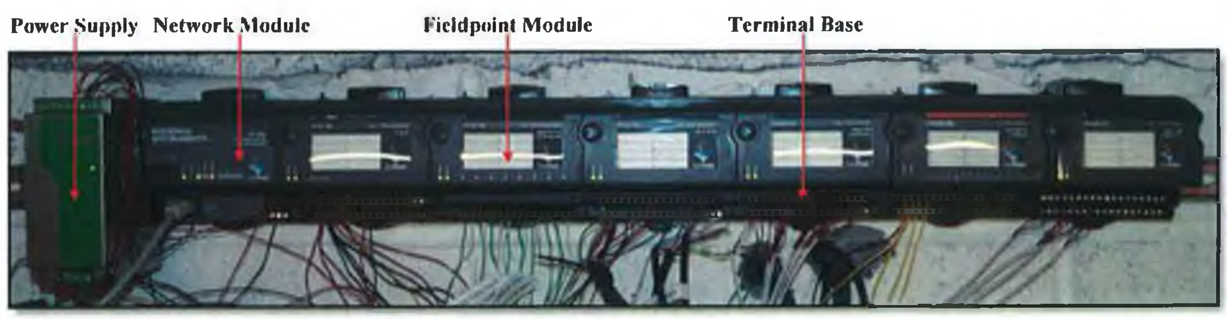


FIGURE 3.27 *HP-IRL* DATA ACQUISITION SYSTEM.

The FieldPoint modules include a variety of isolated analog and digital I/O modules, counter or pulse modules, terminal bases and network interfaces for ease of connection to standard network technology, such as ethernet, serial or wireless. As indicated in Figure 3.27 the FieldPoint modules are attached to terminal bases to provide the communication link between each other, the network module and wiring field connections.

These modules were installed at the following three locations shown in Figure 3.1: in a control box located at the horizontal collector (Location A); in the IiBC plant room (Location B) and adjacent to the ASHP (Location C). The IiBC plant room contains the $GSHP_{HC}$, $GSHP_{VC}$ and gas condensing boilers. Table 3.14 details the list of data modules used in the *HP-IRL* study and their respective locations.

TABLE 3.14 LIST OF *FIELDPOINT* MODULES INSTALLED THROUGHOUT THE *HP-IRL* TEST FACILITY

No.	Module/Item	Number	Function	Location (A, B or C)
1	FP-AI-100	1	Voltage and current analog input module	B: IiBC plant room
2	FP-AI-110	1	Voltage and current analog input module	B: IiBC plant room
3	FP-RTD-124	2	4 Wire RTD input module	B: IiBC plant room
4	FP-RTD-124	7	4 Wire RTD input module	A: Control Box in HP collector profile
5	FP-CTR-500	1	Counter input module	B: IiBC plant room
6	FP-CTR-502	2	Counter input module	A: Control Box in HP collector profile
7	FP-TC-120	1	Thermocouple input module	B: IiBC plant room
8	FP-1601	3	Network module	A, B & C
9	Power supply	3	Supply 24V to the FP modules	A, B & C
10	FP-RTD-124	1	4 Wire RTD input module	C: TESSA Facility
11	FP-CTR-500	1	Counter input module	C: TESSA Facility

The network modules at locations A, B and C were connected to the GMIT's Local Area Network (LAN) with shielded Category 5 network cable. An NI power supply was used as part of the DAQ hardware to supply 24V to the modules. The power supply was run via an Un-interruptible Power Supply (UPS) to prevent power loss or surges. The FieldPoint modules were assigned an IP address by GMIT's network administrator to allow communication between the modules and the *HP-IRL* system PC.

The programming and setup of the FieldPoint modules was carried out through Measurement and Automation eXplorer (MAX) software. The FieldPoint modules are connected to the network module and appear in the MAX software when updated. The modules can be configured to the user's requirements, with appropriate name, measurement range and scanning intervals.

The DAQ system was controlled and monitored using Labview 8.20, running on a *Dell* Optiplex Gx280 PC with a Pentium IV processor (2.8GHz) and 2 GB of RAM. This PC was connected to GMIT's network to facilitate communication between remote PC's in the IiBC plant room and the DAQ hardware. The PC was powered by an UPS to prevent surges and/or power loss.

A total of 18 Labview Virtual Instruments (VI's) and 42 sub-VI's were created to construct and operate the DAQ system. These VI's were used to program and display the sensor outputs via the FieldPoint modules. Labview is made up of a front panel and a block wiring diagram display panel. The front panel is used as the display screen for the live data and control icons to configure each VI and can be seen in Figure 3.28.

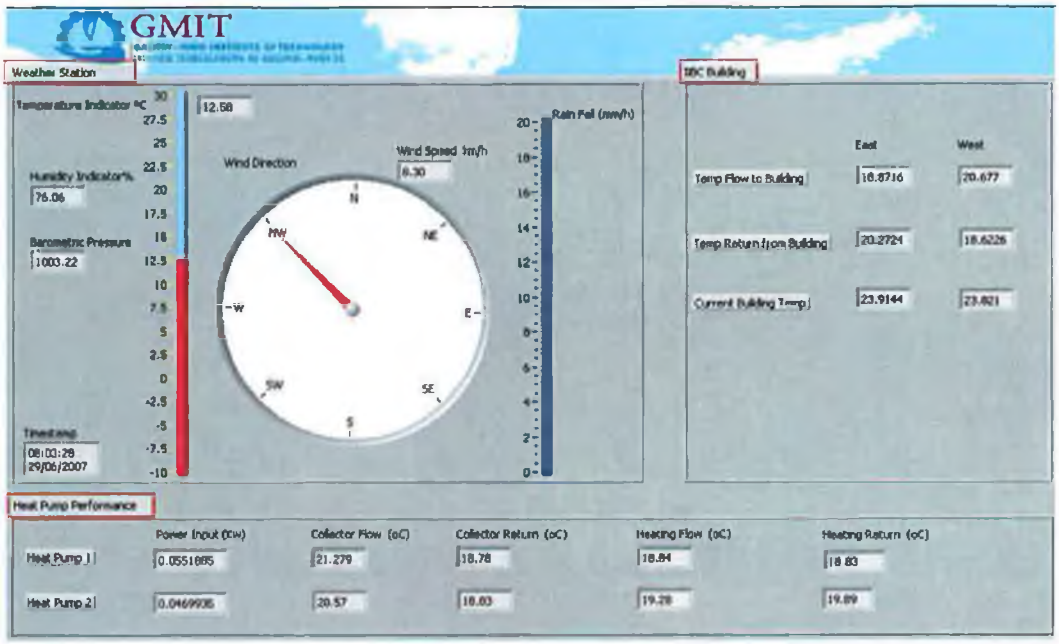


FIGURE 3.28 LABVIEW FRONT PANEL CREATED TO DISPLAY SUMMARY DATA FROM THE WEATHER STATION, THE IIBC AND BOTH GSHPS.

The block diagram is used for the setup and virtual wiring of each VI's configuration which allows the programming of each FieldPoint module, sensor logging and recording intervals and signal manipulation.

3.7 SUMMARY

This chapter details the *HP-IRL* test facilities established to characterise heat pump performance in the Maritime climate of Ireland. The *HP-IRL* facility has the capacity to test two heat pump technologies and three collector designs; $GSHP_{HC}$, $GSHP_{VC}$ and ASHP. The facility targeted a thorough investigation of the collector performance so the potential for higher performance climate-sensitive collector design and source side management may be revealed. A detailed description of the test facility has been presented which identified the location and role of the 111 sensors deployed. Details of the wide-ranging test program undertaken between 2007 and 2010 were also provided, which consisted of 22 individual tests; 9 conducted on the horizontal collector, 8 on the vertical collector and 5 on the air source heat pump.

In total, 747 test days were accumulated, delivering a combined 168,522 kWh's of thermal energy to a functioning office building and ASHP test environment. The key findings of this extensive test program are presented in chapters 4, 5 and 6.

CHAPTER 4 – HORIZONTAL COLLECTOR: EXPERIMENTAL ANALYSIS

This chapter evaluates the data recorded during the three-year long experimental investigation conducted on the horizontal collector described in Section 3.3. The author believes that this is the longest and most intensive, continuous experimental investigation undertaken on a horizontal collector in any climate. Such extensive testing has allowed different interrelated characteristics, such as climate, heat pump duty, ground temperature drawdown during heat pump operation and recovery rates, influence of heat pump duty as well as the influence of ground cover to be quantified for the first time.

4.1 GROUND RESPONSE TO CLIMATE

This section summarises the ground's response to the climate from seasonal and diurnal variation, while identifying the most significant weather variables, in terms of its depth and extent of influence. The study reveals significant differences between influential seasonal and short-term weather events. Note that the heat pump is not operational for any of the results presented in this sub-section. As an introductory indication to the ground's response to climate, Figure 4.1 shows the daily averaged ground temperature at Profile 1 recorded from 2007 to 2010.

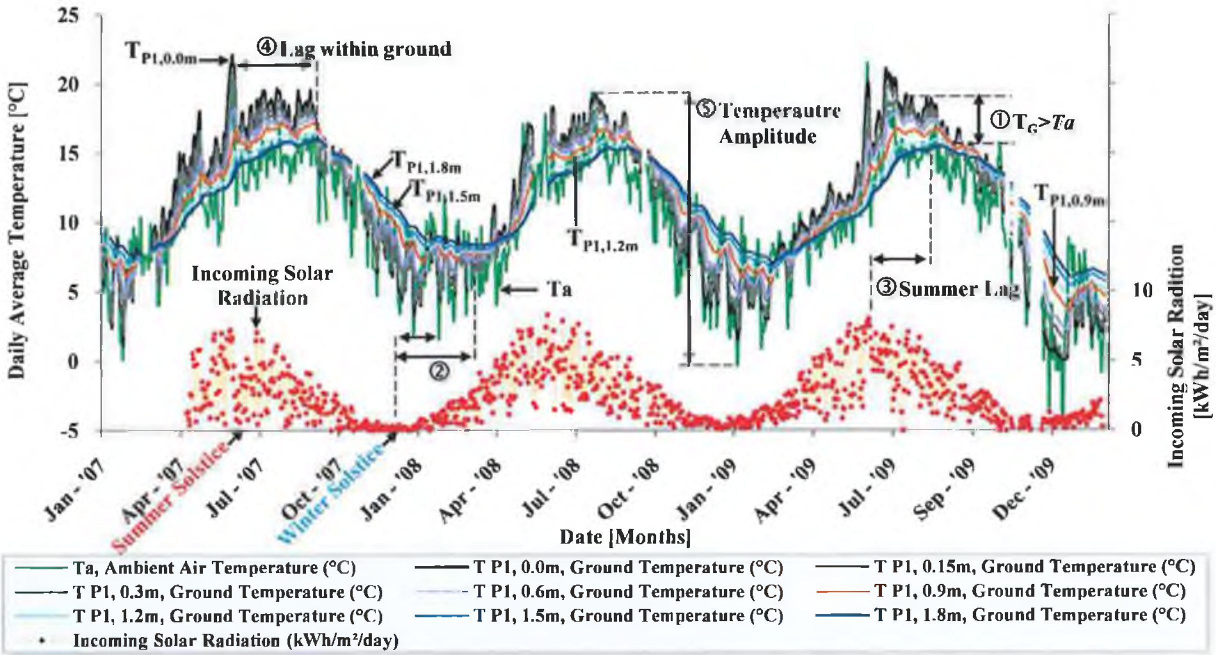


FIGURE 4.1 INFLUENCE OF CLIMATE (SOLAR RADIATION AND AMBIENT AIR TEMPERATURE SHOWN) ON GROUND TEMPERATURE.

The salient features of Figure 4.1 are:

① Ground Temperature (T_G) is greater than ambient air temperature (Ta) in summer

② Winter Lag – after winter solar radiation low (winter solstice) minimum ambient air temperature shows a 5 week (~34 day) lag and ground temperature at 0.9m depth shows a 7 week (~49 day) lag

③ Summer Lag – after the summer solar radiation high (summer solstice) ambient air temperature and ground temperature shows similar lag to winter conditions

④ Lag with depth – lag varies proportional to depth, where ground surface ($T_{P1,0.0m}$) shows negligible lag and at a depth of 0.9m there is a 7 week lag

⑤ The amplitude of annual temperature variation diminishes with depth

Figure 4.2 illustrates the seasonal phase lag between the incoming solar radiation and ambient air temperature at the collector depth of 0.9m.

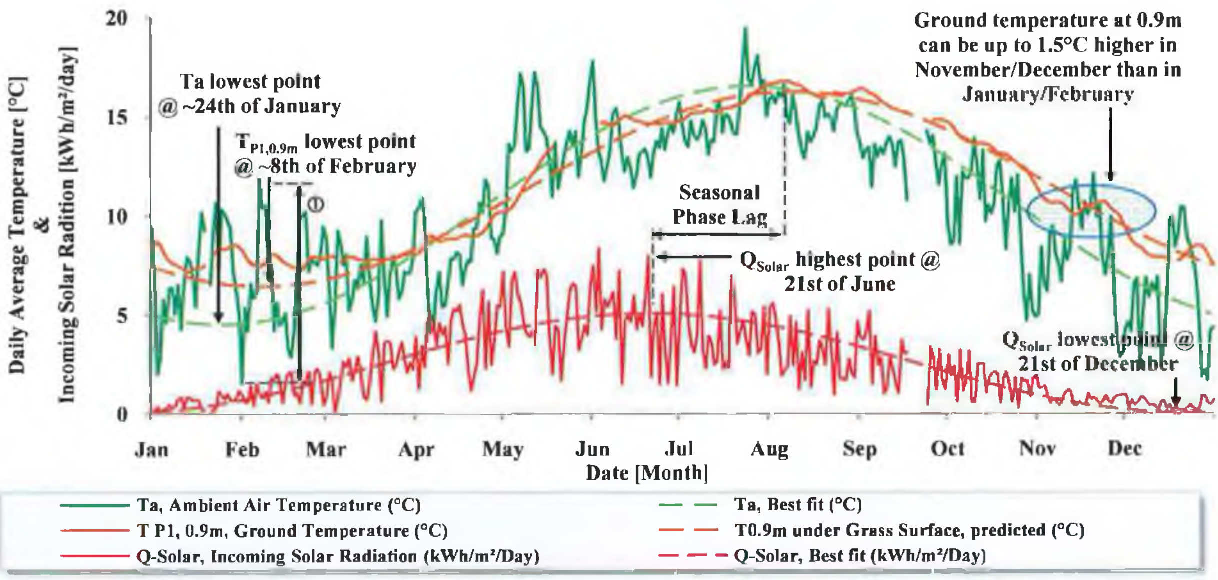


FIGURE 4.2 PROFILE OF DAILY AVERAGED SOLAR RADIATION, AMBIENT AIR TEMPERATURE AND GROUND TEMPERATURE AT 0.9M DURING 2008 VERSUS THAT PREDICTED BY EQUATION 2.1.

Figure 4.2 highlights a feature of the Maritime climate where considerable fluctuation in the ambient air temperature occurs within short periods throughout the year. An example is indicated by ① symbol, where the ambient air temperature changes 10.2°C over six days. While on the upper end of the fluctuation, it is not unusual for the ambient air temperature to fluctuate by 6-8°C within a few days, which drives a corresponding fluctuating heating demand.

Figure 4.3(a) shows the close agreement between the predicted ground temperatures, from Equation 2.1, at a depth of 0.9m with that of the monthly average recorded temperatures in Profile 1. This highlights the predictability of the soil temperature with depth once accurate inputs are available and the impact of short-term weather events are dampened by monthly averages.

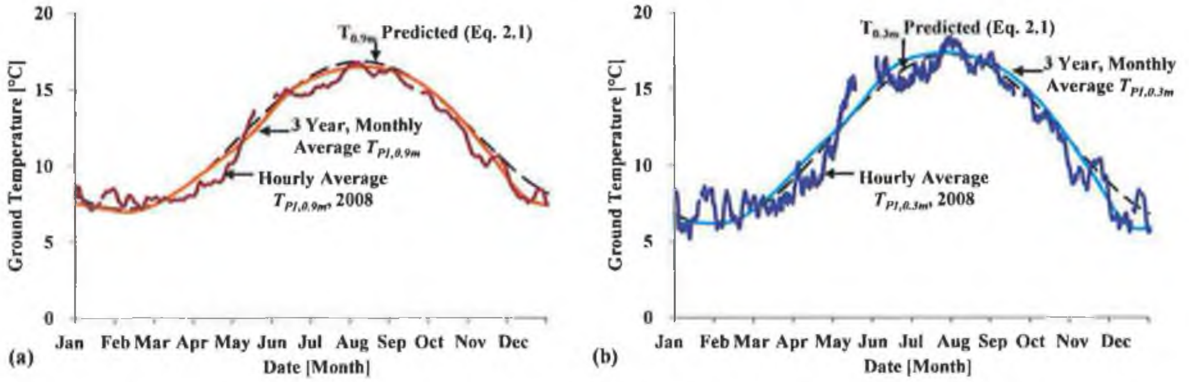


FIGURE 4.3 COMPARISON OF *HP-IRL* MEASURED HOURLY AND MONTHLY AVERAGED GROUND TEMPERATURE OF PROFILE 1AT (A) 0.9M DEPTH BETWEEN 2007 AND 2009 WITH THAT OF THE PREDICTED GROUND TEMPERATURES AT 0.9M AND (B) 0.3M DEPTH BETWEEN 2007 AND 2009 WITH THAT OF THE PREDICTED GROUND TEMPERATURES AT 0.3M.

While it is reassuring to be able to predict ground temperature, based on four relatively accessible parameters, it cannot however account for the impact of rainfall events, severe low temperature occurrences and heat pump operation on either the ground's temperature or thermal capacity. It is also important to highlight that differences between actual and predicted ground temperatures are amplified, with greater diurnal variations, as one approaches the surface as indicated in Figure 4.3(b).

Therefore, while monthly averaged ambient air and ground temperatures are relatively predictable, these mask significant variations over shorter time periods that may have a substantial effect on the horizontal collector and heat pump performance. Furthermore, these climate variations are translated into soil temperature fluctuations to the current collector depth of 0.9m, which cannot be predicted by simulation using Equation 2.1. The question arises if these fluctuations are significant for heat pump operation. A further evaluation of weather events is presented in Section 4.1.2.

Therefore, Figures 4.2 and 4.3 allow the following conclusions to be drawn:

- That weekly or monthly average ambient air, ground temperatures and solar radiation are predictable
- Predicted results are unable however to capture the impact of the high frequency of weather events that differ significantly from the weekly/monthly averages
- Such frequent changes in weather patterns are a feature of *Cool Marine* climates and this raises the question if these non-predictable weather events impact negatively on heat pump performance or could be harnessed to produce a positive influence on heat pump performance

4.1.1 MEASURED GROUND VERTICAL TEMPERATURE GRADIENT

In $GSHP_{HC}$ design, it is desirable to position the collector at a depth that delivers the highest heating season collector fluid return temperature. In order to establish a depth that achieves this aim a vertical profile of the ground temperatures to a depth of 1.8m (Profile 1) was undertaken outside of the collector region using sensors located at eight depths defined in Figure 3.20 and Figure 4.4 presents monthly average temperatures recorded.

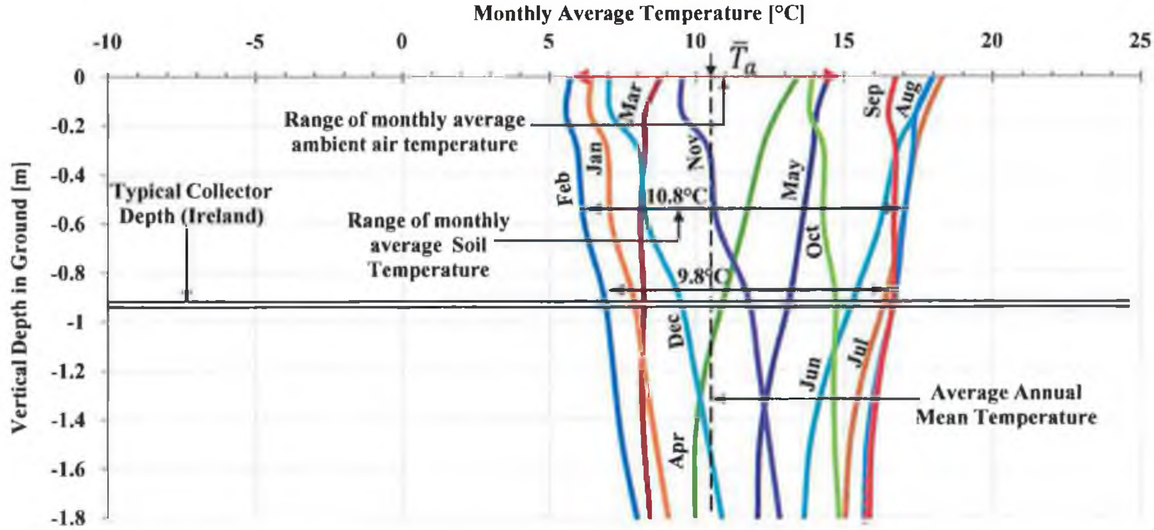


FIGURE 4.4 MONTHLY AVERAGED GROUND TEMPERATURES RECORDED OVER ONE YEAR IN GALWAY, LATITUDE 53°N (P1, 2007).

Figure 4.4 shows that:

- Minimum ground temperatures at all depths occur in February, which will generate the greatest heat pump temperature lift at a time of peak heating demand
- The minimum ground temperature in February increases with depth, suggesting increasing heat pump performance with collector depth
- The average year-round ground temperature at a depth of 0.9m deep during 2007 was almost +12°C, +9°C for 8 month heating season (October to May) and +7°C for January and February
- Monthly averaged *Ta* fluctuated between +6°C in January and +15°C in August 2007
- Monthly average ground surface temperature fluctuated between +6°C in February and +18°C in July 2007

Table 4.1 presents the seasonal average ground temperatures of 0.3m and 0.9m depths in the Maritime Irish climate.

TABLE 4.1 SEASONAL GROUND TEMPERATURES BASED ON 2007, 2008 AND 2009

Season	Months	P1 Seasonal Average Temperature at Depths		Temperature Difference
		0.3m	0.9m	$\Delta T = T_{0.3m} - T_{0.9m}$
Spring	March, April, May	10.6°C	10.0°C	0.6 K
Summer	June, July, August	16.9°C	15.8°C	1.1 K
Autumn	September, October, November	13.2°C	13.7°C	-0.5 K
Winter	December, January, February	5.6°C	6.8°C	-1.2 K

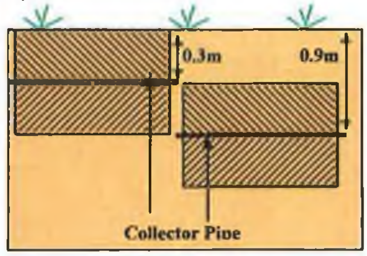


Table 4.1 highlights that the ground temperature at 0.3m depth is higher than 0.9m for the spring and summer seasons by as much as 1.1K, which could theoretically to be translated into to an improved heat pump COP for those seasons.

Figure 4.5 contrasts the measured data in Figure 4.4 with that of a continental climate at latitude 45° N in Ottawa, Canada.

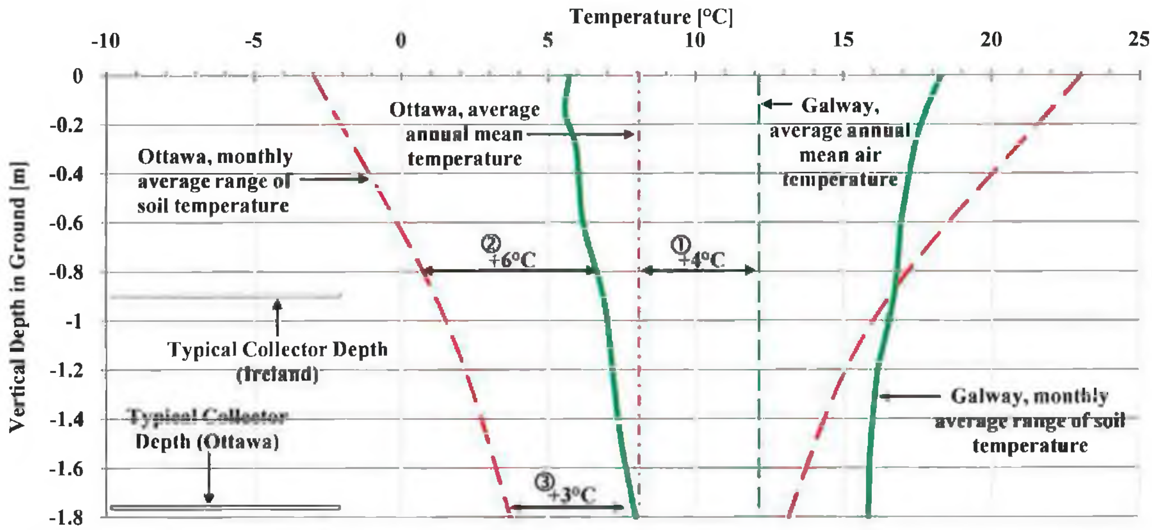


FIGURE 4.5 COMPARISON OF DEPTH DEPENDENCE OF ANNUAL RANGE OF GROUND TEMPERATURES IN GALWAY (IRELAND, 53°N) AND OTTAWA (CANADA, 43°N) (WILLIAMS AND GOLD, 1976).

Despite Ottawa being located at a latitude 8° south of Galway, it is notable that Ireland maintains a:

① +4°C higher and far more compact ambient ground temperature range than Ottawa

② +6°C higher minimum ground temperature at a depth of 0.9m

③ +3°C higher minimum ground temperature at a depth of 0.9m compared with the minimum ground temperature at a depth of 1.8m in Ottawa

The higher ground temperatures generated by the Irish Maritime climate illustrate the potential for high and stable ground temperatures between 0.8m and 1.2m that should translate into a performance advantage for Irish heat pumps. These positive observations reinforce the need for the *HP-IRL* study and the opportunity it presents for optimisation.

4.1.2 IMPACT OF WEATHER ON GROUND CONDITIONS

This section explores the impact of the climate and specific weather events (ambient air temperature, solar radiation, rainfall, wind speed) on ground temperature. Data presented in Figures 4.6 through 4.12 were recorded outside the collector region in the grass covered Profile 1 and are being presented not only to establish the penetration depth and duration of influence associated with weather events, but also to present a novel graphical approach to identifying the impact.

4.1.2.1 SOLAR RADIATION AND AMBIENT AIR TEMPERATURE

Figure 4.6(a) illustrates the ambient air temperature and incoming solar radiation from 2007 to 2010, including a best fit profile for both parameters.

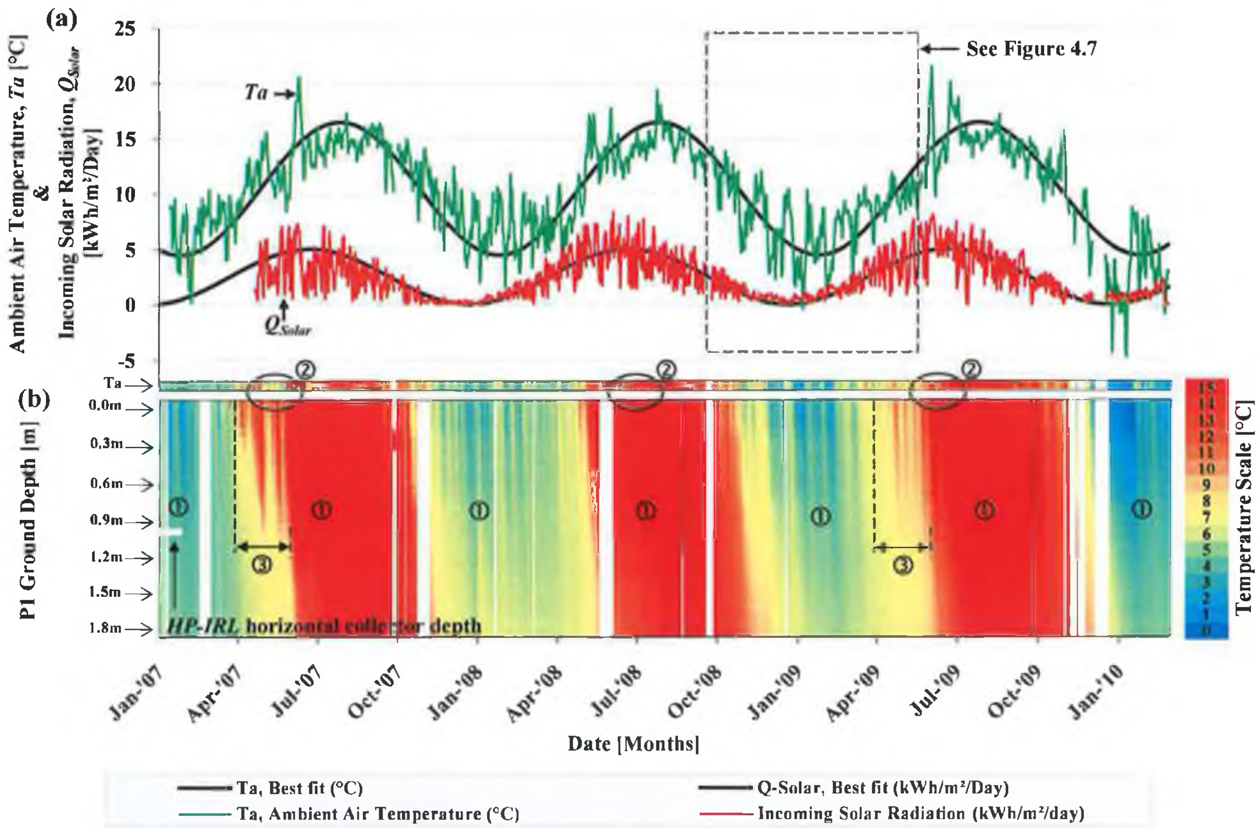


FIGURE 4.6 (A) DAILY AMBIENT AIR TEMPERATURE AND INCOMING SOLAR RADIATION DRIVING THE THERMAL CHARGE AND DISCHARGE OF THE GROUND IN PROFILE 1 (B) PROFILE 1 3-D TRANSIENT TEMPERATURE MAP SHOWING HOURLY GROUND TEMPERATURE VARIATION WITH DEPTH TO 1.8M.

Using a novel three dimensional representation of the transient ground temperature with depth, Figure 4.6(b) shows the impact of climate on the ground temperature to a depth of 1.8m. The three dimensional transient temperature map in Figure 4.6(b) was based on temperature measurements recorded using the vertical temperature profiles in Figure 3.20 and is presented using a grid with a spatial resolution of 0.15m in depth and one hour time intervals. Linear interpolation was used to increase the spatial resolution of the measured data to that displayed in Figure 4.6(b). The white stripes through Figure 4.6(b) represents short periods where the DAQ system was off due to maintenance or upgrade of the PC/Network.

Figure 4.6 highlights:

① Seasonal variation in ground temperature with depth between +5°C and +15°C

② Ground temperature higher in upper surface layer than ambient air temperature in some instances due to the impact of solar gain

③ Temperature lag increases with ground depth

Figure 4.7 is taken from the shorter time period shown in Figure 4.6(a) and illustrates the thermal characteristics of the ground during the 2008/2009 heating season.

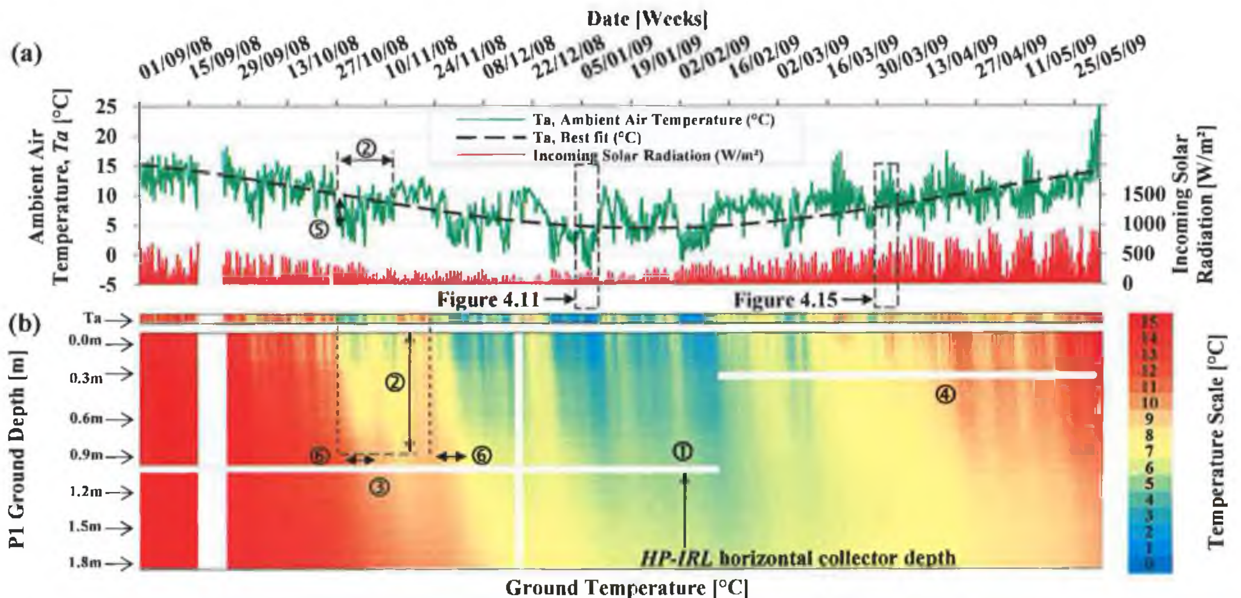


FIGURE 4.7 (A) HOURLY AVERAGED AMBIENT AIR TEMPERATURE AND INCOMING SOLAR RADIATION DRIVING (B) 3-D TRANSIENT TEMPERATURE MAP OF THE GROUND THERMAL CHARGE AND DISCHARGE AT PROFILE 1 DURING HEATING SEASON FROM SEPTEMBER 2008 TO MAY 2009.

Some key observations from Figure 4.7 are:

① Clearly seasonal effects penetrate to the collector depth of 0.9m where average winter ground temperature of +5.1°C, whereas it was +3.4°C at 0.3m

② Effects of two-week long October cold period penetrated 0.9m deep

③ Higher ground temperatures exist at 0.9m and below from September to February

④ Higher ground temperatures at 0.3m and above from September to February

⑤ Cold periods of 4°C below the seasonal norm occur every 3 days in winter, and lasts on average for 1 day (similar results for all three years)

⑥ Estimate of time lag between weather events and it having an impact at 0.9m collector depth

Figure 4.7(a) also highlights two, one-week long periods that characterise typical weather events in winter and spring. The resultant effects on the ground's thermal energy during these weeks are presented in Figures 4.10 and 4.12 in Section 4.1.2.3.

4.1.2.2 RAINFALL

As illustrated in Section 2.3.4.1 moisture content is an important factor in the ground's thermal storage capacity and thermal mobility. Moisture content helps to increase the specific heat capacity of the ground by displacing air, reducing thermal contact resistance between the ground and the collector pipe and boost the thermal conductivity from 0.25 W/m·K for dry soil to 2.5 W/m·K for saturated soil, as shown in Figure 2.8(b).

Figure 4.8 illustrates the moisture content at the collector depth in three profiles, along with the rainfall levels for December 2009 to January 2010.

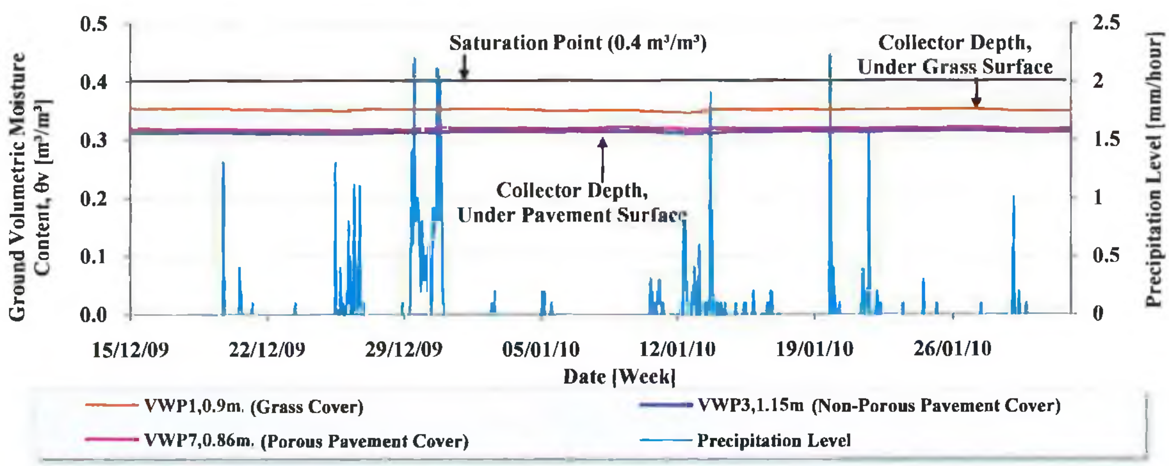


FIGURE 4.8 RECORDED RAINFALL AND HOURLY AVERAGED GROUND VOLUMETRIC MOISTURE CONTENT AT COLLECTOR DEPTH (DECEMBER 2009 TO JANUARY 2010).

It is noticeable from Figure 4.8 that despite the relatively dry ten day period in early January there was a negligible impact on the volumetric moisture content (θv) at the collector depth. Indeed the moisture content remained consistently above 75% (0.3m³/m³ with a saturation point 0.4m³/m³) for both the grass and paved ground covers, with the paved surface yielding a 9% lower moisture content than grass. Figure 4.9 however shows that for the ten day dry period θv drops as low as 0.22m³/m³ at 300mm depth before recovering due to the significant rainfall event on January 11th. Again however the moisture content consistently exceeded 50%, substantially above the critical 25% level identified numerically by Leong *et al.* (1998) as the threshold level for good heat pump performance.

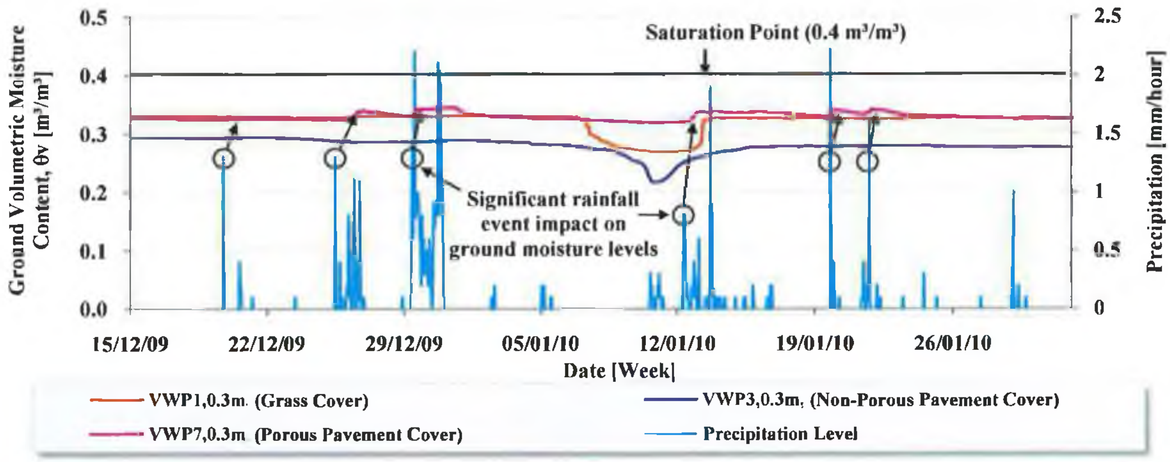


FIGURE 4.9 RECORDED RAINFALL AND HOURLY AVERAGE GROUND VOLUMETRIC MOISTURE CONTENT IN COLLECTOR REGION AT 0.3M DEPTH (DECEMBER 2009 TO JANUARY 2010).

Typically the Maritime climate generates a substantial and consistent rainfall pattern throughout the heating season and will therefore tend to deliver consistently high ground moisture contents. Additionally, with heat extraction the thermal conductivity of the ground should increase as the effect of moisture migration by diffusion pulls moisture towards the colder collector pipe. Not only will rainfall increase the conductive qualities of the ground, with groundwater movement there is the added effect of advection. While this may be a substantial contributor to the successful operation of the heat pump system it can be difficult to quantify and account for in design (RETScreen, 2005).

Conclusions that can be drawn from the rainfall effects on ground thermal capacity are:

- The Irish Maritime climate maintains consistent ground saturation levels, well above the critical 25% level during the peak heating season
- There is some evidence of moisture content reduction at 0.3m depth after 6 dry days, with no impact at 0.9m depth
- A significant rainfall event was defined as having a visible impact on ground moisture content at 0.3m depth, consisting of a precipitation level of 0.5mm per hour and lasting more than 3 hours
- A significant rainfall event occurs on average every 3 days in winter over the three years
- Significant rainfall events occurred on average every 4 days in spring, 3 days in summer and 4 days in autumn over the three years

4.1.2.3 WEATHER SENSITIVE GROUND TEMPERATURE MAPPING

This section explores if the influence of specific weather events can be isolated and quantified.

Figure 4.10 extends the novel 3-D temperature mapping concept in Figure 4.6(b) to allow a more detailed climatic impact on ground temperature analysis to be conducted for January 2008.

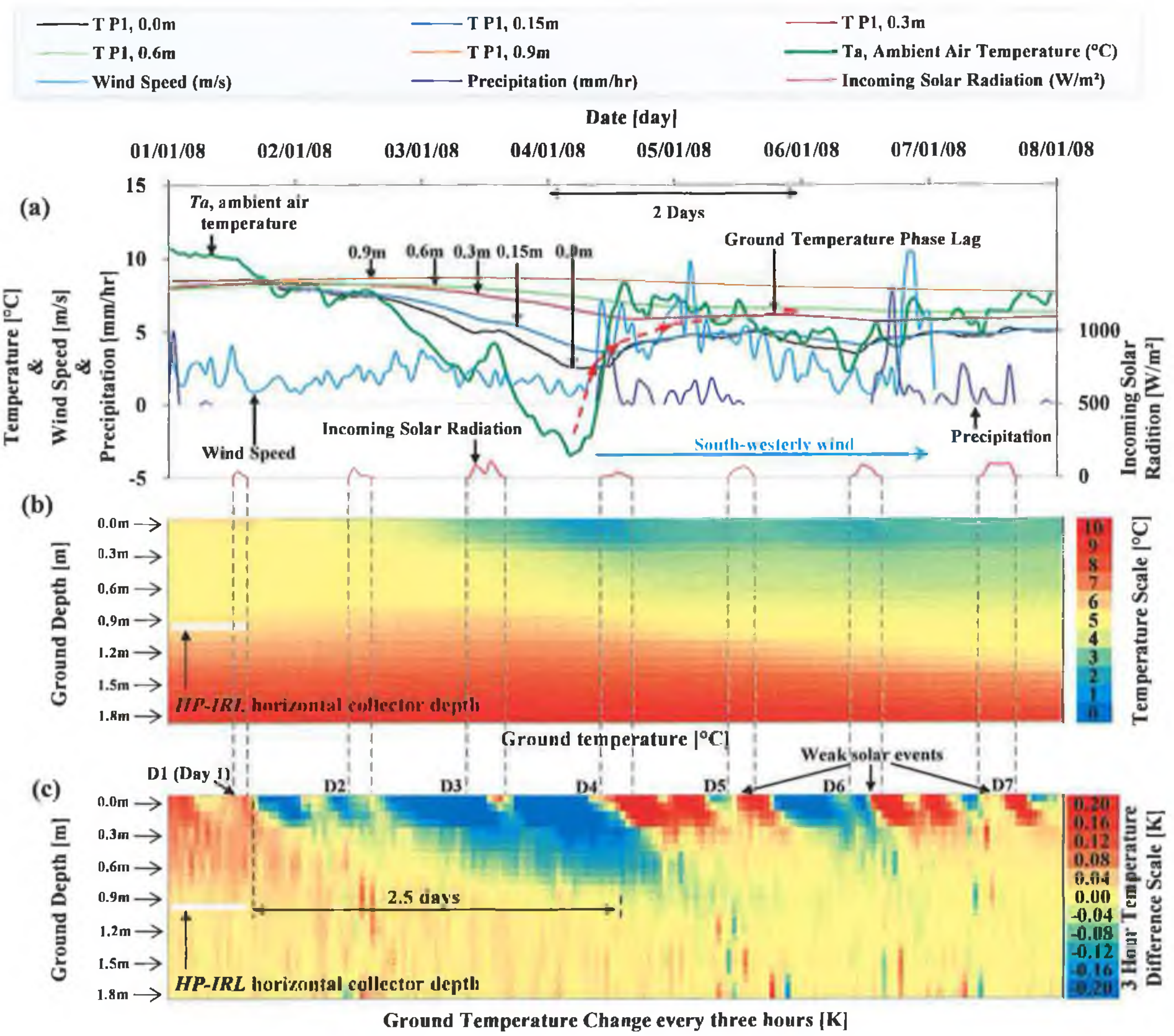


FIGURE 4.10 FIRST WEEK IN JANUARY, 2008, HOURLY AVERAGE (A) TRANSIENT WEATHER PARAMETERS AND PROFILE 1 GROUND TEMPERATURE, (B) 3-D TRANSIENT TEMPERATURE MAP AT PROFILE 1 AND (C) 3-D TRANSIENT TEMPERATURE CHANGE MAP OF THE RATE OF GROUND TEMPERATURE CHANGE EVERY THREE HOURS.

Figure 4.10(a) charts the typical climate variables (ambient air temperature, precipitation and wind speed) and the resultant ground temperature variation with depth. A significant weather event occurs between January 2nd and 4th where the external ambient air temperature continues to drift lower, particularly on January 3rd and 4th.

Figure 4.10(b) illustrates the three dimensional transient temperature map illustrating the impact of the cold period on ground temperature. It is noticeable that this weather event reduces the ground temperature to a depth of 1.5m by January 8th.

Figure 4.10(c) illustrates the rate of ground temperature change every three hours. While the absolute temperature change every three hours is less than 0.2°C this temperature mapping approach allows the impact of climate on ground temperature to be resolved with higher definition. It allows the temperature change and depth of penetration of specific weather parameters to be resolved. For instance:

- Incoming solar radiation: the signature of weak incoming solar radiation during each of the seven days is clearly visible.
- Wind speed: the depth of penetration of the temperature signature is not only influenced by the solar intensity but also by the combined convective influences of the wind speed and ground surface-to-air temperature difference. This is highlighted by the contrasting weak signatures on days 1 through 3 (D1→D3 in Figure 4.10(c)), during low wind, with stronger signatures on days 4 through 7 (D4→D7) that result from higher wind speeds. The warm signature during the night-time between D4 and D5, and D6 and D7 result from this convective effect.
- External ambient air temperature: while wind speed and solar intensity remain relatively stable the steady 13°C fall in external ambient air temperature during days 1 through 4 generates significant ground cooling to a depth of 0.9m.
- Rainfall: three separate rainfall events take place during the seven days and it is possible to identify the faint influence of its effect during early in D1, between D4 and D5 and between D6 and D7 in Figure 4.10(c), showing an increased temperature signature intensity

The main driver for the recovery on the 4th of January in Figure 4.10 is due to the ambient air temperature returning back to +5°C, which is in turn influenced by the warm south-westerly winds. The heightening of the ambient air temperature during the day can correspond to a heightening of the grounds temperature to a depth of 0.6m at night.

Figure 4.10 demonstrates that the combination of climate and ground temperature (Figure 4.10(a)) with three dimensional graphical analysis techniques in Figure 4.10(b) and (c) offers the sensitivity to not only resolve the impact of climate on ground temperature but also the impact of specific weather parameters. This offers significant new insight into the influence of collector design encompassing piping configuration and positioning, soil type and ground cover type as well as operational parameters that enable improved source side management, involving collector thermal energy management and climate sensitive duty cycles informed by ground temperature feedback sensors.

4.1.2.4 CLIMATE INDUCED GROUND THERMAL ENERGY MAPPING

Knowing the transient ground temperature (Figures 4.1 and 4.7), soil type and moisture content (Figures 4.8 and 4.9) enables the ground's fluctuating thermal energy content to be established. The climate driven change in the ground volumetric energy content can be calculated using Equation 4.1.

$$Q_{Depth} = V \cdot C_G \cdot \Delta T \qquad \textit{Equation 4.1}$$

Q_{Depth} is the thermal energy content within a layer of specified thickness at a specified depth and measured in kWh/m², Where V is the volume of collector ground per one square meter of collector area, C_G is the ground volumetric heat capacity (2.29 MJ/m³·K) and ΔT is the temperature difference between the ground temperature at the specific depth and the baseline temperature specified as 0°C.

The volumetric energy content available within a 0.6m thick layer, centred at a depth of 0.3m ($Q_{0.0m\text{-}0.6m}$) and a 0.6m thick sub-layer centred at 0.9m ($Q_{0.6\text{-}1.2m}$) is presented in Figure 4.11. This confirms that during the peak heating season that there is over 30% more energy available at the 0.9m collector depth than at the shallower 0.3m depth and it is also more stable.

By way of reference, a thermal demand Q'_{HC} of a generic domestic dwelling with a $15kW_{th}$ peak demand is also shown in Figure 4.11. The climate driven space heating requirement was proportional to ambient air temperatures between +15.5°C (0%) and -4°C (100%). A generic daily hot water (DHW) demand of 18kWh (6570kWh/annum) was also included.

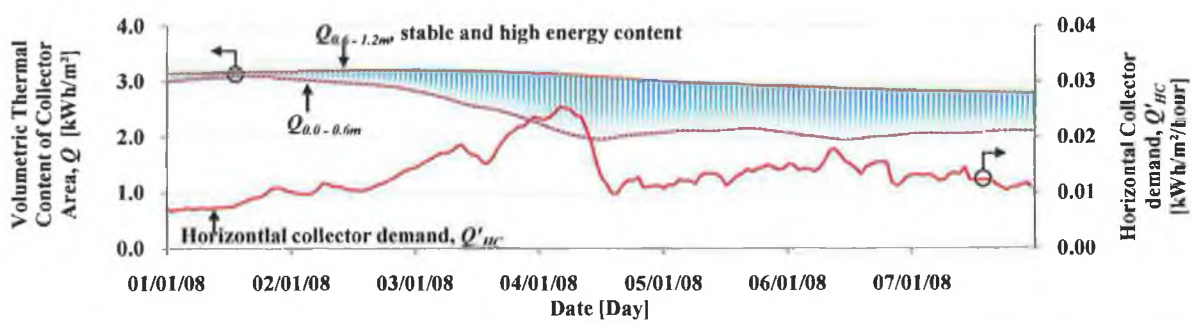


FIGURE 4.11 THERMAL ENERGY CONTENT WITHIN A 600MM THICK LAYER CENTRED AT DEPTHS OF 0.3M AND 0.9M IN PROFILE 1 VERSUS A PROJECTED COLLECTOR THERMAL DEMAND AT PEAK HEATING SEASON, JANUARY, 2008.

Figure 4.11 shows the demand is driven by external ambient air fluctuations peaking on the 4th of January. Contrasting available energy and demand, Figure 4.11 shows that the available energy exceeds estimated demand by more than an order of magnitude at both collector depths, but higher collector/heat pump performance would be generated by the 0.9m deep collector.

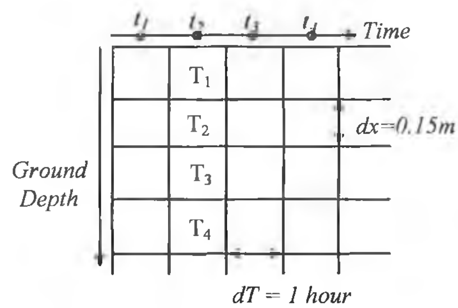
This analysis is repeated for a one week period in early April in Figure 4.12, where the combination of higher external ambient air temperature and solar radiation increase the temperature of the ground surface layer, Figure 4.12(b) signalling the a spring-time thermal rejuvenation period. This is in contrast to Figure 4.11(b) which shows the warmest temperatures are below collector.

Figure 4.12(c) presents the climate driven soil temperature change per three hour period. The influence of the longer hours of daylight and higher intensity of solar radiation inject a stronger solar impulse that penetrates to the collector depth (0.9m) within approximately 12 hours. The strength of the daily solar impulse is influenced by the combination of the weather and the temperature difference between the ambient air temperature and the ground surface layer.

Contrast the strong and deep reaching impulse on day 1 and 2, resulting from the cooler ground temperatures, high wind speed and positive air-ground temperature difference, with those of days 3, 4 and 5 when low wind speed and neutral air-ground temperature differences exist. Figure 4.12(d) converts the ground temperature difference per three hour period into a heat flux which shows positive day-time heat flux impulses of up to 20W/m² entering the ground and negative night-time heat flux leaving the ground. The heat flux is calculated using hourly average ground temperatures and with depth intervals of 0.15m as follows:

$$Q_{Depth} = -\lambda_G \cdot A \cdot \left(\frac{dT}{dx}\right)_t \qquad \textit{Equation 4.2}$$

$$Q''_{Depth} = \frac{Q_{Depth}}{A} = -\lambda_G \cdot \left(\frac{dT}{dx}\right)_t \qquad \textit{Equation 4.3}$$



The depth specific heat flux Q''_{Depth} is measured in W/m², where A is the area and λ_G is the ground thermal conductivity. Figure 4.12(e) illustrates the difference in heat flux at 0.3m and 0.9m depths. It demonstrates the potential for up to 89% of the horizontal collector thermal demand, q''_{HC} (in W/m²) to be satisfied by the solar driven heat flux at 0.3m depth.

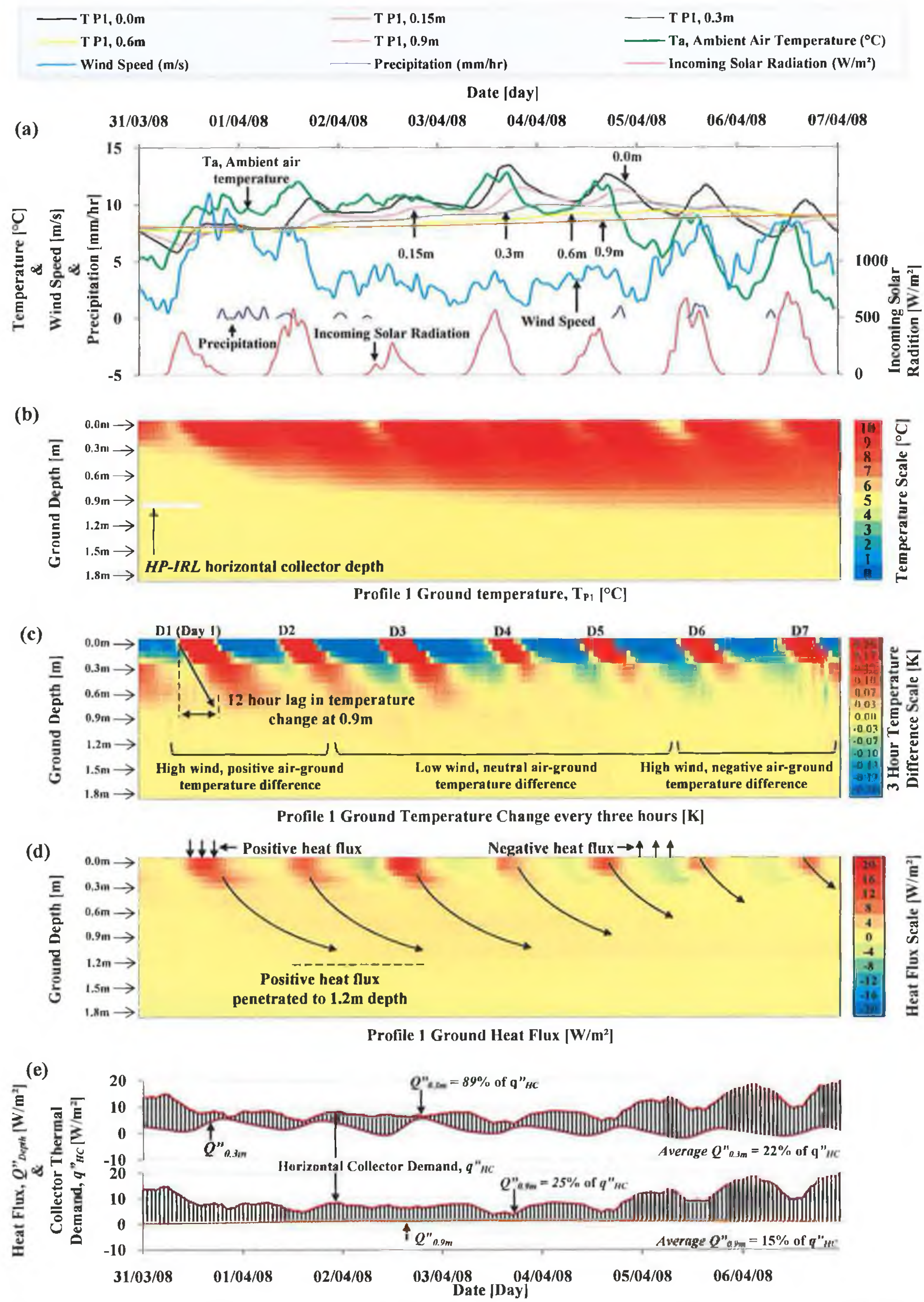


FIGURE 4.12 FIRST WEEK IN APRIL, 2008, HOURLY AVERAGED (A) WEATHER PARAMETERS AND PROFILE 1 GROUND TEMPERATURE, (B) 3-D TRANSIENT TEMPERATURE MAP OF PROFILE 1, (C) 3-D TRANSIENT TEMPERATURE CHANGE MAP SHOWING THE RATE OF GROUND TEMPERATURE CHANGE EVERY THREE HOURS AT PROFILE 1, (D) TRANSIENT GROUND HEAT FLUX MAP AND (E) GROUND HEAT FLUX IN COMPARISON TO THEORETICAL HORIZONTAL COLLECTOR THERMAL DEMAND.

Reflecting the higher ground temperatures near the surface, shown in Figure 4.12(b), Figure 4.13 highlights the higher energy content available at a depth of 0.3m ($Q_{0.0\text{-}0.6m}$) compared with 0.9m depth ($Q_{0.6\text{-}1.2m}$). The projected horizontal collector demand, Q'_{HC}, for the same domestic dwelling as in Figure 4.11 is also presented in Figure 4.13(a).

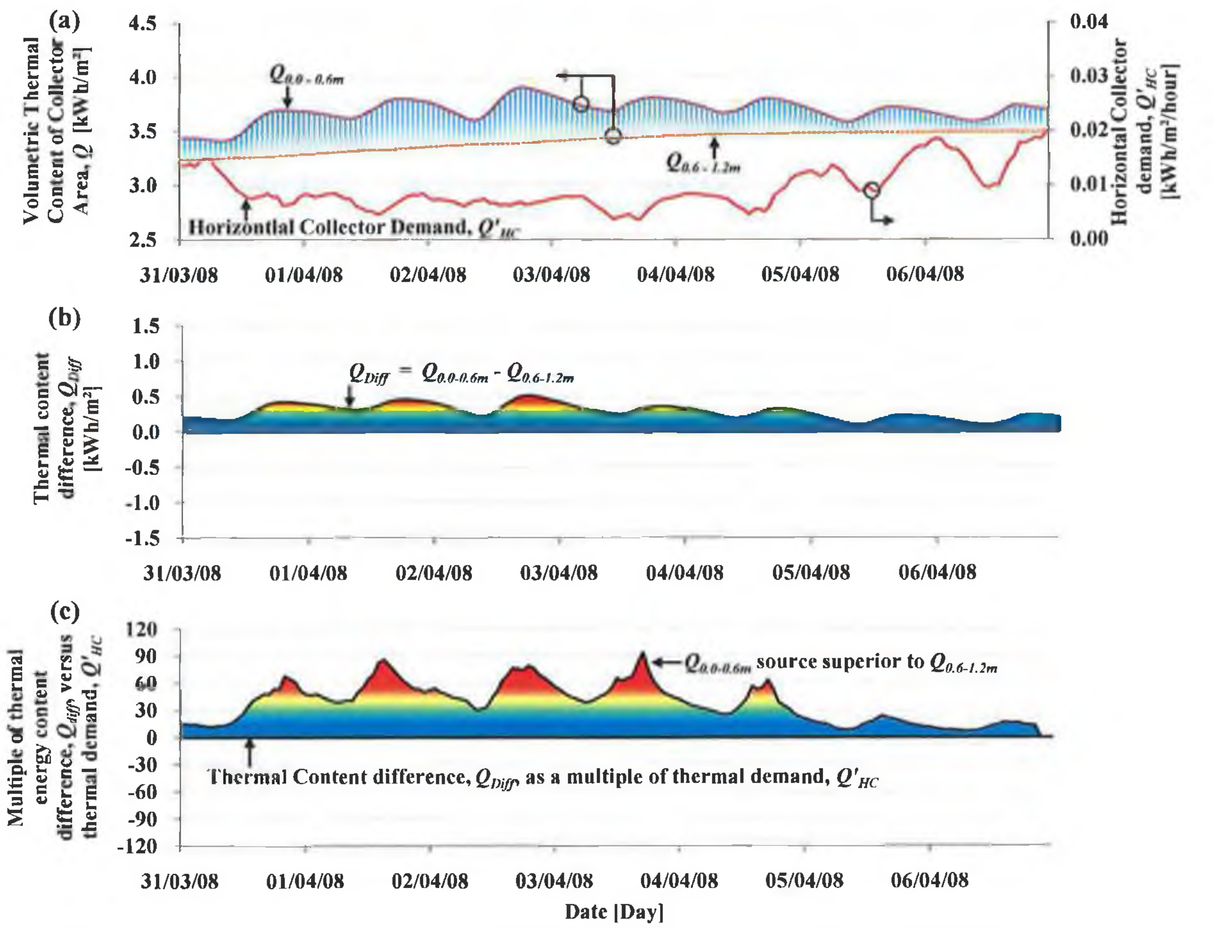


FIGURE 4.13 (A) AVAILABLE THERMAL ENERGY AT 0.3M AND 0.9M DEPTHS VERSUS THEORETICAL COLLECTOR THERMAL DEMAND, (B) VOLUMETRIC THERMAL ENERGY DIFFERENCE BETWEEN $Q_{0.0\text{-}0.6M}$ AND $Q_{0.6\text{-}1.2M}$ AND (C) VOLUMETRIC THERMAL ENERGY DIFFERENCE BETWEEN $Q_{0.0\text{-}0.6M}$ AND $Q_{0.6\text{-}1.2M}$ AS A MULTIPLE OF HORIZONTAL COLLECTOR THERMAL DEMAND, Q'_{HC}.

The volumetric energy content available within a 0.6m thick layer, centred at a depth of 0.3m ($Q_{0.0m\text{-}0.6m}$) and a 0.6m thick sub-layer centred at 0.9m ($Q_{0.6\text{-}1.2m}$) is presented in Figures 4.13(a) and (b). This confirms that during the springtime that there can be potentially as much as 15% more energy available at the 0.3m collector depth than at the deeper 0.9m depth. In terms of thermal provision, Figure 4.13(c) illustrates the ground thermal energy content difference as a multiple of the projected dwelling demand, Q'_{HC}, and shows that the shallower 0.3m layer has a thermal supply that is up to 100 times that of the deeper 0.9m layer. Therefore it clearly evident from Figure 4.13(c) that a higher temperature thermal source lay at the shallower depth of 0.3m during this part of the spring heating season.

Figure 4.14 presents a comparison of the thermal energy at both collector depths of 0.3m and 0.9m for the year 2008 based on daily averaged ground temperatures, illustrating the seasonal variation in thermal capacity at both potential horizontal collector depths of 0.3m and 0.9m.

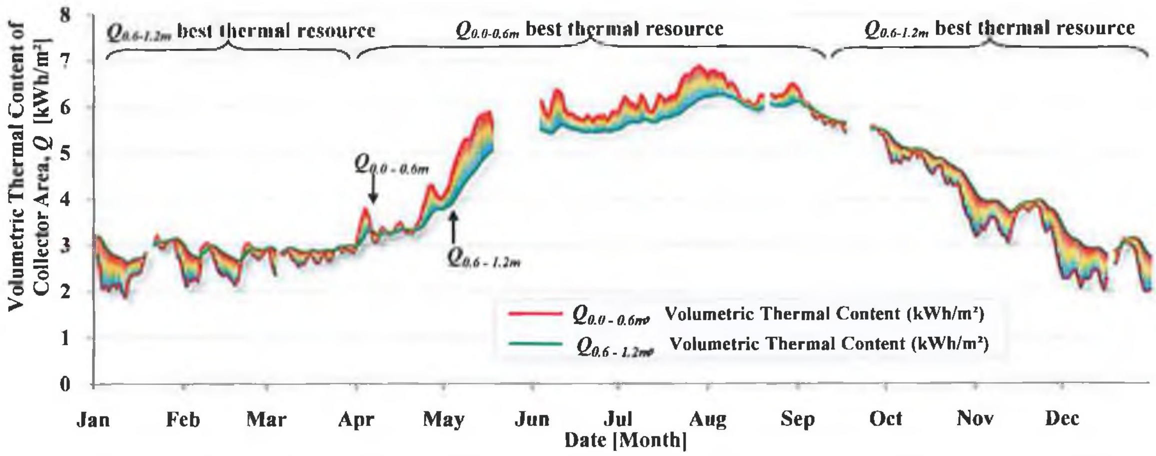


FIGURE 4.14 THERMAL ENERGY CONTENT OF GROUND AT 0.3M AND 0.9M DEPTHS FOR 2008.

Figure 4.15 illustrates the daily change in ground volumetric thermal energy content to a depth of 2m at Profile 1 over 2008.

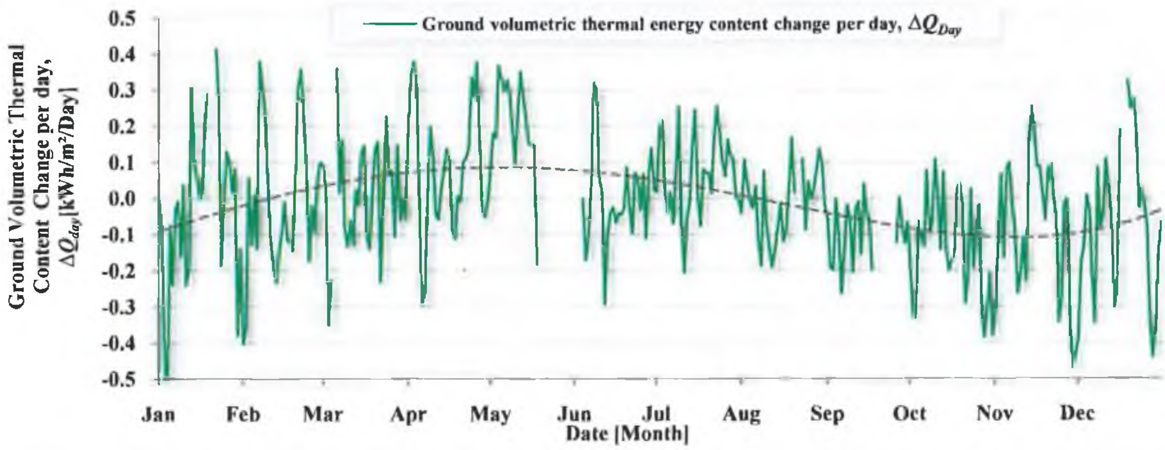


FIGURE 4.15 TOTAL CHANGE IN GROUND VOLUMETRIC THERMAL ENERGY CONTENT AT PROFILE 1 PER DAY OVER 2008.

While it is evident from Figure 4.15 that the ground thermal energy provision increases and decreases with the seasons, there is a substantial variation in the ground thermal content from day to day and potential exists to operate a suitable collector to take advantage of the peaks and use the thermal storage to avoid the troughs. This concept is explored further in Section 5.2.

4.1.3 SUMMARY

This section presented the weather induced fluctuation of the ground temperature and moisture content to a depth of 1.8m outside of the collector region, between 2007 and 2009. This not only allowed external ambient air and ground surface layer temperatures to be compared to that in other countries, but also enabled a suite of novel graphical analytical techniques to be developed to assess the impact of both climate and specific weather events on the ground's thermal capacity. These showed that:

- Maritime climate clearly generates higher average ground temperatures, with Galway's mean temperature being 4°C higher than Ottawa, located 8° further south. The minimum winter soil temperature at 0.9m depth in February was also higher in Galway, only dropping to +6°C (on average) over the three years monitoring in *HP-IRL* versus +1°C in Ottawa.
- Cold periods of 4°C below the seasonal norm occurred every 3 days in winter, and lasts on average for 1 day over the three years.
- Monthly average ground temperatures are predictable to an average accuracy of ±0.3°C and that seasonal effects are negligible below 15m depth, but unpredictable hourly averages may fluctuate by ±2.5°C from the monthly average at 0.9m. While seasonal effects are predictable based on monthly average ground temperatures, the impact of weather events are not. Measurements have shown that weather events of two weeks duration can affect ground temperature to 0.9m depth, and longer duration weather events can influence ground temperature to a depth of 1.5m and more, below the standard collector depths.
- Moisture: ground dries out from surface layer, where Figures 4.8 and 4.9 show that there is a moisture deficit on the surface layer at 300mm depth, with an average 5% higher moisture content at 900mm depth.
- Graphical technique showed that the combination of climate and ground temperature measurement with three dimensional graphical analysis techniques offer the sensitivity to not only resolve the impact of climate on ground temperature but also the impact of specific collector design, soil type and ground cover type, as well as operational parameters that enable improved source side management informed by ground temperature feedback sensors.

This analysis showed that the ground's surface layer (<1m deep) is being constantly thermally charged and discharged by a combination of; solar radiation (positive), air-ground surface temperature difference (positive/negative), which is amplified by both wind speed and rainfall. In winter-time ambient air temperature, wind speed and rainfall levels dominate to 2m as illustrated in Figure 4.10. In spring-time, solar radiation levels dominate ambient air temperatures and ground temperatures, the effects of which penetrate to a depth of 1m as illustrated in Figure 4.12.

The following climate sensitive parameters were recognised as influential in horizontal collector design:

- The ground temperature lag is proportional with depth, which is positive for deep collectors in autumn and winter, but negative in spring and summer
- Sunny spring days can generate a significant temperature impulse to a depth of 300mm that could potentially be used boost collector performance
- Link between the localised weather events, both geographically and in time, demand feedback sensors to monitor the status of the source

The latter is significant, as it seeks to establish the degree by which, in both time and depth, the Maritime climate influences parameters such as ground temperature and moisture content that impact on heat pump collector performance. It therefore queries which climate parameters or combination of parameters, such as sunshine, ambient air temperature, wind speed or rainfall have the greatest positive or negative impact.

4.2 GSHP$_{HC}$ TEST PROGRAM (2007 – 2009)

This section presents selected data from the 293 days that the horizontal collector operated during this three year long *HP-IRL* study. It identifies the rationale for each of the nine individual tests conducted and presents detailed results and key findings.

4.2.1 GSHP$_{HC}$ TEST PROGRAM RATIONALE

The goal of the comprehensive test program was to identify heat pump performance during different seasons and applications. Hence the test program reflected a wide range of demands (10% - 100% duty) and test durations (1 – 69 days). The horizontal collector was operated and monitored for the nine individual test periods identified as "HC#" in Figure 4.16 and Table 4.2. Figure 4.16 plots ambient air temperature and ground temperature at the collector depth along with the collector fluid return temperature and heat pump duty.

The operational duty was dictated by the ambient air temperature, but for the purpose of analysis and discussion it was categorised as follows:

- Low demand: 10-40% (5 months – from May through September, allowing for domestic hot water demand in summer)
- Moderate demand: 40-70% (4 months – October, November, March and April)
- Intensive demand: 70-100% (3 months – December, January and February)

The test duration is characterised as follows:

- Short-term: Period of less than one week
- Medium-term: Periods between one week and one month
- Long-term: Periods longer than one month

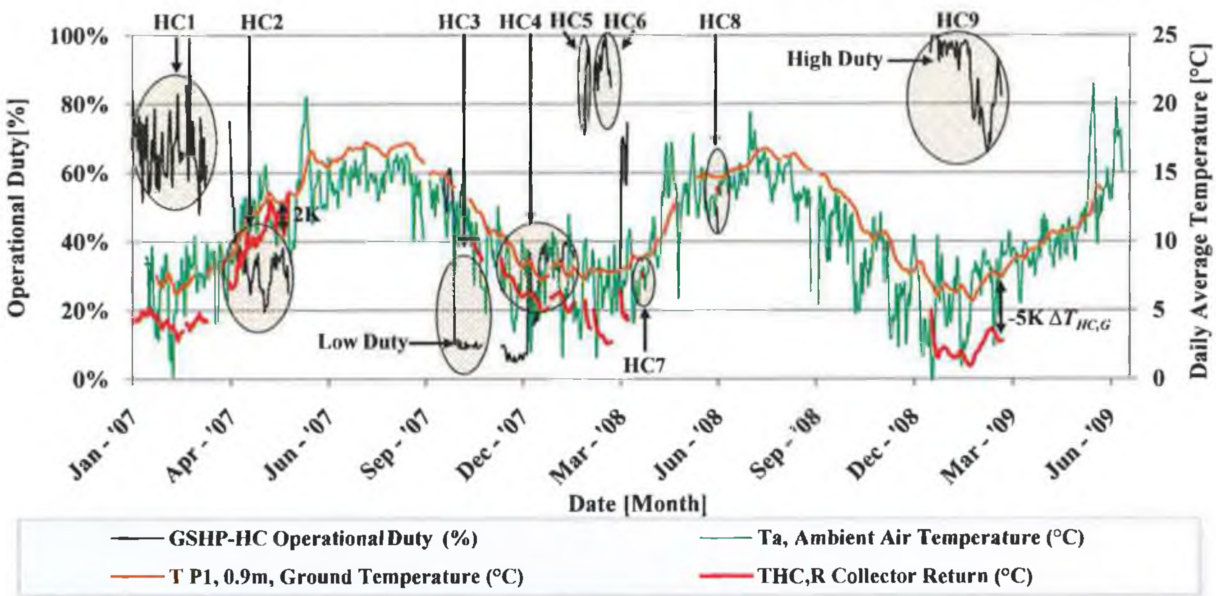


FIGURE 4.16 TIMING OF THE NINE $GSHP_{HC}$ TEST PERIODS (2007-2009) AND CORRESPONDING DUTY.

Table 4.2 presents the demand, duration and a brief description of each test.

TABLE 4.2 CHARACTERISTICS OF NINE $GSHP_{HC}$ TEST PERIODS (2007-2009)

Test Period	Demand	Term	Duration	Application	Description
HC1	Moderate	Long	69 days	Domestic/Commercial	First IiBC heating season observational period with moderate thermal extraction rates.
HC2	Low	Long	55 days	Domestic	Prolonged steady state low level thermal extraction, indicative of autumn/spring time domestic dwelling utilisation.
HC3	Moderate	Medium	11 days	Domestic	Fixed daily extract and recovery periods, indicative of domestic dwelling utilisation.
HC4	Low	Long	68 days	Domestic	Comparative heat pump operation period with the $GSHP_{HC}$ and $GSHP_{VC}$ in simultaneous operation.
HC5	Intensive	Short	6 days	Domestic/Commercial	Steady-state thermal extraction and subsequent recovery period, indicative of extreme utilisation.
HC6	Intensive	Medium	16 days	Domestic/Commercial	Steady-state thermal extraction and subsequent recovery period, indicative of extreme utilisation.
HC7	Low	Short	24 hours	Domestic	Recording localised collector profile thermal extraction and recovery temperature gradients.
HC8	Moderate	Short	15 hours	Domestic	Recording localised collector profile thermal extraction and recovery temperature gradients.
HC9	Intensive	Long	66 days	Commercial	Prolonged steady-state intensive thermal extraction, indicative of peak winter utilisation (commercial application).

Some of the expected initial observations from Figure 4.16 are:

- The ground temperature at the collector depth follows the changing ambient air temperature, dampening in amplitude (with depth) and increasing lag time
- The heat pump operational duty is proportional to ambient air temperature (see HC1, HC2 and HC9 in Figure 4.16)
- The drawdown on the collector temperature ($\Delta T_{HC,G}$) is a function of the heat pump operational duty (see HC1 through HC4)

The ground temperature drawdown ($\Delta T_{HC,G}$) is the difference between the horizontal collector fluid return temperature ($T_{HC,R}$) and the farfield temperature ($T_{HC,\infty}$) at collector depth ($\Delta T_{HC,G} = T_{HC,R} - T_{HC,\infty}$). The farfield temperature ($T_{HC,\infty}$) was recorded at a depth of 0.9m in the reference Profile 1 ($T_{P1,0.9m}$).

4.2.2 Initial findings

Over the course of the nine test program the horizontal collector operated for 293 days between 2007 to 2009, delivering 69,514 kWh of energy (250 GJ), which is equivalent to five years of space heating for a domestic dwelling (12,000 – 15,000kWh/annum). The energy extracted by the collector generated an overall drawdown of -3.5K on the ground source farfield temperature and the heat pump delivered an average heat pump sink temperature of +49.1°C. Table 4.3 summarises the results recorded within each test period.

Table 4.3 Summary of key $GSHP_{HC}$ test period results (2007-2009)

Test #	Dates	Days	Operational time per hour (Total time in operation)	Total thermal extraction (kWh)	Average collector area extract rate	Average collector pipe extract rate	Coefficient Of Performance*, COP [-]
HC1	01/01/07 – 10/03/07	69	69% (55%)	10,193 kWh	18.1 W/m²	5.2 W/m	2.8 (3.1)
HC2	31/03/07 – 24/05/07	55	34% (32%)	5,242 kWh	10.1 W/m²	2.9 W/m	3.0 (3.3)
HC3	08/11/07 – 19/11/07	11	59% (45%)	1,342 kWh	15.0 W/m²	4.3 W/m	3.1 (3.4)
HC4	07/12/07 – 12/02/08	68	33%	5,943 kWh	8.8 W/m²	2.5 W/m	2.8 (3.1)
HC5	22/02/08 – 27/02/08	6	93%	1,239 kWh	23.6 W/m²	6.8 W/m	2.8 (3.0)
HC6	01/03/08 – 17/03/08	16	93%	2,811 kWh	22.4 W/m²	6.4 W/m	2.7 (2.9)
HC7	14/04/08 – 15/04/08	1	35%	107 kWh	10.4 W/m²	3.0 W/m	3.3 (3.6)
HC8	23/06/08 – 24/06/08	1	48%	105 kWh	16.2 W/m²	4.6 W/m	3.4 (3.7)
HC9	05/01/09 – 11/03/09	66	89%	14,164 kWh	21.1 W/m²	6.0 W/m	2.7 (2.9)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The COP improved proportionally with a reduction in the overall heat pump temperature lift (ΔT_{HP}), which is the difference between the source temperature ($T_{HC,R}$) and the sink temperature ($T_{HP,F}$). This value varied from 25K to 49K over the test program and COP varied between 2.7 and 3.5. The overall average SPF was 2.90.

In order to evaluate the heat pump COP against temperature lift (ΔT_{HP}) and the *HP-IRL* facility the test results from *HP-IRL* were compared in Figure 4.17 with results published by the European test centre *Arsenal Research* as per EN-14511 for the same *Solterra* heat pump using fluid input temperatures ($T_{HC,R}$) of -5°C, 0°C and +5°C. These input temperatures are indicative of those experienced in colder continental climates and are lower than the equivalent in Maritime climate regions. Indeed, the literature values the heating season range of ground temperatures within the 0°C and +15°C range (Zogou and Stamatelos, 1998) and this can be directly compared with the Maritime climate ground temperature range of +5°C and +15°C shown in Figure 4.1.

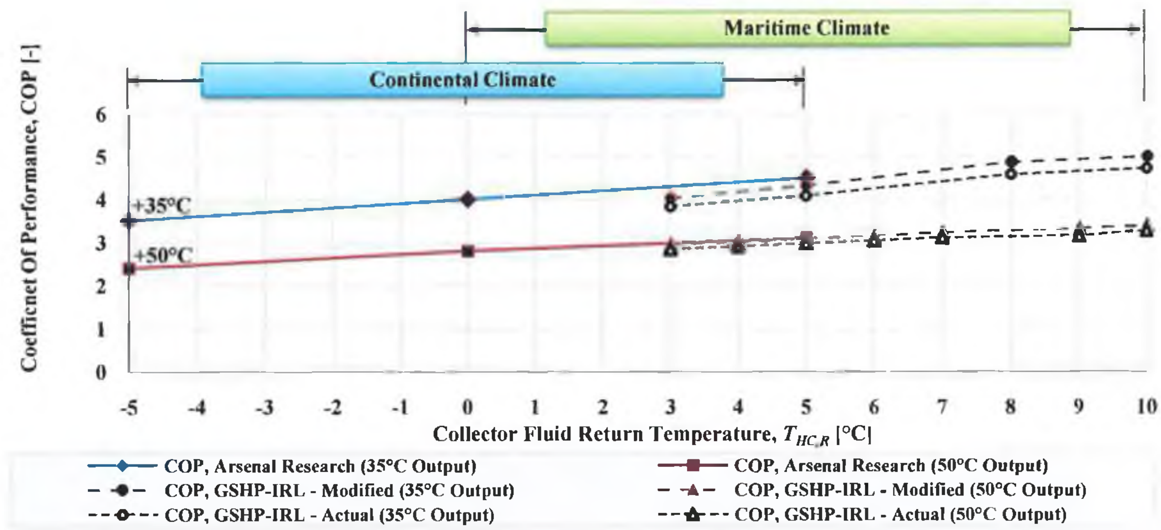


FIGURE 4.17 COMPARISON OF HEAT PUMP PERFORMANCE AS MEASURED USING INTERNATIONAL STANDARD EN-14511 AND DATA FROM *HP-IRL* STUDY.

The *HP-IRL* test result show both recorded and modified COP's. The modified COP take into account the electrical power consumption of the compressor and the pumping power required to overcome resistance losses in the evaporator as per EN-14511 and is summarised in Table 4.4. Whereas the *HP-IRL* recorded results account of the collector pumping power, and is a more accurate reflection of actual installed performance. The modified results for sink temperatures of +35°C and +50°C were within 4% and 0.3% respectively of those presented by Arsenal Research, adding credibility to the accuracy of the *HP-IRL* facility. Thus, the heat pump performance test results compared within an average 2.1% of the *Solterra* heat pump performance results carried out by *Arsenal Research* as per test standard EN 14511.

While both sets of data compared well, it was noted that standard EN-14511 only calls for a portion (pressure loss across the evaporator) of the collector side fluid pumping power to be included in the COP calculation. The EN-14511 standard only includes the power required to overcome the resistance of the evaporator plate heat exchanger, averaging at in this instance at 100W, but the *HP-IRL* horizontal collector consumed close to the 440W power rating. Studies have shown that collector fluid pumping power range betweens 14W and 40W per

1kW of heat pump capacity (Kavanaugh and Rafferty, 1997), and the *HP-IRL* study falls within this range with a pumping requirement of 29W/kW delivered. Furthermore, in estimating the thermal extraction from the collector, it must be noted that heat is added by the circulation pump to the brine solution due to frictional resistance losses in the collector pipe and across the evaporative heat exchanger. This contribution has not been allowed for in the literature.

TABLE 4.4 IMPACT OF INCLUDING PUMPING POWER IN HEAT PUMP COP EVALUATION

Test	Modified COP, *HP-IRL* (as per EN-14511)	COP, Arsenal Research	% Difference
B5°C/W35°C	4.3	4.5	-4.0%
B5°C/W50°C	3.1	3.1	-0.3%

Test	COP, *HP-IRL* (+ All Pumping Power)	COP, Arsenal Research	% Difference
B5°C/W35°C	4.1	4.5	-9.1%
B5°C/W50°C	3	3.1	-3.9%

Therefore, when collector pumping power, not included in EN 14511 compliant results, are included the average difference in COP against the standard test results increase to 6.5%. The percentage differences between the +35°C and +50°C outputs reflect the fact that the collector pumping power remains the same for both outputs, but it represents a larger proportion of the total electrical consumption at an output of +35°C and thus has a greater impact on the disparity. Figure 4.18 illustrates the measured improvement in *HP-IRL* COP with collector fluid return temperature.

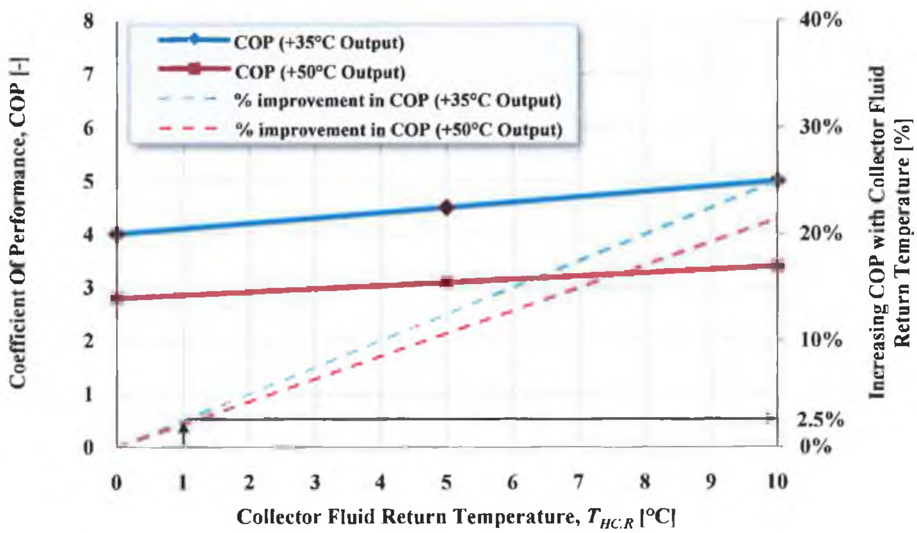


FIGURE 4.18 PERCENTAGE IMPROVEMENT IN *HP-IRL* COP WITH INCREASED COLLECTOR FLUID RETURN TEMPERATURE.

Significantly, Figure 4.18 highlights a 2.5% increase in COP with each 1°C increase in source temperature. This establishes the likely contribution that effective source side management could have on heat pump performance.

4.3 EXPERIMENTAL ANALYSIS

This section presents the key findings from the test program which highlights the impact of both collector duty and duration on the collector region, temperature, as well as the influence of climate and surface cover on heat pump performance.

4.3.1 DUTY CYCLE CHARACTERISTICS

A typical signature of a heat pump operational cycle on collector ground temperature is shown in Figure 4.19. The shaded area represents the extent by which the heat extraction lowers the ground temperature in the vicinity of the collector and this can be sub-divided into the drawdown, steady-state and recovery periods.

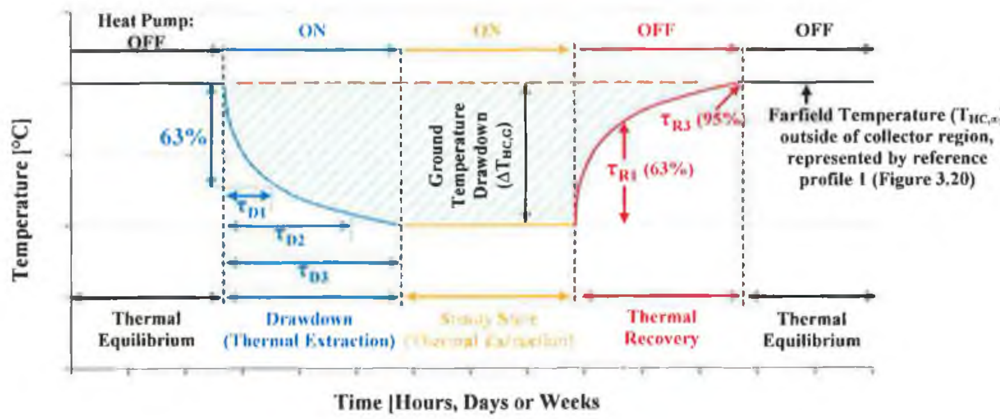


FIGURE 4.19 SCHEMATIC OF THERMAL EXTRACTION AND RECOVERY CHARACTERISTICS, TO DEFINE DRAWDOWN, STEADY-STATE AND RECOVERY PERIODS ASSOCIATED WITH HEAT PUMP COLLECTOR OPERATION.

The "*drawdown period*" is characterised by a lowering of the ground temperature adjacent to the collector represented by the collector return temperature ($T_{HC,R}$) relative to the farfield temperature at the same depth ($T_{HC,\infty}$). Drawdown commences when the heat pump is activated and was defined to end once the ground temperature within the collector region reached 95%, or three time constants ($3\tau = \tau_{D3}$), of its steady-state value. The time constant, τ, represents the time taken to reach 63% of its final value. The third (τ_{D3}) was used as a convenient way of consistently identifying the end of the drawdown period across a number of tests, even though it equates to the time that the collector fluid return temperature only achieves 95% of the eventual steady state ground temperature drawdown ($\Delta T_{HC,G}$). This "*steady-state*" period refers to that portion of heat pump operation where the $\Delta T_{HC,G}$ remains relatively constant. The "*recovery*" period refers to the time taken for the ground temperature within the collector region to reach 95%, or three time constants ($3\tau = \tau_{R3}$), of its fully recovered state relative to the farfield temperature at the same depth ($T_{HC,\infty}$).

The analysis approach helps to illustrate how:

- drawdown rates and steady-state ground temperature difference between inside and outside the collector region are influenced by heat pump duty
- recovery rates are influenced by climate
- the Thermally Affected Zone (TAZ) surrounding the horizontal collector piping develops
- climate impacts on the upper surface temperature field, both in extent and depth
- weather conditions, duty and duration affect heat pump performance (COP)

The following sub-sections expand on the parameters identified in Figure 4.19.

4.3.1.1 DRAWDOWN

Recalling that the recommended ground temperature drawdown ($\Delta T_{HC,G}$) identified in Section 2.3.3.2 should not exceed -12K for base load conditions and -18K for peak load conditions (VDI 4640 / Part 2, 2001; Reub and Sanner, 2001) and this internationally recognised drawdown is compared against the measured data of the *HP-IRL* study.

As the ground temperature drawdown is a measure of both the collector and heat pump performance, Figure 4.20 presents an initial illustration of the transient ground temperature drawdown the moderate and intensive test periods HC4 and HC9.

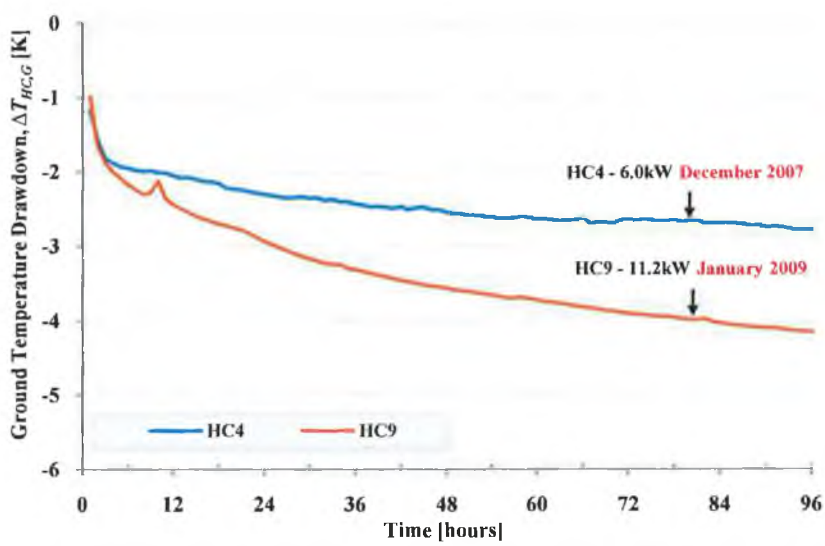


FIGURE 4.20 HOURLY AVERAGED GROUND TEMPERATURE DRAWDOWN FOR TEST PERIODS HC4 AND HC9.

While the presentation of the profiles is complicated by the start condition, driven by both climate differences and previous ground temperatures Figure 4.20 illustrates the ground temperature drawdown variation with thermal extract rate of test periods HC4 and HC9. The intensive (11.2kW) test period HC9 generates a ground temperature drawdown of -4.2K after

90 hours, compared with the moderate (6kW) test period HC4 that generates a ground temperature drawdown of only -2.8K after 90 hours.

4.3.1.2 STEADY-STATE OPERATION

Illustrating the move from the drawdown period into steady-state operation, Figures 4.21 and 4.22 present the entire 17 and 66 day long HC6 and HC9 test periods, with subsequent collector region thermal rec

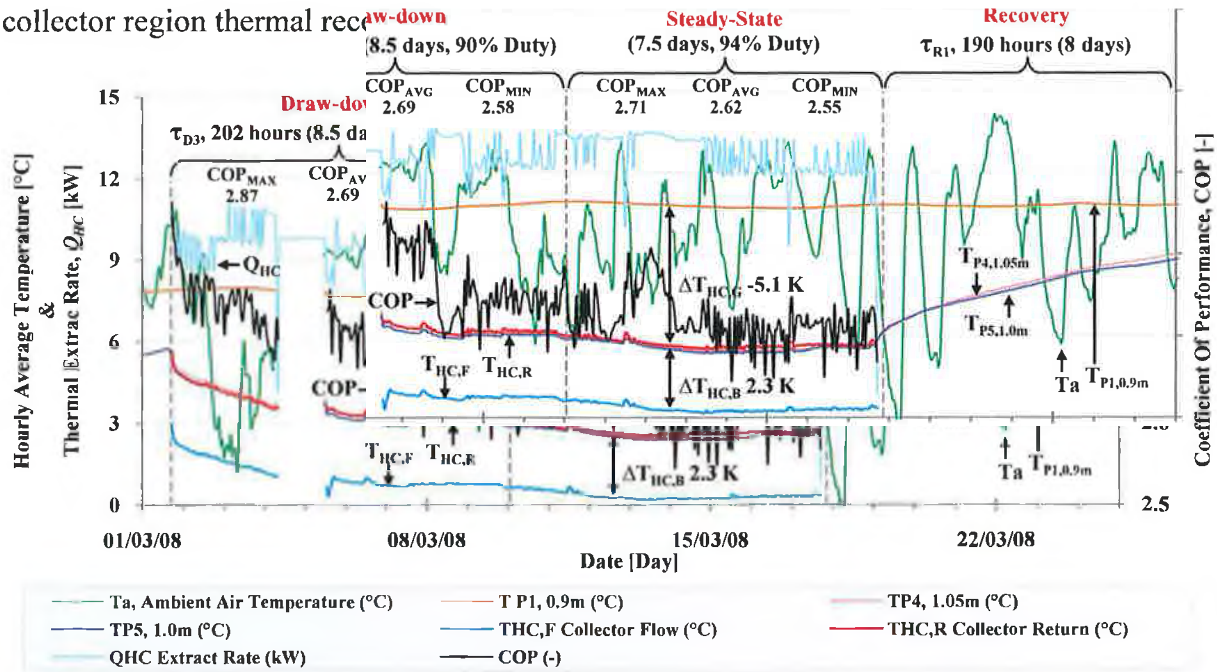


FIGURE 4.21 HOURLY AVERAGED THERMAL DRAWDOWN, STEADY-STATE AND RECOVERY FOR TEST PERIOD HC6.

This intensive test period, with an average collector thermal extraction of 9.8kW, reached *HP-IRL* defined "steady-state" conditions (τ_{D3}) after 8.5 days heat pump operation and was continued for 7.5 days. This test illustrates the robust thermal performance of the horizontal collector under consistently high and constant thermal extraction (9.8kW) over a medium-term (16 day) test duration. This steady-state test period produced the largest ground temperature drawdown, $\Delta T_{HC,G}$ over the 3-year long test program, recorded at -5.1K.

It should be noted that the heat pump did not deliver a variable output and the variation in Q_{HC} shown in Figure 4.21 is due to the variation in heat pump duty each hour. The heat pump's collector fluid flow ($T_{HC,F}$) and return ($T_{HC,R}$) temperatures are recorded every minute, and plotted as hourly average values when the heat pump is operational. The ambient air temperature (T_a) and ground temperature at the control profile ($T_{P1,0.9m}$) are recorded every five minutes and presented in Figure 4.21 as hourly averaged values.

Table 4.5 presents the primary test parameters for test period HC6 and Table 4.6 presents a summary of the secondary indicators of heat pump performance during HC6.

TABLE 4.5 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD HC6 (MEDIUM-TERM, INTENSIVE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{HC,R}$	Average $T_{HC,\infty}$	Total kWh Solar Thermal incoming	Total kWh extracted from HC
HC6	01/03/08 – 17/03/08	16	93%	9.8 kW	+3.1°C	+7.8°C	13,428	3,709

TABLE 4.6 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD HC6

Test #	Fraction of thermal energy extracted over incoming solar energy	Average $\Delta T_{HC,G}$	Average collector pipe extract rate [W/m]	Average $GSHP_{HC}$ extract rate per m² of collector	Coefficient Of Performance, COP [-]
HC6	0.28	- 4.7 K	6.4 W/m	23.3 W/m²	2.65 (2.91)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period HC6 was +49.8°C, with a temperature lift (ΔT_{HP}) of +46.7 K, delivering a COP_{AVG} of 2.7. The average $\Delta T_{HC,G}$ for test period HC6 was -4.7K. If a steady-state $\Delta T_{HC,G}$ of -12K defined by the VDI 4640 standard (VDI 4640 / Part 2, 2001; Reub and Sanner, 2001) was imposed on the *HP-IRL* collector design, it is estimated that for the same output temperature of +49.8°C, the collector return temperature would fall from +3.1°C to -4.2°C, generating a temperature lift (ΔT_{HP}) of +54K, and the COP would be reduced from 2.7 to 2.2

As a test of collector endurance the $GSHP_{HC}$, under test period HC9 a prolonged intensive level thermal extraction over 66 days was performed from January, 2009. Steady-state (τ_{D3}) was reached after 6 days and a 8.7kW level of extraction was maintained for 60 days. The results of this extreme test are presented in Figure 4.22 and the results of which are summarised in Tables 4.7 and 4.8.

TABLE 4.7 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD HC9 (LONG-TERM, INTENSIVE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{HC,R}$	Average $T_{HC,\infty}$	Total kWh Solar Thermal incoming	Total kWh extracted from HC
HC9	05/01/09 – 11/03/09	66	89%	9.1 kW	+2.2°C	+6.7°C	31,683	14,164

TABLE 4.8 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS OBTAINED FROM TEST PERIOD HC9

Test #	Fraction of thermal energy extracted over incoming solar energy	Average $\Delta T_{HC,G}$	Average collector pipe extract rate [W/m]	Average $GSHP_{HC}$ extract rate per m² of collector	Coefficient Of Performance, COP [-]
HC9	0.45	- 4.5 K	6.0 W/m	21 W/m²	2.66 (2.92)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period HC9 was +49.1°C, with a temperature lift (ΔT_{HP}) of +46.9 K, delivering a COP_{AVG} of 2.7.

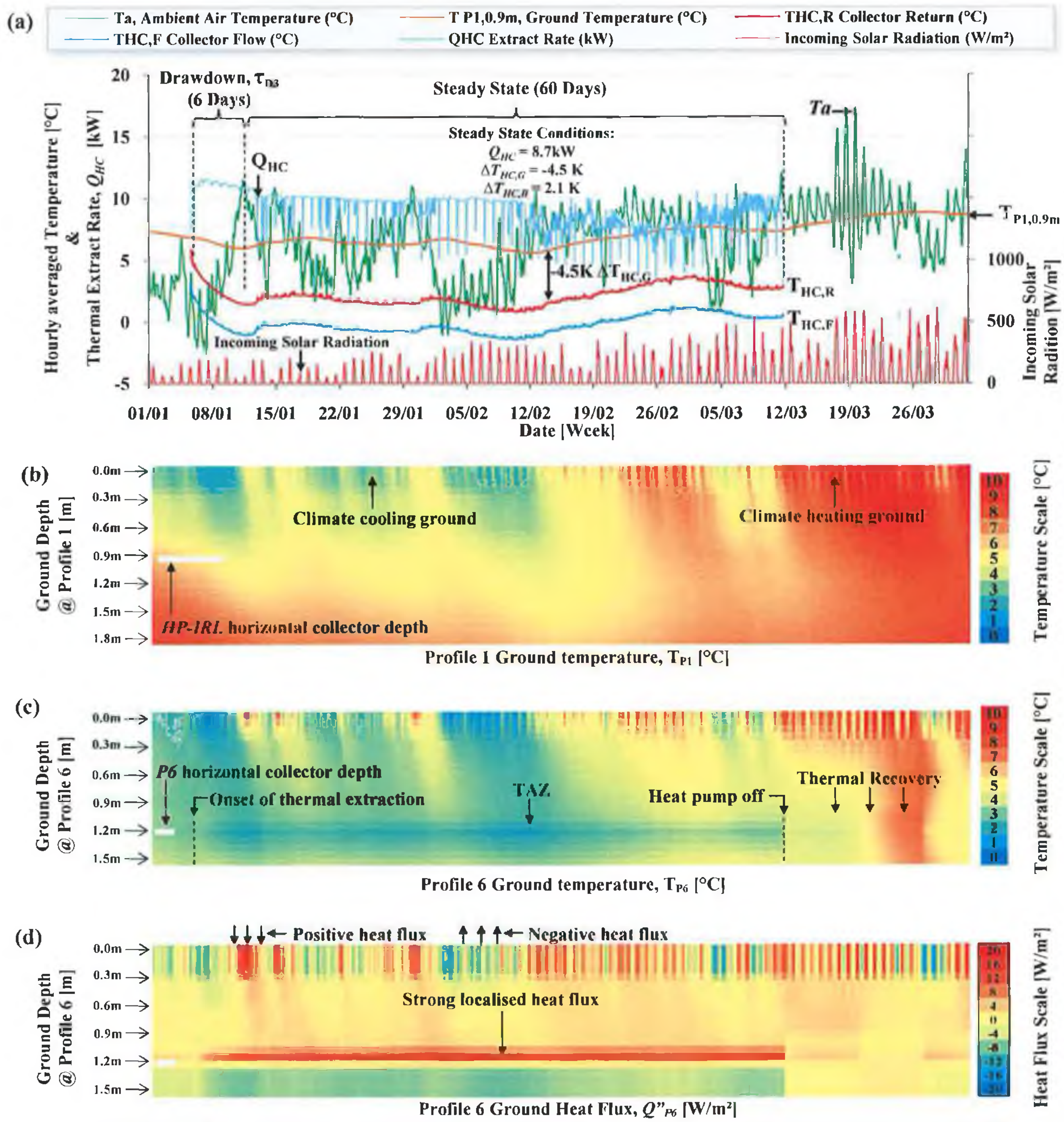


FIGURE 4.22 TEST PERIOD HC9, LONG-TERM GROUND THERMAL DRAWDOWN AND STEADY STATE EXTRACTION (A) CLIMATE AND GROUND TEMPERATURE, (B) 3-D TEMPERATURE MAP IN REFERENCE PROFILE 1, (C) 3-D TEMPERATURE MAP IN COLLECTOR REGION SHOWING TAZ AND (D) 3-D HEAT FLUX MAP WITH POSITIVE DOWNWARD HEAT FLUX IN RED, NEGATIVE UPWARD HEAT FLUX IN BLUE.

Figure 4.22(a) shows:

- It takes 6 days to achieve steady-state drawdown and is maintained thereafter while delivering 8.7kW over 60 days
- A steady-state ground temperature drawdown of -4.7K
- The collector fluid temperature rise is constant at 2.1K

Figure 4.22(b) shows the transient 3-D temperature map for the control Profile 1. This graph captures:

- The transition from winter to spring in mid-February when the cooling influence of the climate gives way to a warming influence in the upper ground layers.
- The influence of climate to 1.5m, but with diminishing effect

Figure 4.22(c) shows the first transient 3-D temperature map within the collector region at Profile 6. Notable features include:

- The localised cooling effect of the collector as it draws heat from above and below
- The formation of a TAZ about the collector compared to the 3-D temperature map of Profile 1
- How the TAZ contracts once the collector is turned off

In Figure 4.22(d) a downward heat flux is indicated as positive, denoted as red, an upward heat flux is negative and blue. Observations from Figure 4.22 include:

- Intense heat flux close to the collector pipe at approximately 12W/m² from above and below the collector
- The recovery of the ground thermal energy in the collector region is driven by climatic influences in mid-March, as opposed to geothermal gradients below the collector.

From the nine tests conducted, test periods HC3, HC4, HC6 and HC9 developed steady-state ground temperature drawdown conditions and details of which are presented in Table 4.9.

TABLE 4.9 DRAWDOWN, STEADY-STATE THERMAL EXTRACTION AND RECOVERY FOR HC3, 4, 6 AND 9

	Draw-down					Steady-state								Recovery	
	Q_{HC} (kW)	τ_{D3} (hours)	$T_{HC,\infty}$ (°C)	$T_{HP,F}$ (°C)	COP_{AVG} (-)	Steady-state duration after τ_{D3} (hours)	Q_{HC} (kW)	$\Delta T_{HC,G}$ (K)	$T_{HC,m}$ (°C)	$T_{HC,R}$ (°C)	$T_{HP,F}$ (°C)	ΔT_{HP} (K)	COP_{AVG} (-)	τ_R – hours	$T_{HC,\infty}$ (°C)
HC3	NA	NA	NA	NA	NA	50	6.2	-3.2	+12.0	+8.8	+49.5	40.7	2.98	τ_{R1} - 163	+11.3
HC4	6.0	60	+10.4	+49.4	3.03	1550	3.8	-2.3	+8.2	+6.0	+48.9	42.9	2.82	56% - 246	+7.5
HC6	9.6	202	+7.8	+49.6	2.69	182	9.6	-5.1	+7.8	+2.7	+50.0	47.3	2.62	τ_{R1} - 190	+7.8
HC9	10.3	380	+6.5	+47.0	2.76	1181	8.7	-4.7	+6.7	+2.1	+49.8	47.7	2.62	τ_{R3} - 846	+8.8

Note: Steady-state conditions are taken at a time equal to three time constants $\tau_{D3}=3\tau$, or 95% of final steady-state condition.

The results of the steady-state thermal extraction are illustrated in Figure 4.23.

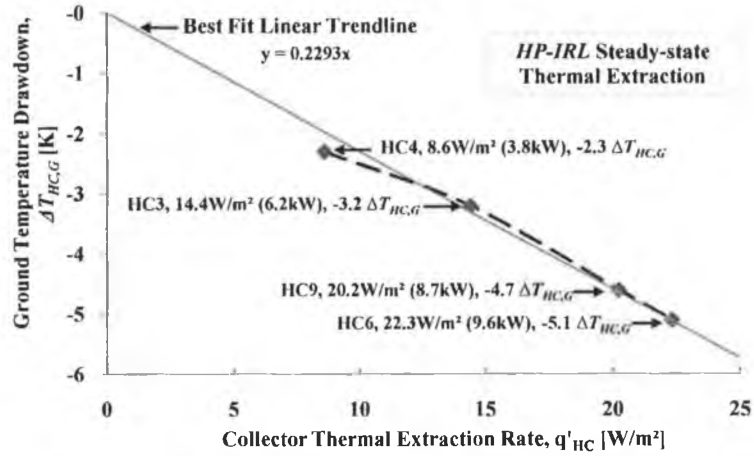


FIGURE 4.23 GROUND TEMPERATURE DRAW-DOWN ($\Delta T_{HC,G}$) RECORDED UNDER STEADY-STATE THERMAL EXTRACTION CONDITIONS DURING TEST PERIODS HC3, HC4, HC6 AND HC9.

The results indicate under steady-state thermal extraction the ground temperature drawdown is a relatively stable linear function of collector thermal extraction in watts per meter squared of collector area, given by 0.2293 (extraction).

As indicated in Table 2.3 the horizontal collector design guidelines do not give any indication as to the performance of the horizontal collector in operation and are missing a parameter that can allow collector design to take into account the collector performance in relation to heat pump duty. As a means of characterising performance of horizontal collector designs a new parameter is required that combines both the thermal extraction rate (W/m²) and ground temperature drawdown ($\Delta T_{HC,G}$). This parameter is desirable as an indicator of a collector design's performance both pre and post installation and also as a unified collector performance indicator for comparing the performance of horizontal collectors generally. This parameter is called the Collector Performance Indicator (*CPI*). This *CPI* parameter provides a new means of assessing the impact of climate driven duty cycle on the climate sensitive collector region. In doing so, it can also reflect the impact of collector design and sources side management. The *CPI* is characterised as follows.

$$CPI = \frac{\Delta T_{HC,G}}{q'_{HC}} \qquad \textit{Equation 4.4}$$

Using the results of the *HP-IRL* horizontal collector performance in Figure 4.23 the CPI for the highest steady-state thermal extraction recorded in test period HC6 is as follows:

$$CPI = \frac{\Delta T_{HC,G}}{q'_{HC}} = \frac{5.1}{22.3} = 0.23\,K/(W/m^2)$$

For the *HP-IRL* horizontal collector design the *CPI* of 0.23K/(W/m²) indicates a ground temperature drawdown of 0.23K per unit demand (W) of a unit collector area (m²).

4.3.1.3 RECOVERY

The ground's rate of thermal recovery is a key component of the horizontal collector efficiency as it draws in energy from the surrounding ground. While the recovery of the collector region is driven from below the collector during the intensive heating months of December, January and February, during the moderate months of spring the recovery is increasingly driven by climatic parameters such as ambient air temperature and incoming solar radiation as indicated by Figure 4.22(d).

Figure 4.24 illustrates the thermal recovery of the collector region at the collector depth for test periods HC4 to HC9. The temperature recovery, $\Delta T_{HC,R}$, is the difference between the control profile temperature ($T_{P1,0.9m}$) and the collector temperature ($T_{P5,1.0m}$).

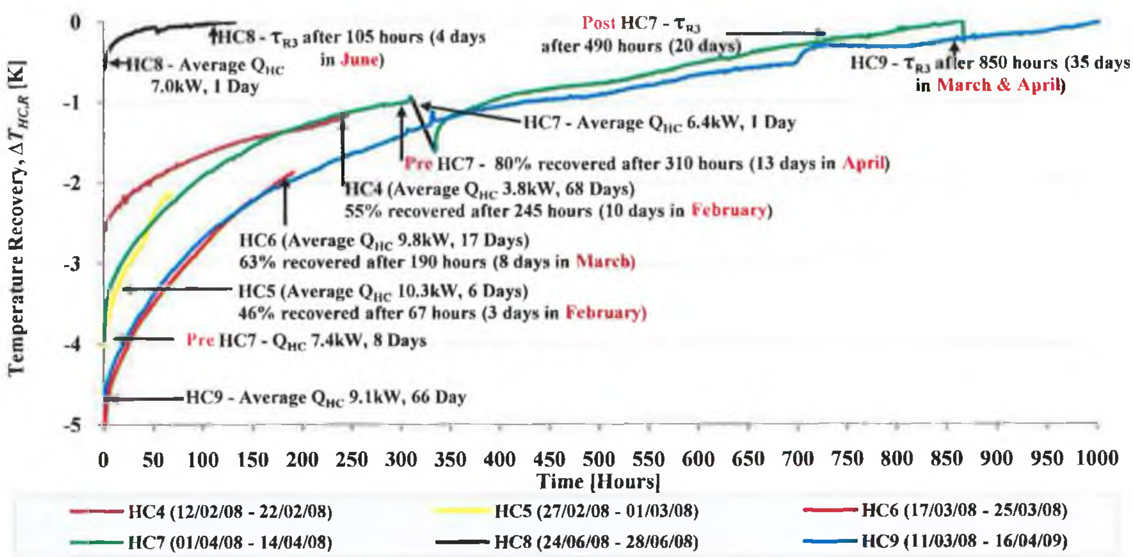


FIGURE 4.24 THERMAL RECOVERY OF THE COLLECTOR REGION WITHIN PROFILE 5 AFTER TEST PERIODS HC4 THROUGH HC9.

Due to the relatively stable ground thermal properties, the thermal recharging characteristics remain quite consistent throughout the heating season with variation occurring due to duration and rate of thermal extraction prior to recovery. High extract rates and over long durations have the effect of prolonging the time in which the ground reaches 95% recovery, τ_{R3}.

The ground surrounding the collector recharges from both above, from warm ambient air and solar radiation, and from below the collector, where the core temperature below 15m remains at a relatively stable +10.5°C. Depending on the time of the year, the proportion of thermal recharging from above and below varies. During the winter months the heat flux is predominantly from below.

4.3.2 IDENTIFYING COLLECTOR THERMALLY AFFECTED ZONE

The Thermally Affected Zone (TAZ) identified in Figure 4.22 refers to the ground surrounding the collector pipe whose temperature has been affected by thermal extraction. The TAZ expands radially from each horizontal collector pipe (Greene *et al.*, 2010). The extent or volume of this zone depends on both the thermal extraction rate and its duration (Greene *et al.*, 2010). The distance from the collector pipe to the outer edge of the TAZ is referred to as the farfield radius, r_f. As the expanding farfield radii of two parallel collector pipes converge the expansion of the TAZ becomes linear both towards the collector surface and below the collector, but remains radial along the outside edge of the horizontal collector. The distance from the collector pipe to the outer edge of the TAZ is referred to as the distance to farfield, D_f. Beyond D_f, the collector has no influence.

The TAZ has been explored in Figure 4.25 and contrasts the short-term (15 hours) moderate extraction rate (48%) HC8 test, with the short-term (6 day) intensive extraction rate (93%) HC5 test in Figure 4.26 and the long-term (66 day) intense extraction rate (89%) HC9 test shown in Figures 4.31 and 4.32.

The primary results of test periods HC5 and HC8 are presented in Table 4.10 and Table 4.6 presents a summary of the secondary indicators of heat pump performance during HC8.

TABLE 4.10 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD HC5, HC8

Test #	Dates	Days (Hours)	Test period Duty	Average Collector Extract Rate	Average $T_{HC,R}$	Average $T_{HC,*}$	Total kWh Solar Thermal incoming	Total kWh extracted from HC
HC5	22/02/08 – 27/02/08	6	93%	10.3 kW	+4.1°C	+7.8°C	3,736	1,239
HC8	23/06/08 – 24/06/08	1 (15)	48%	7.0 kW	+13.4°C	+14.4°C	204	105

TABLE 4.11 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD HC8 (SHORT-TERM, MODERATE)

Test #	Fraction of thermal energy extracted over incoming solar energy	Average $\Delta T_{HC,G}$	Average collector pipe extract rate [W/m]	Average $GSHP_{HC}$ extract rate per m² of collector	Coefficient Of Performance, COP_{AVG} [-]
HC8	0.51	- 1.0 K	4.6 W/m	16.2 W/m²	3.35 (3.68)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period HC8 was +48.4°C, with a temperature lift (ΔT_{HP}) of +35.0K, delivering a COP_{AVG} of 3.4. Figure 4.25(a) shows the steady 7kW thermal extraction from the horizontal collector maintaining a -1K difference between the ground farfield temperature ($T_{P1,0.9m}$) and the horizontal collector return fluid temperature ($T_{HC,R}$). This small $\Delta T_{HC,G}$ is due to the short 15 hour long thermal extraction period. Figure 4.25(b) represents the temperature difference between the control Profile 1 and with the collector at Profile 5, with five minute time intervals. Note there was no temperature difference between both profiles before the test commenced.

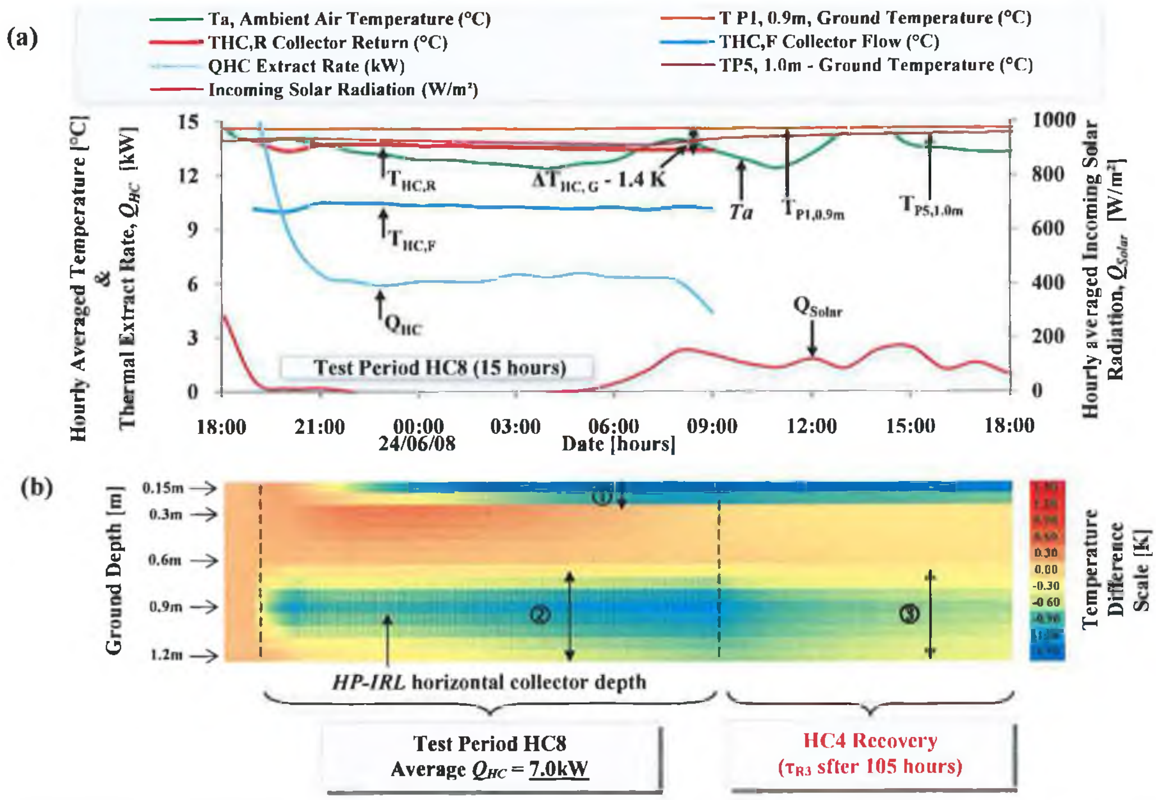


FIGURE 4.25 (A) SHORT-TERM, MODERATE THERMAL EXTRACTION FROM HORIZONTAL COLLECTOR AND (B) TAZ AROUND THE HORIZONTAL COLLECTOR DURING TEST HC8 ESTABLISHED BY SUBTRACTING GROUND TEMPERATURES AT PROFILE 1 FROM GROUND TEMPERATURES AT PROFILE 5 (P5-P1).

This novel illustration captures the growth of the TAZ around the collector pipe during the 15 hour thermal extraction and subsequent thermal recovery. Some observations from Figure 4.25(b) are:

① There is a larger surface cooling at Profile 5, the effects of which penetrate to a depth of 0.3m. This is attributed to the difference in surface covers, where P1 has a grass surface cover and P5 has a shrubbery surface cover and is insulated from the incoming solar radiation

② During thermal extraction, the TAZ grows to 0.3m above and below the horizontal collector

③ In recovery, the TAZ remains visible, but reduces in intensity and is fully recovered to τ_{R3} after 105 hours

Test period HC5 was employed to establish the short-term (six day) response of the collector to a near maximum thermal extraction rate during the peak heating season in February 2008. It was conducted over a period of six consecutive days with an intensive thermal extraction rate of 10.2 kW (93% duty) and the results of which are shown in Figure 4.26(b) along with 3-D transient temperature difference map, indicating the temperature difference between the reference profile 1 and with the collector region at profile 5. Table 4.12 presents a summary of the secondary indicators of horizontal collector and heat pump performance in test period HC5.

TABLE 4.12 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD HC5 (SHORT-TERM, INTENSIVE)

Test #	Fraction of thermal energy extracted over incoming solar energy	Average $\Delta T_{HC,G}$	Average collector pipe extract rate [W/m]	Average $GSHP_{HC}$ extract rate per m^2 of collector	Coefficient Of Performance, COP [-]
HC5	0.33	- 3.7 K	6.7 W/m	23.6 W/m²	2.78 (3.05)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) is similar to that for HC8, at +49°C, but the delivered COP_{AVG} dropped from 3.3 to 2.8. This can be explained by the higher temperature lift, ΔT_{HP}, of +44.9K brought about by the colder collector return temperature of +4.1°C, down from +13.4°C during summer operation in HC8. It is also notable that the intense extract rate (93%, or 10.3kW) generated a $\Delta T_{HC,G}$ of -3.7K between the collector fluid return temperature and the surrounding ground farfield temperature, as opposed to -1K at 7kW shown in Figure 4.25 for test period HC8.

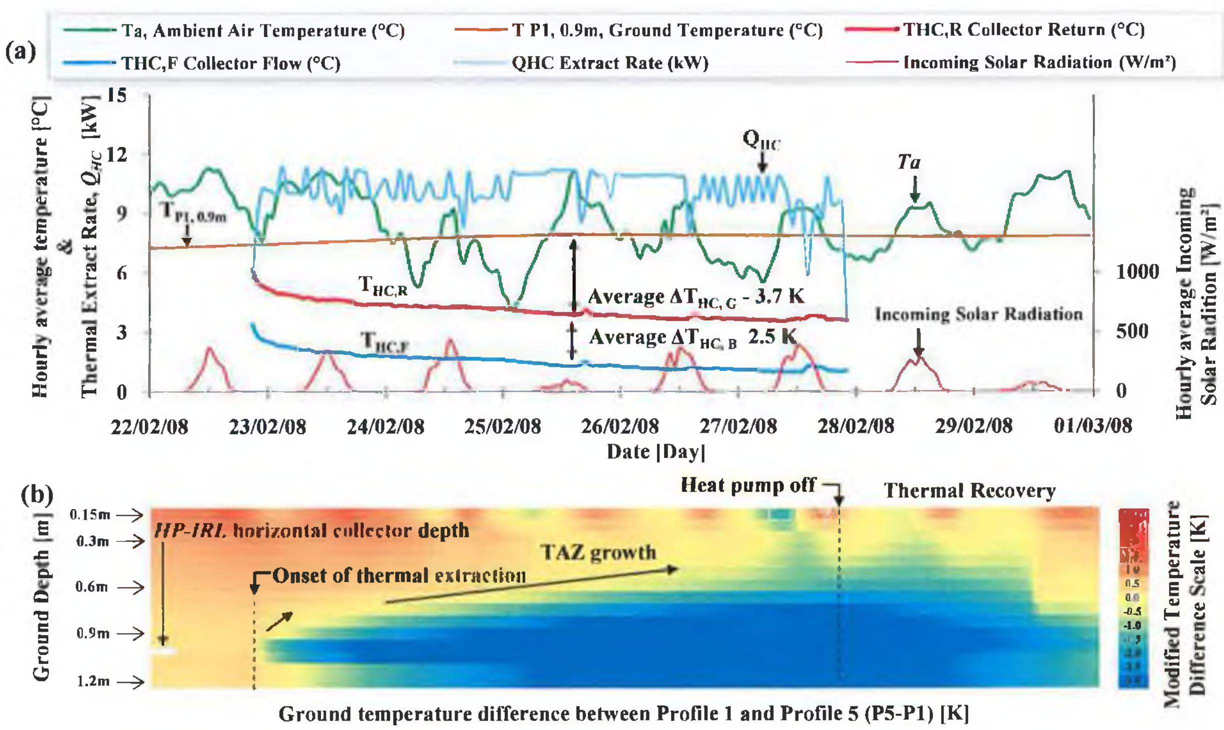


FIGURE 4.26 (A) SHORT-TERM, INTENSIVE THERMAL EXTRACTION (93%) DURING TEST PERIOD HC5 AND (B) 3-D TEMPERATURE DIFFERENCE MAP ILLUSTRATING THE TAZ AROUND THE HORIZONTAL COLLECTOR DURING TEST HC8, ESTABLISHED BY SUBTRACTING GROUND TEMPERATURES AT PROFILE 1 FROM GROUND TEMPERATURES AT PROFILE 5 (P5-P1).

What is noticeable from Figure 4.26(b) is the rapid initial expansion of the TAZ and a gradual increase thereafter, coming close to the ground surface after five days of intense thermal extraction. In a complementary analysis of the TAZ at various depths, Figure 4.27 further illustrates how the TAZ diminishes with vertical distance above the collector pipes.

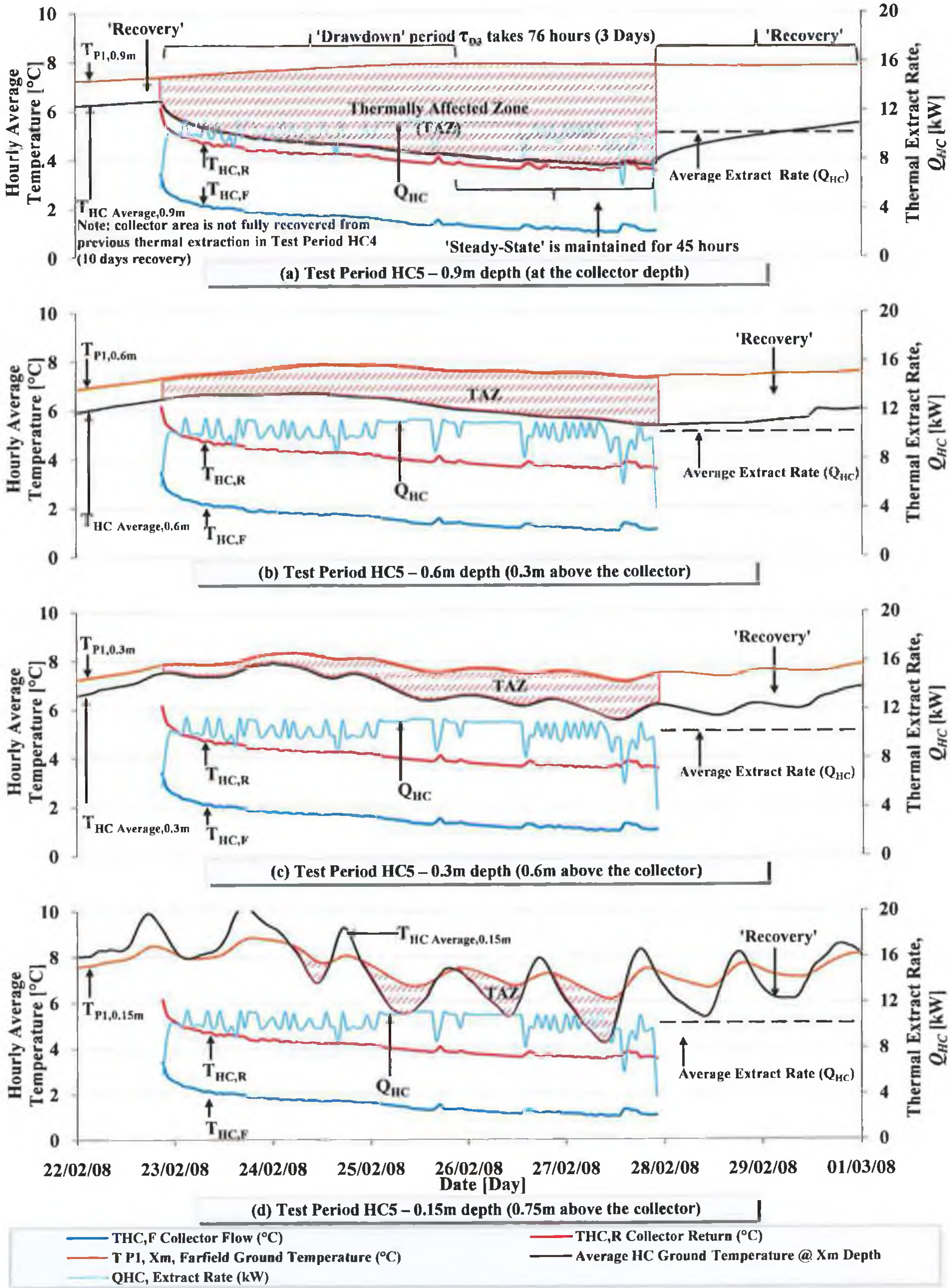


FIGURE 4.27 TEST PERIOD HC5 THERMAL DEGRADATION OF COLLECTOR AREA AT DEPTHS BETWEEN 0.15M AND 0.9M.

The collector was in recovery for a period of ten days prior to the commencement of test HC5 but it can be seen from the start of the test data in Figure 4.27 the ground within the collector region was not fully recovered, in comparison to the control profile, P1. The initial temperature difference at the start of the test period was equalised in the 3-D map in Figure 4.26(b) to allow illustration clarity to the TAZ expansion. Using a different approach Figure 4.28 illustrates the expansion of the TAZ with time in response to collector thermal extraction.

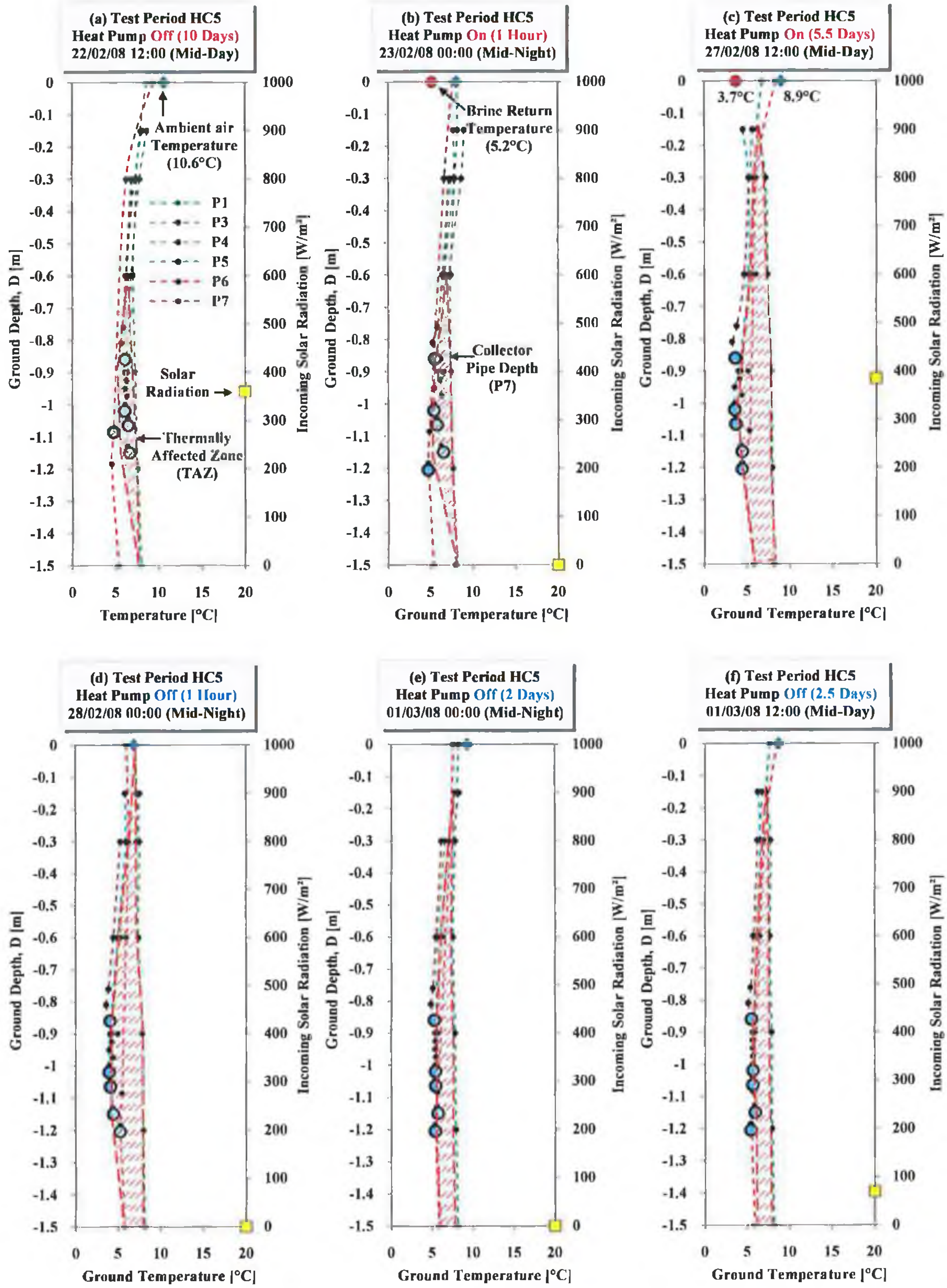


FIGURE 4.28 VERTICAL PROFILE OF ALL COLLECTOR PROFILE'S THERMALLY AFFECTED ZONE GENERATED DURING TEST PERIOD HC5.

Figure 4.29 illustrates the extent of the TAZ and thermal recovery at Profile 5 after test period HC4 and also the effect of thermal extraction during test period HC5.

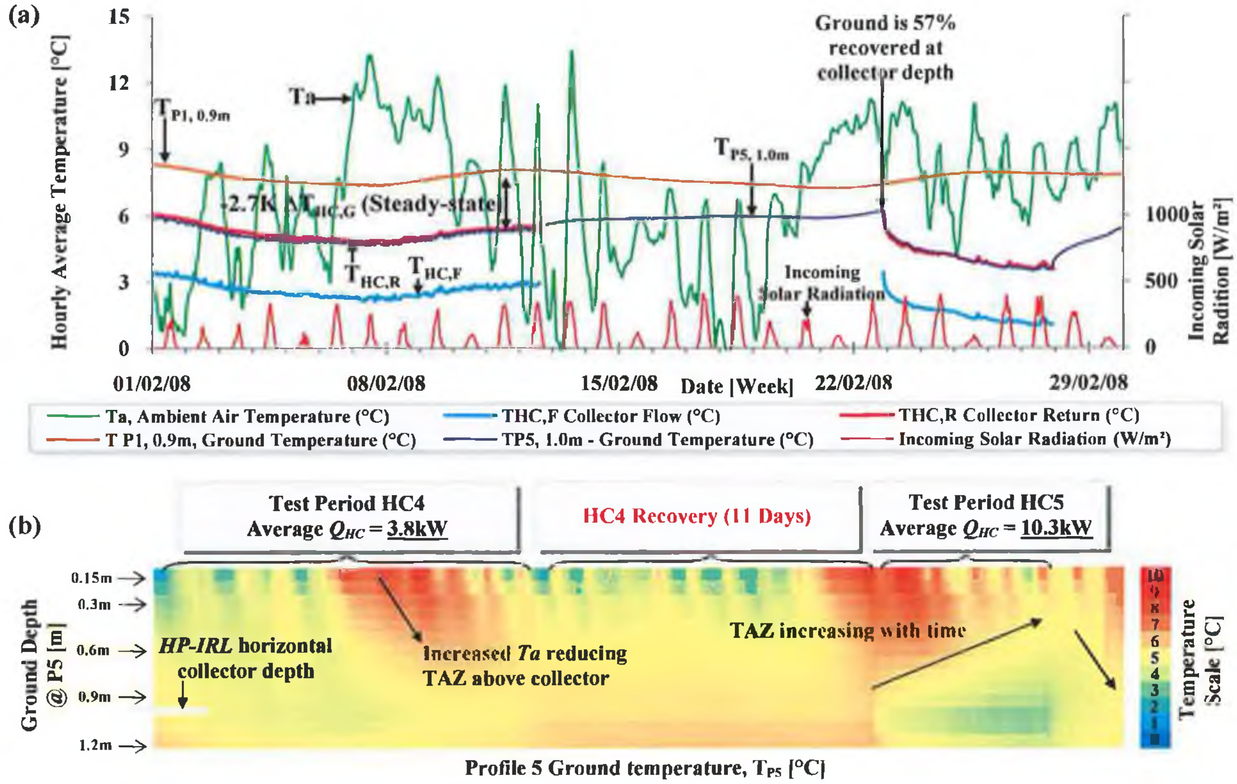


FIGURE 4.29 (A) GROUND THERMAL EXTRACTION AND RECOVERY FOR TEST PERIODS HC4 AND HC5 AND (B) 3-D TRANSIENT TEMPERATURE MAP AT PROFILE 5.

As illustrated in Figure 4.29(b), test period HC4 has a low thermal extraction of 3.8kW and the TAZ is less pronounced around the collector, indeed there is a notable decrease in the TAZ at the end of the test period which coincided with an increased ambient air temperature. It is noticeable from Figure 4.29 that the ground's recovery after test period HC4 coincides with a drop in the ambient air temperature, and the recovery is inhibited as a result, regaining 57% (below τ_{R1}, 63%) of its steady-state temperature in P5 after 11 days, just prior to the start of test period HC5.

In a continuation of the characterisation of the TAZ around the horizontal collector, the effect of an intensive period of thermal extraction (66 days, 89% duty) with a resultant ground thermal content depletion, Figure 4.30 illustrates the growth of the TAZ within the collector region in test period HC9. From Figure 4.30(a) it can be seen that there is no noticeable TAZ at the start of the test. This is due the collector being off for nine months before the test commenced. There is however some variation seen in the profile temperatures throughout the test period. These variations are due to differences in the ground thermal properties at the various profiles and differences in ground surface covers. There is a considerable difference in temperature between profile 3, under a brick pavement, and the other profiles measured shown in Figure 4.30(e).

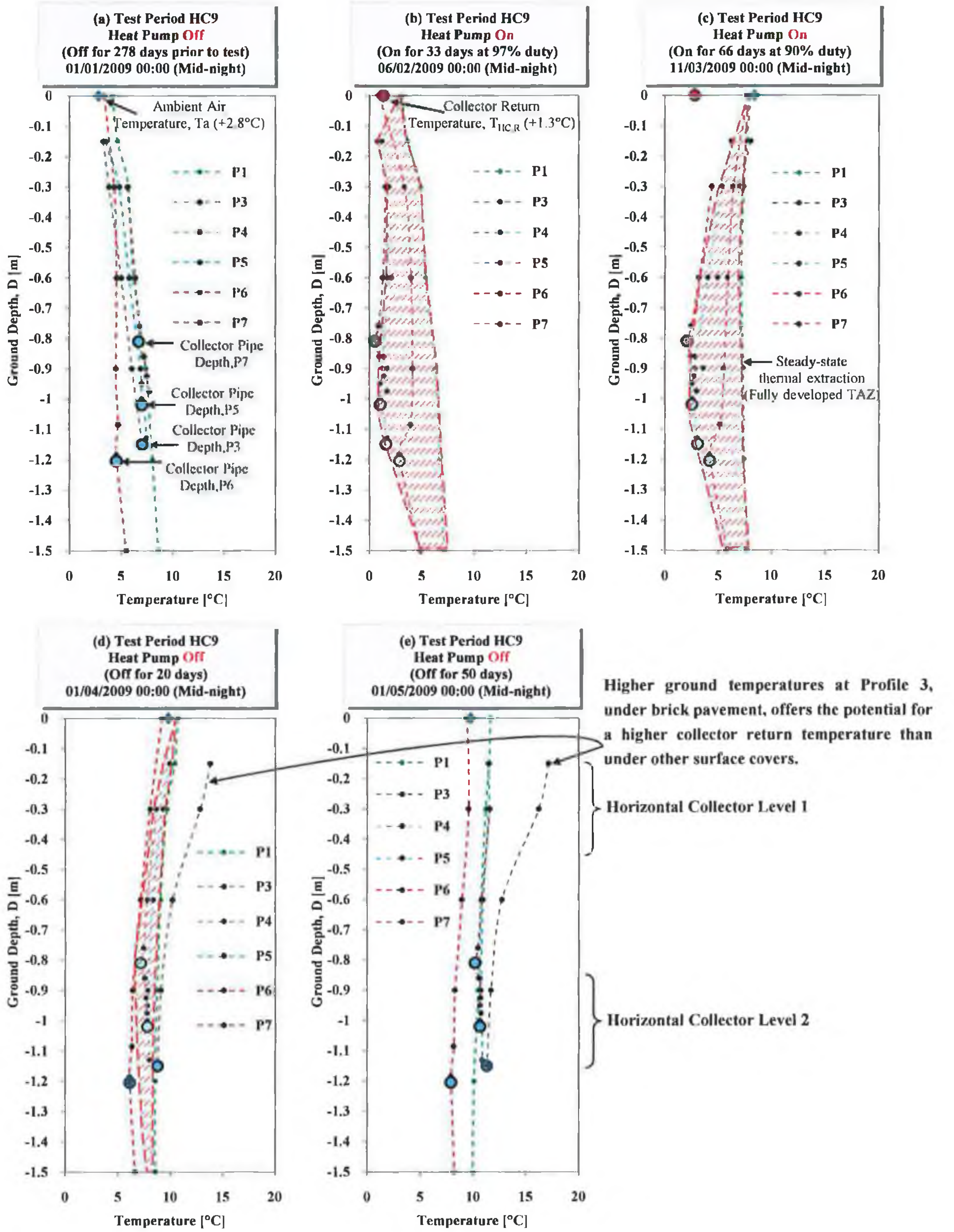


FIGURE 4.30 LONG-TERM EXPANSION AND CONTRACTION OF THE THERMALLY AFFECTED ZONE FROM TEST PERIOD HC9.

As the measurement date in Figure 4.30(e) is May 2008, the strengthening of the solar radiation is an important influence in the temperature rise under the exposed pavement profile. This aspect of the collector surface influence is discussed in more detail in Section 4.3.5.

The ground temperature difference between the reference profile and within the TAZ reaches a maximum of -3K after 33 days of thermal extraction and reduces down to -2.7K between 33 (06/02/09) and 66 days (11/03/09). This improvement of the overall TAZ ground temperature difference from day 33 to day 66 is indicative of the slightly reduced thermal extraction from 9.9kW to 8.3kW, along with the increased levels of solar energy incident onto the collector area.

Figure 4.31(a) and (b) presents the TAZ expansion due to test period HC9 thermal extraction and Figure 4.32(a) and (b) the TAZ contraction due to thermal recovery. Figures 4.31(a) and 4.32(a) presents the change in temperature within the collector region (Profile 5) over the test period HC9. Figures 4.31(b) and Figure 4.32(b) represents the difference in temperature between the control, Profile 1, and the representative of the collector, Profile 5, at the corresponding time and depths for instance: $T_{P5,0.9m}$ - $T_{P1,0.9m}$.

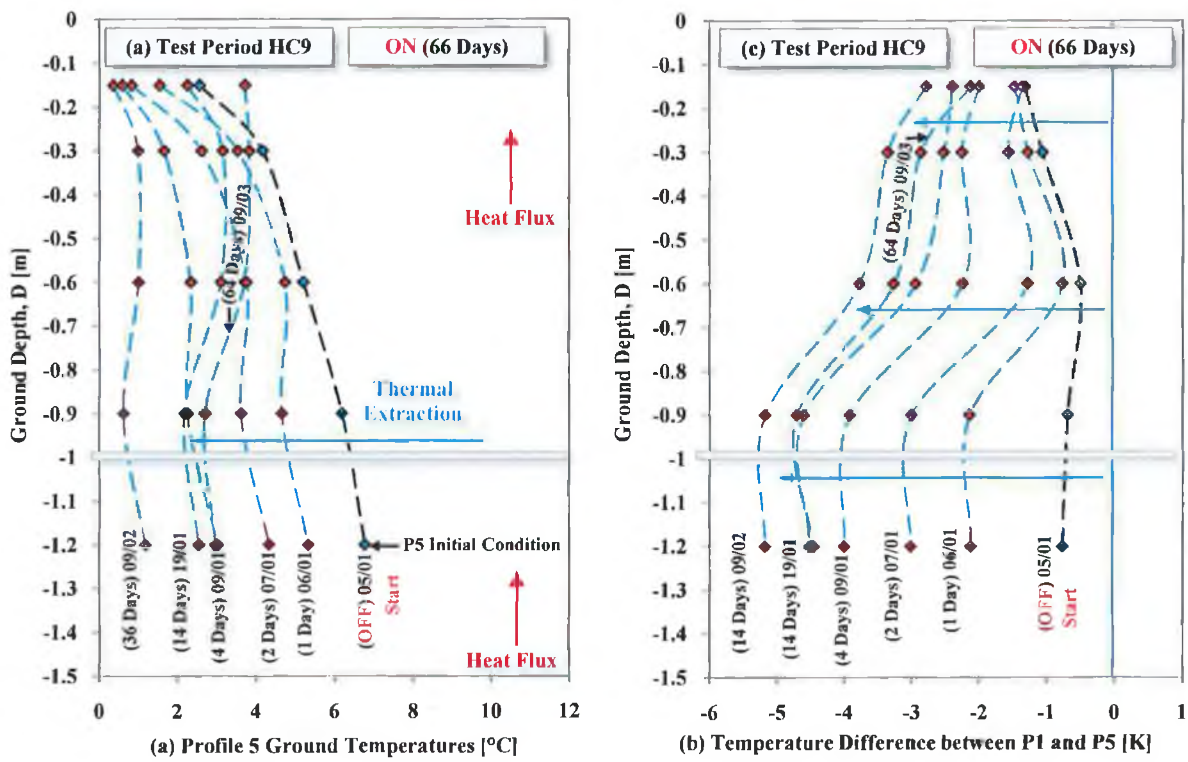


FIGURE 4.31 LONG-TERM EXPANSION OF THE THERMALLY AFFECTED ZONE FROM JANUARY TO MARCH 2009 DURING TEST PERIOD HC9.

Figure 4.31(a) shows that, from the temperature gradients between the various depths, with thermal extraction in January and February the heat flux is predominantly from below, and for March and April the ground recovery is facilitated from both above and below the collector, as shown in Figure 4.32(a).

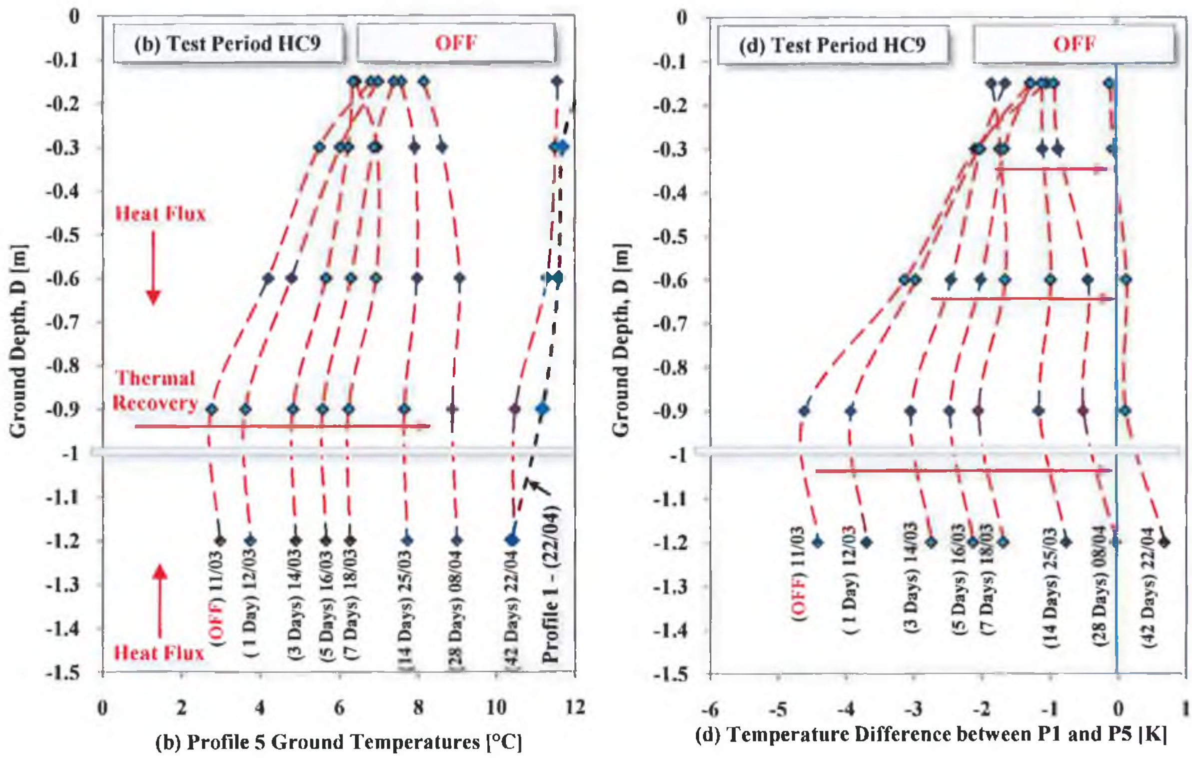


FIGURE 4.32 LONG-TERM RETRACTION OF THE THERMALLY AFFECTED ZONE FROM MARCH TO MAY 2009 SUBSEQUENT TO THE CESSATION OF TEST PERIOD HC9.

Figures 4.31 and 4.32 highlights the effect of heat pump duty, both thermal extraction rate and duration, has on ground temperature within the collector region, showing the growth and retraction of the TAZ. It is important to note that this test far exceeds the extraction rates that characterise domestic heat pump duty. However, the test was important in establishing a 'worst case scenario' in collector thermal extraction, highlighting the TAZ volumetric boundary.

In summary, this section identified the characteristics of the TAZ around the horizontal collector during thermal extraction and utilised three graphical techniques that illustrated:

- it takes approximately five days for the TAZ to extend to the surface during HC5, as shown in Figure 4.26
- how different ground and surface cover types impact of the TAZ, shown in Figure 4.30(d) and (e)

The following section characterises heat pump and collector performance with variation in duty cycle.

4.3.3 IMPACT OF DUTY CYCLE CHARACTERISTICS ON COLLECTOR PERFORMANCE

This section evaluates the implications of variations in duty cycle on the ground temperature drawdown, thermal recovery and COP. Test period HC3 shown in Figure 4.33 illustrates a 59% duty cycle profile similar to domestic applications, where the system was operational for only 14 hours and off for 10 hours per day. This duty cycle run in November, 2007 and maintained for six days. It was followed by a 24 hour, 100% duty cycle for a further five days.

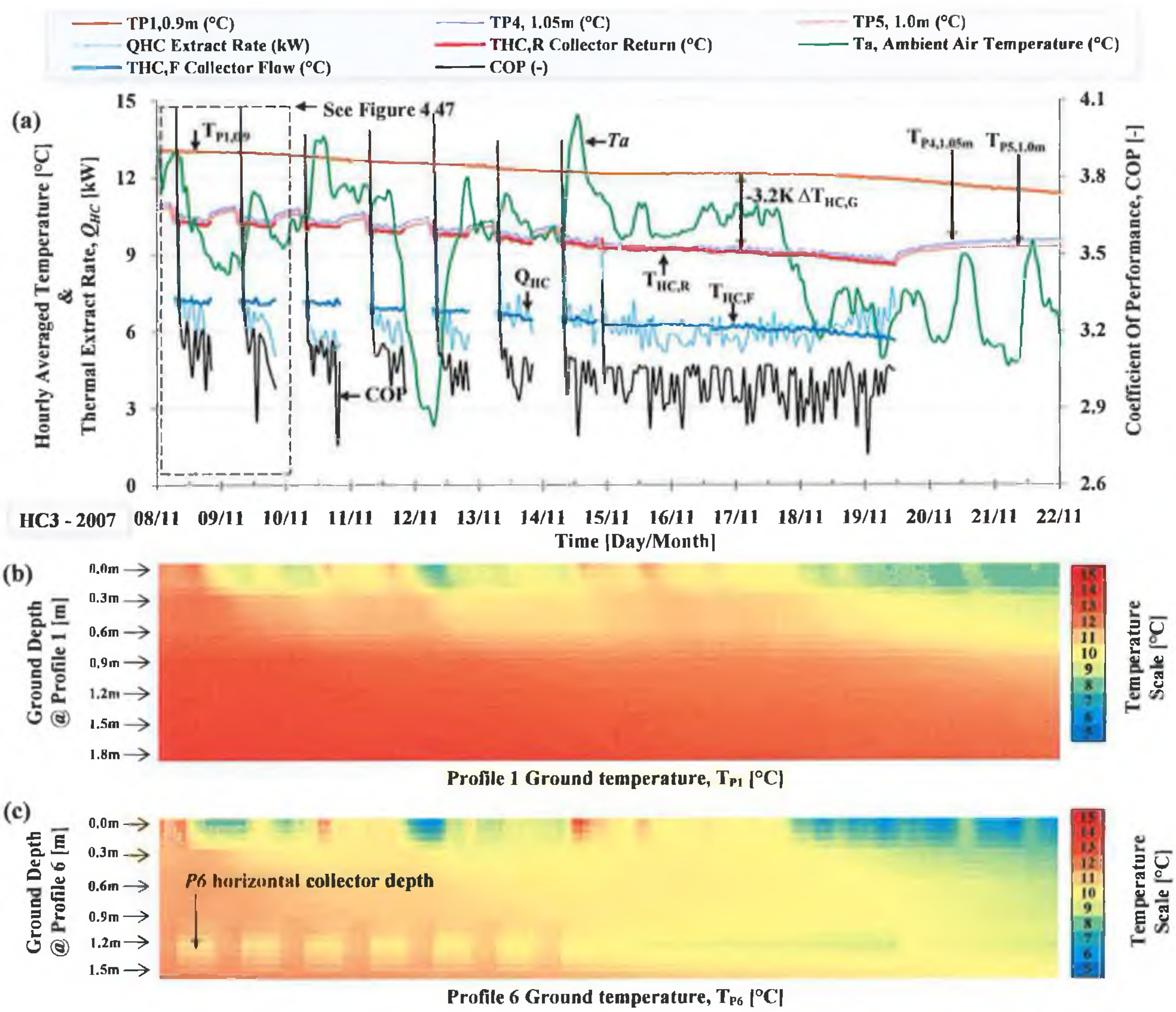


FIGURE 4.33 (A) HOURLY AVERAGED HEAT PUMP AND COLLECTOR PERFORMANCE IN TEST PERIOD HC3, (B) PROFILE 1 3-D REFERENCE TRANSIENT TEMPERATURE MAP AND (C) PROFILE 6 3-D TRANSIENT TEMPERATURE MAP OF GROUND TEMPERATURE DRAWDOWN AND RECOVERY IN THE HORIZONTAL COLLECTOR REGION.

Figure 4.33(a) shows typical climate, ground and collector fluid temperature fluctuations over the 14 day long test period. Given the time of year, it is not surprising that the ambient air temperature decreases steadily, which is reflected in all other parameters, as ground temperature at collector depth in reference profile ($T_{P1,0.9m}$), collector fluid flow and return temperatures ($T_{HC,F}$ an d $T_{HC,R}$) and COP also decrease at the same rate.

Figure 4.33(b) shows the impact of the advancing winter with colder ambient external air temperatures on the 12th of November and from the 18th to the 22nd of November depleting the thermal energy levels in the upper 1m deep layer of the reference profile 1.

Tables 4.13 and 4.14 present primary and secondary results from test period HC3 respectively.

TABLE 4.13 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD HC3 (MEDIUM-TERM, MODERATE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{HC,R}$	Average $T_{HC,\infty}$	Total kWh Solar Thermal incoming	Total kWh extracted from HC
HC3	08/11/07 – 19/11/07	11	59%	6.5 kW	+9.4°C	+11.5°C	1,835	1,342

TABLE 4.14 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD HC3

Test #	Fraction of thermal energy extracted over incoming solar energy	Average $\Delta T_{HC,G}$	Average collector pipe extract rate [W/m]	Average $GSHP_{HC}$ extract rate per m² of collector	Coefficient Of Performance, COP [-]
HC3	0.73	- 2.1 K	4.3 W/m	15.0 W/m²	3.10 (3.40)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period HC3 was +49.1°C, with a temperature lift (ΔT_{HP}) of +39.7K, delivering a COP_{AVG} of 3.1. The average $T_{HC,G}$ was -2.1K.

Figure 4.34 expands on the heat pump performance during the first two days of HC3.

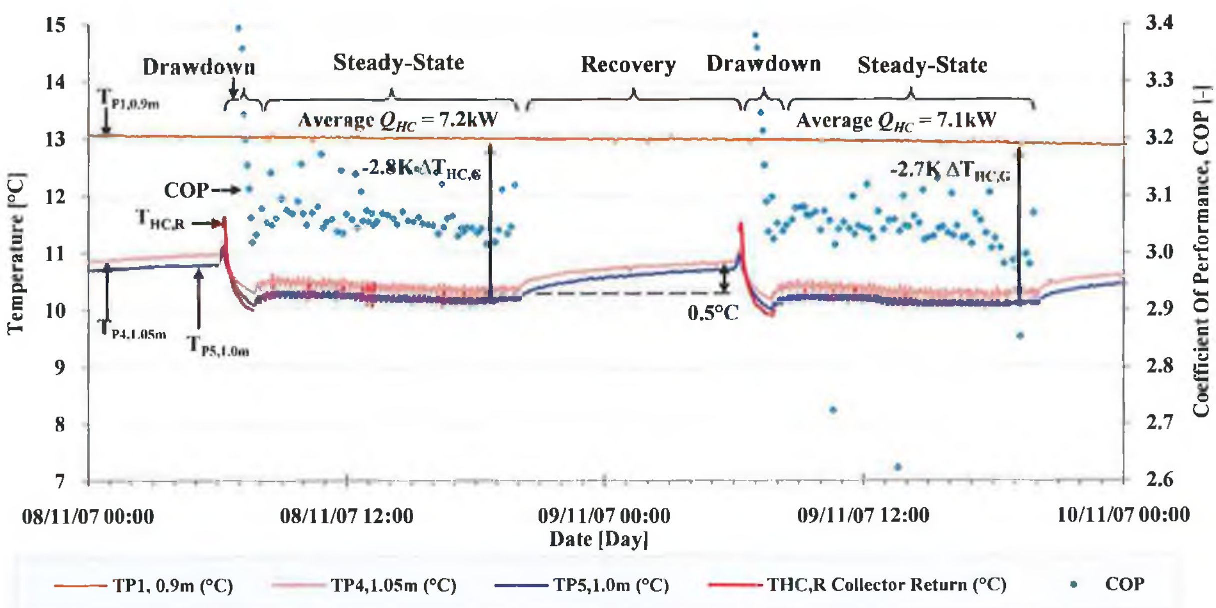


FIGURE 4.34 DRAWDOWN AND RECOVERY OF THE GROUND TEMPERATURE IN THE $GSHP_{HC}$ COLLECTOR REGION DURING NOVEMBER, 2007 (HC3).

The collector brine temperature and COP were recorded at one minute intervals and ground temperate at five minute intervals and the ambient external air temperature is an hourly average. Figure 4.34 therefore illustrates the interaction between the heat pump performance, collector and climate and in doing so demonstrates:

- negligible thermal resistance between the collector ground temperature within 100mm of collector and collector fluid return temperature
- rapid 2 hour drawdown followed by 12 hours of steady-state operation 2.8°C below the reference temperature at profile 1
- small collector region thermal recovery of 0.5°C in the 10 hours between periods of thermal extraction
- the collector fluid return temperature ($T_{HC,R}$) achieves a similar ground temperature to that achieved during previous cycle
- ambient external air temperature generates a slow drop in ground temperature which increases the heat pump temperature lift (ΔT_{HP}) and reduces COP

To illustrate the effect of varying the thermal extract rate on the $\Delta T_{HC,G}$ Figure 4.35 presents a detailed overview of the performance characteristics of both the $GSHP_{HC}$ and the impact of its operation on the ground temperature during February 2008.

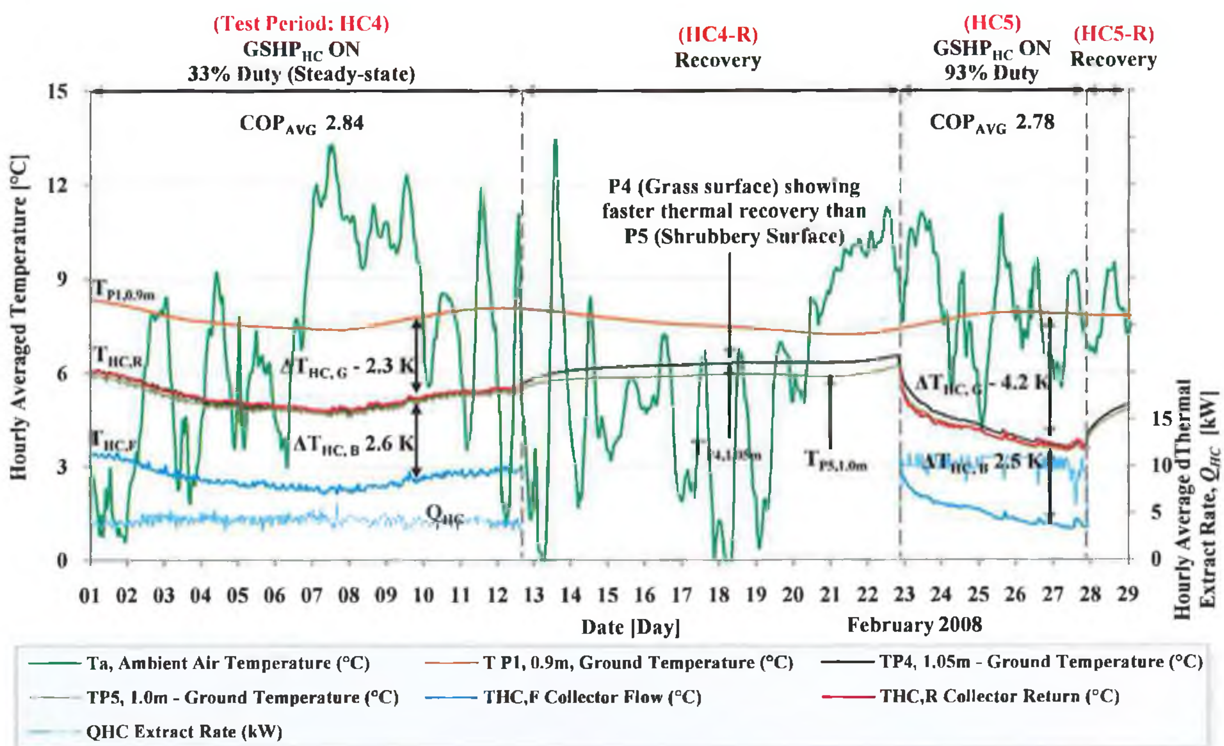


FIGURE 4.35 DRAWDOWN AND RECOVERY OF THE GROUND TEMPERATURE IN THE COLLECTOR REGION OF THE $GSHP_{HC}$ DURING TEST PERIODS HC4 AND HC5.

Test period HC4 ran for 68 days but just the last 12 days are represented in Figure 4.35. The average heat extraction rate was 3.8kW (33% duty) and the $\Delta T_{HC,G}$ remained relatively stable at -2.3K. Table 4.15 provides a detailed analysis of heat pump performance for both test periods shown in Figure 4.35.

TABLE 4.15 SUMMARY OF THE IMPACT OF VARYING THE HEAT EXTRACTION RATE DURING FEBRUARY 2008, AS PRESENTED IN FIGURE 4.35

Test #	Duration (Days)	Thermal Extract	$\Delta T_{HC,G}$	$\Delta T_{HC,B}$	Average $T_{HC,R}$	Average $T_{HP,F}$	COP_{AVG}	COP_{MAX}	COP_{MIN}
HC4	68	3.8kW	- 2.3K	2.6K	+6.1°C	+48.9°C	2.84 (3.12)*	3.15 (3.45)*	2.12 (2.32)*
HC5	6	10.3kW	-3.7K	2.5K	+4.1°C	+49°C	2.78 (3.05)*	2.93 (3.21)*	2.68 (2.94)*

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

It is notable that the ground temperature taken from P1 at a depth of 0.9m in Profile 1 ($T_{P1,0.9m}$) displays a small 1°C sensitivity to the 12°C fluctuation in the ambient air temperature during test period HC4. This thermal dampening effect of the ground's thermal mass minimises the impact of air temperature fluctuations, which is helpful for typical short periods of unusually low air temperature.

Between Day 12 and Day 23 (period HC4-R) the heat pump was turned off and the ground allowed to recover, reducing the $\Delta T_{HC,G}$ to less than one third (-0.8K) of the value established during operation. On day 23 the heat pump was reactivated and requested to deliver a higher output that averaged approximately 10.3kW during test period HC5. It is notable that the $\Delta T_{HC,G}$ increased from -2.5K to -4.2K, reflecting the higher thermal extract rate (Q_{HC}).

It is also noticeable that the negligible thermal resistance between the ground and the collector fluid, where it can be seen that the temperature of the ground directly beside the collector pipe in Profiles 4 and 5 are of similar temperatures to that of the collector fluid, which agrees with the findings of O'Connell and Cassidy (2004).

Also a notable feature in Figure 4.35 is the recovery variation between Profiles 4 and 5, where Profile 4 under grass indicates a faster recovery than that of Profile 5, which is under shrubbery.

A prominent condition that can affect the performance of the horizontal collector is that which is imposed by the climate. The following section quantifies the potential impact of climatic parameters on collector performance.

4.3.4 IMPACT OF WEATHER ON COLLECTOR PERFORMANCE

As outlined in Sections 2.3.1 and 4.1.2, a ground condition that can affect horizontal collector performance is moisture content. Figure 4.36 highlights how the Maritime climate provides relatively frequent significant rainfall events that maintain moist ground conditions. From this Maritime climate rainfall provision, Figures 4.8 and 4.9 showed that the ground moisture content remain around saturation levels, and this is assumed for all tests conducted.

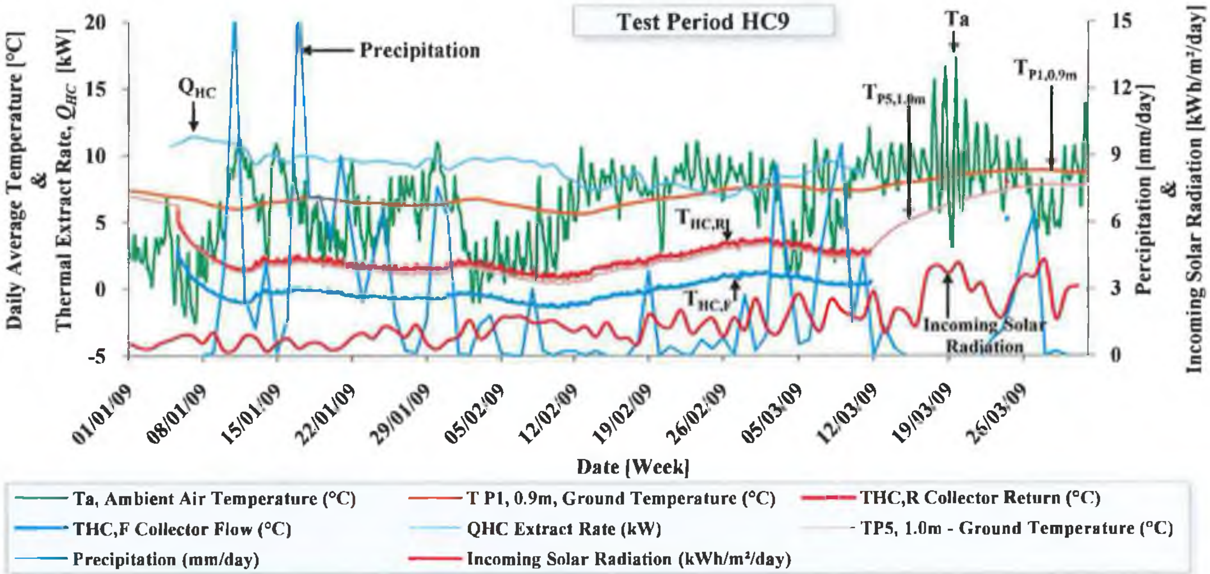


FIGURE 4.36 PRECIPITATION LEVELS OVER 3-MONTH LONG TEST PERIOD HC9.

As can be seen from Figure 4.36 the ambient air temperature and incoming solar radiation are also important climatic parameters that affect collector performance. As the ambient air temperature and solar radiation levels increase, so does the collector return temperature ($T_{HC,R}$). This indicates a positive influence of the climate on collector performance during winter and spring and results from Sections 4.1 and 4.2 indicate these positive influences also increase closer to the surface in spring and summer. Thus, the interface between the collector and the climate is an important factor in maximising the collector performance, the impact of which depends on the depth of the collector, season, weather and choice of ground surface as indicated in Figure 4.30(e). This therefore shows that:

- Ambient air temperature seasonally influences collector ground temperature at 0.9m depth
- Incoming solar radiation begins to increase collector ground temperature at 0.9m depth in March
- Weather parameters have a minimal short-term impact at the collector depth of 0.9m, as shown in Figure 4.36. However, depending on surface cover type significant influence is shown to penetrate to 0.5m depth

The Maritime climate offers a relatively high and stable winter ground temperature and over the course of three years of testing ground temperatures at a depth of 0.9m did not go below +3.7°C. Indeed the lowest collector flow ($T_{HC,F}$) and return ($T_{HC,R}$) temperatures never went below -2.2°C and +0.8°C respectively, even under the intensive and long-term thermal extraction test period HC9. Typical secondary refrigerant freeze protection in Ireland is facilitated for fluid temperatures down to -15°C, and this is achieved with brine volumetric mix of 30% ethylene glycol and 70% water. If the secondary refrigerant freeze point should be at least 5K below the mean heat pump collector fluid temperature (Rawlings *et al.*, 2004) this would indicate a freeze point of -7°C for Maritime climate collectors is adequate. Reducing the volumetric percentage of glycol would facilitate a reduced secondary refrigerant freeze point. As water is less viscous and has a higher specific heat capacity than glycol a reduction of glycol in the secondary refrigerant will also reduce pumping power requirements, increase heat capacity and increase heat transfer efficiency. As the margin of safety is reduced, careful management of the source would be required in order to ensure no breach of the higher freeze temperature occurs.

4.3.5 IMPACT OF GROUND SURFACE COVER

This section examines the impact ground surface cover illustrated in Figure 3.17 has on the ground temperature and thermal recovery rates.

The ground surface material above the collector can play a significant part in maximising the thermal capacity of the collector volume, where for example small shrubs can offer a layer of thermal insulation against excessive convective cooling, or alternatively where a brick surface can increase the ground temperature by absorbing a greater proportion of incident solar radiation. The brick's capacity to increase the ground temperature is revealed in Figure 4.37, where an infra-red thermal image taken in April 2009 of the collector surface indicates that the brick surface to be almost 8°C warmer than the grass.

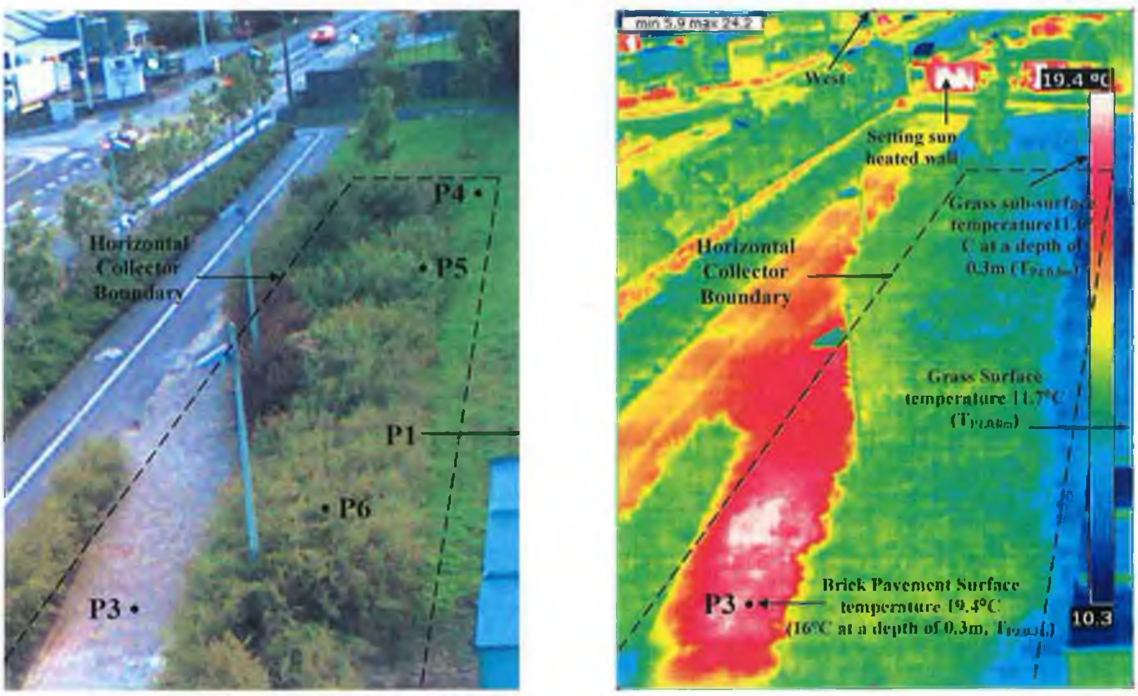


FIGURE 4.37 THERMAL IMAGE OF HORIZONTAL COLLECTOR SURFACE, AT 10PM ON THE 18TH OF APRIL 2009.

This observation is substantiated by the PT100 temperature measurement presented from Profiles 1, 3 and 5 in Figure 4.38 which shows that the brick covered surface achieves substantially higher sub-surface ground temperatures than under shrubbery or grass, during spring and summer. Little differences exist in autumn and winter.

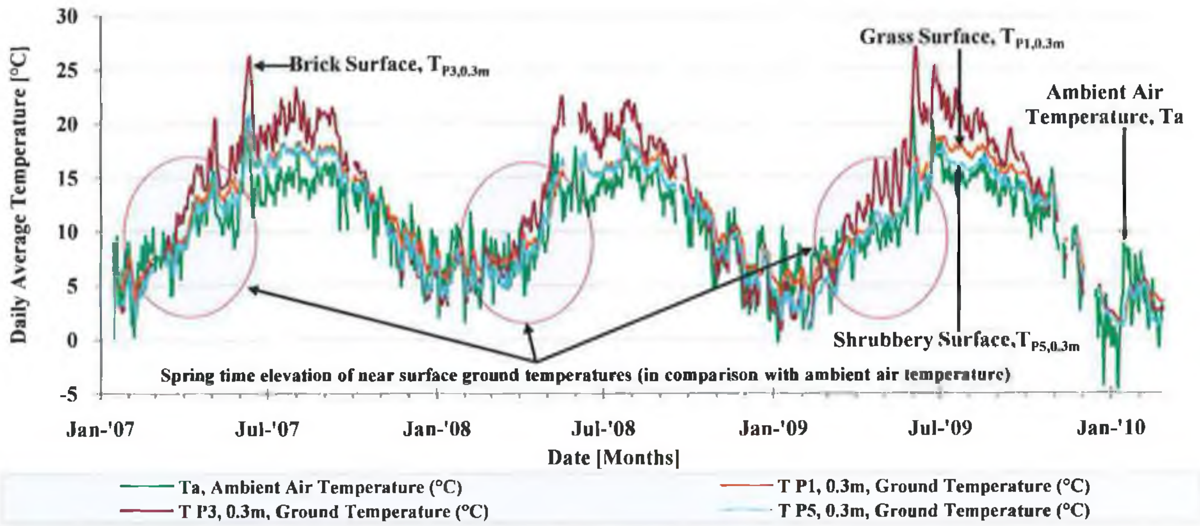


FIGURE 4.38 GROUND TEMPERATURES AT 0.3M UNDER GRASS, BRICK AND SHRUBBERY SURFACES.

Figure 4.39 shows that seasonal advantage in spring and summer translates into a year-round advantage with brick pavement ($T_{P3,0.3m}$) displaying a 2.8°C higher annual average temperature than the ambient air temperature (*Ta*), and 1.2°C higher than grass.

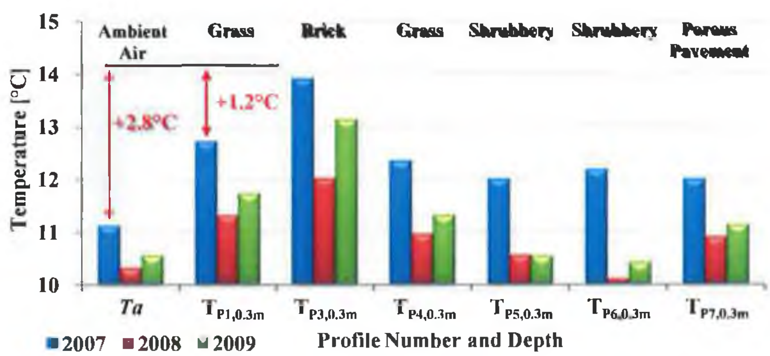


FIGURE 4.39 ANNUAL AVERAGED GROUND SURFACE TEMPERATURES AT 0.3M DEPTH.

Figures 4.40 and 4.41 illustrates the spring-time thermal advantage a near surface collector, positioned at a depth of 0.3m under a brick surface, may have when compared to a deeper collector at 0.9m, under a grass surface.

For January and February the temperature under the grass at 0.9m is an average 0.8K warmer than at 0.3m under the grass surface. However, as can be seen in Figure 4.40, the temperature profile is reversed from mid-march, where at a depth of 0.3m has an average 1K thermal advantage over the ground at 0.9m depth.

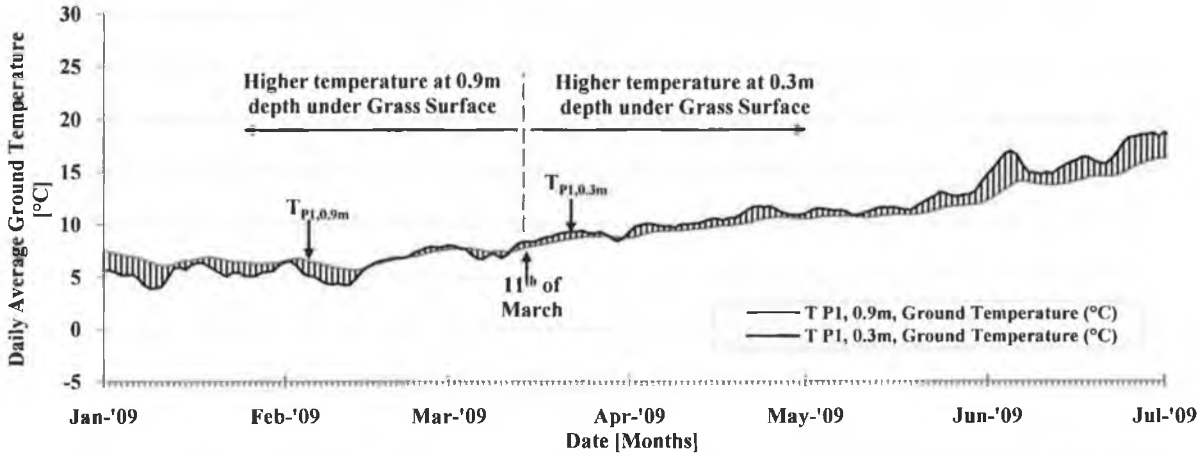


FIGURE 4.40 WINTER, SPRING AND SUMMER GROUND TEMPERATURE DIFFERENCES UNDER GRASS SURFACE AT 0.3M AND 0.9M DEPTH.

If the ground temperature of the ground at a depth of 0.9m under a grass surface is compared with the ground temperature at a depth of 0.3m under a brick pavement, as presented in Figure 4.41, the spring-time thermal advantage at the near surface location is more pronounced.

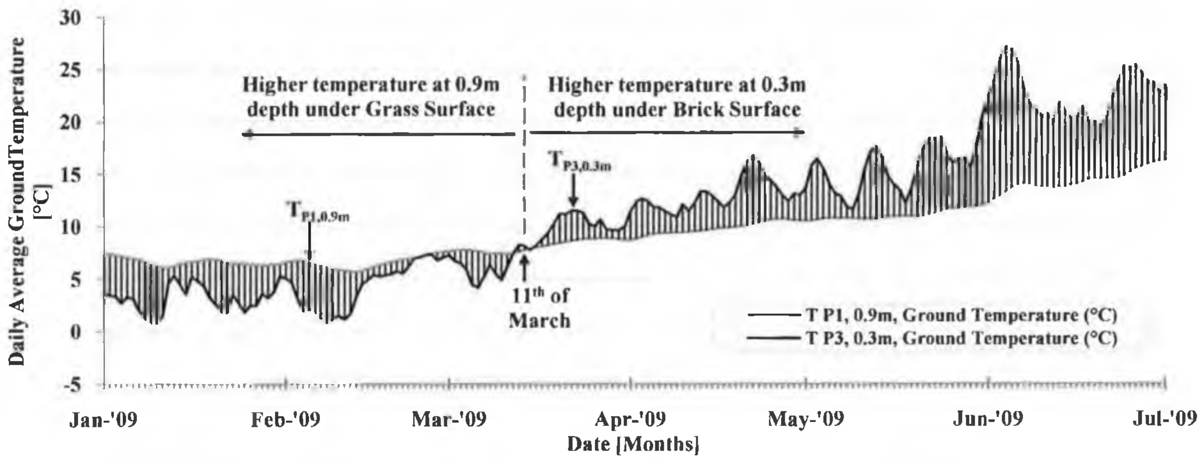


FIGURE 4.41 WINTER, SPRING AND SUMMER GROUND TEMPERATURE DIFFERENCES UNDER GRASS SURFACE (0.9M) AND UNDER BRICK PAVEMENT SURFACE (0.3M).

Figure 4.41 shows that between January and February the temperature under the brick pavement is an average 2.8K cooler than under the grass surface cover. However, this temperature profile is reversed from mid-march, where up until June the average temperature under the brick pavement is an average 4.7K higher. This increase in source temperature, if utilised correctly during spring and summer, could have a significant effect on horizontal collector performance.

While Figures 4.37 to 4.41 indicate that there is potential for substantial improvements in the thermal provision under a brick pavement, or indeed tarmac surface, its permeability that allows the ingress of rainfall is a key element controlling adequate sub-surface moisture content. The moisture content could be maintained under a pavement/tarmac surface by utilising porous material, or by piping rainfall run-off under the surface.

There is potential for a near surface (0.3m) horizontal collector that could be installed above the standard horizontal collector (0.9m), offering a higher thermal resource during the springtime, through source side thermal management. Indeed there is also potential to utilise a source side thermal management of a multi-source horizontal collector that is located under both grass (0.9m) and pavement/driveway (0.3m). Driveways are not currently utilised as part of general horizontal collector design.

4.4 SUMMARY

The chapter has presented a detailed analysis of all the parameters that impact on the thermal characteristics of the horizontal collector region, the depth to which they impact and their seasonal characteristics. It has illustrated the potential to accentuate the many positive aspects of the climate with variations in collector depth. In assessing the performance of the horizontal collector under Maritime climatic conditions, new insights have been revealed into the impact of climate, collector design (depth, ground cover), drawdown, thermally affected zone and heat pump duty (steady-state and thermal recovery).

Over the course of the nine test program the horizontal collector operated for 293 days from 2007 to 2009, delivering 69,514 kWh of energy (250 GJ). This is the equivalent of five years of space heating for a domestic dwelling (12,000 – 15,000kWh/annum). The energy extraction by the collector generated an overall drawdown of -3.5K on the ground source farfield temperature and the heat pump delivered an average heat pump sink temperature of +49.1°C and a COP that ranged from 2.7 to 3.4. The performance of the $GSHP_{HC}$ over all nine test periods is presented in Table 4.16.

TABLE 4.16 SUMMARY OF AVERAGED TEST PERIOD PERFORMANCE INDICATORS FOR THE $GSHP_{HC}$

Test #	Days (Hours)	Collector Extract Rate (Duty)	$T_{HC,\infty}$	$T_{HC,R}$	$\Delta T_{HC,G}$	$T_{HP,F}$	ΔT_{HP}	Collector pipe extract rate	Collector extract rate per m² of collector area	Coefficient Of Performance, COP [-]*
HC1	69	7.8kW (55%)	+6.7°C	+4.0°C	-2.7K	+48.9°C	44.9K	5.2 W/m	18.1 W/m²	2.82 (3.10)
HC2	55	4.3kW (32%)	+11.5°C	+9.2°C	-2.3K	+49.4°C	40.2K	2.9 W/m	10.1 W/m²	3.02 (3.31)
HC3	11	6.5kW (59%)	+11.5°C	+9.4°C	-2.1K	+49.1°C	39.7K	4.3 W/m	15.0 W/m²	3.10 (3.40)
HC4	68	3.8kW (33%)	+8.4°C	+6.1°C	-2.3K	+48.9°C	42.8K	2.5 W/m	8.8 W/m²	2.84 (3.12)
HC5	6	10.3kW (93%)	+7.8°C	+4.1°C	-3.7K	+49.0°C	44.9K	6.8 W/m	23.6 W/m²	2.78 (3.05)
HC6	16	9.8kW (93%)	+7.8°C	+3.1°C	-4.7K	+49.8°C	46.7K	6.4 W/m	22.4 W/m²	2.65 (2.91)
HC7	1 (24)	6.4kW (35%)	+8.8°C	+7.5°C	-1.3K	+46.4°C	38.9K	3.0 W/m	10.4 W/m²	3.27 (3.61)
HC8	1 (15)	7.0kW (48%)	+14.4°C	+13.4°C	-1.0K	+48.4°C	35.0K	4.6 W/m	16.2 W/m²	3.35 (3.68)
HC9	66	9.1kW (89%)	+6.7°C	+2.2°C	-4.5K	+49.1°C	46.9K	6.0 W/m	21.1 W/m²	2.66 (2.92)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

Over the course of the nine test periods the average COP varied from 2.65 (HC6) to 3.35 (HC8), with heat pump temperature lifts (ΔT_{HP}) of 46.7K and 35.0K respectively. Since the heat pump sink temperature ($T_{HP,F}$) was typically around +49°C for all test periods, the variation in performance is due to the fluctuations in the collector return temperature ($T_{HC,R}$). The overall average SPF was 2.90.

The average $T_{HC,R}$ for the three years of testing was +4.1°C, with the lowest recorded return temperature of +0.8°C recorded in test period HC9. The greatest hourly average ground temperature drawdown ($\Delta T_{HC,G}$) of -5.2K and was recorded in test period HC6. The test period with the greatest average thermal extraction was HC5, extracting 6.8W/m of collector pipe and 23.6W/m² of collector area. The lowest ground temperature recorded at 0.9m depth was +3.7°C, recorded on the 13th of January 2010.

From the tests conducted, the following observations were made:

- *HP-IRL* COPs recorded to the EN-14511 standard matched those of the independent test laboratory Arsenal Research to within 2.2%.
- COPs recorded to the EN-14511 allow the performance of different heat pumps to be compared, but do not reflect heat pump performance when collector pumping power is included. In this case COP variation increased to 6.5%. Additional collector pumping power can reduce the EN-14511 standard test results by up to 10%.
- Positive influence of milder Maritime climate on heat pump performance which showed that collector ground temperature did not drop below +3.7°C.
- Ground temperature drawdown a function of the duty cycle and for steady-state conditions ranged between -2.3°C and -5.1°C between 33% and 94% respectively.
- With source side management, potential exists to improve collector thermal performance with a reduction in the horizontal collector secondary refrigerant freeze point from an existing -15°C to -7°C.
- A new parameter developed to indicate a climate sensitive performance of horizontal collectors called the Collector Performance Indicator (*CPI*). For the *HP-IRL* horizontal collector design, the *CPI* of 0.23K/(W/m²) indicates a ground temperature drawdown of 0.23K per unit demand (W) of collector area (m²).
- A suite of graphical analysis tools have been developed that allow the impact of both climate and specific weather events to be identified and this showed that more thermal energy content available in upper 500mm for spring and summer seasons than below 500mm.
- Ground thermal recovery is a function of the climate, ground type and surface cover.

- Type of ground cover is influential in thermal absorption with varying sensitivities to solar radiation with the ground at 300mm below the brick surface displaying a 4°C higher temperature in spring/summer.
- The recorded average annual temperatures at 0.3m depth under the grass and brick surfaces were +11.9°C and +13.0°C respectively, where the averaged *Ta* was only 10.5°C
- New methods for analysing the fluctuating Thermally Affected Zone around the collector have been presented.

Based on these findings the next Chapter explores the potential for enhanced $GSHP_{HC}$ performance with optimised climate sensitive collector design and source side management.

CHAPTER 5 – HORIZONTAL COLLECTOR: MODELLING AND DESIGN

Using the test results from Chapter 4, this chapter identifies empirical models that are capable of predicting heat transfer rates between the ground and the collector fluid and uses these to develop a simulation tool for horizontal collector design. This tool was used to develop a climate sensitive collector design that minimises the ground temperature drawdown and increases the collector fluid return temperature to yield a predicted 8% improvement in annual heat pump performance.

5.1 COLLECTOR HEAT TRANSFER

This section identifies the empirical models that are capable of predicting heat transfer rates between the ground and the collector fluid. Figure 5.1 illustrates an idealised horizontal collector region, along with key design features such as collector pipe length and depth, along with performance indicators such as collector fluid inlet ($T_{HC,F}$) and outlet ($T_{HC,R}$) temperatures and the farfield temperature.

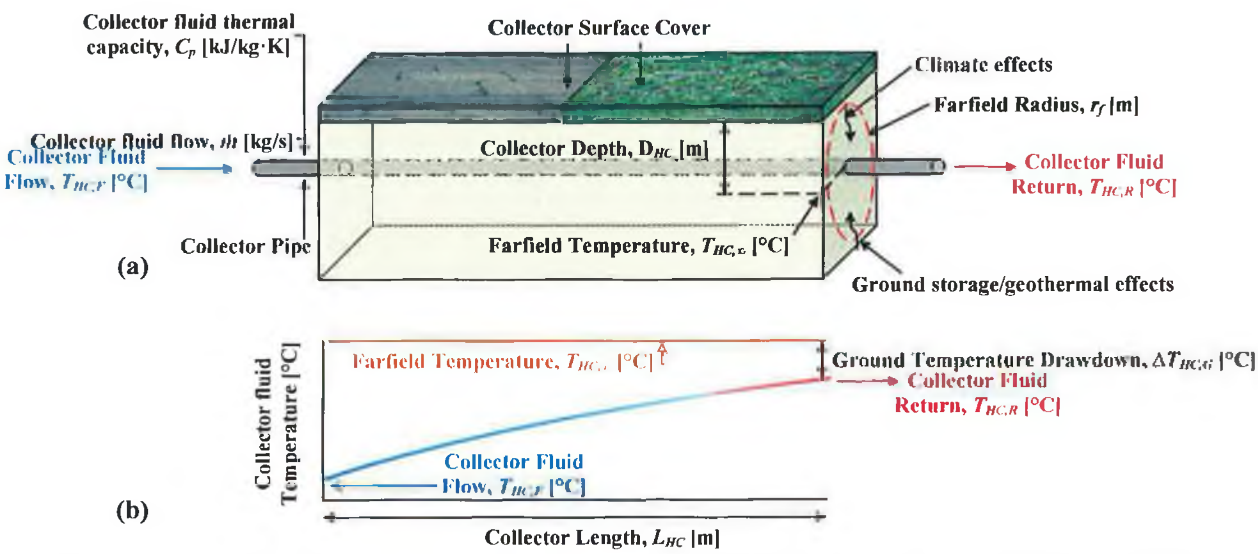


FIGURE 5.1 HORIZONTAL COLLECTOR (A) DESIGN AND PERFORMANCE INDICATORS AND (B) COLLECTOR FLUID TEMPERATURE CHANGE WITH COLLECTOR LENGTH.

Based on heat transfer analysis presented in Appendix K for internal fluid flows, the collector flow and return temperatures $T_{HC,F}$ and $T_{HC,R}$ can be estimated using Equation 5.1.

$$\frac{T_{P1,0.9}-T_{HC,R}}{T_{P1,0.9}-T_{HC,F}} = exp\left(-\frac{L_{HC}}{\dot{m}\cdot C_p\cdot R_{Total}}\right) \qquad \textit{Equation 5.1}$$

Where; $T_{P1,0.9}$ is the ground's farfield temperature at the collector depth ($T_{HC,\infty}$), $\dot{m}$ is the collector fluid mass flow, C_P is the collector fluid's specific thermal capacity, L_{HC} is the horizontal collector loop length and R_{Total} is the total collector thermal resistance. Rearranging Equation 5.1 allows the collector fluid return temperature, $T_{HC,R}$, to be determined:

$$T_{HC,R} = T_{P1,0.9} - \left(T_{P1,0.9} - T_{HC,F}\right) \cdot \left(exp\left(-\frac{L_{HC}}{\dot{m} \cdot C_p \cdot R_{Total}}\right)\right) \qquad \textit{Equation 5.2}$$

Drawing on the steady-state recorded data from test periods HC3, HC4, HC6 and HC9 shown in Figure 5.2(a), Figure 5.2(b) compares the results of test period HC9 with the predicted steady-state thermal extraction using Equation 5.2.

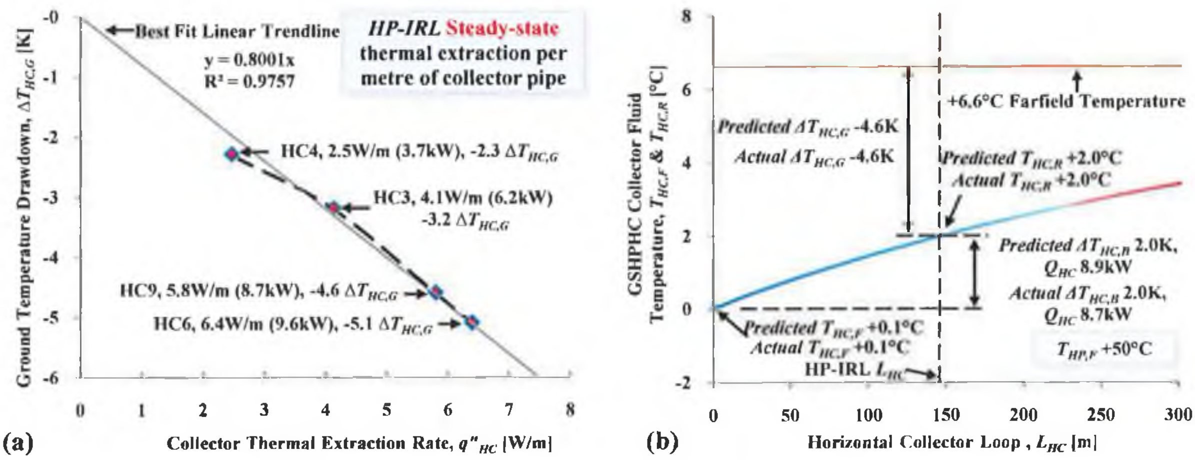


FIGURE 5.2 (A) GROUND TEMPERATURE DRAWDOWN ($\Delta T_{HC,G}$) VERSUS COLLECTOR THERMAL EXTRACTION RATE PER METER OF COLLECTOR PIPE (1500M) RECORDED UNDER STEADY-STATE THERMAL EXTRACTION CONDITION IN TEST PERIODS HC3, HC4, HC6 AND HC9 AND (B) LONG-TERM STEADY STATE PREDICTED VARIATION TO THE COLLECTOR FLUID TEMPERATURE ALONG THE COLLECTOR LENGTH WHEN EXPOSED TO A UNIFORM FARFIELD GROUND TEMPERATURE OF +6.6°C VERSUS TEMPERATURES RECORDED IN TEST PERIOD HC9.

Figure 5.2(b) displays the effective increase in the horizontal collector fluid temperature ($T_{HC,F}$ to $T_{HC,R}$) along the horizontal collector length where the collector is operating under typical winter ground temperatures ($T_{HC,\infty}$ = +7°C). Predicted results in Figure 5.2(b) corresponds accurately with the measured experimental results of the long-term test period HC9 (Average Q_{HC}: 8.7kW, Average $\Delta T_{HC,G}$: -4.6K), details of which are taken from test results presented in Section 4.3.1. An average diffusivity (α) of 1.1E-6 m²/s and ground thermal conductivity (λ_G) of 2.3 W/m·K are used, assuming uniform moisture content, ground material and for maximum thermal extraction the TAZ will envelope a distance to farfield of 1m above the collector to the surface and 6m below. However, under more moderate thermal extraction duties that would reflect general heat pump utilisation in the Maritime climate the TAZ will remain within the region of between 0.3m and 1m above and below the collector as characterised in Figures 4.25 to 4.28.

5.2 DETERMINING OPTIMAL HORIZONTAL COLLECTOR DESIGN

This section utilises the heat transfer characteristics outlined in Section 5.1 to design efficient collectors that minimise heat pump performance under the Irish Maritime climate. The key parameters include the collector length, spacing and depth.

5.2.1 COLLECTOR LENGTH

As the COP increases with collector source temperature the only variable that can be adjusted to increase the return temperature and not impact on the thermal energy delivered is the collector length. Based on the analysis presented in Appendix H Table 5.1 details the collector pressure drop and pumping power requirement of a 1500m long collector along with the recovered (absorbed) thermal energy from the circulation pumps and the resulting total parasitic power lost from pumping.

TABLE 5.1 COLLECTOR ELECTRICAL PUMPING REQUIREMENT FOR A 1500M LONG HORIZONTAL COLLECTOR

Collector Components	Pressure	Pumping Power (η_{circ}=0.6)	Percentage of Total Power
Heat Pump Evaporator, ΔP_{HE}	33 kPa	60 W	14%
Flow & Return to Collector Manifold, ΔP_{pipe2}	3 kPa	5 W	1%
Spiral Loops, ΔP_{pipe1}	25 kPa	46 W	11%
In-Line Loops, ΔP_{pipe1}	173 kPa	315 W	74%
Total	**234 kPa**	425 W	
Total Pumping Power of 425W of which 128W is thermally recovered			

Collector Length 1500m
∴ **Parasitic Pumping Power = 0.2W/m (q'_{pump})**

Table 5.1 quantifies the parasitic power lost per meter length of collector pipe and therefore the effect of increasing the collector length on increasing both the flow resistance and pumping power. Since pumping power increases by 0.2W/m collector length (q'_{pump}), in increasing the collector length a compromise must be reached between an increased COP from an increased collector fluid return temperature and an increased pumping power. In order to justify a length increase, any additional length must raise the collector fluid return temperature enough to overcome the negative effect of increased pumping power. Therefore, the collector length should be maximised to generate a maximised COP. To establish the conditions under which the COP is maximised, Equation 5.3 illustrates the increase in collector fluid return temperature per meter of collector length ($\Delta T_{HP,Lift}$) required to impact positively on the COP.

$$\Delta T_{\mathrm{HP,Lift}} = \frac{q'_{\mathrm{pump}} \cdot \mathrm{COP}}{\left(\frac{Q_{\mathrm{HC}}}{\Delta T_{\mathrm{HC,B}}}\right)} = \frac{(0.2\mathrm{W/m})(3)}{\left(\frac{11000W}{2.8K}\right)} = 1.5\ x\ 10^{-4}\ K/m \qquad \textit{Equation 5.3}$$

Where; $\Delta T_{HP,Lift}$ is the minimum allowable rise in fluid temperature per meter of collector pipe (K/m), Q_{HC} is the overall collector thermal extraction (kW), $\Delta T_{HC,B}$ is the temperature drop across the heat pump evaporator (K) with a nominal COP of 3. As a result, any increase in the collector length must raise the collector fluid return temperature by 1.5 x 10^{-4} K/m so that the negative influence of increasing pumping power per meter of collector pipe is reversed by heat absorbed per additional meter length. If an increase in collector length does not reach this thermal absorption rate the COP will begin to reduce.

Having identified models capable of predicting the thermal dynamics of the *HP-IRL* horizontal collector, the variation in heat transfer characteristics and pumping power consumption with collector length is conducted to establish optimum collector length.

Taking this into consideration and using the EN-14511 test standard results for the *Solterra* heat pump, the predicted performance evaluation illustrated in Figures 5.3 and 5.4 shows that the optimal length of horizontal collector required to deliver a heat pump output temperature of +50°C and +35°C is 170m and 160m respectively.

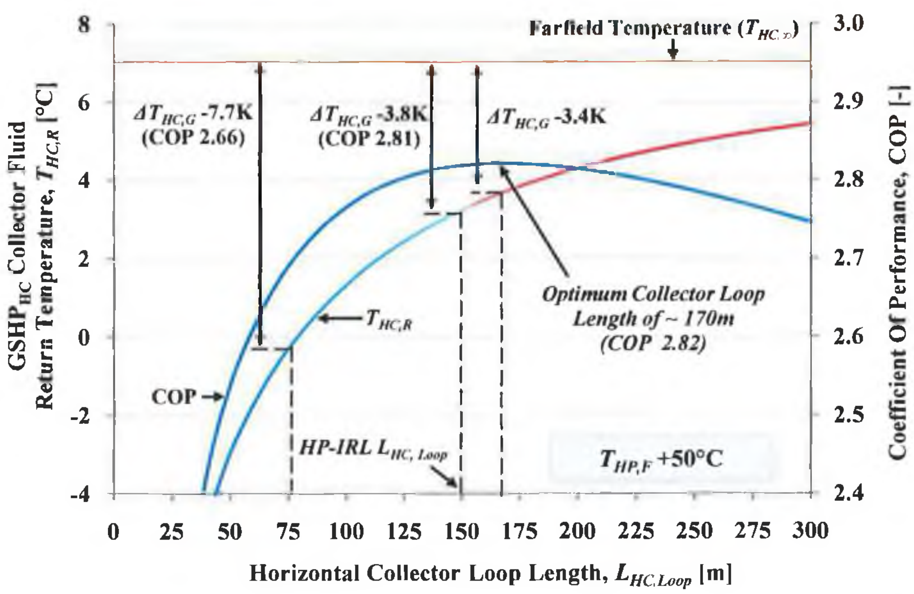


FIGURE 5.3 VARIATION IN THE COLLECTOR FLUID RETURN TEMPERATURE AND RESULTING COP WITH COLLECTOR LENGTH FOR +50°C OUTPUT TEMPERATURE.

In both Figures 5.3 and 5.4 the ground temperature drawdown on the *HP-IRL* collector loop length (150m) and the optimum length collector is compared with the 75m collector loop length recommended in the VDI-4640 standard and illustrated in Figure 2.6. At an output of +35°C, there is a potential 6% difference in COP between the optimum length (160m, COP 3.89) and that associated by the VDI 4640 standard length (75m, COP 3.66).

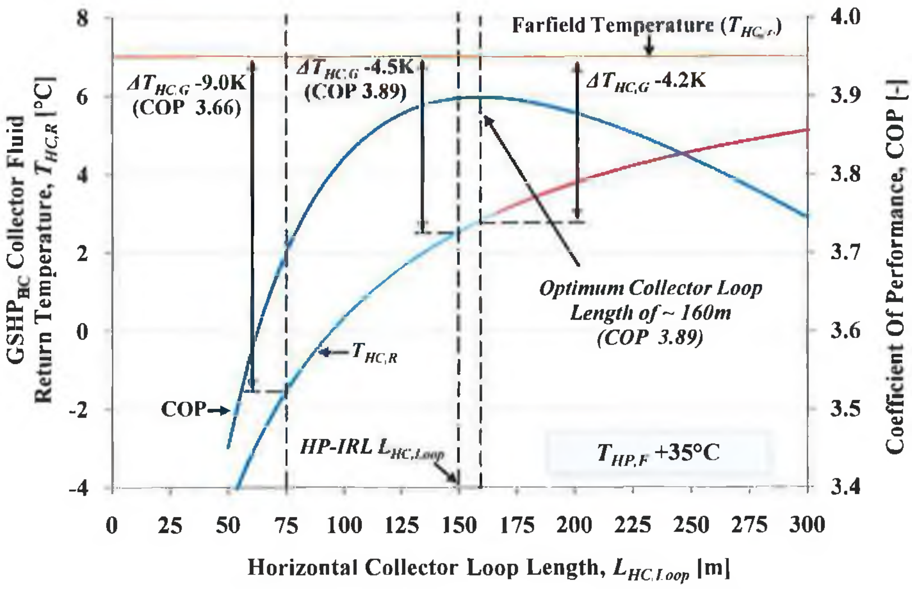


FIGURE 5.4 VARIATION IN THE COLLECTOR FLUID RETURN TEMPERATURE AND RESULTING COP WITH COLLECTOR LENGTH FOR +35°C OUTPUT TEMPERATURE.

From this model, the optimum collector length for the *Solterra* heat pump can be taken as 150m per collector loop, 1,500m overall, with a maximum thermal extraction rate of 7.3W/m. The VDI-4640 standard collector delivers a maximum thermal extraction rate of 14.6W/m.

5.2.2 COLLECTOR SPACING

In this *HP-IRL* study the collector pipes are placed 0.3m apart, which is at the lower end of the 0.3 to 0.8m spacing recommended by both Kavanaugh and Rafferty (1997) and the VDI 4640 standard (2001). Details of the *HP-IRL* GSHP$_{HC}$ are shown in Table 5.2.

TABLE 5.2 HORIZONTAL COLLECTOR DIMENSIONS

Parameter	Value
Total Horizontal Collector Area	430 m²
Total Collector Pipe Length	1500 m
Collector Pipe Outer Diameter, D_o	0.032 m
Distance between Collector Pipes	0.3 m

Ideally, to enable maximum thermal extraction there should be no thermal interaction between the collector pipes. According to Hart and Couvillion (1986), using an approximation equation for determining the farfield radius ($r_f = 4 \cdot \sqrt{\alpha \cdot t}$), thermal interaction between adjacent pipes spaced 0.3m apart starts after just 15 minutes, regardless of the thermal extraction rate. If the pipes were spaced 0.8m apart, the time to reach farfield (r_f= 0.4m) would be 2.5 hours and a spacing f 4m would have to be implemented to eliminate interaction. This indicates that some form of thermal interaction will occur when the heat pump is in operation, regardless of collector pipe separation and thermal extract rate. Thus, a compromise in collector pipe spacing of between 0.3m and 0.8m is recommended in the literature.

5.2.3 Collector depth

The literature reflects the consensus that horizontal collector performance increases with depth due to higher source temperature with depth over the winter heating season (Mihalakakou *et al.*, 1996; IGSHPA *et al.*, 1997; Kavanaugh and Rafferty, 1997; VDI 4640 / Part 2, 2001; ASHRAE, 2003 - 2006; EHPA, 2005; Brown, 2009). Eleven of the twelve guidelines reviewed in Table 2.3 recommended horizontal collector depths that ranged from 0.9m (EHPA, 2005) to 3.0m (Hepbasli, 2004) with the average between 1.2m and 1.5m.

However, as indicated in Section 4.3.5, there is evidence that shallower collectors positioned less than 0.5m deep can operate successfully in the Irish Maritime climate, particularly during spring and summer. Indeed performance can be further improved by using a ground cover such as brick with high solar absorptivity.

Using *HP-IRL* data, Figure 5.5 illustrates the potential difference in ground temperature with depth under brick and grass covers.

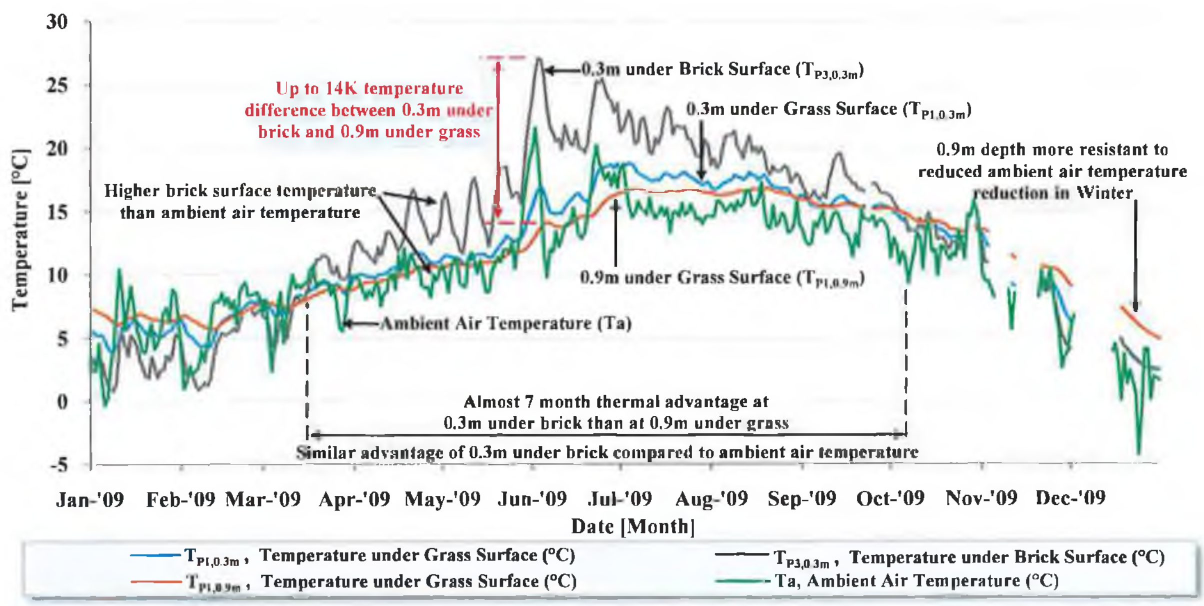


Figure 5.5 Variations in the ground temperature under brick and grass surface covers during 2009.

It is noticeable from Figure 5.5 that better protection from the harshest winter temperature fluctuations are shown a depth of 0.9m, maintaining higher source temperatures than at 0.3m under brick cover. However, it is the brick surface cover that performs best for almost seven months during spring, summer and early autumn. Illustrating this point further, Figure 5.6 shows the predicted and actual ground temperatures under the brick and grass during 2009.

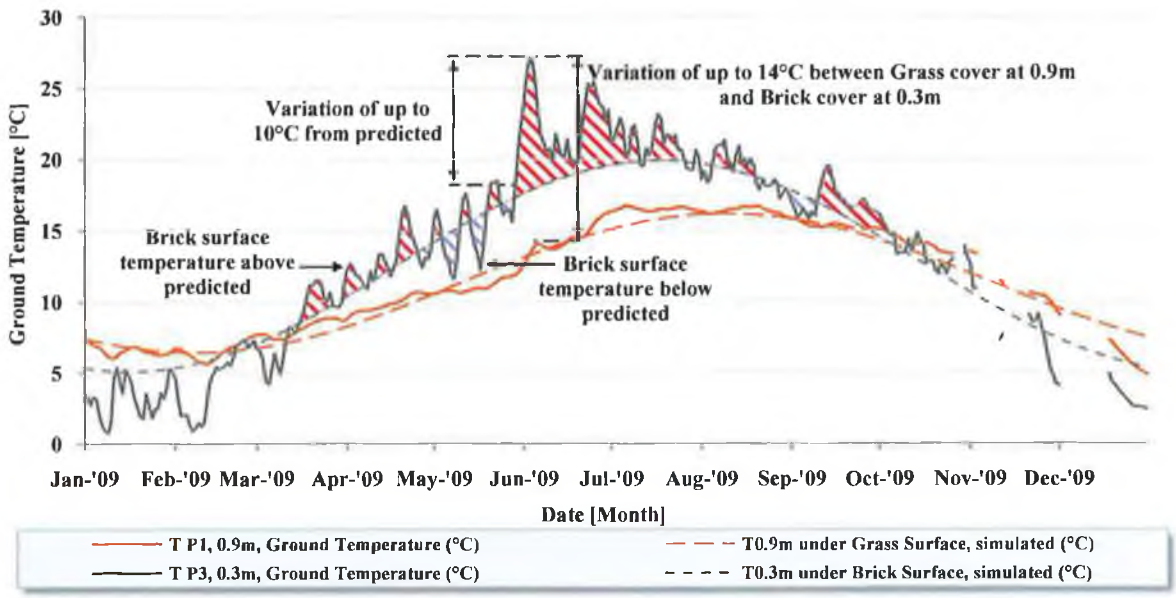


FIGURE 5.6 DAILY AVERAGED MEASURED AND PREDICTED GROUND TEMPERATURES UNDER A GRASS AND BRICK COVERED SURFACE AT DEPTHS OF 0.3M AND 0.9M DURING 2009.

Figure 5.6 clearly illustrates higher ground temperatures under the brick pavement during the spring-summer-autumn period, compared with those at 0.9m under grass. While a peak difference of 14°C was observed in early June, the brick pavement achieved a 4.2°C higher average temperature than grass between April and October 2009. Based on Figure 4.18, this alone could translate into a 10% higher COP during this period. Also observable from Figure 5.6 is the peaks in temperature under the brick surface, which could potentially be exploited further with thermal storage.

5.3 GROUND SOURCE HEAT PUMP PERFORMANCE SIMULATION

This section seeks to develop a simple, excel based tool to simulate horizontal collector ground source heat pump performance given climate, ground temperature and heat pump performance based on source temperature.

Using the ground and collector fluid temperature prediction models presented, the simulation tool capable of evaluating heat pump performance based on various collector depth, length, surface covers, demand profile and sink temperatures, based on recorded ground temperatures and weather data between 2007 and 2009, and characteristic heat pump performance from EN 14511 was constructed. A screen-grab of the excel file data inputs are presented in Appendix L.

A typical domestic dwelling application was used for thermal demand purposes. The building had a pro-rata space heating demand of 0-15kW (0%-100% duty) with external ambient air temperatures of between +15.5°C and -4°C respectively. Based on this criterion, the *HP-IRL* recorded hourly averaged external ambient air temperatures for the three years 2007, 2008 and 2009 were used to determine space heating demand. A constant Domestic Hot Water (DHW) demand of 18kWh per day (6570kWh/annum) was also applied. A flow diagram for the simulation tool is presented in Figure 5.7.

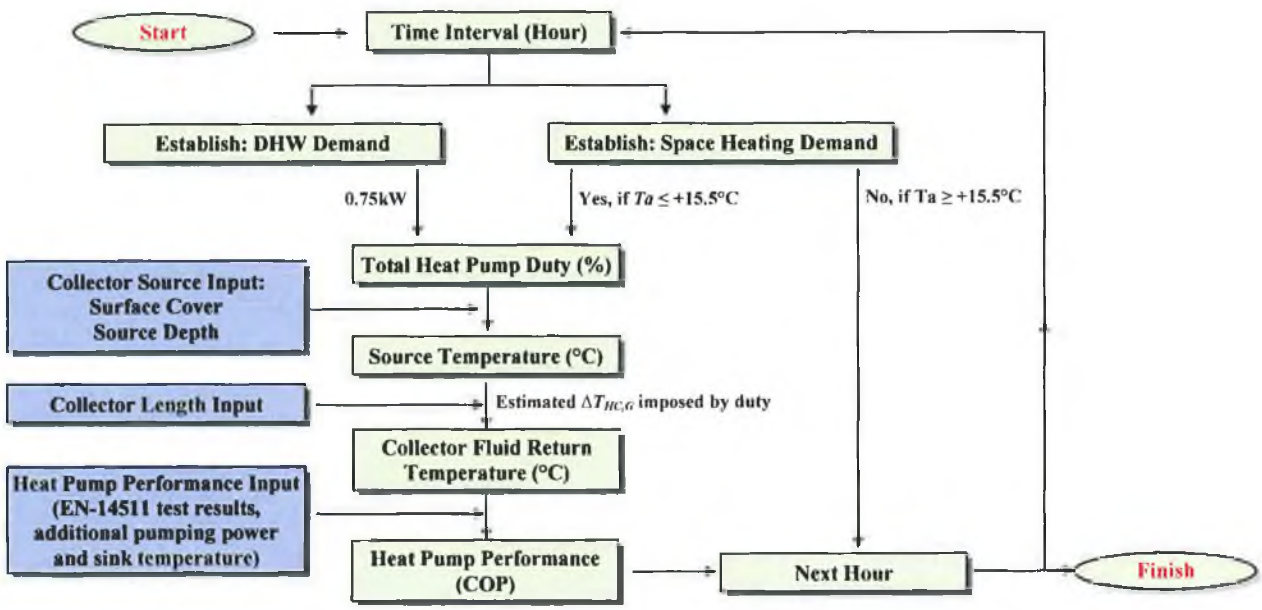


FIGURE 5.7 HEAT PUMP PERFORMANCE SIMULATION FLOW DIAGRAM

No.1 in Table 5.3 presents the three year average farfield temperature at various depths under the grass surface, with No.2 showing the weighted average farfield temperature averaged over the periods of time when the heat pump was operational. For the purpose of presenting the average farfield temperature when the heat pump is in operation, a weighting is applied to the source temperature that is proportional to the heat pump duty. Therefore a source temperature will have a greater weighting when the heat pump duty is 100% than if the duty is only 10%.

TABLE 5.3 THREE-YEAR AVERAGED GROUND TEMPERATURES UNDER GRASS SURFACE

No.	Ground Depth under Grass Surface	0.3 m	0.6 m	0.9 m	1.2 m	1.5 m	1.8 m
1	Three year average farfield ground temperature	+11.8°C	+11.8°C	+11.8°C	+11.6°C	+11.6°C	+11.8°C
2	Three year weighted average farfield ground temperature	+8.7°C	+8.8°C	+9.4°C	+9.5°C	+9.7°C	+10.0°C

The three-year weighted average ambient air temperature was +7°C. It is noticeable from Table 5.3 the weighted average ground temperature differs by 1.3°C between depths of 0.3m and 1.8m. This would translate into a 3% improvement in COP based on Figure 4.18.

Table 5.4 presents the simulated $GSHP_{HC}$ performance with the horizontal collector located at various depths under the grass surface. The ground temperature drawdown, $\Delta T_{HC,G}$ was proportional to the thermal demand and modulates between 0 and -6K as per Figure 5.2(a).

The simulation uses the *Solterra* heat pump performance data as shown in Table 3.4, adjusted to include pumping power, with an optimum collector length of 1500m and 0.3m pipe spacing.

TABLE 5.4 PREDICTED *SOLTERRA* HEAT PUMP PERFORMANCE AT VARIOUS DEPTHS UNDER GRASS SURFACE USING *HP-IRL* RECORDED GROUND TEMPERATURES, SINK TEMPERATURES OF +50°C AND +35°C, 1500M COLLECTOR LENGTH AND A 0.3M PIPE SPACING

+50°C Sink Temperature, $T_{HP,F}$						
Ground Depth under Grass Surface	**0.3 m**	**0.6 m**	**0.9 m**	**1.2 m**	**1.5 m**	**1.8 m**
Three year weighted average farfield ground temperature	+8.7°C	+8.8°C	+9.4°C	+9.5°C	+9.7°C	+10.0°C
Average Collector Return Temperature, $T_{HC,R}$	+5.9°C	+6.0°C	+6.5°C	+6.6°C	+6.8°C	+7.1°C
Lowest Collector Return Temperature, $T_{HC,R}$	-2.4°C	-1.7°C	-0.0°C	+0.9°C	+1.7°C	+2.2°C
Average Drawdown Temperature, $\Delta T_{HC,G}$	-2.8 K	-2.8 K	-2.9 K	-2.9 K	-2.9 K	-2.9 K
Maximum Drawdown Temperature, $\Delta T_{HC,G}$	-5.3 K	-5.5 K	-5.8 K	-6.0 K	-6.2 K	-6.3 K
Average Heat Pump Temperature Lift, ΔT_{HP}	+44.1K	+44.0K	+43.5K	+43.4K	+43.2K	+42.9K
Average Extract Rate, Q_{HC}	3.1kW	3.1kW	3.2kW	3.2kW	3.2kW	3.3kW
Seasonal Performance Factor, SPF	**3.05**	**3.06**	**3.09**	**3.10**	**3.11**	**3.13**

+35°C Sink Temperature, $T_{HP,F}$						
Ground Depth under Grass Surface	**0.3 m**	**0.6 m**	**0.9 m**	**1.2 m**	**1.5 m**	**1.8 m**
Three year weighted average ground temperature	+8.7°C	+8.8°C	+9.4°C	+9.5°C	+9.7°C	+10.0°C
Average Collector Return Temperature, $T_{HC,R}$	+5.6°C	+5.8°C	+6.3°C	+6.4°C	+6.6°C	+6.9°C
Lowest Collector Return Temperature, $T_{HC,R}$	-3.4°C	-2.6°C	-0.9°C	+0.1°C	+0.9°C	+1.5°C
Average Drawdown Temperature, $\Delta T_{HC,G}$	-3.0 K	-3.0 K	-3.1 K	-3.1 K	-3.1 K	-3.1 K
Maximum Drawdown Temperature, $\Delta T_{HC,G}$	--6.3 K	-6.4 K	-6.7 K	-6.8 K	-6.9 K	-7.0 K
Heat Pump Temperature Lift, ΔT_{HP}	+29.4K	+29.2K	+28.7K	+28.6K	+28.4K	+28.1K
Average Extract Rate, Q_{HC}	3.4kW	3.4kW	3.4kW	3.4kW	3.4kW	3.4kW
Seasonal Performance Factor, SPF	**4.01**	**4.02**	**4.05**	**4.06**	**4.07**	**4.09**

Table 5.4 shows that the grass surface delivers a 2.6% improvement in heat pump performance between depths 0.3m and 1.8m, with a sink temperature of +50°C. A COP improvement of 31% is achieved if the delivered temperature is reduced from +50°C to +35°C, with COP increasing from 3.09 to 4.05. The 1500m long horizontal collector system indicated a collector performance (*CPI*) of 0.21K/(W/m²).

A comparison of the predicted ground temperature drawdown resulting from Equation 5.2 for 750m and 1500m long collectors is illustrated in Figure 5.8.

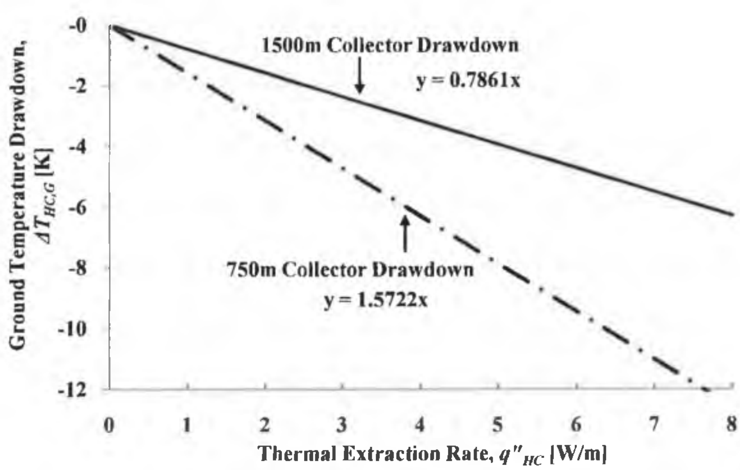


FIGURE 5.8 PREDICTED GROUND TEMPERATURE DRAWDOWN WITH THERMAL EXTRACTION RATE PER M OF COLLECTOR PIPE FOR 750M AND 1500M COLLECTORS WITH 0.3M PIPE SPACING.

Repeating the simulation for a VDI-4640 standard recommended collector length of 750m and using the ground temperature drawdown indicated in Figure 5.8 yielded the results shown in Table 5.5 for a range of collector depths under a grass surface.

TABLE 5.5 PREDICTED *SOLTERRA* HEAT PUMP PERFORMANCE AT VARIOUS DEPTHS UNDER GRASS SURFACE USING RECORDED GROUND TEMPERATURES, SINK TEMPERATURES OF +50°C AND +35°C, 750M COLLECTOR LENGTH AND A 0.3M PIPE SPACING

+50°C Sink Temperature, $T_{HP,F}$						
Ground Depth under Grass Surface	**0.3 m**	**0.6 m**	**0.9 m**	**1.2 m**	**1.5 m**	**1.8 m**
Three year average ground temperature	+8.7°C	+8.8°C	+9.4°C	+9.5°C	+9.7°C	+10.0°C
Average Collector Return Temperature, $T_{HC,R}$	+3.5°C	+3.6°C	+4.1°C	+4.2°C	+4.4°C	+4.6°C
Lowest Collector Return Temperature, $T_{HC,R}$	-6.1°C	-5.5°C	-4.1°C	-3.3°C	-2.6°C	-2.2°C
Average Drawdown Temperature, $\Delta T_{HC,G}$	-5.2 K	-5.2 K	-5.3 K	-5.3 K	-5.4 K	-5.4 K
Maximum Drawdown Temperature, $\Delta T_{HC,G}$	-9.1 K	-9.4 K	-9.9 K	-10.2 K	-10.5 K	-10.7 K
Average Heat Pump Temperature Lift, ΔT_{HP}	+46.5K	+46.4K	+45.9K	+45.8K	+45.6K	+45.4K
Average Extract Rate, Q_{HC}	2.9kW	2.9kW	3.0kW	3.0kW	3.0kW	3.0kW
Seasonal Performance Factor, SPF	**2.95**	**2.96**	**2.98**	**3.00**	**3.01**	**3.03**

+35°C Sink Temperature, $T_{HP,F}$						
Ground Depth under Grass Surface	**0.3 m**	**0.6 m**	**0.9 m**	**1.2 m**	**1.5 m**	**1.8 m**
Three year weighted average ground temperature	+8.7°C	+8.8°C	+9.4°C	+9.5°C	+9.7°C	+10.0°C
Average Collector Return Temperature, $T_{HC,R}$	+3.0°C	+3.1°C	+3.5°C	+3.7°C	+3.9°C	+4.1°C
Lowest Collector Return Temperature, $T_{HC,R}$	-8.3°C	-7.5°C	-6.0°C	-5.2°C	-4.4°C	-3.9°C
Average Drawdown Temperature, $\Delta T_{HC,G}$	-5.7 K	-5.7 K	-5.8 K	-5.8 K	-5.9 K	-5.9 K
Maximum Drawdown Temperature, $\Delta T_{HC,G}$	-11.1 K	-11.3 K	-11.8 K	-12.0 K	-12.3 K	-12.4 K
Average Heat Pump Temperature Lift, ΔT_{HP}	+32.1K	+31.9K	+31.5K	+31.3K	+31.1K	+30.9K
Average Extract Rate, Q_{HC}	3.2kW	3.2kW	3.2kW	3.2kW	3.3kW	3.3kW
Seasonal Performance Factor, SPF	**3.96**	**3.98**	**3.99**	**4.00**	**4.01**	**4.03**

Notable from Tables 5.4 and 5.5 at a depth of 0.9m and sink temperature of +50°C, the 1500m long collector achieves a 3.7% higher COP than the shorter 750m collector. This performance increase was due to the 1500m long collector delivering a 1.6°C higher average collector return temperature than the 750m long collector. It is also noticeable that, for the shorter collector, the collector fluid return temperature drops as low as -4.1°C at the 0.9m depth and remains below 0°C for 970 hours (40 days) in total over the three years. This indicates that the collector region could freeze substantially, inhibiting thermal recovery. The 1500m long collector at a depth of 0.9m achieves a minimum collector fluid return temperature of 0.5°C, generating minimal ground freezing.

The *CPI* for the VDI-4640 standard collector was 0.43K/(W/m²) and is therefore less thermally effective than the 1500m long horizontal collector which indicated a collector performance of 0.21K/(W/m²).

5.4 PERFORMANCE CHARACTERISATION OF AN ALTERNATIVE MARITIME CLIMATE HORIZONTAL COLLECTOR DESIGN

Based on observations made in Sections 4.1, 4.3.5 and 5.2.3 that showed an average 4.2°C higher ground temperature under brick pavement during the spring, summer and early autumn of 2009 compared to grass covered regions, the simulation tool was deployed to evaluate an alternative split level collector presented in Figure 5.9.

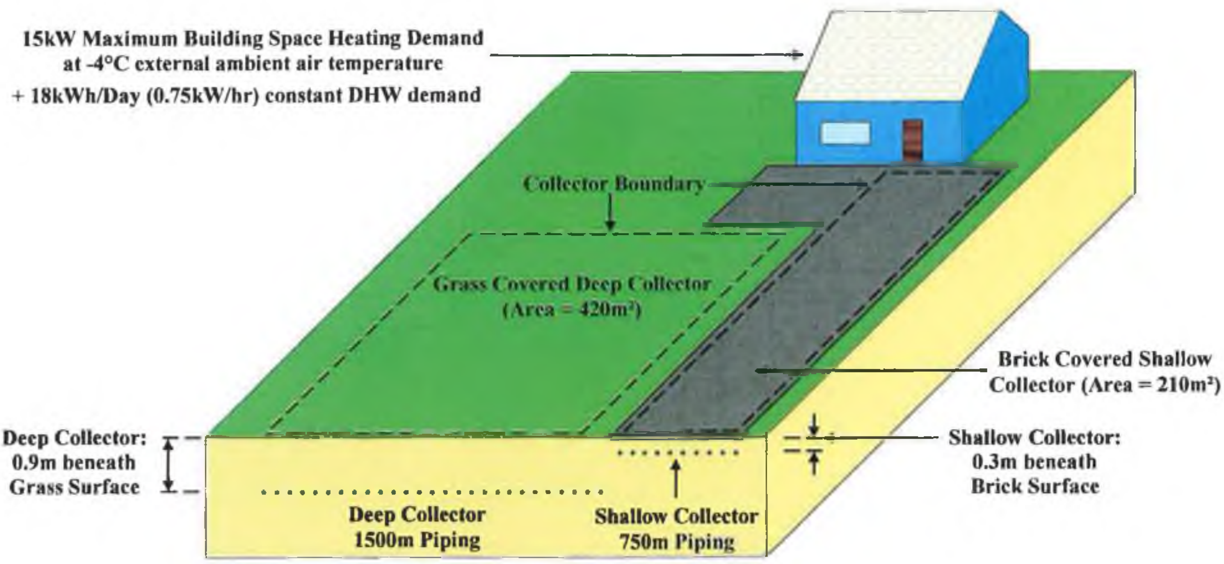


FIGURE 5.9 ALTERNATIVE, MARITIME CLIMATE SENSITIVE SPLIT LEVEL HORIZONTAL COLLECTOR DESIGN.

The split level collector consists of two separate collectors, a deep collector identical to the *HP-IRL* collector covering 420m² (1500m long pipe) and positioned 0.9m beneath a grass surface and a 210m² shallow collector of (750m long pipe) area positioned 0.3m beneath a brick surface. Both collectors have a *CPI* of 0.21K/(W·m²).

As the shallow collector is half the deep collector length it is limited to a heat pump duty of 50% or less. The activation of either collector is determined by the highest source side temperature which was achieved by comparing farfield temperatures of both source depths over three years data (2007-2009) from profile 3 (brick) and profile 1 (grass).

This split level collector design attempts to capitalise on the higher ground temperature under brick, especially during spring, summer and early autumn periods and Table 5.6 presents the results of a simulation run using *HP-IRL* recorded ground and ambient air temperatures for 2007, 2008 and 2009 for the reference dwelling shown in Figure 5.9.

The decision tree for system operation and collector selection is shown in Figure 5.10.

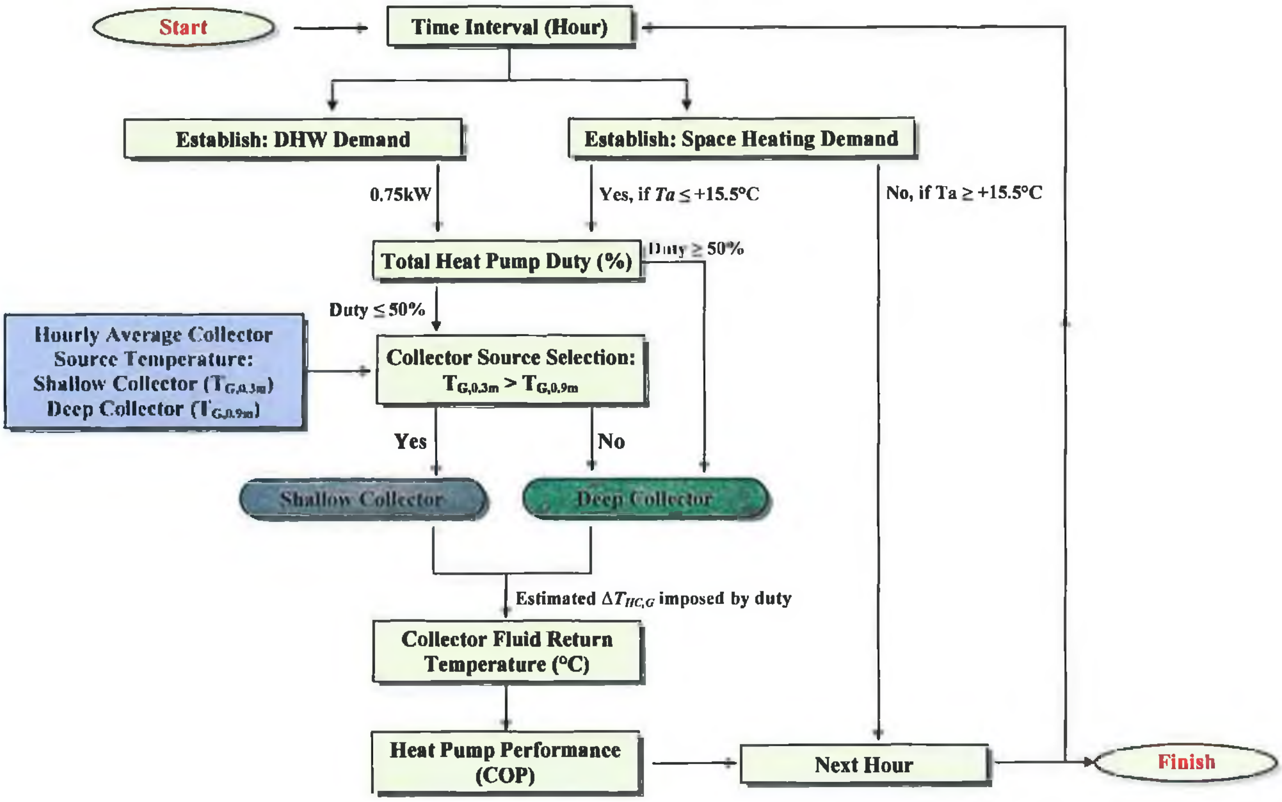


FIGURE 5.10 SPLIT LEVEL HORIZONTAL COLLECTOR GSHP SIMULATION DECISION TREE BASED ON THREE YEARS OF HOURLY AVERAGED TEMPERATURE DATA (2007-2009).

TABLE 5.6 PREDICTED *SOLTERRA* HEAT PUMP PERFORMANCE WHEN USED WITH SPLIT LEVEL HORIZONTAL COLLECTOR IN FIGURE 5.9

Heat Pump Operational Parameter	*VDI-4640 Standard, Single Source Collector*	*Single Source, Optimal Length Collector*	*Split Level Collector*
Collector Depth	0.9m	0.9m	0.3m & 0.9m
Collector Area	350m²	420m²	640m²
Collector Length	750m	1500m	2250m
Sink Temperature, $T_{HP,F}$	+50.0°C	+50.0°C	+50.0°C
Average Extract Rate, Q_{HC}	3.0kW	3.2kW	3.5kW
Average Thermal Supply, Q_{HP}	8.9kW	9.9kW	11.3kW
Three year average ground temperature	+9.4°C	+9.4°C	+11.1°C
Average Collector Return Temperature, $T_{HC,R}$	+4.1°C	+6.5°C	+8.4°C
Lowest Collector Return Temperature, $T_{HC,R}$	-4.1°C	-0.9°C	-0.0°C
Average Drawdown Temperature, $\Delta T_{HC,G}$ (Maximum)	-5.3 K (-9.9 K)	-2.9 K (-5.8 K)	-2.8 K (-5.8K)
Average Heat Pump Temperature Lift, ΔT_{HP}	+45.9K	+43.5K	+41.7K
Seasonal Performance Factor, SPF	2.98	3.09	3.22
Performance Improvement on VDI Standard Collector Size:	-	3.7%	8.1%

The shallow collector delivered a substantial contribution of 26% of thermal demand and operated 57% of the heat pump on-time. Table 5.6 shows that at +8.4°C, the split level collector return temperature ($T_{HC,R}$) is 4.3°C higher than the VDI-4640 standard collector. This delivers for the VDI-4640 standard collector and split level collector seasonal performance factors of 2.98 and 3.22 respectively, corresponding to an 8.1% increase on the system performance for the split level collector.

5.5 SUMMARY

The detailed characterisation of the horizontal collector operation in the Maritime climate presented in Chapters 4 and 5 have significantly increased understanding of horizontal collector design and performance. Test data from three years of testing has highlighted:

- The recorded average annual external ambient air temperature, *Ta*, was +10.6°C
- The weighted average heating season external ambient air temperature was +7.0°C
- The weighted average heating season ground temperature under the grass surface at 0.3m and 0.9m depths was +8.7°C +9.4°C respectively

The experimental test data allowed a series of heat transfer models to be evaluated. These models were combined into an excel based heat pump performance tool which showed:

- Optimum collector length of 1500m (7.3W/m) offers 3.7% improved seasonal performance over standard 750m (14.6W/m) collector size
- Optimum collector length absorbs 7.3W/m (VDI-4640 standard 14.6W/m)
- New horizontal collector performance indicator, *CPI*, delivers a benchmark value of 0.21K/(W/m^2) for optimised collector, in comparison to the *CPI* for the VDI-4640 standard collector which was only 0.43K/(W/m^2)
- Under a grass surface, a 1.2m deep collector shows a 1.6% improvement in seasonal performance over a 0.3m deep collector
- A split level collector positioned at 0.9m depth beneath a grass surface and 0.3m beneath a brick surface, utilising source side management, indicates a SPF improvement of 8.1% over a standard horizontal collector

While the horizontal collector design simulation tool may portray limited sophistication, it creates opportunities to develop and further refine the climate sensitive collector design with source side management techniques including sensoring/feedback, weather, ground temperature, heat flux measurement and thermal storage the potential exists for an 8-15% increase in performance.

CHAPTER 6 – $GSHP_{VC}$ AND ASHP: PERFORMANCE EVALUATION

The three-year long *HP-IRL* study also afforded an opportunity to experimentally investigate the performance of three vertical collectors described in Section 3.4. As the climate plays a diminished role in the performance of this collector type, attention on understanding the impact of operational parameters such as duty cycle on ground temperature drawdown/recovery rates and long term ground thermal depletion, not quantified by the standard Thermal Response Test (TRT).

6.1 VERTICAL COLLECTOR THERMAL CHARACTERISTICS

While the vertical collector consists of three 100m deep boreholes it is worth recalling observations made in Section 3.4 about the possible climate influences, geothermal gradients and soil type before reviewing experimental results.

6.1.1 CLIMATE INFLUENCES

To assess the potential of the climate to influence vertical collector performance it is possible to subdivide the collector depth into sections that have varying degrees of exposure to the climate. Table 3.9 shows that the *HP-IRL* vertical consisted of:

- 3% (24m) of collector pipe within the building which should absorb/release a negligible amount of thermal energy during operation
- 20% (150m) that fluctuates throughout the year similar to that of the horizontal collector at 1m depth
- 11% (84m) that fluctuates partially with the season to a depth of 15m
- 66% (510m) remains at a stable temperature throughout the year

Figure 6.1 illustrates the dampening of mean temperature variation with depth based on measurements to 1.8m, where the monthly average change in temperature at the surface was 14K and this diminishes to zero at 15m depth. Similar observations were published by Mihalakakou (1997) and Eugster (2000).

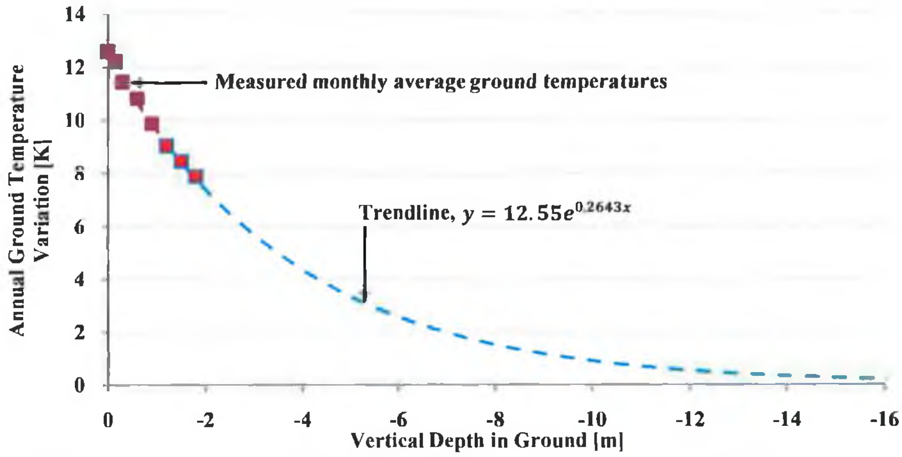


FIGURE 6.1 EXTRAPOLATION OF MEASURED MONTHLY AVERAGED DATA TO PREDICT GROUND TEMPERATURE VARIATION FROM MEAN TO A DEPTH OF 16M IN GALWAY.

The influence of seasonal climatic effects is shown for borehole 1 in Figure 6.2, where the temperature sensor located on the pipe wall at 5m deep, clearly shows the impact of winter and summer ambient air temperature changes during the heat pump off periods. However, no such climatic influence was evident at either 50m or 95m. Instead the influence of the 0.01°C/m geothermal gradient was noted with the sensor at 95m consistently reading 0.5°C higher than at 50m.

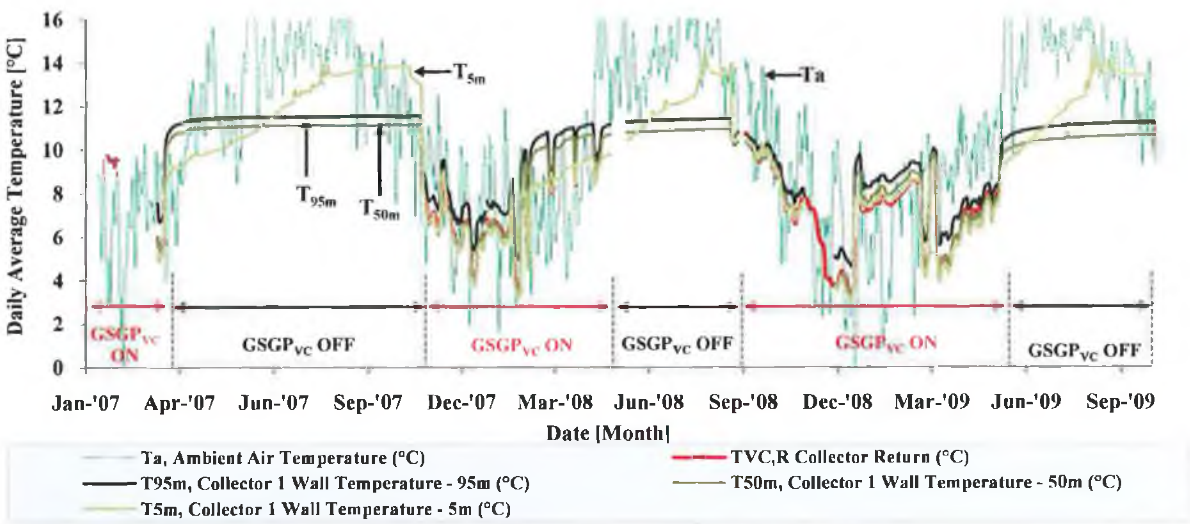


FIGURE 6.2 VERTICAL COLLECTOR (BOREHOLE 1) WALL TEMPERATURES AT 5M, 50M AND 95M BETWEEN 2007 AND 2009.

The thermal properties of the ground material surrounding the pipe are another factor influencing vertical collector thermal performance. Section 3.4 revealed that 55% of the entire *HP-IRL* vertical collector is surrounded by Limestone ($\lambda \approx 4$ W/m·K) between depths of 2m and 100m, 38% in soil/clay ($\lambda \approx 2.3$ W/m·K) from the surface to 2m and 6% within the IiBC ($\lambda \approx 0.025$ W/m·K).

When the GSHP$_{VC}$ was turned off for the summer the wall temperatures of the vertical collector were continually recorded during a 148 day long recovery period over the three years. At the end the wall temperature were recorded and are presented in Table 6.1.

TABLE 6.1 VERTICAL COLLECTOR GROUND TEMPERATURE RECOVERY

Test #	Year	Summer recovery duration (days)	Temperature after recovery period		
			5m	50m	95m
1	2007	148	+13°C	+11.1°C	+11.5°C
2	2008	148	+13.8°C	+10.9°C	+11.4°C
3	2009	148	+13.4°C	+10.7°C	+11.2°C

As indicated in Figure 6.2 and Table 6.1 over the course of the three heating seasons there was a 0.3°C reduction in the ground recovery at T95m, and 0.4°C at T50m. Combining these results as a portion (69%) of the overall collector there is a 0.25°C (0.08°C per year) temperature penalty, T_P (defined in Section 2.4.3) on the *HP-IRL* GSHP$_{VC}$, which is considered negligible.

6.1.2 GEOTHERMAL GRADIENTS

Geothermal gradients can play a significant role in the thermal performance of a vertical collector and heat pump system. While ambient air temperature is the baseline temperature that drives ground temperature from a depth of 15m, any temperature increase with depth below 15m is attributable to geothermal gradients. As described in Section 3.2 the 30 year average ambient air temperature for Galway is +10.2°C. Figure 6.3 shows the recorded collector pipe wall temperature at 5m, 50m and 95m over a period of six months following test period VC1 which are indicative of the ground temperatures at the specified depths.

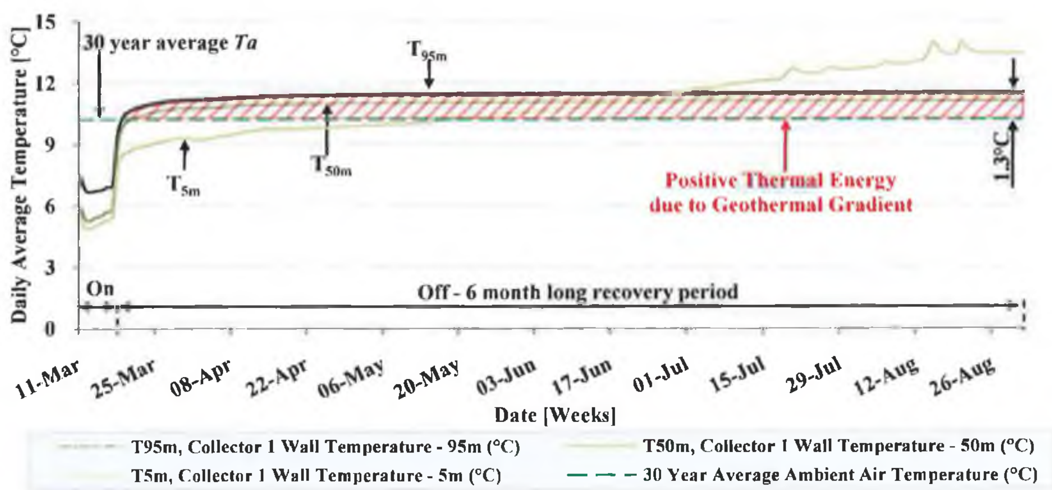


FIGURE 6.3 GEOTHERMAL GRADIENTS ALONG THE *HP-IRL* VERTICAL COLLECTOR DURING THE RECOVERY PERIOD FOLLOWING TEST VC1 (MARCH-AUGUST 2007).

Figure 6.3 shows a 1.4°C (±0.3°C) temperature difference between the 30 year average ambient air temperature and the stable ground temperature at a depth of 95m (+11.5°C).

From Table 6.1, there is also a 0.5°C difference between the temperatures at 50m and 95m. This reveals the influence of the geothermal gradient of 0.01°C/m identified in Section 2.4.3 which translates into a geothermal gradient of 1.1°C per 100m. The results compare well with the geothermal gradient map of Ireland developed by CSA (Table 2.6), with southern Ireland having a gradient of 1.6°C per 100m. Meanwhile, the pipe wall temperature at 5m depth shown in Figure 6.3 recovered to +13°C due to climatic influences.

6.1.3 Farfield temperature

As with the horizontal collector, it is also important to establish the "farfield" temperature ($T_{VC,\infty}$) from which the collector draws its energy. A recommended method for establishing this is to record and average the fluid temperature in the vertical collector within "off" periods through the year. This method of attaining a mean or effective ground farfield temperature for the GSHP$_{VC}$ collector is achieved by running the circulation pump without the heat pump being activated until the $T_{VC,R}$ stabilises and an example of such a test conducted in Luleå, Sweden, is presented in Figure 6.4.

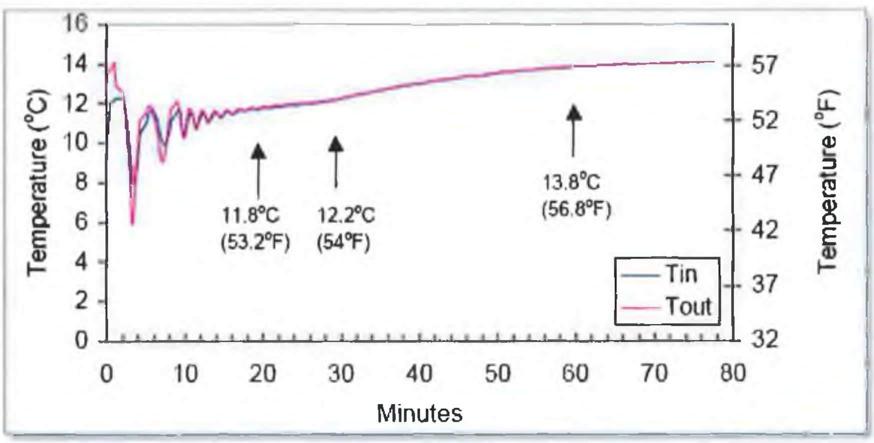


Figure 6.4 Vertical collector farfield ground temperature in Luleå Sweden (Gehlin, 2002).

Figure 6.5 shows the *HP-IRL* vertical collector farfield ground temperature test results.

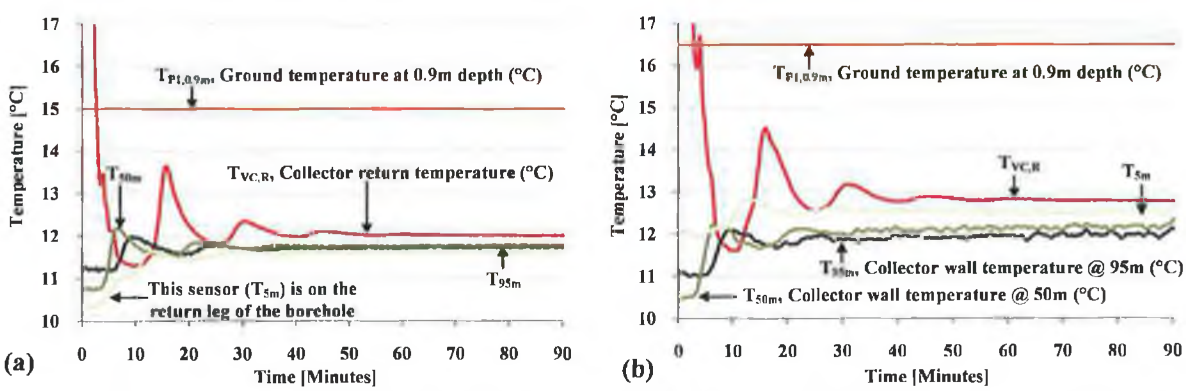


Figure 6.5 *HP-IRL* vertical collector (borehole 1) farfield ground temperature in (a) May 2008 and (b) July 2009.

Figure 6.5(a) shows the *HP-IRL* vertical collector farfield ground temperature recorded on the 29th of May 2008 after the $GSHP_{VC}$ was off for a period of two months prior to the test. Figure 6.5(b) shows the *HP-IRL* vertical collector farfield ground temperature recorded on the 28th of July 2009 after the $GSHP_{VC}$ was off for a period of two months prior to the test.

The steady state $T_{VC,R}$ values shown in Figures 6.5(a) and (b) are +12.0°C and +12.8°C respectively with higher return temperatures in July mainly attributable to higher ground temperatures at 1m deep, to which 20% of the collector is exposed.

From this analysis, values ranged between +10°C (January), +12°C (May), +12.8°C (July) and +10.4°C (November). Recording the maximum and minimum $T_{VC,R}$ values in this way allows the temperature range of $T_{VC,\infty}$ over an entire year to be established. Taking the maximum (+12.8°C) and minimum (+10°C) recordings, the amplitude of the annual farfield temperature fluctuates by 2.8°C (±1.4°C) and the annual average effective farfield temperature is therefore +11.4°C. This is represented graphically in Figure 6.6 as the predicted effective farfield temperature.

However, a more accurate assessment of the borehole farfield temperature can be conducted utilising the measured temperature data for the various depths at which the vertical collector operates. Using temperature data from Table 6.2 and weighting temperatures at the various depths by length of piping exposed to that temperature a measured effective farfield temperature can be generated, and as shown as the measured effective farfield temperature in Figure 6.6.

TABLE 6.2 GROUND/BOREHOLE TEMPERATURE VARIATION WITH DEPTH AROUND THE VERTICAL COLLECTOR (2007–2009)

Region	Depth (m)	Collector Piping Length (m)	Maximum Temperature	Minimum Temperature	Average Temperature
1	1m	150m (20%)	+17°C	+5.5°C	+11.3°C
2	1m to 15m	84m (11%)	+14°C	+8.5°C	+11.4°C
3	15m to 50m	210m (28%)	-	-	+11.1°C
4	50m to	300m (40%)	-	-	+11.5°C
Total Borehole annual weighted average farfield Temperature:					**+11.4°C**

All vertical collector wall temperatures at 5m, 50m and 95m were recorded when the heat pump was off for at least one month prior to testing. The control profile temperature sensor $T_{P1,0.9m}$ is used to represent the temperature at 1m depth, at which depth the vertical collector is situated running from the IiBC building to the boreholes.

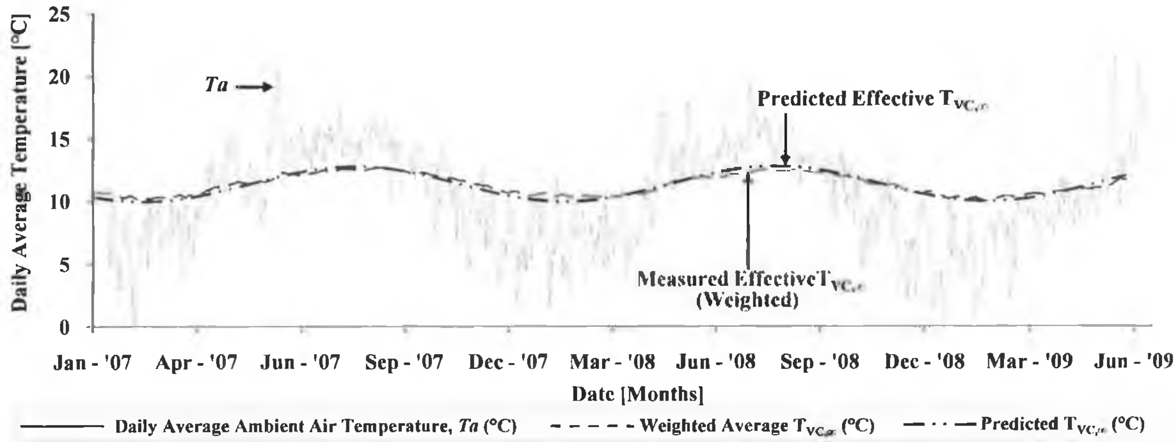


FIGURE 6.6 ESTIMATING *HP-IRL* VERTICAL COLLECTOR EFFECTIVE FARFIELD TEMPERATURE.

As 20% of the vertical collector is situated in the ground at a depth of 1m (Region 1), running from the building to the three boreholes and is uninsulated, there is a partial climatic influence on the overall farfield temperature that shows a variation of farfield temperature ($T_{VC,\infty}$) with changes in ambient air temperature (Ta).

This section has utilised temperature sensors located in the ground at 1m depth ($T_{P1,0.9m}$) and sensors located on the vertical collector pipe wall to identify the climatic influences on collector return temperature as 31% of the overall collector length is within 15m of the ground surface. This has allowed a year-round effective farfield temperature of 11.4°C to be established and varies between 10°C and 12.8°C. For the purpose of determining the vertical collector ground temperature drawdown ($\Delta T_{VC,G}$) the measured (weighted) effective farfield temperature shown in Figure 6.6 is used.

6.2 GSHP$_{VC}$ TEST PROGRAM (2007 – 2009)

This section presents the extent of the 289 day long test program undertaken using the vertical collector. It identifies the rationale for each of the eight individual tests conducted within this period and presents key findings.

6.2.1 GSHP$_{VC}$ TEST PROGRAM RATIONALE

The goal of the comprehensive test program was to identify heat pump performance during different seasons and applications. Hence the test program reflected a wide range of demands (10% - 100%) and run times (1 – 96 days) and the eight individual test periods identified in Figure 6.8 were discussed. The timing and duration of each test period along with the heat pump operation time, or duty, is presented in Figure 6.7. It is evident that the heat pump was operated during autumn, winter and spring and for duties between 10% and 100%.

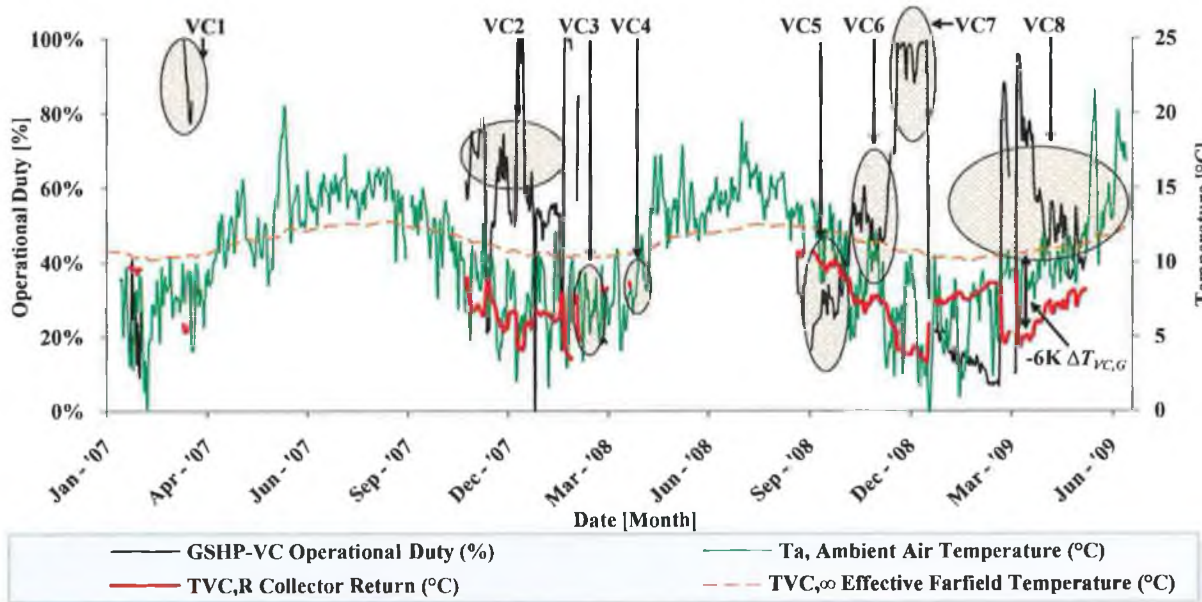


FIGURE 6.7 TIMING OF THE EIGHT GSHP$_{VC}$ TEST PERIODS (2007-2009) AND CORRESPONDING DUTY.

Table 6.3 presents the demand and duration of each test period along with a brief description of the analysis conducted during this period.

TABLE 6.3 CHARACTERISTICS OF EIGHT GSHP$_{VC}$ TEST PERIODS (2007-2009)

Test Period	Demand	Term	Duration	Application	Description
VC1	Intensive	Short	8 days	Domestic/Commercial	First IiBC heating season observational period with intensive thermal extraction rates.
VC2	Moderate	Long	96 days	Domestic	Comparative heat pump operation period with the GSHP$_{HC}$ and GSHP$_{VC}$ in simultaneous operation.
VC3	Intensive	Short	3 days	Domestic/Commercial	Evaluating the BHE thermal behaviour under a intensive thermal extraction over 3 days. Indicative of a typical winter period of reduced ambient air temperature, inducing a high domestic building thermal demand.
VC4	Low	Short	1 days	Domestic/Commercial	Evaluating the BHE thermal behaviour under a low intensity, one day thermal extraction period.
VC5	Low	Long	43 days	Domestic	Prolonged steady state low level thermal extraction, indicative of autumn/spring time domestic dwelling utilisation.
VC6	Moderate	Long	45 days	Domestic	Steady state moderate level thermal extraction, indicative of extreme winter time domestic dwelling utilisation.
VC7	Intensive	Medium	30 days	Commercial	Steady state intensive level thermal extraction.
VC8	Moderate	Long	62 days	Commercial	Prolonged steady-state moderate thermal extraction.

Some of the expected initial observations from Figure 6.7 are:

- The effective farfield temperature varies by ±1.4°C the year
- The heat pump operational duty is proportional to ambient air temperature as shown by VC5, VC6 and HC7 in Figure 6.7
- The drawdown on the collector temperature ($\Delta T_{VC,G}$) is a function of the heat pump duty as evidenced by comparing VC5, 6 and 7 in Figure 6.7

The ground temperature drawdown ($\Delta T_{VC,G}$) is the difference between the vertical collector fluid return temperature ($T_{VC,R}$) and the effective farfield temperature ($T_{VC,\infty}$) along the borehole ($\Delta T_{VC,G} = T_{VC,R} - T_{VC,\infty}$). $T_{VC,\infty}$ was defined in Section 6.1.3.

6.2.2 INITIAL FINDINGS

Over the course of the eight test program the vertical collector operated for 289 days between 2007 to 2009, delivering 74,011 kWh of energy (266 GJ), which is the equivalent of five years of space heating for a domestic dwelling (12,000 – 15,000kWh/annum). The energy extraction by the collector generated an overall drawdown of -4.1K on the ground source farfield temperature and the heat pump delivered an average heat pump sink temperature of +49.5°C and a COP ranged from 2.8 to 3.4. The overall average SPF was 2.95 for the $GSHP_{VC}$ compared to the $GSHP_{HC}$ SPF of 2.90.

Table 6.4 outlines the vertical collector operational results recorded within each test period.

TABLE 6.4 SUMMARY OF KEY $GSHP_{VC}$ TEST PERIOD RESULTS (2007-2009)

Test #	Dates	Days	Operational time	Total thermal extraction (kWh)	Average collector pipe extract rate [W/m]	Coefficient Of Performance, COP [-]*
VC1	11/03/07 – 18/03/07	8	87%	1,573 kWh	25.2 W/m	2.85 (3.16)
VC2	19/11/07 – 22/02/07	96	62%	16,207 kWh	19.4 W/m	2.90 (3.20)
VC3	27/02/08 – 01/03/08	3	84%	641 kWh	25.7 W/m	2.77 (3.05)
VC4	14/04/08 – 15/04/08	1	34%	115 kWh	12.9 W/m	3.36 (3.72)
VC5	11/09/08 – 23/10/08	43	28%	3,686 kWh	9.9 W/m	3.18 (3.40)
VC6	24/10/08 – 08/12/08	45	52%	6,744 kWh	16.8 W/m	3.01 (3.20)
VC7	09/12/08 – 06/01/09	30	94%	6,723 kWh	27.1W/m	2.80 (2.99)
VC8	24/03/09 – 25/05/09	62	44%	9,787 kWh	15.5 W/m	2.77 (3.04)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-14511 Test Standard

The COP improves with a reduction in the overall heat pump temperature lift (ΔT_{HP}), which varied from 38.8K (Test period VC4) to 44.8K (Test period VC8).

6.3 $GSHP_{VC}$ EXPERIMENTAL EVALUATION

The transient response of the vertical collector to variations in both the heat pump duty cycle and test duration the resulting impact on heat pump performance are described in this section. The following sub-sections present the effect of duty and duration on the vertical collector ground temperature drawdown, steady-state operation and ground temperature recovery.

6.3.1 DRAWDOWN

This section evaluates the ground temperature drawdown under various operational loads. As outlined previously the recommended ground temperature drawdown, $\Delta T_{VC,G}$, should not exceed -12K for base load conditions and -18K for peak load conditions (VDI 4640 / Part 2, 2001; Reub and Sanner, 2001) and this internationally accepted drawdown is compared against measured data from this *HP-IRL* study.

As the ground temperature drawdown is a measure of both collector and heat pump performance, Figure 6.8 presents the transient ground temperature drawdown profiles for six of the *HP-IRL* test periods, spanning low intensity, short-term tests (VC4) to moderate intensity, long-term tests (VC8).

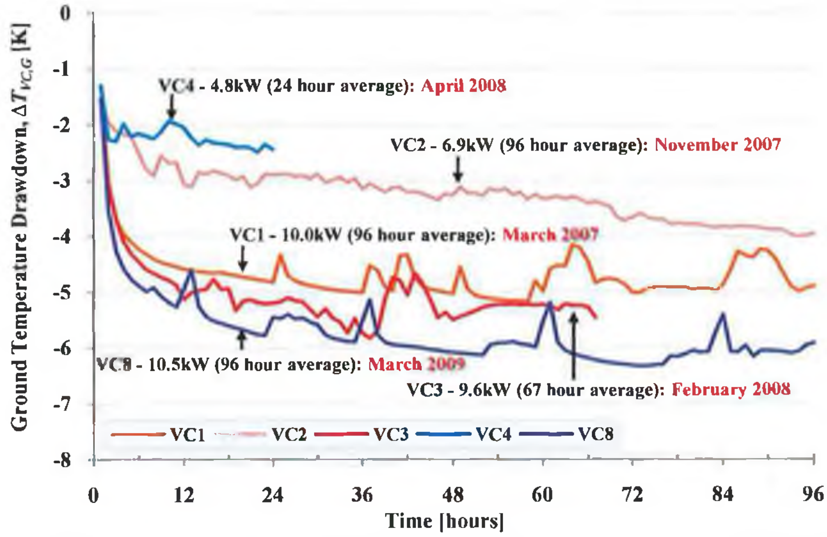


FIGURE 6.8 HOURLY AVERAGED GROUND TEMPERATURE DRAWDOWN FOR FIVE PERIODS VC1, 2, 3, 4 & 8.

Figure 6.8 illustrates the variation in drawdown with thermal extraction rate, where the intensive (10.5kW) test period VC8 generates a ground temperature drawdown of -6.1K after 60 hours, compared with just -3.3K for the moderate (6.9kW) test period VC2.

Figure 6.9 characterises the ground temperature drawdown for test periods VC5, VC6 and VC7, which have low, moderate and intensive thermal extraction rates respectively.

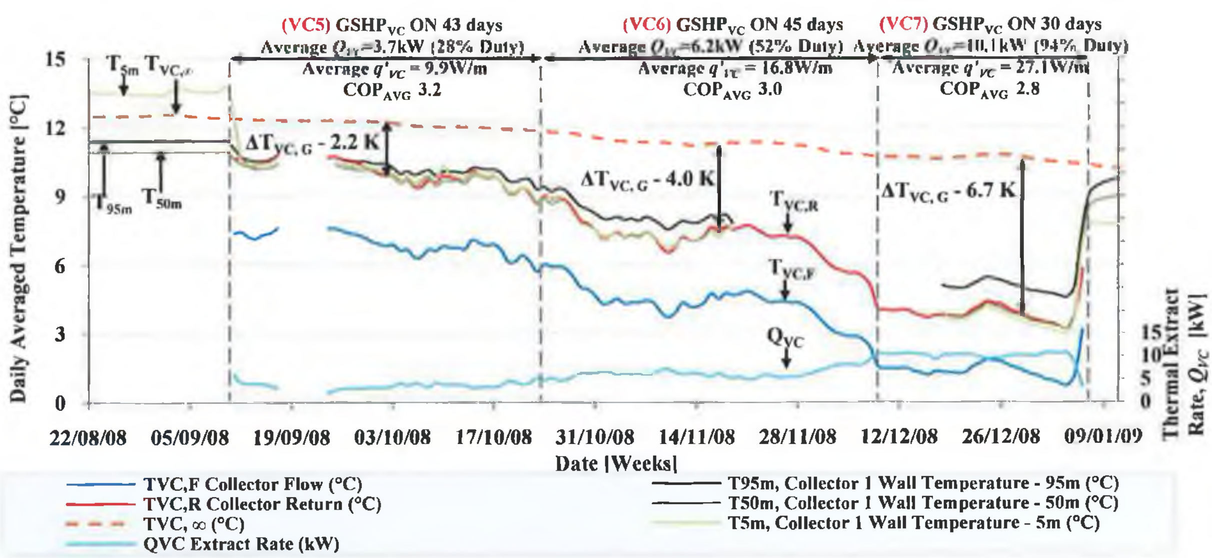


FIGURE 6.9 DAILY AVERAGED GROUND TEMPERATURE DRAWDOWN DURING THE CONSECUTIVE LOW, MODERATE AND INTENSIVE VERTICAL COLLECTOR THERMAL EXTRACTION PERIODS VC5, 6 AND 7.

Table 6.5 summarises the recorded test parameters during test periods VC5, 6 and 7.

TABLE 6.5 SUMMARY OF THE IMPACT OF VARYING THE HEAT EXTRACTION RATE FROM THE VERTICAL COLLECTOR DURING TEST PERIOD VC5, 6 AND 7 IN FIGURE 6.9

Test #	Collector Extract Rate, Q_{VC}	Borehole Extract Rate, q'_{VC}	Average $\Delta T_{VC,G}$	Average $\Delta T_{VC,B}$	Average $T_{VC,R}$	Average $T_{HP,F}$	Coefficient Of Performance, COP [-]*
VC5	3.7kW	9.9W/m	- 2.2K	3.0K	+9.9°C	+49.5°C	3.18 (3.40)
VC6	6.2kW	16.8W/m	-4.0K	2.9K	+7.3°C	+49.7°C	3.01 (3.20)
VC7	10.1kW	27.1W/m	-6.7K	2.5K	+3.9°C	+49.2°C	2.80 (2.99)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

Observable from Figure 6.9 is the step increase in ground temperature drawdown with increased thermal extraction. The maximum ground temperature drawdown recorded was –7.4K was recorded during test period VC7, with a thermal extract rate of 11.4kW (30.6W/m). This compares favourably with the -12K drawdown recognised by the standards.

6.3.2 STEADY-STATE OPERATION

This section presents steady-state behaviour for the range of thermal extraction spanning 4.7kW to 9.6kW. Figure 6.10 illustrates the results of the three day test period VC3, and Table 6.6 presents a summary of the key findings.

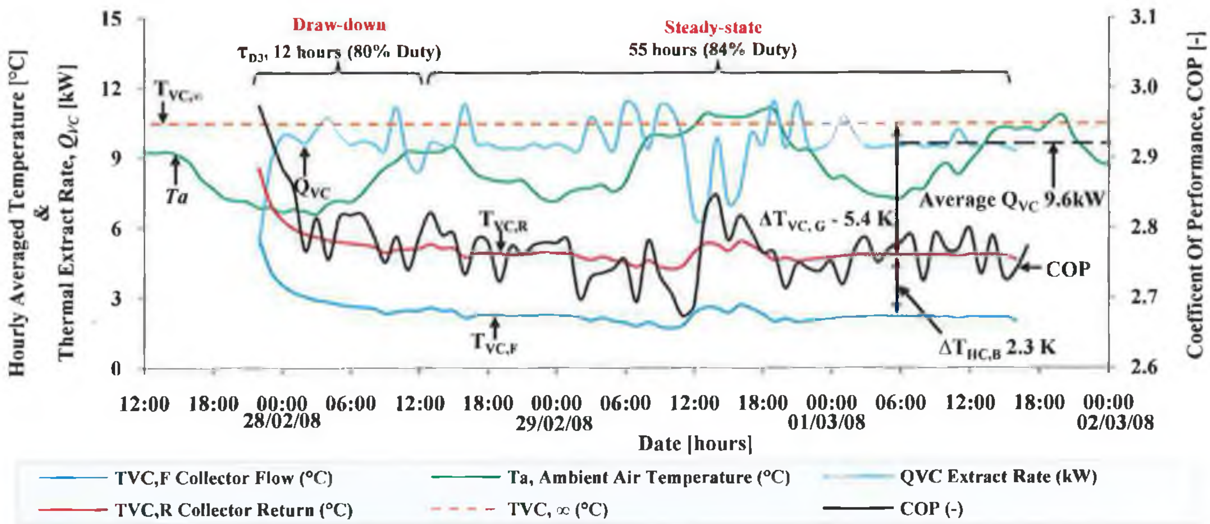


FIGURE 6.10 HOURLY AVERAGED THERMAL DRAWDOWN AND STEADY-STATE PERFORMANCE DURING TEST PERIOD VC3.

Figure 6.10 shows the average 9.6kW thermal extraction from the vertical collector, maintaining an average -5.4K $\Delta T_{VC,G}$ during the 55 hour long steady state thermal extraction.

The primary results of test period VC3 are presented in Table 6.6 and Table 6.7 presents a summary of the secondary indicators of heat pump performance.

TABLE 6.6 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD VC3 (SHORT-TERM, INTENSIVE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{VC,g}$	Average $T_{VC,\infty}$
VC3	27/02/08 – 01/03/08	3	84%	9.6 kW	+5.0°C	+10.4°C

TABLE 6.7 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD VC3

Test #	Average $\Delta T_{VC,G}$	Average borehole extract rate [W/m]	Total kWh extracted from the Vertical Collector	Coefficient Of Performance, COP [-]*
VC3	- 5.4K	25.7 W/m	640	2.77 (3.05)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period VC3 was +49.8°C, with a temperature lift (ΔT_{HP}) of +44.8 K, delivering a COP$_{AVG}$ of 2.8. The average $\Delta T_{VC,G}$ for test period VC3 was -5.4K.

Test period VC3 illustrates the thermal performance of the vertical collector under intensive thermal extract rates (9.6kW) over short term durations, and characterises the near maximum steady-state condition the vertical collector will endure.

Figure 6.11 shows the ground temperature drawdown of the vertical collector along with the vertical collector pipe wall temperatures at depths of 5m, 50m and 95m during the seven day test period VC1.

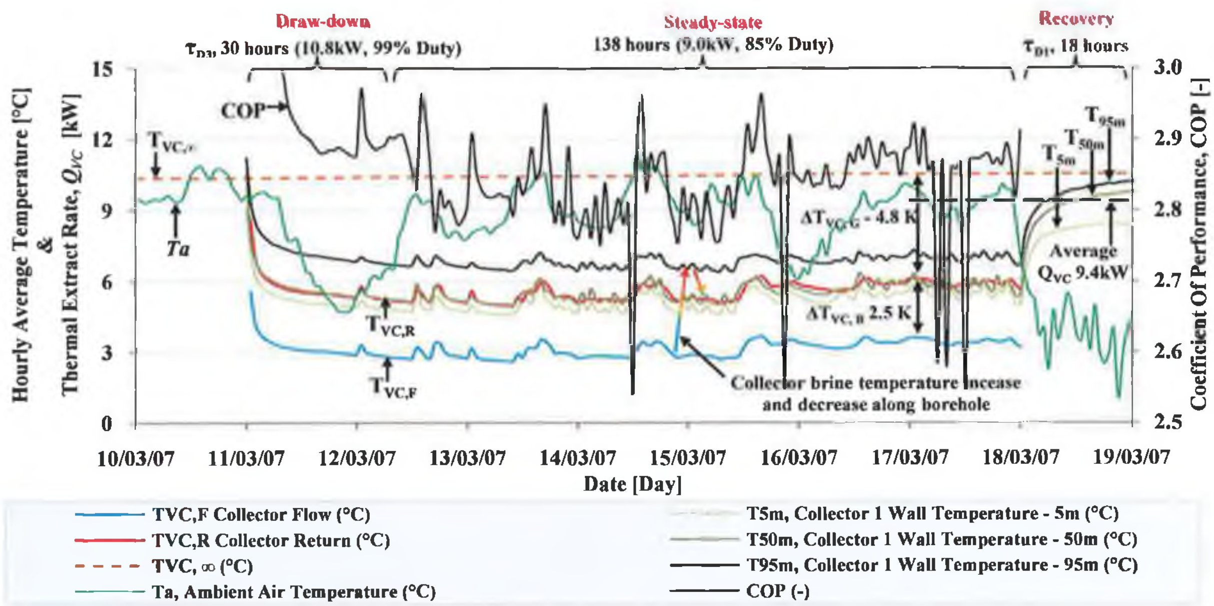


FIGURE 6.11 HOURLY AVERAGED THERMAL DRAWDOWN, STEADY-STATE AND RECOVERY PERIODS DURING TEST PERIOD VC1.

Figure 6.11 shows the average 9.4kW thermal extraction from the vertical collector, maintaining a $\Delta T_{VC,G}$ of -4.8K steady state thermal extraction for six days. The primary results of test period VC1 are presented in Table 6.8 and Table 6.9 presents a summary of the secondary indicators of heat pump performance.

TABLE 6.8 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD VC1 (SHORT-TERM, INTENSIVE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{VC,R}$	Average $T_{VC,\infty}$
VC1	11/03/07 – 18/03/07	7	87%	9.4 kW	+5.6°C	+10.4°C

TABLE 6.9 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD VC1

Test #	Average $\Delta T_{VC,G}$	Average borehole extract rate [W/m]	Total kWh extracted from the Vertical Collector	Coefficient Of Performance, COP [-]*
VC1	- 4.8K	25.2 W/m	1,573	2.85 (3.16)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during test period VC1 was +48.9°C, with a temperature lift (ΔT_{HP}) of +43.3K, delivering a COP_{AVG} of 2.9.

From Figure 6.11 it can be seen that the vertical collector wall temperatures at 5m and 50m are similar to that of the collector fluid return temperature, $T_{VC,R}$. The temperature sensor T_{5m} is located on the return leg of the borehole. This shows that the ground surrounding the collector at these points is 1°C lower than the collector fluid. However, it is noticeable that the collector wall temperature at 95m remains 2°C higher than $T_{VC,R}$. As there was no insulation placed between flow and return legs of the borehole a portion of this temperature difference between T_{95m} and $T_{VC,R}$ may result from thermal interaction between the collector pipes within the borehole itself, where the flow pipe down the borehole cools the return pipe coming up. A suggested method for avoiding this thermal interference is to insulate between the upward and downward flow pipes at the top of the collector, ranging in depth from 5 to 15m (Gehlin, 2002).

Therefore the reasons for the reduced temperature from the optimum temperature at T_{95m} are:

- Thermal interaction between downward and upward flow legs of collector pipe
- Heat loss to ground at <15m deep, corresponding to 31% of collector length
- Uncertainty if collector fluid actually reaches T_{95m}

Therefore there is thermal savings to be made by not only insulating the collector return pipes from 15m deep to the surface, it is also important to insulate the return leg from the boreholes to the building. This has the potential to raise the collector return temperature by 1 - 2°C, corresponding to a 2.5 - 5% improvement in COP.

From the eight tests conducted, test periods VC1, 2, 3 and 4 developed steady-state ground temperature drawdown conditions and details of which are presented in Table 6.10.

TABLE 6.10 DRAWDOWN, STEADY-STATE THERMAL EXTRACTION AND RECOVERY FOR VC1, 2, 3 AND 4

	Draw-down					Steady-state								Recovery (T_{50m})	
	Q_{HC} (kW)	τ_{D3} (Hours)	$T_{VC,\infty}$ (°C)	$T_{HP,F}$ (°C)	COP_{AVG} (-)	Steady-state duration (hours)	Q_{HC} (kW)	$\Delta T_{HC,G}$ (K)	$T_{VC,D}$ (°C)	$T_{VC,R}$ (°C)	$T_{HP,F}$ (°C)	ΔT_{HP} (K)	COP_{AVG} (-)	τ_R (Hours)	$T_{VC,\infty}$ (°C)
VC1	10.0	30	+10.1	+47.1	2.98	138	9.0	-4.9	+10.2	+5.6	+49.3	43.7	2.84	τ_{R1} - 18	+10.2
VC2	7.1	120	+11.2	+49.5	2.83	310	8.1	-4.3	+11.0	+6.8	+49.6	42.8	2.78	τ_{R1} - 54	+10.0
VC3	9.4	12	+10.0	+49.8	2.84	55	9.6	-5.2	+10.0	+4.8	+49.8	45.0	2.76	τ_{R1} - 40	+10.1
VC4	4.9	14	+10.7	+46.8	3.41	10	4.7	-2.4	+10.7	+8.3	+48.0	39.7	3.29	τ_{R1} - 26	+10.7

Note: Steady-state conditions are taken at a time equal to three time constants $\tau_{D3}=3\tau$, or 95% of final steady-state condition.

The results of the steady-state thermal extraction are illustrated in Figure 6.12.

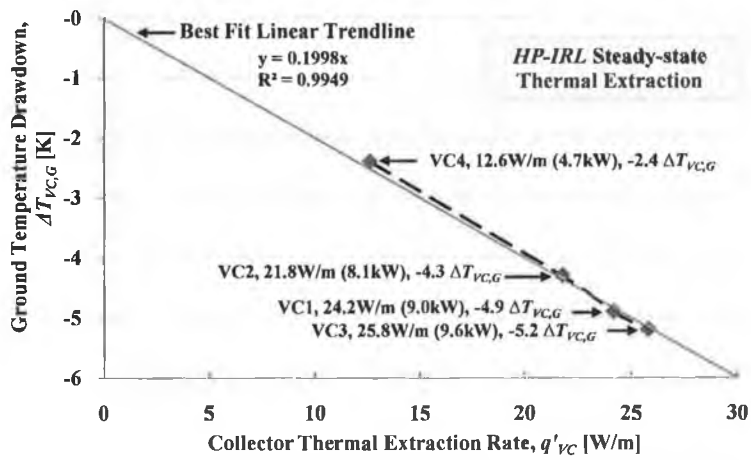


FIGURE 6.12 GROUND TEMPERATURE DRAWDOWN ($\Delta T_{VC,G}$) RECORDED UNDER STEADY-STATE THERMAL EXTRACTION CONDITIONS DURING TEST PERIODS VC1, 2, 3 AND 4.

The results indicate under steady-state thermal extraction the ground temperature drawdown is a linear function of collector thermal extraction. It also shows a much lower ground temperature drawdown than the nominal -12K indicated in the VDI 4640 standard.

6.3.3 RECOVERY

The ground's thermal recovery, and rate of recovery, is a key component of the vertical collector efficiency as it draws in energy from the surrounding area.

Figure 6.13 illustrates the thermal recovery of the vertical collector for test periods VC1 to VC8. The temperature recovery, $\Delta T_{VC,R}$, is the difference between the initially established temperature of +11.1°C at 50m depth shown in Table 6.1 and the actual collector wall temperature at T_{50m} which is the collector midpoint.

What is noticeable from Figure 6.13 is the slow recovery after test period VC8 in comparison the fast recovery after test period VC1. VC1 sustained a thermal extraction rate of 9.4kW for 8 days and was the initial test period, in comparison VC8 was the last test period and had sustained a thermal extraction rate of 5.8kW for 62 days. The slowest ground temperature recovery is shown to be subsequent to test period VC8. This illustrates the effect of a prolonged moderate intensity thermal extraction, along with the temperature penalty of 0.25°C after three years of operation.

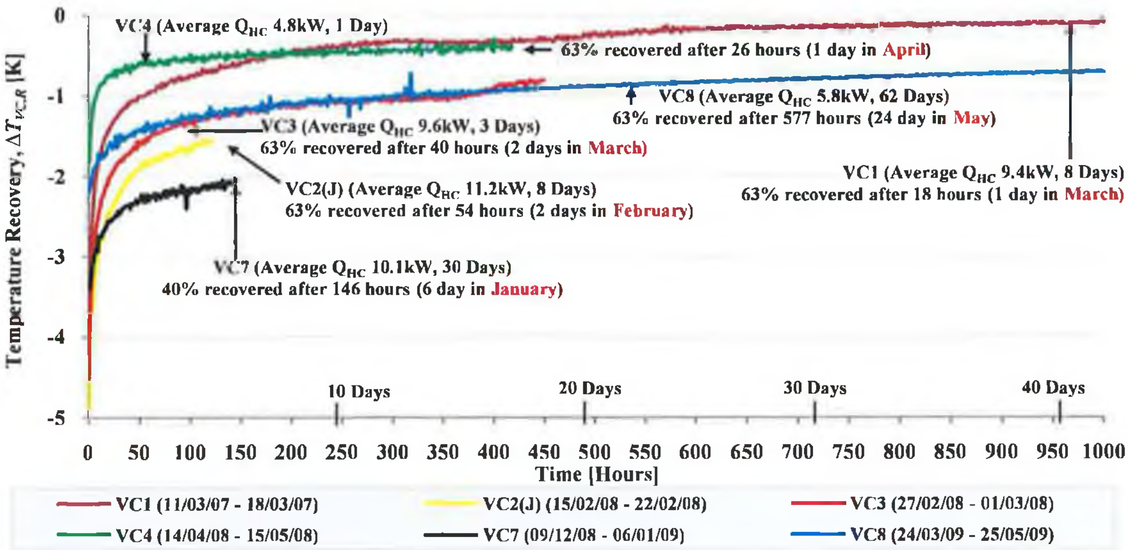


FIGURE 6.13 THERMAL RECOVERY OF THE VERTICAL COLLECTOR AT A DEPTH OF 50M AFTER TEST PERIODS VC1 TO VC8.

As test period VC1 was the first evaluation of the collector performance, Figure 6.14 presents drawdown and recovery during and after VC1 and shows that T_{50m} adequately reflects the recovery rate of the borehole.

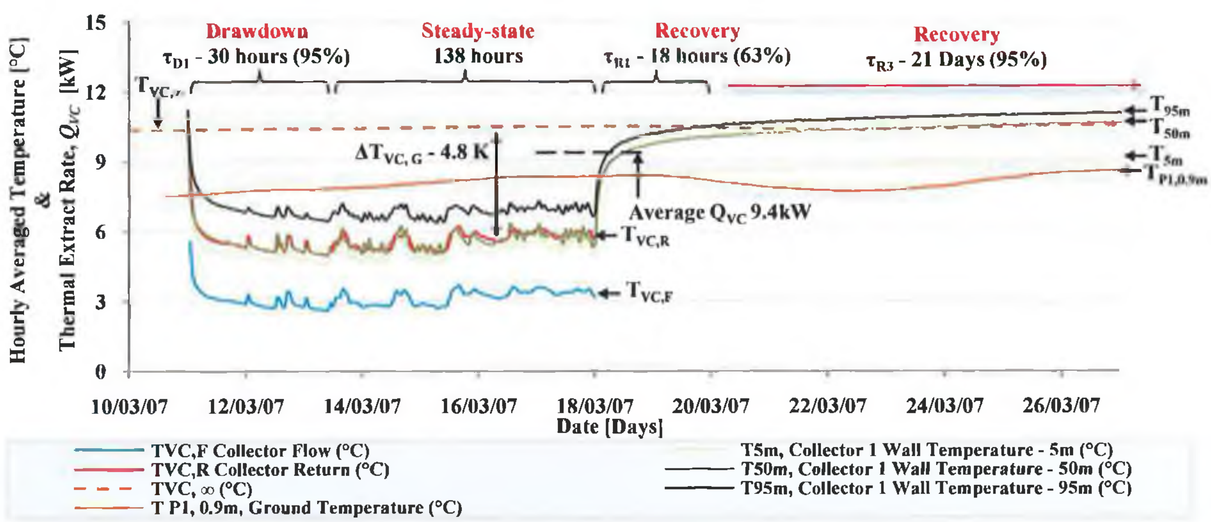


FIGURE 6.14 HOURLY AVERAGED BOREHOLE THERMAL DRAWDOWN, STEADY-STATE AND RECOVERY AFTER INTENSIVE TEST PERIOD VC1.

The 1 day (30 hour) borehole thermal recovery to τ_{R1} after the initial test period VC1 shown in Figure 6.14 is contrast with the 24 day borehole thermal recovery to τ_{R1} after the final test period VC8 shown in Figure 6.15. The ability of the borehole to recover thermally is affected by the prolonged thermal extraction period of 63 days with an average thermal extract rate of 5.8kW and compounded by the temperature penalty effects of three years operation.

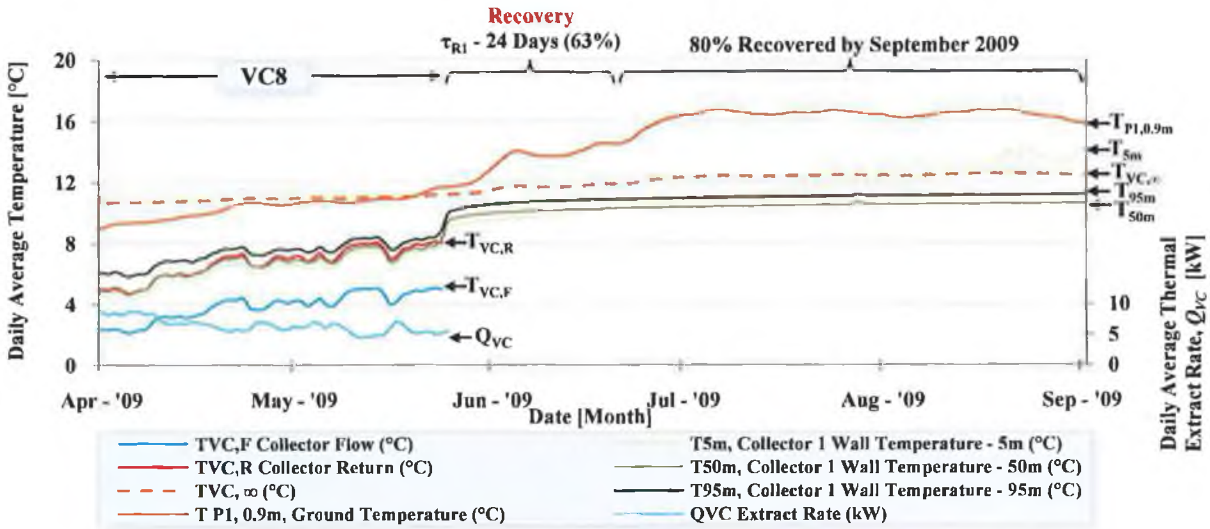


FIGURE 6.15 TEST PERIOD VC8 AND VERTICAL BOREHOLE THERMAL RECOVERY AFTER THREE CONSECUTIVE YEARS OPERATION.

6.3.4 LONG-TERM THERMAL DRAWDOWN AND RECOVERY

This section mimics the variable thermal demand profile of a typical domestic dwelling and monitors the collector response to long-term thermal extraction imposed during test period VC2. This test was run for 96 days and delivered an average duty of 62%.

Figure 6.16 breaks down test period VC2 into ten time periods from VC2(A) to VC2(J) with varying thermal extraction rates and ground temperature drawdown.

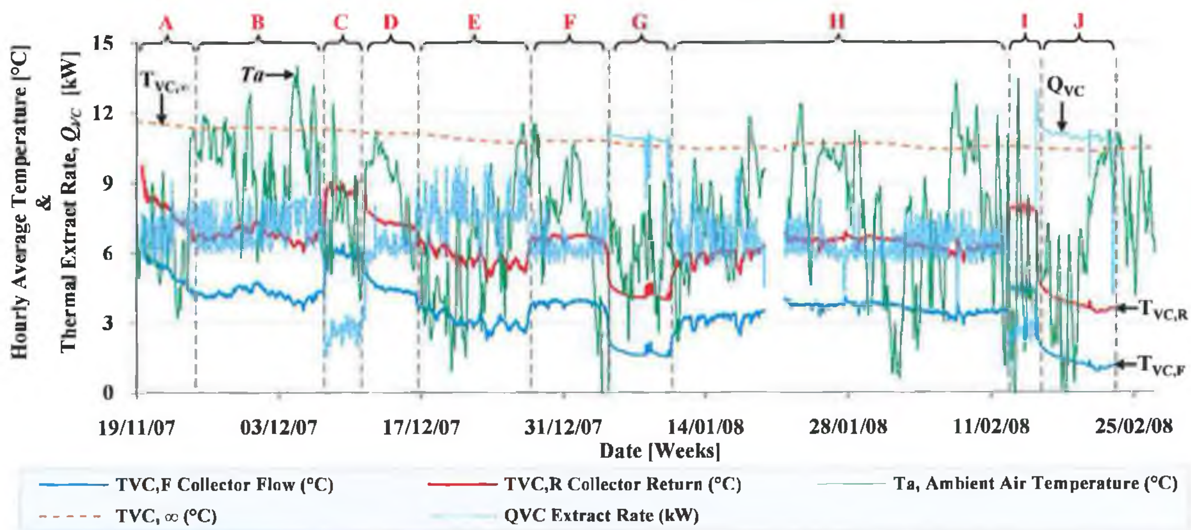


FIGURE 6.16 LONG-TERM VARIABLE THERMAL EXTRACTION DURING TEST PERIOD VC2.

Noticeable from Figure 6.16 is the change in farfield temperature from 11.6°C at the beginning of the test period to 10.4°C at the end. Figure 6.16 shows the 10 consecutive and different thermal extraction rates that generate ground temperature drawdown's ($\Delta T_{VC,G}$) that vary between -2.6K (VC2-C) and -6.4K (VC2-J) for thermal extraction rates of 2.6kW and

11kW respectively. The variable extraction rates reflect changes in external ambient air temperature and Table 6.11 gives a breakdown of the test period VC2.

TABLE 6.11 TEST PERIOD VC2 GROUND TEMPERATURE DRAWDOWN AND STEADY-STATE PERFORMANCE STATISTICS UNDER VARIOUS THERMAL EXTRACT RATES

Test #	Test Duration	kWh Extracted	Average Q_{VC}	Steady-state Q_{VC}	Average $\Delta T_{VC,G}$	Steady-state $\Delta T_{VC,G}$
VC2(A)	104 hours (4 days)	720	6.9kW	-	-3.5K	-
VC2(B)	324 hours (14 days)	2,646	7.1kW	7.1kW	-4.5K	-4.5K
VC2(C)	96 hours (4 days)	250	2.6kW	2.6kW	-2.6K	-2.6K
VC2(D)	118 hours (5 days)	735	6.2kW	6.2kW	-3.7K	-4.0K
VC2(E)	260 hours (11 days)	2,027	7.8kW	7.8kW	-5.1K	-5.1K
VC2(F)	192 hours (8 days)	1,224	6.4kW	6.4kW	-4.2K	-4.2K
VC2(G)	146 hours (6 days)	1,578	10.8kW	10.6kW	-6.3K	-6.4K
VC2(H)	795 hours (33 days)	5,134	6.5kW	6.5kW	-4.3K	-4.3K
VC2(I)	64 hours (3 days)	165	2.6kW	2.6kW	-2.6K	-2.6K
VC2(J)	180 hours (8 days)	1,987	11.0kW	11.0kW	- 6.4K	-6.7K

The results of Table 6.13 show the test period average and steady-state thermal extraction rate and ground temperature drawdown. The steady-state condition was τ_{D3}, and shows a consistent increase in ground temperature drawdown with thermal extraction.

Figure 6.17 takes a closer look at the relationship between ground temperature ($T_{VC,\infty}$), collector return temperature ($T_{VC,R}$) and the thermal extraction rate (Q_{VC}) for extraction periods H, I and J in Figure 6.16.

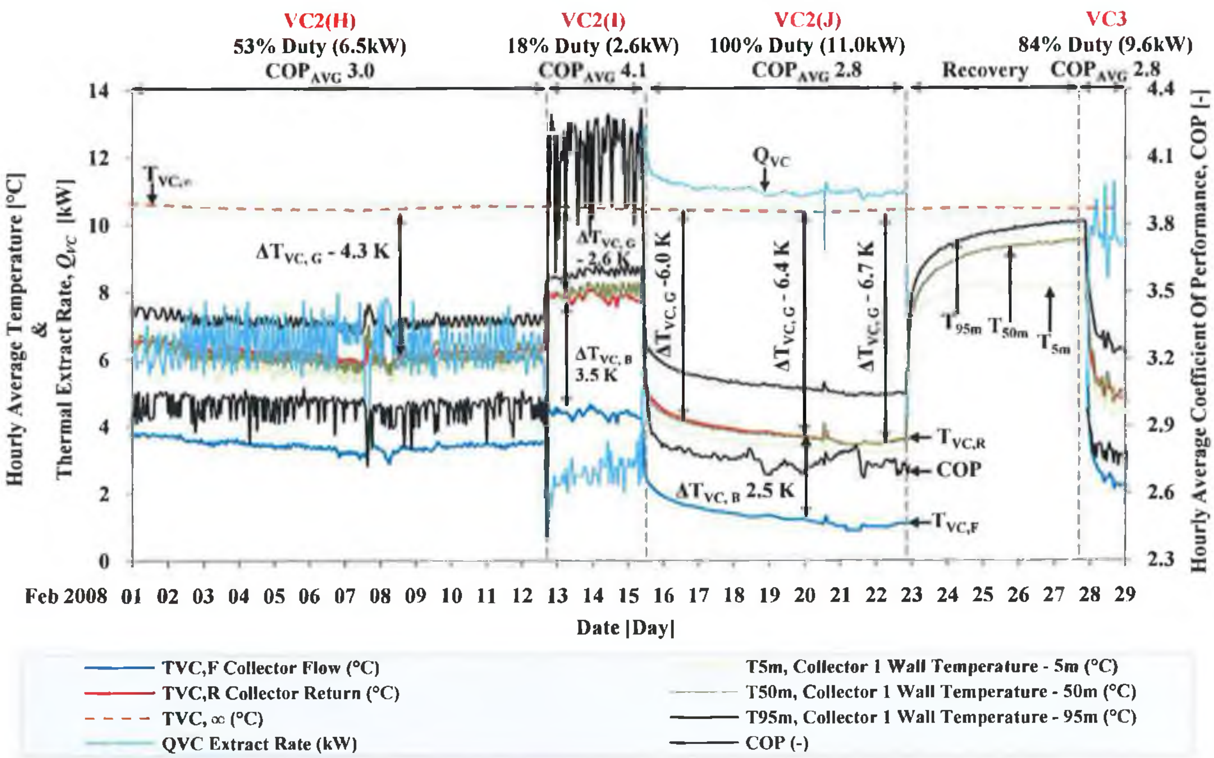


FIGURE 6.17 THERMAL DRAWDOWN AND RECOVERY OF THE GROUND SURROUNDING THE GSHP$_{VC}$ VERSUS HEAT PUMP PERFORMANCE DURING VC 2 AND VC3.

Noticeable is the rapid recovery of the borehole at 95m depth in comparison to 5m depth between test period VC2(J) and VC3, which is attributable to the impact of colder winter ambient air temperatures penetrating down to 5m. Table 6.12 provides a detailed analysis of heat pump performance for the four periods of vertical collector operation shown in Figure 6.17. Average COPs ranged between 2.8 at +50°C output (VC2-J) and 4.1 at +35°C output (VC2-I).

TABLE 6.12 SUMMARY OF THE IMPACT OF VARYING THE HEAT EXTRACTION RATE FROM THE VERTICAL COLLECTOR DURING FEBRUARY 2008, AS PRESENTED IN FIGURE 6.17

Test #	Extract Rate, Q_{VC}	$\Delta T_{VC,G}$	$\Delta T_{VC,B}$	Average $T_{VC,R}$	Average $T_{HP,F}$	Average ΔT_{HP}	COP_{AVG}	COP_{MAX}	COP_{MIN}
VC2(H)	6.5kW	- 4.3K	2.8K	+6.2°C	+50.0°C	43.8K	2.98	3.08	2.72
VC2(I)	2.6kW	-2.6K	3.5K	+7.8°C	+35.1°C	27.3K	4.07	4.31	3.59
VC2(J)	11.0kW	-6.4K	2.5K	+3.9°C	+49.2°C	45.3K	2.76	3.35	2.67
VC3	9.6kW	-5.4K	2.6K	+5.0°C	+49.8°C	44.8K	2.77	2.97	2.67

The primary results of the total test period VC2 are presented in Table 6.13 and Table 6.14 presents a summary of the secondary indicators of heat pump performance.

TABLE 6.13 SUMMARY OF PRIMARY RESULTS OBTAINED FROM TEST PERIOD VC2 (LONG-TERM, MODERATE)

Test #	Dates	Days	Test period Duty	Average Collector Extract Rate	Average $T_{VC,R}$	Average $T_{VC,\infty}$
VC2	19/11/07 – 22/02/08	96	62%	7.2 kW	+5.9°C	+10.7°C

TABLE 6.14 SUMMARY OF SECONDARY HEAT PUMP PERFORMANCE INDICATORS FOR TEST PERIOD VC2

Test #	Average $\Delta T_{VC,G}$	Average borehole extract rate [W/m]	Total kWh extracted from the Vertical Collector	Coefficient Of Performance, COP [-]*
VC2	- 4.8K	19.4 W/m	16,206	2.90 (3.20)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

The average heat pump flow temperature ($T_{HP,F}$) during the total test period VC2 was +49.5°C, with a temperature lift (ΔT_{HP}) of +43.6K, delivering a COP_{AVG} of 2.9.

6.3.5 LINE SOURCE EVALUATION OF VERTICAL COLLECTOR

Numerical models of the vertical collector operation has generally utilised line source theory, as it offers the capacity to model constant heat injection/extraction per unit length of a small diameter pipe in an infinite medium (Gehlin, 2002). There are two main approaches utilised: the line and cylindrical source models. The cylindrical source model was established by Carslaw and Jaeger and later presented by Ingersoll (Ingersoll and Plass, 1948; Ingersoll *et al.*, 1948, 1954). However, for this vertical collector application, the line source model is more appropriate since it assumes a constant heat injection/extraction per unit length of a heat source, or sink, via a small diameter pipe in an infinite medium. This method was first proposed by Ingersoll and Plass (Ingersoll and Plass, 1948) who elaborated on Kelvin's original line source theory (Kelvin, 1882). For an approximation of the heat extracted by a heat exchanger (vertical collector), a transient process can be approximated by the following equation where the fluid temperature, $T_f(t)$, can be developed as a function of time:

$$T_f(t) = \frac{Q}{4\cdot\pi\cdot\lambda_G}\cdot\left(\ln\left(\frac{4\cdot\alpha\cdot t}{r_o^2}\right)-\gamma\right) + Q\cdot R_{Total} + T_\infty \qquad \text{Equation 6.1}$$

or

$$T_f(t) = \frac{Q}{4\cdot\pi\cdot\lambda_G\cdot L}\cdot\ln(t) + \left[\frac{Q}{L}\left[\frac{1}{4\cdot\pi\cdot\lambda_G}\left[\ln\left[\frac{4\cdot\alpha}{r_o^2}\right]-\gamma\right] - R_{Total}\right] + T_\infty\right] \qquad \text{Equation 6.2}$$

where: T_f is the bulk mean return temperature of the brine solution, Q is the extraction power, λ_G is the ground thermal conductivity, α is the ground thermal diffusivity, t is the time, R_{Total} is the thermal resistance as a function of distance between the brine circulating fluid (T_f) and the farfield temperature, γ is Euler's constant (0.5772) and T_∞ is the ground farfield temperature.

According to Gehlin (Gehlin, 2002) the application of this equation can be simplified to a linear relation between $T_f(t)$ and $\ln(t)$:

$$T_f(t) = y\cdot\ln(t) + m \qquad \text{Equation 6.3}$$

where: y and m are constants. Constant y is also proportional to the thermal conductivity and is therefore the slope of the curve and is calculated by means of determining the slope of the mean fluid temperature (T_f) against the natural log of the time parameter:

$$y = \frac{Q}{4\cdot\pi\cdot\lambda_{eff}} \quad or \quad \lambda_{eff} = \frac{Q}{4\cdot\pi\cdot y\cdot H} \qquad \text{Equation 6.4}$$

where: H is the length of the heat exchanger (m), y is the inclination of the curve of temperature versus logarithmic time (K/s), Q is the heat injection/extraction rate and λ_{eff} is the effective ground thermal conductivity (W/m·K).

A detailed review of the Kelvin line source theory by Yavuzturk (1999) outlines the various applications of the theory and a novel use of the theory in transient operation. Most evaluation models of the collector heat extraction are used simply to establish thermal capacity under steady state conditions. However, the collector is generally not utilised in steady-state operation and therefore the development of a transient model is desirable.

In order to facilitate the use of the line source evaluation the thermal performance of a vertical collector is typically characterised using Thermal Response Tests (TRT) (Gehlin, 2002; Sanner, 2007). While the typical TRT was not conducted in this *HP-IRL* study a transient temperature profile of both the vertical collector and the associated heat pump performance were recorded in a similar way to the TRT during test period VC1 and the results are presented in Figures 6.18 and 6.19 for two similar tests with extraction rates of 11kW.

Figure 6.18(a) shows the results of the initial 24 hours of test period VC1 the TRT replica test and Figure 6.18(b) shows the recorded results of the initial 78 hours of test period VC2(J), during typical operation.

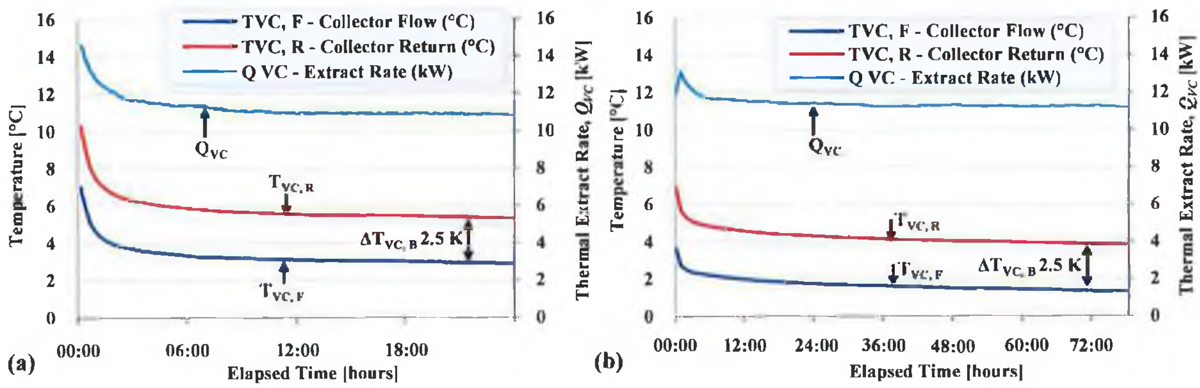


FIGURE 6.18 MEASURED THERMAL EXTRACTION IN TEST PERIODS (A) OVER ONE DAY, VC1 (11/03/07) AND (B) OVER THREE DAYS, VC2(J) (15/02/08 – 18/02/08) AND ITS IMPACT ON THE COLLECTOR FLUID TEMPERATURES.

Figure 6.19(a) presents the results of the test periods VC1 and VC2(J) as a natural log progression using an average temperature of both $T_{VC,F}$ and $T_{VC,R}$. Figure 6.19(b) presents the natural log temperature progression of the collector wall temperatures along both vertical collector wells during test period VC2(J).

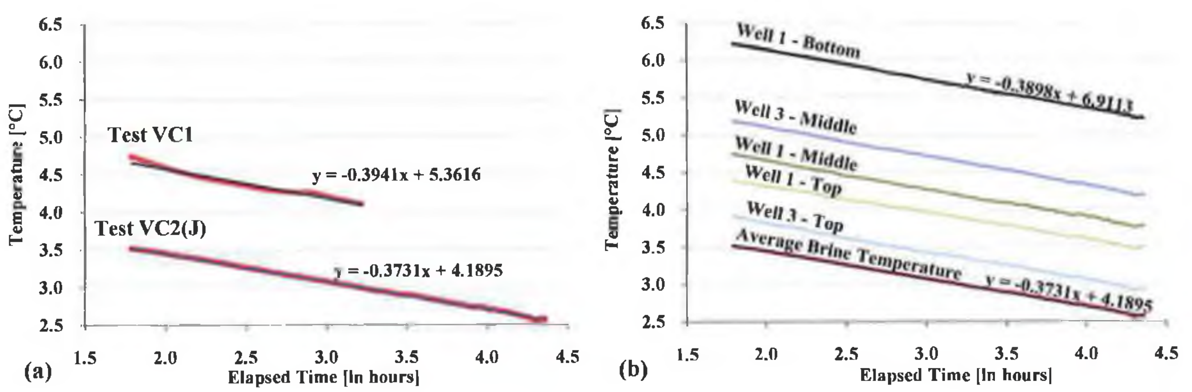


FIGURE 6.19 NATURAL LOG PROGRESSION OF THERMAL EXTRACTION FOR (A) TWO TIME PERIODS VC1 AND VC2(J) AND (B) COLLECTOR WALL TEMPERATURES DURING TEST PERIOD VC2(J).

Figure 6.19(b) shows similar thermal characteristics of both instrumented $GSHP_{VC}$ boreholes.

From Equation 6.4 and using the test results from VC1 in Figure 6.19(a) the effective ground thermal conductivity around the vertical collector is:

$$\lambda_{VC,eff} = \frac{10978W}{4 \cdot \pi \cdot 0.3941 \cdot 375m} = 5.9\ W/m \cdot K$$

Using the test results from VC2(J) in Figure 6.19(a) the effective ground thermal conductivity around the vertical collector is:

$$\lambda_{VC,eff} = \frac{11173W}{4 \cdot \pi \cdot 0.3731 \cdot 375m} = 6.3\ W/m \cdot K$$

The presence of groundwater can have an effect on the heat extraction capacity of the vertical collector and must be taken into consideration. Where ground water flow is slow or stagnant then the heat conduction and convection can be approximated using an effective thermal conductivity value but this simplification is not applicable if there is significant groundwater movement. In this *HP-IRL* study it is noticeable the $\lambda_{VC,eff}$ is higher than what would be expected from the vertical collector if it was extracting conductively from limestone alone, where from Figure 2.13 $\lambda_{VC,eff}$ might be expected to range between 1.5 W/m·K and 5.5 W/m·K. As the line source model method used does not differentiate between thermal convection and conduction it must be acknowledged that the $\lambda_{VC,eff}$ calculated in this *HP-IRL* study may not result from purely conductive based heat transfer, since an aquifer was exposed below 90m when drilling. Indeed, Chiasson *et al.* (2000) showed that it is difficult to adapt results from current design guidelines and software tools utilising the line source theory to account fully for the effect of groundwater movement. This conclusion was reached due to the typically unquantifiable nature of groundwater movement.

Groundwater flow has a major effect on the thermal characteristics of the vertical collector and has been highlighted as an important factor in the thermal estimation of the collector effectiveness and calls have been made to study its long term effect on collector performance (Gehlin, 2002). Furthermore, groundwater flow influence may be reduced considerably if the circulating brine is below 0°C causing the thermal flow path to be reduced due to freezing.

6.3.6 $GSHP_{VC}$ PERFORMANCE EVALUATION SUMMARY

Over the course of the three years of heat pump performance analysis the hourly averaged COP varied from 2.77 (VC3) to 3.36 (VC4), with heat pump temperature lifts (ΔT_{HP}) of 44.8K and 38.8K respectively. The overall average SPF was 2.95. The heat pump sink temperature ($T_{HP,F}$) was typically around +49°C for all test periods other than VC4, with variation in performance due to the fluctuations in the collector return temperature ($T_{HC,R}$). The average $T_{VC,R}$ for the three years of testing was +6.9°C, with the lowest recorded return temperature of +3.0°C recorded in test period VC7, generating hourly average ground temperature drawdown ($\Delta T_{VC,G}$) of -7.4K. The test period with the greatest average thermal extraction was VC7, extracting 27.1W/m of borehole.

The performance of the vertical collector for all eight test periods is presented in Table 6.15.

TABLE 6.15 SUMMARY OF AVERAGED TEST PERIOD PERFORMANCE INDICATORS FOR THE $GSHP_{VC}$

Test #	Days (Hours)	Collector Extract Rate (Duty)	$T_{VC,\infty}$	$T_{VC,R}$	$\Delta T_{VC,G}$	$T_{HP,F}$	ΔT_{HP}	Collector pipe extract rate	Coefficient Of Performance, COP [-]*
VC1	8	9.4kW (87%)	+10.4°C	+5.6°C	-4.8K	+48.9°C	43.3K	25.2 W/m	2.85 (3.16)
VC2	96	7.2kW (62%)	+10.7°C	+5.9°C	-4.8K	+49.5°C	43.6K	19.4 W/m	2.90 (3.20)
VC3	3	9.6kW (84%)	+10.4°C	+5.0°C	-5.4K	+49.8°C	44.8K	25.7 W/m	2.77 (3.05)
VC4	1 (24)	4.8kW (34%)	+10.6°C	+8.5°C	-2.1K	+47.3°C	38.8K	12.9 W/m	3.36 (3.72)
VC5	43	3.7kW (28%)	+12.1°C	+9.9°C	-2.2K	+49.5°C	39.6K	9.9 W/m	3.18 (3.40)
VC6	45	6.2kW (52%)	+11.3°C	+7.3°C	-4.0K	+49.7°C	42.4K	16.8 W/m	3.01 (320)
VC7	30	10.1kW (94%)	+10.6°C	+3.9°C	-6.7K	+49.2°C	45.3K	27.1 W/m	2.80 (2.99)
VC8	62	5.8kW (44%)	+11.0°C	+7.5°C	-3.5K	+50.0°C	42.5K	15.5W/m	2.77 (3.04)

*Unbracketed data reflects the actual COP including collector pump power; Bracketed data reflects the COP as per EN-24511 Test Standard

This study identified that climate influences collector in two ways: weather events impacts the surface level collector element, seasonal weather changes affects collector down to 15m depth and geothermal gradients effect below 15m. In this *HP-IRL* study 31% of the vertical collector was in some way influenced by seasonal variations in ambient air temperature.

Some key observations of the *HP-IRL* vertical collector heat pump performance testing are:

- The vertical collector performed resiliently under the three years of sustained thermal extraction, showing minimal signs of thermal degradation of the source
- During the three month peak winter heating season of 2008/2009 the vertical collector recording a minimum collector return temperature of 3.0°C
- A temperature penalty of 0.25°C recorded over the three years of thermal extraction
- Influence of climate varies the farfield temperature between +10°C in winter and +12.8°C in summer, with an average temperature of +11.4°C
- Compared with the horizontal collector, the vertical collector displayed no sensitivity to weather events other than seasonal effects which yielded a year-round farfield temperature variation of ±1.4°C, but did show greater sensitivity to duty with a maximum ground temperature drawdown of -7.4K versus -5.1K for the horizontal collector
- Ground temperature drawdown rates are proportional to the thermal extraction rates, showing higher drawdown rates per kW capacity than the horizontal collector
- $GSHP_{VC}$ generated a SPF of 2.95 over the three years operation, outperforming the $GSHP_{HC}$ by 1.7% which had an SPF of 2.90
- Vertical collector showed faster drawdown and recovery rates that the horizontal collector. A performance comparison of both collectors is presented in Chapter 7
- Ground thermal recovery rates vary with depth along the borehole
- The results of the thermal response test determined the effective ground thermal conductivity to be between 5.9 and 6.3 W/m·K, 10% higher than expected

Potential exists to boost the winter-time collector return temperature by insulating the return pipe of the borehole from a depth of 15m back to the building. This could offer a potential improvement of 2°C on the return temperature and increase the COP by 5%.

6.4 ASHP TEST PROGRAM (2008 – 2009)

This section presents key findings from a six month long ASHP test program conducted as part of the *HP-IRL* study. It identifies the rationale for each of the five individual tests conducted during the heating season and the heat pump performance achieved.

6.4.1 ASHP TEST PROGRAM RATIONALE

The goal of the *HP-IRL* ASHP test program was to monitor heat pump performance under various heating demands and weather conditions. Four individual test periods were run for duties ranging between 60 and 87% and heat pump performance is illustrated in Figure 6.20 based on hourly averaged data.

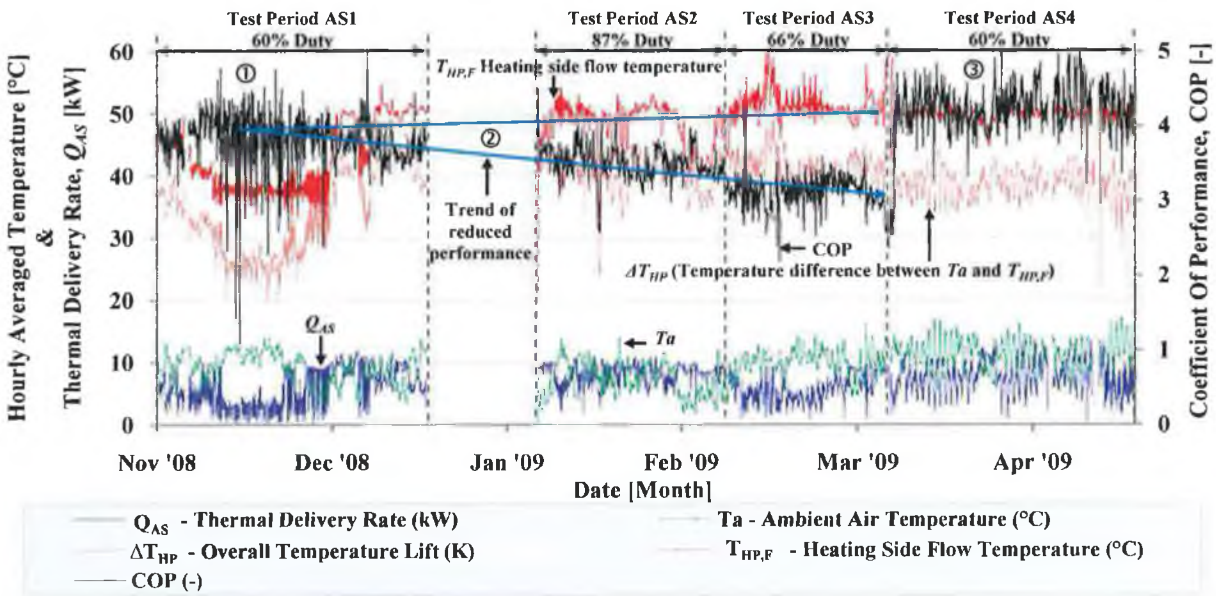


FIGURE 6.20 TIMING OF FOUR OF THE ASHP TEST PERIODS (2008-2009) AND CORRESPONDING PERFORMANCE.

The heat pump duty cycle was configured to achieve stable water temperature in a series of thermally uninsulated aquaculture water tanks. The heat loss from the water tanks was a function of ambient external air temperature and as a result ASHP duty fluctuated with this weather condition. The key performance characteristics of the overall ASHP test program are summarised in Table 6.16.

TABLE 6.16 SUMMARY RESULTS FROM THE ASHP TEST PROGRAM (2008-2009)

Overall ASHP Test Period	Days	Average Operational Time	On-Time Average Q_{AS}	Overall Average Thermal Delivery Rate, Q_{AS}	Weighted Average $Ta_{,\infty}$	Weighted Average $T_{HP,F}$	Total kWh's delivered (Thermal)	Total kWh's consumed (Electrical)	SPF
01/11/08 – 07/05/09	165	67%	9.5kW	6.9kW	+8.9°C	+47.8°C	24,997	6,690	3.7

Over the course of the five test program the ASHP operated for 165 test days, delivered 24,997 kWh of energy (90 GJ), which is the equivalent of two years of space heating for a domestic dwelling (12,000 – 15,000kWh/annum). The heat pump delivered an average heat pump sink temperature of +47.8°C and a COP that ranged from 2.6 to 4.7 and an SPF of 3.7. Both the COPs and SPF compare favourably with both GSHPs.

Table 6.17 presents the demand and duration of the test periods, along with a brief description of the evaluation.

TABLE 6.17 CHARACTERISTICS OF THE FIVE ASHP TEST PERIODS (2008-2009)

Test #	Demand	Term	Duration	Description	kWh Delivered	COP_{AVG} [-]
AS1	Moderate	Long	48 days	Initial observational period with moderate system demand.	6,475	3.83
AS2	Intensive	Long	35 days	ASHP performances evaluation under a long term intensive duty demand, evaluating start-up and steady-state operational dynamics.	6,548	3.52
AS3	Moderate	Medium	29 days	ASHP performances evaluation under a medium term moderate duty demand, evaluating steady-state operational dynamics.	3,862	3.11
AS4	Moderate	Long	51 days	ASHP performances evaluation under a long term moderate duty demand, evaluating steady-state and intermittent operational dynamics.	8,008	4.26
AS5	Moderate	Short	2 days	ASHP performance under typical domestic duty cycle and an output temperature of +35°C	104	4.23

Some initial observations from Figure 6.20 are:

① The initial test period AS1 shows a large variation in COP which corresponds with a high ambient air temperature, causing cycling of the system with fan operation and no heat pump operation

② From the middle of test period AS1 the COP steadily deteriorates through to test period AS4 due to a filter blockage impeding the fluid flow of the hot side of the heat pump ($T_{HP,F}$)

③ After a filter change test period AS4 was conducted showing a consistently high COP averaging at 4.3, with a heat pump output temperature of +49.4°C and an average ambient outside air temperature of +10.9°C

It is noticeable from Table 6.17 that there was a steady deterioration in COP from approximately 4 to 3 between AS1 and AS3. This resulted from a filter blockage on the heating side fluid flow that impeded flow, increased pumping power and heat pump cycling with only fans in operation, while also doubling the temperature difference across the condenser from 10K to 20K. The position of the filter in relation to the ASHP is shown in Figure 3.26. Once the filter was cleaned at the end of test period AS3 normal operating conditions resumed.

The following section profiles the performance characteristics of the *HP-IRL* study air source heat pump operation under Irish Maritime climate conditions. The transient response of the ASHP to variations in the heat pump duty cycle and the resulting impact on heat pump performance is described.

6.4.2 ASHP PERFORMANCE EVALUATION

Figure 6.21 displays the ASHP operational data for the test period AS1 conducted over 48 days between 01/11/09 and 18/12/09. This test period revealed the performance of the system operating with heat pump temperature lifts (ΔT_{HP}) that ranged between 42K and 28K, with COP's ranging from 3.7 to 4.0.

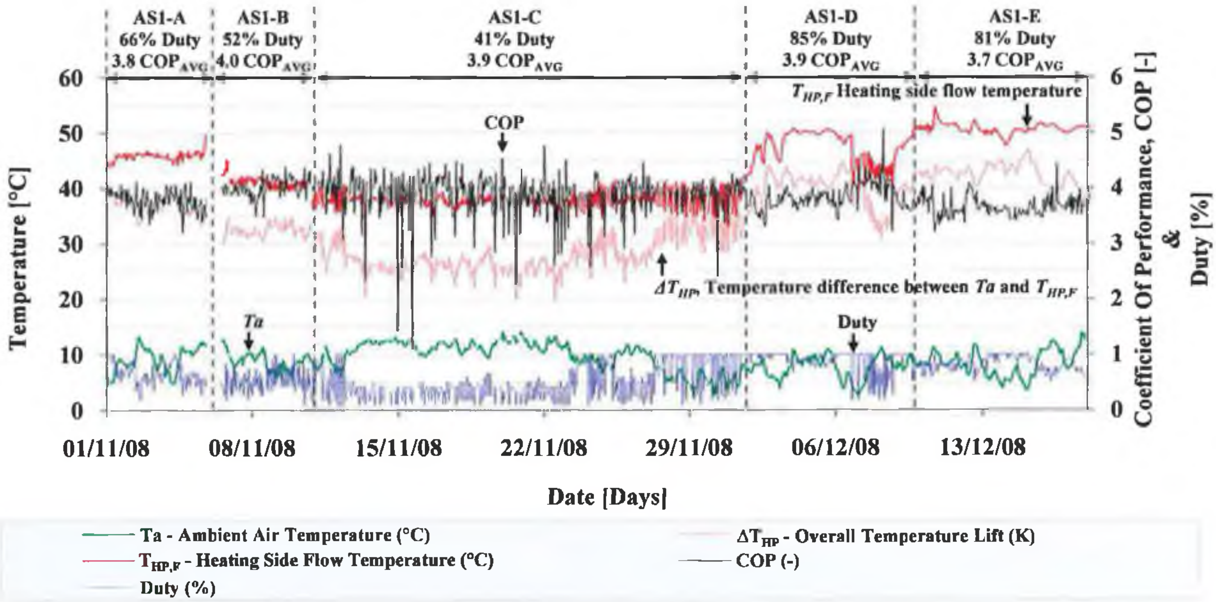


FIGURE 6.21 HOURLY AVERAGED ASHP PERFORMANCE CHARACTERISTICS DURING TEST PERIOD AS1.

Test period AS1 is subdivided into five sub-sections where the duty cycle changes to reflect the climate. Table 6.18 presents the variation in performance for each of those five sub-test periods.

TABLE 6.18 TEST PERIOD AS1 SUMMARY PERFORMANCE RESULTS

Test #	Dates	Days	Operational Time	On-Time Average Q_{AS}	Overall Average Thermal Delivery Rate, Q_{AS}	Weighted Average Ta	Weighted Average $T_{HP,F}$	ASHP ΔT_{HP}	COP$_{AVG}$
AS1-A	01/11/08 – 05/11/08	5	66%	8.9kW	5.9kW	+8.5°C	+45.7°C	37.1K	3.79
AS1-B	06/11/08 – 11/11/08	5	52%	9.3kW	4.9kW	+8.6°C	+40.9°C	32.3K	4.02
AS1-C	11/11/08 – 30/11/08	20	41%	8.4kW	4.0kW	+9.2°C	+37.5°C	28.3K	3.88
AS1-D	30/11/08 - 09/12/08	9	85%	9.9kW	8.6kW	+7.0°C	+45.3°C	38.3K	3.86
AS1-E	09/12/08 – 18/12/08	9	81%	9.5kW	7.7kW	+8.2°C	+50.4°C	42.1K	3.67
AS1	01/11/08 – 12/12/08	48	60%	9.1kW	6.0kW	+8.2°C	+44.3°C	36.1K	3.83

What is again noticeable from Figure 6.21 is the close relationship between the heat pump thermal delivery (Q_{AS}) and the ambient outside air temperature (Ta). However, also notable from Table 6.18 is the relative insensitivity of the heat pump COP with heat pump temperature lift. This is in part due to the high ambient air temperature requiring a low heat pump duty that means the compressor cycles on and off but the evaporator fans are continuously on and consuming power. From Figure 6.21 it can be seen that no single moment can be isolated as the start of the filter blockage, rather it increases progressively with time. Figure 6.22 illustrates the negligible influence of relative humidity and ambient air temperature on heat pump performance during the 48 day long AS1.

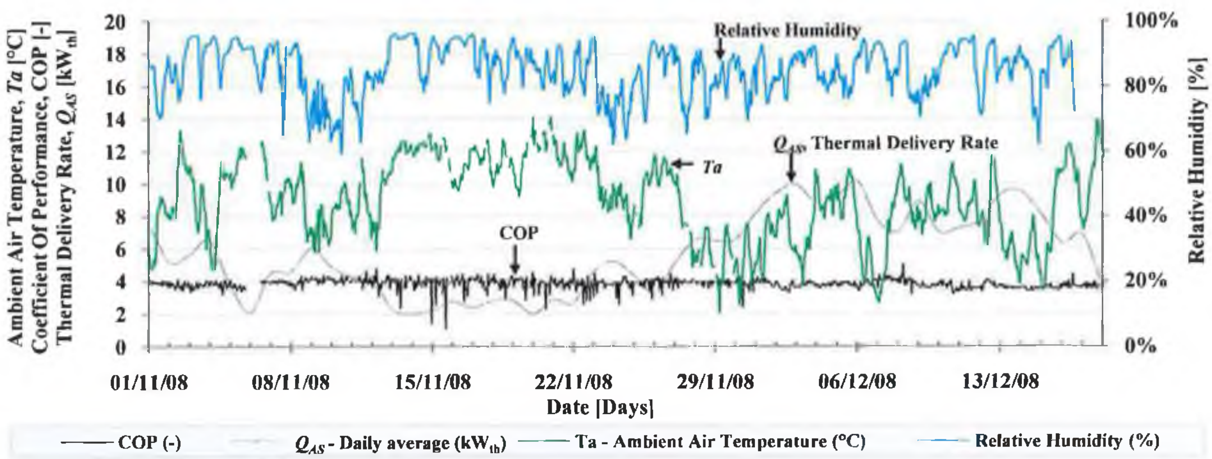


FIGURE 6.22 HOURLY AVERAGED WEATHER PARAMETERS AND THEIR INFLUENCE UPON ASHP PERFORMANCE DURING TEST PERIOD AS1.

While the hourly averaged thermal demand (Q_{AS}) varies with external ambient air temperature (*Ta*) the heat pump delivers a consistently high COP of 3.83. COP shows a marked insensitivity to external ambient air temperature, even below +5°C. Relative humidity fluctuated between 60% and 95%, but the heat pump displayed a stable performance throughout.

One of the coldest days of testing occurred on the 14th of December with an average air temperature of +5.8°C. This is an ambient air temperature that can induce a large defrost requirement. The ASHP operated for 96% of the day delivering a consistent 9.9kW$_{th}$ throughout the day. This resulted from the ASHPs dual evaporator which allows it to operate even during defrosting. This feature allows the frost affected evaporator to defrost, but more importantly it uses heat from the condenser to accelerate defrosting. If the thermal demand is high, this defrost can also take place in both evaporators simultaneously, with the evaporators are kept from frosting with supplemental heat injection from the hot gas side. This defrost energy is not fully lost to the environment, as is the case in standard reverse cycle defrost, as the continuous operation of the evaporator reabsorbs any excess thermal energy. This process keeps the evaporator on the verge of frosting.

A detailed illustration of test period AS2 is shown in Figure 6.23 and Table 6.19 summarises the results. This test period performed a prolonged high average output temperature of +47.4°C for 35 days that corresponded to and average overall temperature lift of 40.4K.

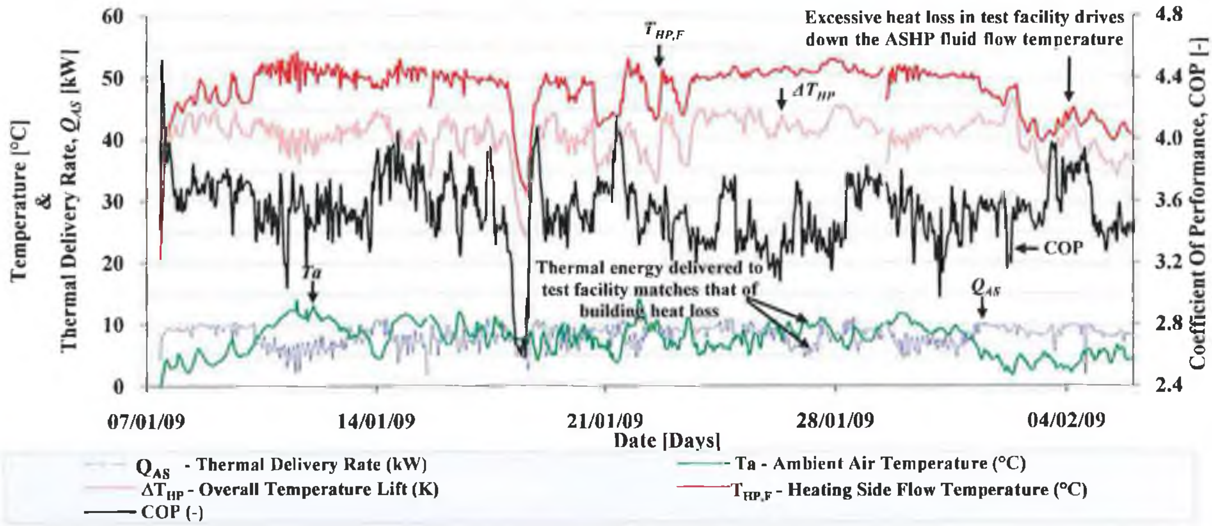


FIGURE 6.23 HOURLY AVERAGED ASHP PERFORMANCE CHARACTERISTICS DURING TEST PERIOD AS2.

TABLE 6.19 TEST PERIOD AS2 SUMMARY PERFORMANCE RESULTS

Test #	Dates	Days	Operational Time	On-Time Average Q_{AS}	Overall Average Thermal Delivery Rate, Q_{AS}	Weighted Average Ta	Weighted Average $T_{HP,F}$	ASHP ΔT_{LIFT}	COP_{AVG}
AS2	07/01/09 – 10/02/09	35	87%	9.3kW	8.1kW	+7.0°C	+47.4°C	40.4K	3.52

Some observations from Figure 6.23 are:

- Thermal energy delivery, Q_{AS}, to the water storage tanks with changes in ambient external air temperature, suggesting a large heat loss from the test facility
- Overall COP has started to become impaired by the restricted fluid flowrates

Figure 6.24 shows the thermal progression of the ASHP as it heats the water tanks from +15°C up to +50°C, illustrating the coefficient of performance of the *HP-IRL* ASHP from start-up to maximum output temperature. All recording intervals in Figure 6.24 are 5 minutes.

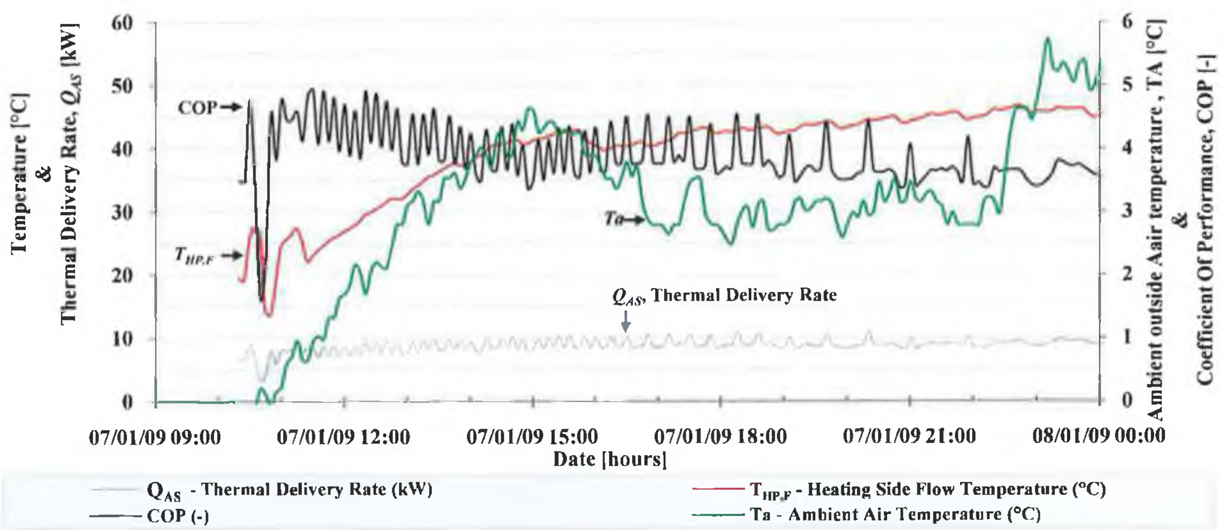


FIGURE 6.24 FIVE MINUTE RECORDING INTERVALS OF ASHP START-UP PERFORMANCE FOR TEST PERIOD AS2.

Prior to the commencement of test period AS2 the aquaculture test facility water tanks were allowed to cool down to an ambient temperature of 22°C. Figure 6.24 shows the ASHP operated continuously for 22 hours from start-up without interruption and ran for a further 48 hours with just 20 minutes down-time. The heat pump provided a consistent thermal supply of 9.1kW and the COP ranged from 4.8 to 3.4.

Subsequent the filter replacement after test period AS3, test period AS4 commenced and a detailed illustration of 20 days of the 51 day long test period is shown in Figure 6.25. This test period generated a stable 20 day high average output temperature of +49.4°C with and overall average temperature lift of 38.4K.

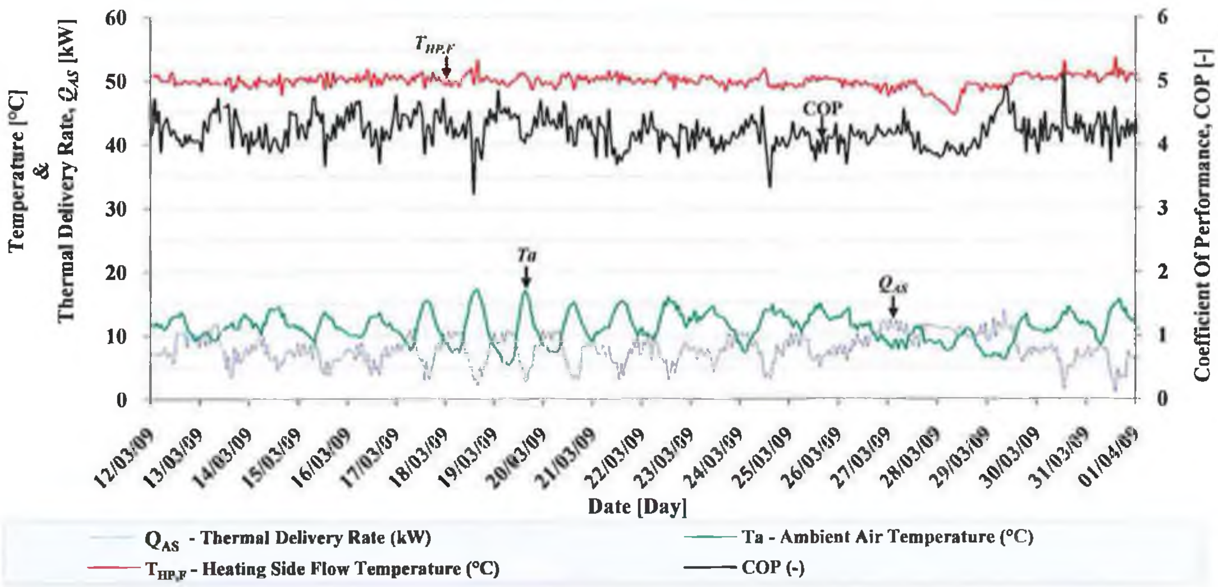


FIGURE 6.25 HOURLY AVERAGED ASHP PERFORMANCE CHARACTERISTICS DURING TEST PERIOD AS4.

Having identified and eliminated the performance inhibiting factor of the blocked filter this test period gives the best indication of the ASHP performance. As the ambient air temperature (*Ta*) remained relatively high for the period (+10.9°C) the coefficient of performance was a high 4.3 due to minimal defrost related thermal losses. It was also high in comparison the other test periods as the initial problems of reduced flow-rates on the heating side were corrected.

Table 6.20 displays the performance characteristics of the air source heat pump during test period AS4.

TABLE 6.20 TEST PERIOD AS4 SUMMARY PERFORMANCE RESULTS

Test #	Dates	Days	Average Operational Time per hour (Total time in operation)	On-Time Average Q_{AS}	Overall Average Thermal Delivery Rate, Q_{AS}	Weighted Average Ta	Weighted Average $T_{HP,F}$	ΔT_{HP}	COP_{AVG}
AS4	11/03/09 – 01/05/09	51	68% (60%)	11.0kW	7.5kW	+10.9°C	+49.4°C	38.4K	4.26

The results of test period AS4 reveal a consistently high COP of 4.3 with an output temperature of +49.4°C and compared favourably with both GSHPs under similar conditions.

Narrowing the focus of ASHP application, test period AS5 looked at ASHP performance under typical domestic dwelling conditions where the duty cycle concentrates around the night rate electricity time period of between 12pm and 9am. As shown in Figure 6.26, the output temperature was set at +35°C to mimic thermal provision for utilisation with an underfloor heating system.

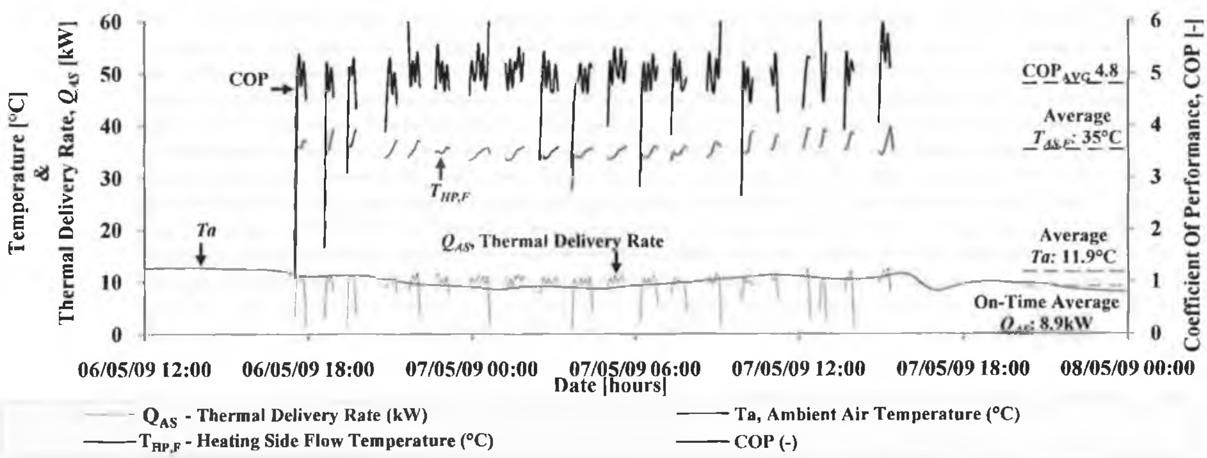


FIGURE 6.26 FIVE MINUTE RECORDING INTERVAL OF THE ASHP PERFORMANCE CHARACTERISTICS DURING TEST PERIOD AS5 WITH +35°C HEAT PUMP FLOW TEMPERATURE ($T_{HP,F}$).

Although the ambient outside air temperature is a mild +11.9°C, Figure 6.26 illustrates the potentially high performance capability of an ASHP operating under Ireland's moderate Maritime Climate.

6.4.3 ASHP PERFORMANCE EVALUATION SUMMARY

Over the course of the six month heat pump performance evaluation COP variations occurred due to systemic problems of reduced fluid flow on the heating side and also due to variations in ambient air temperature. The average daily COP over the course of testing varied from 2.65 to 4.81, with heat pump temperature lifts (ΔT_{HP}) of 45.5K and 20.6K respectively. The heat pump sink temperature ($T_{HP,F}$) averaged +47.8°C for the test program. The performance of the ASHP for three test periods is presented in Table 6.21.

TABLE 6.21 SUMMARY OF KEY ASHP TEST PERIOD RESULTS (2008-2009)

Test #	Dates	Days	kWh Delivered	Overall Average Thermal delivery, Q_{AS} (Duty)	Ta	$T_{HP,F}$	$\Delta T_{HP,B}$	ΔT_{HP}	COP_{AVG}
AS1	01/11/08 – 18/12/08	48	6,475	6.0kW (60%)	+8.2°C	+44.3°C	10.8K	35.4K	3.83
AS4	11/03/09 – 01/05/09	57	8,112	7.5kW (60%)	+10.9°C	+49.4°C	10.3K	38.0K	4.26
AS5	06/05/09 – 07-05/09	2	104	4.5kW (50%)	+11.9°C	+35.3°C	7.9K	23.2K	4.81
Total	**01/11/08 – 07/05/09**	**165**	**24,997**	**6.9kW (67%)**	**+8.9°C**	**+47.8°C**	**12.1K**	**38.9K**	**3.74**

The average ambient outside air temperature, *Ta,* for the six months of winter and spring testing was +8.9°C, with the lowest recorded hour average *Ta* during testing being +1.7°C recorded on the 2nd of February 2009, generating an hour average COP of 3.5.

Some key observations of the *HP-IRL* air source heat pump performance testing are:

- Heat pump performance showed no obvious negative sensitivity to either ambient outside air temperatures or relative humidity fluctuations
- ASHP delivers thermal supply regardless of weather conditions without interruption for defrosting
- Hourly average heat pump performance varied from 2.6 to and 5.2, and varies from 3.1 to 4.8 during test periods AS3 and AS5 respectively
- Heating side fluid filter became progressively blocked over test periods AS2 and AS3, degrading the ASHP capacity and performance. This highlighted the importance of monitoring newly installed heat pump systems for an initial period after installation and commissioning. The *HP-IRL* GSHP systems encountered similar brine filter blockages after the first six months of operation, as debris within the closed loop pipework was flushed out.
- Once the initial commissioning problem of the blocked filter was resolved, the ASHP delivered a consistently high COP, even at +50°C output temperature, that exceeded the GSHPs COP by over 30%
- The ASHP system delivered a SPF of 3.74
- ASHP system is an attractive option that demands further evaluation of precise amount of thermal energy lost in defrost, along with evaluation in sub zero ambient external air temperatures

The following chapter provides a techno-economic assessment of the three respective heat pumps.

CHAPTER 7 – TECHNO-ECONOMIC EVALUATION

Having identified the performance characteristics of three heat pump collector types, this chapter contrasts all three from a performance, environmental and economic perspective.

7.1 HEAT PUMP PERFORMANCE

This section compares the performance characteristics of each of the three collector types studied.

7.1.1 HEAT PUMP TEST PROGRAM

Performance data was accumulated during the *HP-IRL* study which ran for 747 days and delivered over 168,552kWh of thermal energy, equivalent to 12 years of typical domestic dwelling use. Table 7.1 presents a breakdown of this supply per collector type and test year.

TABLE 7.1 SUMMARY *HP-IRL* TEST RESULTS (2006–2009)

Winter Heating Season Test Results		$GSHP_{HC}$	$GSHP_{VC}$	ASHP
Total Test Days	2006-2009	293 Days	289 Days	165 Days
Total kWh Delivered	2006-2009	69,514	74,011	24,997
# of Equivalent Years Domestic Energy Use	2006-2009	5	5	2
Combined Total Test Program SPF	2006-2009	2.90	2.95	3.74
Total kWh delivered/Heating Season	2006/2007	23,348	2,492	-
	2007/2008	23,623	26,010	-
	2008/2009	22,543	45,519	24,997
Average Heat Pump Duty/Heating Season	2006/2007	52%	50%	-
	2007/2008	42%	60%	-
	2008/2009	86%	46%	67%
Average Source, ($T_{HC,R}$ *or* Ta)/Sink Temperature, ($T_{HP,F}$)	2006/2007	+5.3°C/+45.6°C	+6.6°C/+48.0°C	-
	2007/2008	+5.0°C/+48.7°C	+6.0°C/+49.4°C	-
	2008/2009	+2.1°C/+49.0°C	+6.2°C/+49.7°C	+8.9°C/+47.8°C
Average Ground Temperature Drawdown, ΔT_G	2006/2007	-2.8K	-3.7K	NA
	2007/2008	-3.1K	-4.2K	NA
	2008/2009	-4.5 K	-4.6K	NA
Seasonal Performance Factor, SPF	2006/2007	2.98	3.07	-
	2007/2008	2.94	3.05	-
	2008/2009	2.77	2.88	3.74

Table 7.1 shows the 2K average difference between the collector source temperature of the horizontal ($T_{HC,R}$) and vertical ($T_{VC,R}$) collectors over the course of the three winter heating seasons. As indicated in Figure 4.23 this temperature difference can result in a 5% higher performance from the vertical collector over the horizontal collector. This can be explained by the higher source temperatures available to the vertical collector during the heating season shown in Figure 7.1. Much emphasis is placed on the thermal stability that the ground provides for GSHPs, but this advantage over ASHPs is only applicable if the temperature

swings as in continental climates. However, the air source temperatures recorded during the 2008/2009 heating season and presented in Table 7.1 shows that the ambient air temperature during the test periods of operation was almost 3°C higher than the borehole.

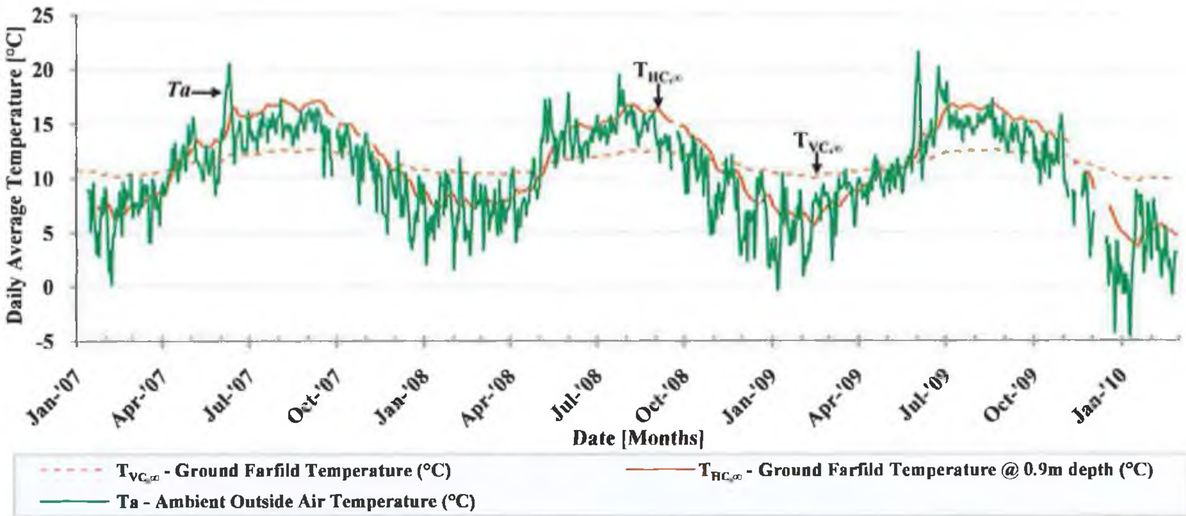


FIGURE 7.1 HEAT PUMP SOURCE TEMPERATURE COMPARISON BETWEEN JANUARY 2007 AND FEBRUARY 2010.

From Figure 7.1 it is noticeable that the greatest annual variation occurs in ambient outside air temperature with variations of up to 26K from summer to winter, with the horizontal collector source temperature at 0.9m depth ($T_{HC,\infty}$) showing a 13K variation and the vertical collector source temperature showing the smallest variation of 2.5K. It is also worth highlighting that over the 3 years the average air, vertical and horizontal collector source temperatures were +10.5°C, +11.4°C and +11.9°C respectively. The average winter farfield temperature was +10.9°C and +7.9°C for the vertical and horizontal collectors respectively. The average winter ambient outside air temperature was +7.3°C.

In Figure 7.2, the total test period COP_{DAY} results of both the horizontal and vertical collectors are contrasted with the ambient outside air temperature.

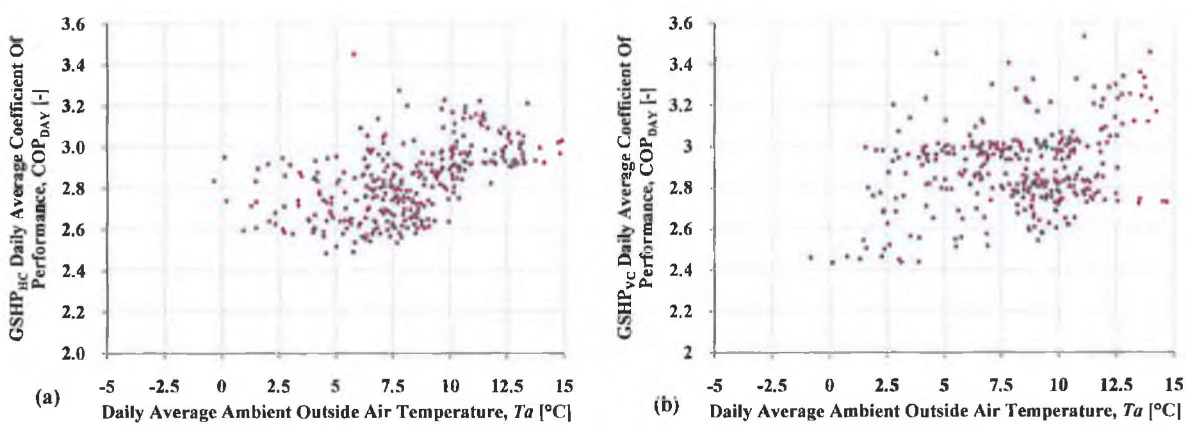


FIGURE 7.2 COP_{DAY} VALUES FOR (A) $GSHP_{HC}$ AND (B) $GSHP_{VC}$ SYSTEMS OVER THE TOTAL TEST PERIOD FROM THE 1ST OF JANUARY 2007 TO THE 1ST OF MARCH 2010.

While both sets of data reveal a large spread of COPs with regard to Ta, the COPs for the horizontal collector Figure 7.2(a) does show a positive upward trend with ambient air temperature, whereas no similar trend was evident for the vertical collector in Figure 7.2(b). This non correlation with the ambient outside air temperature is due in part to both the large thermal capacity of the IiBC building into which the heat pumps were supplying thermal energy, but also due to the test programs that were imposed on the system regardless of the buildings thermal needs. The overall test period SPF for the horizontal collector was 2.90 and 2.95 for the vertical collector, where the $GSHP_{HC}$ was operational for 293 days and the $GSHP_{VC}$ was operational for 289 days.

The following section characterises the evaluation of each collector type's source temperature.

7.1.2 GSHP SOURCE TEMPERATURE

As discussed in Chapter 2 there is a limited understanding of ground temperature drawdown due to GSHP operation in the literature. The most common calculation method for estimating the collector fluid temperature was developed on a procedure presented by the IGSHPA (1988) (RETScreen, 2005). This method showed, for a given ambient air bin temperature, T_{bin}, the temperature of the collector return ($T_{HC,R}$) entering the heat pump is as follows:

$$T_{HC,R} = T_{HC,R-min} + \left(\frac{T_{HC,R-max} - T_{HC,R-min}}{T_{a,max} - T_{a,min}}\right)(T_{bin} - T_{a,min})$$

Where $T_{a,max}$ and $T_{a,min}$ represents the design maximum and minimum external ambient air temperatures respectively, and this is graphically illustrated in Figure 7.3(a). From the IGSHPA (1988) literature the collector return temperature maximum and minimum corresponds with the ground temperature (T_G) as follows:

$$T_{HC,R-min} = T_{G,min} - 8°C \qquad \text{and:} \qquad T_{HC,R-max} = T_{G,max} + 11°C$$

Where $T_{G,max}$ and $T_{G,min}$ represents the maximum and minimum ground temperatures at the collector depth. However, this method has certain limitations in evaluating the entering water temperature as it neglects the natural variations in heat pump duty, the 'buffering' effect of the ground's thermal capacity on the collector fluid return temperature and the drawdown of the ground's temperature with short, medium and prolonged duty cycles.

This is illustrated in the context of the Irish Maritime climate in Figure 7.3(b).

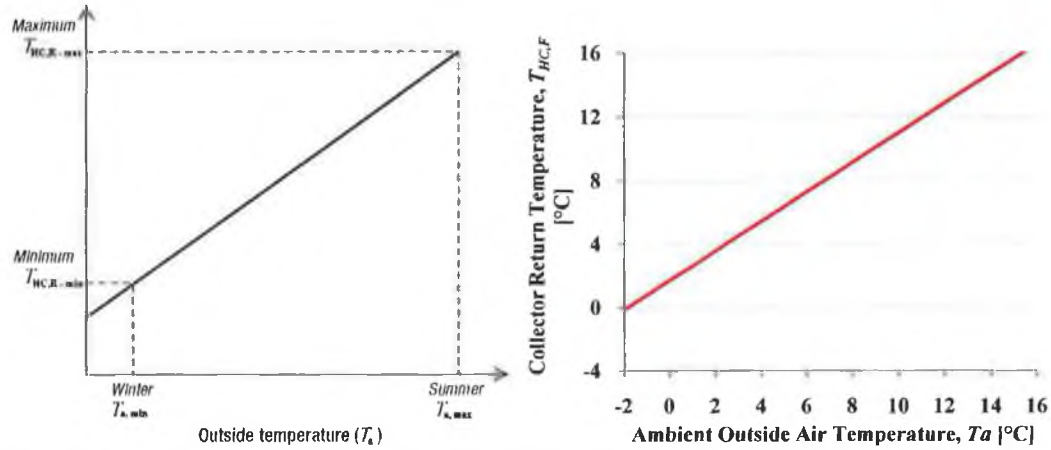


FIGURE 7.3 ENTERING WATER TEMPERATURE AS (A) A FUNCTION OF EXTERNAL AIR TEMPERATURE (RETSCREEN, 2005) AND (B) ENTERING WATER TEMPERATURE AS A FUNCTION OF EXTERNAL AIR TEMPERATURE FOR IRISH CLIMATIC CONDITIONS.

From the *HP-IRL* study, some examples of variable extract rates and the subsequent effect on the heat pump's collector fluid return temperature can be seen in Figures 7.4(a) and 7.4(b) where the daily averaged fluid return temperature to the heat pump varies between +3°C and +12°C for the horizontal collector ($T_{HC,R}$) and between +4°C and +9°C for the vertical collector ($T_{VC,R}$).

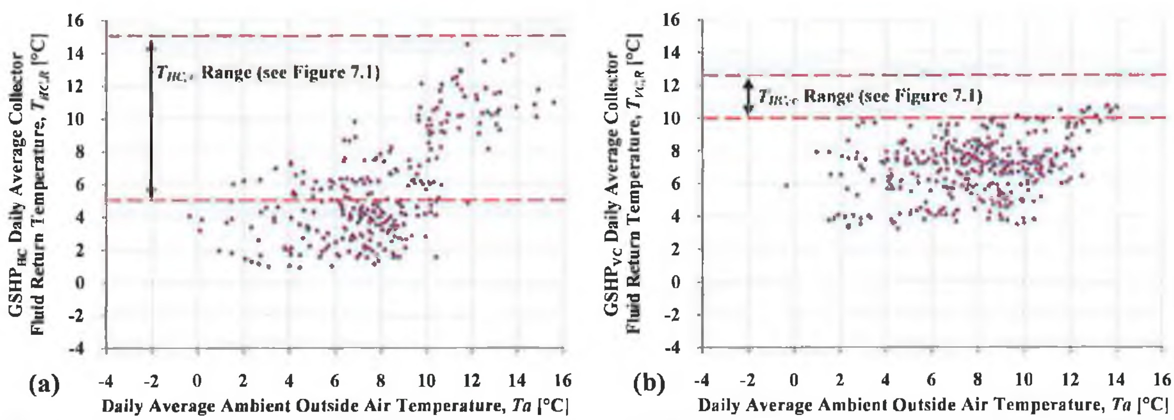


FIGURE 7.4 DAILY AVERAGED COLLECTOR FLUID RETURN TEMPERATURE AS A FUNCTION OF AMBIENT AIR TEMPERATURE FOR *HP-IRL* (A) GSHP$_{HC}$ AND (B) GSHP$_{VC}$ OPERATION FROM 2007-2009.

For the GSHP$_{VC}$, there is greater predictability in the collector return temperature shown in Figure 7.4(b) than the GSHP$_{HC}$ shown in Figure 7.4(a). From the *HP-IRL* data shown in Figure 7.4 it can therefore be recognised that the use of simplified methods of estimating the collector fluid temperature ($T_{HC,R}$), as shown by the linear relationship with ambient air temperature in Figure 7.3, may not accurately reflect the robust and variable nature of the heat pump operation, the transient effect of thermal extraction on the ground temperature drawdown and consequent irregular collector fluid return temperature.

The following subsection takes a closer look at the ground temperature drawdown of the horizontal and vertical collector for comparison.

7.1.3 GROUND TEMPERATURE DRAWDOWN COMPARISON

Using the *HP-IRL* test facility, Figure 7.5 compares the rate at which the ground temperature surrounding the collector reduces when both collectors extract approximately 6.4kW and 10.5kW. While both collectors display similar ground temperature drawdown rates at the lower 6kW extract rate, the drawdown rate of the vertical collector accelerates beyond that of the horizontal collector by as much as a factor of 2 at full load (10.5kW) leading to a 66% greater overall drawdown. This highlights the sensitivity of the vertical collector to high thermal extraction rates and the ability of the horizontal collector to deliver a temperature that is close to the 'farfield' at full load over short time periods.

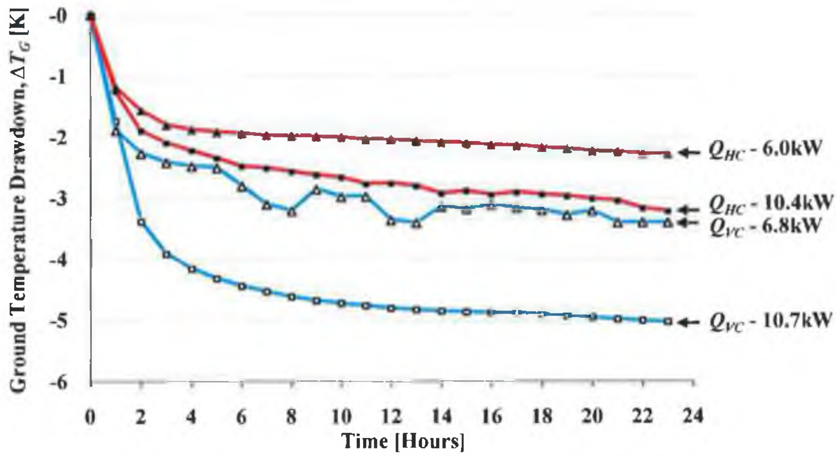


FIGURE 7.5 COMPARING THE DRAWDOWN OF THE GROUND TEMPERATURE IN THE VICINITY OF BOTH THE HORIZONTAL COLLECTOR ($\Delta T_{HC,G}$) AND THE VERTICAL COLLECTOR ($\Delta T_{VC,G}$) VERSUS EXTRACT RATES (Q), RESULTS WERE OBTAINED FROM TESTS CARRIED OUT ON VARIOUS DATES IN 2007.

The variation in ground temperature drawdown shown in Figure 7.5 illustrates the much smaller change in the horizontal collector drawdown temperature with thermal extraction over a short period of time of 24 hours or less. However, this is only a temporary advantage, which diminishes once longer term continuous thermal extraction is undertaken, as indicated by the steady state drawdowns in Figure 7.6.

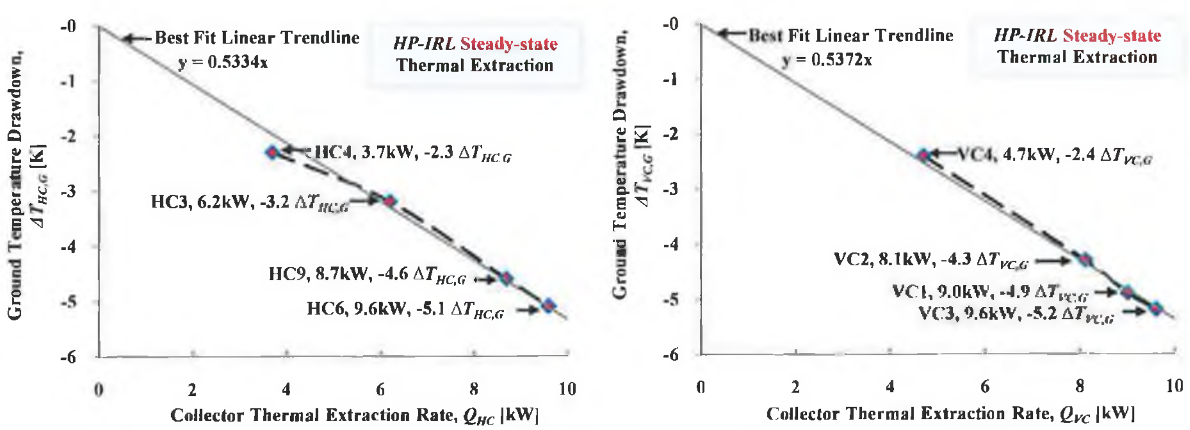


FIGURE 7.6 STEADY-STATE GROUND TEMPERATURE DRAWDOWN VERSUS THERMAL EXTRACTION RATE FOR THE (A) HORIZONTAL COLLECTOR AND (B) VERTICAL COLLECTOR.

Figure 7.6 highlights that given time to reach steady-state, the ground temperature drawdown versus thermal extraction rates are identical for both the horizontal and vertical collector. However, it takes up to 7 times longer for the horizontal collector to reach steady-state as indicated in Tables 4.9 and 6.10 (VC1 versus HC6), due to dispersed nature of its collector compared with the localised vertical collector.

7.1.4 ASHP SOURCE TEMPERATURE

The positive aspect of an air source heat pump is the direct link between the air source and the primary refrigerant at the evaporator. The direct expansion (DX) system is its equivalent in GSHP terms, where the refrigerant is run through the collector and returns directly to the compressor. However, there is one distinct advantage that an ASHP has over a $GSHP_{DX}$ in that there is little, if any, drawdown on the source temperature for the ASHP. Ambient air is pulled through the evaporator by fans, cooled and expelled where it then disperses into the atmosphere. Windy conditions or the natural momentum in the exhaust air ensure that the cold air is removed from the vicinity of the evaporator, which is a typical condition in the Irish Maritime climate.

This in effect means there is no equivalent ΔT_G for an air source and the ambient air temperature is the source temperature. However, a quasi-drawdown exists for ASHP systems in the form of parasitic losses incurred during evaporator defrosting, which does not apply to GSHP systems. Nevertheless, modern defrosting strategies, such as the one incorporated in the *IRL-HP* study ASHP (TC-MACH), introduce elegant defrost management techniques that reduce the impact on COP to as little as 5%. Pumping power for GSHPs can impact on the COP by up as much as 15%.

7.2 ENVIRONMENTAL CONSIDERATIONS

Growing awareness of the links between energy consumption, CO_2 emissions and possible climate change necessitates a review of carbon emissions associated with heat pump operation. The potential for economic savings will always remain the driving force for change. Recent energy price volatility has underpinned the use of more sustainable, and cheaper, alternative sources such as heat pumps and future carbon emission taxes may enhance the incentive. The use of fossil fuels to deliver heating in domestic dwellings, commercial and industrial buildings is one of the largest sources of CO_2 emissions worldwide. Heat pumps offer a reliable and well developed alternative heating technology that can sustain thermal comfort with a substantial reduction in CO_2 emissions and can deliver CO_2 free heating if used in conjunction with renewably derived electricity. As electricity generation inevitably moves towards renewably driven sources the amount of CO_2 emitted as a by-product of electrical generation will continually decrease with time, making heat pumps more CO_2 efficient which is not possible with fossil fuel boilers. Eventually, if all the electricity generated is from a renewable source, heat pump heating systems will be CO_2 neutral. Therefore, heat pumps offer one of the most efficient, effective and versatile heating system on the market for reducing carbon emissions both now and into the future.

The following sub-sections characterise the environmental aspects of heat pump operation.

7.2.1 ELECTRICITY GENERATED CO_2 EMISSIONS

The amount of CO_2 released as a result of operating a heat pump is directly linked to the primary energy source of the electricity consumed, the efficiency by which the electricity was generated and the distribution losses incurred. Figure 7.7 illustrates the position Ireland hold in Europe regarding carbon emissions associated with electricity production.

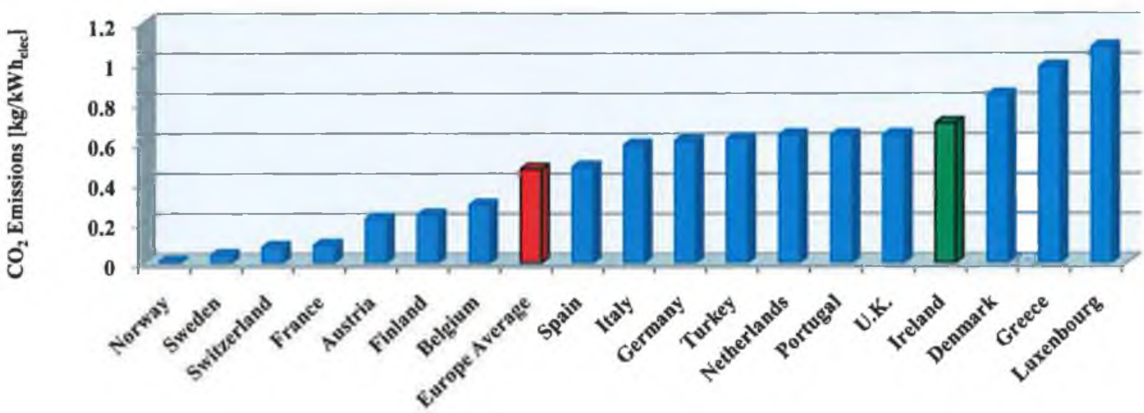


FIGURE 7.7 CO_2 EMISSIONS FROM ELECTRICITY GENERATION BY EU COUNTRIES (SEI, 2007A).

The most recent data in Figure 7.8 shows that 12% of Ireland's electricity was derived from renewable sources in 2008, with 8.1% delivered from wind and 3.2% from hydro (SEI, 2009).

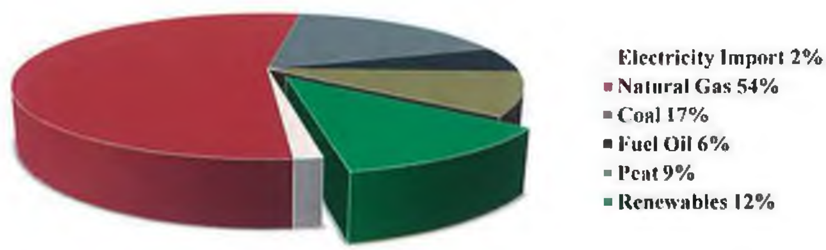


FIGURE 7.8 ELECTRICITY GENERATION IN IRELAND BY SOURCE (SEI, 2009).

While hydro power has negligible amounts of related CO_2 emissions, Ireland has very few sites offering additional capacity. By contrast, Ireland possesses a large wind resource and it has recently become government policy to target the delivery of 40% of Ireland's electricity from renewable sources by 2020 (DCENR, 2010).

7.2.2 EMISSIONS ASSOCIATED WITH HEAT PUMP OPERATION

Demmel and Alefeld (1994) assessed the potential CO_2 emissions reduction that could accrue from the use of heat pumps based on improved electricity generating efficiency and higher percentage of green electricity on the national/international grid. They concluded that when heat pumps are used in conjunction with modern gas-fired combined cycle power plants significant reductions on carbon emissions can be achieved even with COPs of between 2.5 and 3 (Demmel and Alefeld, 1994). This, combined with the steady increase in efficiency of modern heat pumps and the way the heat is collected, has further enhanced the suitability of heat pump in reducing CO_2 emissions. Environmental benefits of using heat pumps are potentially major, but in order to bring to fruition these benefits a number of factors must be resolved; system design falls short of its commercial promise; fails to account for hidden capital costs (trench digging may be an additional cost for example); tradesmen unskilled in the use of heat pumps; lack of government policy; and lack of economy of scale (Dowlatabadi and Hanova, 2007). Ireland lacks in each of the aforementioned factors, yet the technology offers substantial potential to reduce CO_2 emissions as outlined below. Based on the current electricity primary fuel mix in Ireland, Figure 7.9 indicates how heat pumps compare with other heating systems in terms of CO_2 emissions. The heating system's emissions are indicative of the fuel type used, the utilisation efficiency and transmission losses.

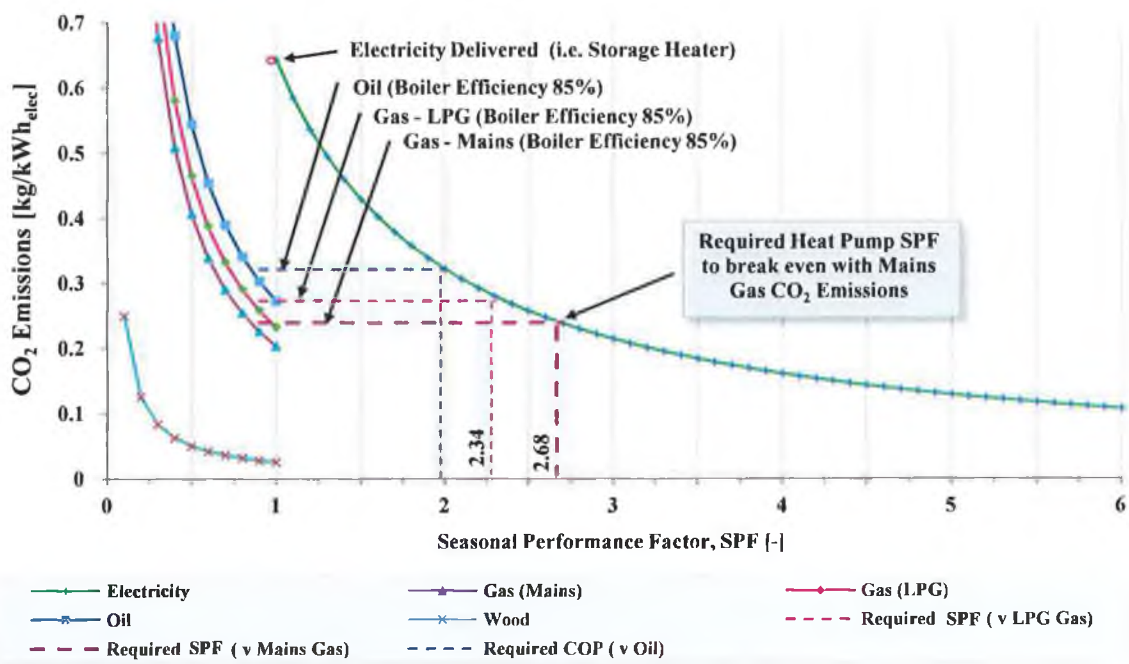


FIGURE 7.9 CO_2 EMISSIONS FROM VARIOUS HEATING SYSTEMS AS A FUNCTION OF HEAT PUMP SPF.

From a primary energy consumption perspective, Figure 7.10(a) further illustrates the capacity of heat pumps to reduce CO_2 emissions by utilising primary energy more efficiently than other space heating systems. Since Ireland's power generating efficiency is higher than the 'EU average' shown in Figure 7.10(a), a seasonal performance factor (SPF) of just 2.1 produces the equivalent primary energy ratio (PER) of any alternative fuel utilised at a typical efficiency of 85%. As primary energy is the full amount of energy imbedded in the fuel before conversion, the PER is the efficiency by which this energy is transformed into thermal energy. Thus, a heat pump achieving a SPF of 2.1 matches any other heating system in its use of fuel.

With regard to the role of heat pumps in the climate change debate Demmel and Alefeld (1994) recognised the increasing likelihood that CO_2 emissions associated with the use of heat pumps will be further reduced in future with improved electricity generation efficiency and as the percentage of green electricity on the grid increases. From the Irish government's target of achieving 40% renewable electricity supply by 2020, Figure 7.10(b) illustrates the predicted reduction in CO_2emissions from mains delivered electricity up to 2020.

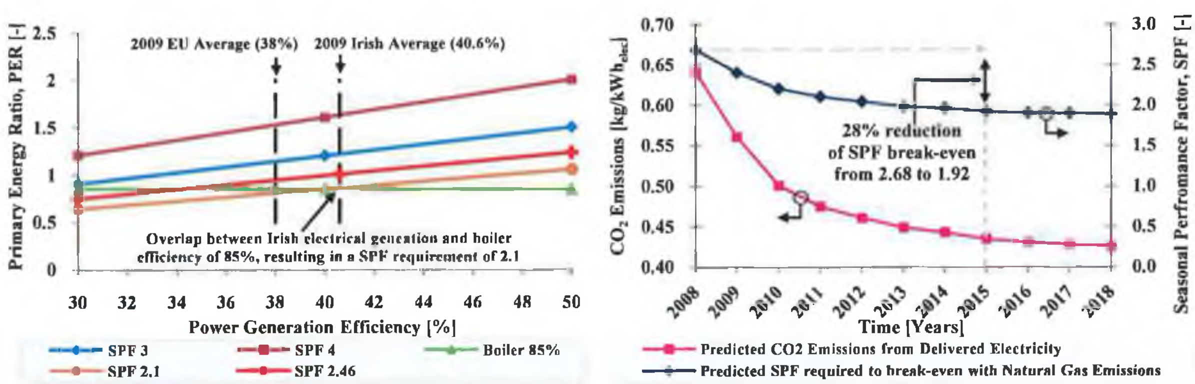


FIGURE 7.10 (A) PER GAINS THROUGH THE USE OF HEAT PUMPS, EU AVERAGE ELECTRICAL POWER GENERATION EFFICIENCY VERSUS IRELAND AVERAGE AND (B) PREDICTED CO_2 EMISSIONS FORECAST FOR ELECTRICITY GENERATION IN IRELAND AND RESULTANT HEAT PUMP SPF REQUIRED TO BREAK-EVEN (SEI, 2009).

As a result of this, from Figure 7.10(b) it can be seen that in 2015 a heat pump with a SPF of 1.92 will have the same CO_2 emissions as a gas condensing boiler. Consequently, a heat pump system with a SPF of 4 or above will produce 50% less emissions than that of a gas condensing boiler with an 85% conversion efficiency, further increasing the environmental advantage of heat pumps.

In summary, in a direct comparison with a conventional boiler, a moderately high efficiency heat pump system will reduce the use of primary energy and will also reduce the amount of hazardous gas emissions produced both locally and nationally.

7.3 HEAT PUMP COST SURVEY - 2009

A survey was conducted between May and July 2009 to identify the cost of procuring and installing a heat pump system for a domestic dwelling. The system design specification stipulated a maximum heating load of 10kW at the design ambient outside air temperature of -4°C. Target suppliers were selected from SEI's list of registered heat pump installers (SEI, 2007b) and 33 companies were invited to quote.

The survey requested potential suppliers to quote for all heat pump options provided and to include a capital and installation cost breakdown. There was a 55% response rate with 18 replies, and 16 valid quotes.

The responses are categorised as follows; 16 supplied quotes for a $GSHP_{HC}$ (100%), 5 quoted for a $GSHP_{VC}$ (31%) and 2 quoted for an ASHP (13%). Table 7.1 summarises the survey results.

TABLE 7.1 SUMMARY OF *HP-IRL* HEAT PUMP COST SURVEY RESULTS

		$GSHP_{HC}$	$GSHP_{VC}$	ASHP
Average Heat Pump Capacity		11.7 kW	11.8 kW	13.8 kW
Average Total System Cost per kW of Nominal Heat Pump Capacity		€1,221	€1,673	€1,182
Average Collector Cost*		€2,499	€5,420	-
Average Collector Cost per kW of Nominal Heat Pump Capacity		€213	€459	-
Total Heat Pump Installation Cost	**Average**	€14,303	€19,747	€16,330
	Maximum	€18,589	€25,893	€17,656
	Minimum	€11,781	€13,082	€15,005
Collector Length	**Average**	740m	175m	-
	Maximum	1200m	150m	-
	Minimum	520m	200m	-
Collector Area (Recommended)	**Average**	477m²	-	-
	Maximum	780m²	-	-
	Minimum	400m²	-	-

*The $GSHP_{HC}$ quotes include materials but excludes the cost of preparing and excavating the collector region

While the average size of the $GSHP_{HC}$ was 11.7kW (117% of maximum heat load) and the $GSHP_{VC}$ was 11.8kW (118%) one company quoted for a heat pump with a rated capacity of 8kW (80%). The latter's undersized heat pump may have been in error, or indeed confidence in the system's ability to perform at the upper end of its output scale under the milder Irish Maritime climate. The $GSHP_{HC}$ is the cheapest system quoted, costing an average 72% of the $GSHP_{VC}$, while the ASHP system is on average 83% of the $GSHP_{VC}$ cost. This lower $GSGP_{HC}$ cost does not however take into account the additional cost of labour and equipment costs for installing the horizontal collector. Digger hire can range from €200 to €300 per day, along with similar labour costs. From the survey results, the ASHP is the cheapest system per kW of nominal capacity, but the $GSHP_{HC}$ was the cheapest overall system. From the survey results, and including an additional €500 for installation, the horizontal collector system is just 55% of the cost of a vertical collector.

For the $GSHP_{HC}$ the average collector length supplied was 740m and with a heat pump COP of 3 this corresponds to a thermal extraction demand of 9W/m. The average recommended area required was 477m² and corresponding to a thermal extraction of 14W/m².

Table 7.2 shows the type and size of the heat pumps quoted. It shows a sizable array of 22 heat pump technologies on offer, representing 18 heat pump manufacturers. Of the 22 heat pump models quoted only 12 were registered with the Sustainable Energy Association of Ireland (SEAI) as part of the *Greener Home Scheme* (SEI, 2010).

TABLE 7.2 ALL *HP-IRL* HEAT PUMP SURVEY TYPES, SIZES AND COSTS

Manufacturer	Type	Brine to Water	Air to Water	Direct Expansion (DX)	Capacity	Heat Pump Only Cost (Ex Delivery, Installation and VAT)	No. of Quotes	SEI Registered System (2010)
Daikin	Altherma		✓		16kW	-	2	✓
Dimplex	SI14ME	✓			14kW	-	1	✓
Dimplex	LA16MS		✓		16kW	€10,462	2	✓
Geostar	i450	✓			13kW	€5,530	1	✓
Heliotherm	HP08E-WEB			✓	9kW	-	1	✓
ISARA	C HE	✓			13kW	-	1	
IVT	Greenline C11 HT Plus	✓			11kW	€9,650	1	✓
MasterTherm	-		✓		12kW	-	1	
NIBE	Fighter 1140	✓			12kW	€11,889, €8,400	2	✓
NIBE	Fighter 1240	✓			12kW	€10,600, €8,880	2	✓
NIBE	Fighter 2005		✓		11kW	€9,050	1	
Ochsner	GMSW	✓			10kW	€9,267	1	✓
Ochsner	GMLW		✓		14kW	€12,387	1	✓
Polar Bear	Phinx PASRW		✓		11kW	€5,742	1	✓
Remko	-		✓		16kW	€6,970	1	
Sanyo	-		✓		11kW	-	1	
Sirocco	HP1AW16		✓		16kW	-	1	
Stiebel Eltron	WPF S	✓			10kW	-	1	✓
Thermia	Comfort	✓			16kW	-	1	
Water Furnace	-	✓			12kW	€7,200	1	
Weider	-	✓			12kW	-	1	
ZEPHYRA	C HE		✓		13kW	-	1	

From the supplied quotations the most popular heat pump manufacturer was the Swedish based *NIBE* having been included in 19% of the quotes, with *Dimplex* (12%) and *Dalkin* (4%) being the next most popular.

Other heat pump system variables that must to be carefully considered are the control system, in terms of quality and programmability, and the size of the buffer tank that will be supplied with the system. While it was not possible to decipher the type and cost of the heat pump control systems from the survey, Table 7.3 shows the cost of buffer tanks and underfloor heating system that were also offered as part of the quote.

TABLE 7.3 *HP-IRL* SURVEY RESULTS FOR UNDERFLOOR HEATING AND BUFFER TANK COSTS

	Average	Maximum	Minimum
Buffer Tank Size	275 Litre	500 Litre	160 Litre
Buffer Tank Cost	€2,316	€4,200	€690
Underfloor Heating Cost (delivery only)	€9,614	€12,466	€5,465
Underfloor Heating Cost per square meter	€33	€43	€22

The *HP-IRL* cost survey delivered a snapshot of heat pump system costs in Ireland that offers insight into the variation in both cost and system capacity associated with the three heat pump systems. From the quotations there is a sizable array of system requirements that require a large degree of scrutiny in their evaluation. Determining what is and is not part of the design is not easily achieved, with a decision on the most efficient and cost effective system being over complicated. For the standard domestic dwelling heat loss (10kW) provided to the companies for quotation, the heat pump system size varies from 9kW to 16kW and cost wise from €11,781 to €25,893.

Differentiation of heat pump performance is based on the heat pump system, along with the difference between the sink and source temperatures. The heat pump system performance must be tested in accordance to the standard EN 14511 and adjusted for the additional collector fluid pumping requirements. The sink temperature is evaluated based on thermal requirements for space heating and/or domestic hot water and can vary from +35°C for underfloor heating only, up to +65°C for domestic hot water supply via a heat pump desuperheater. This parameter of heat pump operation is quantifiable based on the demand profile and space heating thermal delivery type, and is also the same for all three heat pump types. The only parameter that can affect the heat pump performance that is has a variable exposure to climatic influences is the source temperature as outlined in Section 7.1.

7.4 SUMMARY

This chapter outlined the thermal source, economic and environmental performance of all three heat pump collector technologies operating within the Irish Maritime climate and the findings are summarises as follows:

- The average farfield ground temperatures for the *HP-IRL* vertical collector was +11.4°C, and varied by 2.5K throughout the year. The typical average winter heating season farfield temperature was +10.9°C
- The average farfield ground temperatures for the *HP-IRL* horizontal collector was +11.9°C, and varied by 13K throughout the year. The typical average winter heating season farfield temperature was +7.9°C.
- The average ambient outside air temperatures for the *HP-IRL* ASHP performance evaluation was +8.9°C. The typical average winter heating season ambient outside air temperature was +7.3°C.

- The horizontal and vertical collector show similar rates of ground temperature drawdown under steady-state conditions, but the horizontal collector drawdown is less reactive to changes in the rate of thermal extraction.
- The average SPF over the three winter heating seasons was 2.90, 2.95 and 3.74 for the $GSHP_{HC}$, $GSHP_{VC}$ and ASHP respectively.
- For Ireland, a heat pump requires an SPF of 2.1 to break even with an 85% efficient boiler in terms of PER.
- In terms of kg of CO_2 per kWh_{th} delivered, a heat pump requires an SPF of 2.7 to break even with an 85% efficient gas condensing boiler utilising natural gas. With improvements in electricity production, along with more renewable energy derived electricity, by 2015 this will potentially reduce to 1.9.
- Extensive heat pump cost survey of 33 companies performed, showing contemporary heat pump costs, manufacturers and designs.
- For a 10kW design heat loss, an average capacity heat pump of 11.7kW, 11.8kW and 13.8kW was stipulated for a $GSHP_{HC}$, $GSHP_{VC}$ and ASHP respectively.
- The average cost of the $GSHP_{HC}$, $GSHP_{VC}$ and ASHP systems was €14,303, €19,747 and €16,330 respectively - the cost of installing the horizontal collector was not represented in the quotations and is estimated to at €500 to the overall cost, still leaving it the cheapest system.
- An installed horizontal collector costs only 55% that of a vertical collector.
- The average cost per kW of heat pump capacity was €1,221, €1,673 and €1,182 for a $GSHP_{HC}$, $GSHP_{VC}$ and ASHP respectively.
- Of all the heat pump quotations received, system costs including installation, ranged from €11,781 to €25,893.

CHAPTER 8 – CONCLUSIONS AND RECOMMENDATIONS

This *HP-IRL* study aimed to develop a comprehensive test facility for the functional characterisation of GSHP and ASHP technologies in the Irish Maritime climate to support climate sensitive performance analysis, collector design and source side management.

This study delivered:

- A thorough review of the pertinent literature relating to: international heat pump characterisation facilities and test standards; climate classifications and factors that affect heat pump performance; operational dynamics of heat pumps and collector regions; best practice methods of assessing heat pump performance and design of experiments along with heat pump collector design criteria.
- A comprehensive test facility employing 111 sensors and supporting data acquisition was designed, built and commissioned to enable the performance evaluation of functioning $GSHP_{HC/VC}$ and ASHP technologies within the Irish Maritime climate.
- A 747 day long experimental test program was carried out on both heat pump technologies and three collector types to identify the influence of different interrelated parameters on performance such as: the climate and specific weather events; ground material and ground cover; the impact of duty cycle and consecutive year thermal extraction on source temperature; the influence of heat pump duty on source temperature, ground temperature drawdown, thermally affected zones around the collector pipes and recovery rates. This yielded a collector performance indicator for horizontal collector performance sensitivity to be established.
- The potential to maximise horizontal collector performance in Maritime climate conditions was assessed using a model to simulate the impact of varying collector length, depth, surface covers and configurations. This allowed a new climate sensitive horizontal collector design and operational parameter to be developed.
- A review of the economic costs and environmental benefits of heat pump installation and utilisation in Ireland was conducted.

The main Conclusions and Recommendations are listed in the following sections.

8.1 Conclusions

Conclusions are subdivided by heat pump collector type, but the influence of climate is considered at the outset.

8.1.1 Climate

Since *HP-IRL* targeted deeper insights of climate sensitive heat pump performance it simultaneously monitored climate, heat pump duty and collector performance. This showed that:

- The mild and moist Maritime climate is positively disposed to both $GSHP_{HC}$ and ASHP than other climate types at the same latitude.
- Heat pump duty cycle is consistently inversely proportional to ambient air temperature and higher duty cycles impact most on ground temperature.

The following conclusions were drawn concerning horizontal collectors:

- Monthly average ground temperatures are predictable to an average accuracy of ±0.3°C at a depth of 1m but unpredictable hourly averages may fluctuate by ±2.5°C from the monthly average due to weather events. At the ground surface the hourly averages may fluctuate between ±10°C from the monthly average. This poses both a challenge and an opportunity for source side management.
- Ground moisture content impacts on thermal conductivity and capacity and was noted to dry out from the surface layer during dry periods of more than 6 days and there is minimal impact on ground moisture content at 0.3m depth, with no impact at 0.9m depth. Substantial drying was not noted below 0.3m as frequent rainfall events maintained soil moisture above the critical 25%.
- A novel three dimensional graphical analysis technique was developed to resolve the impact of specific weather events on the ground temperature field. This analysis showed that the ground's surface layer (<1m deep) is being constantly thermally charged and discharged by a combination of; solar radiation (positive), air-ground surface temperature difference (positive/negative) and these effects are amplified by both wind speed and rainfall and their impact changes with the seasons, ground cover and ground type.

Considering the vertical collector, *HP-IRL* showed:

- Weather events impacted on ground temperature to a depth of 2m
- Seasonal effects were evident to a depth of 15m

Considering the ASHP, *HP-IRL* showed:

- Winter time ambient external air temperature fluctuated between 0°C and +12°C.
- Consistently high relative humidity levels in the range of 60-95% were recorded throughout the heating season, which induce high defrost requirements.

8.1.2 HORIZONTAL COLLECTOR GROUND SOURCE HEAT PUMP

The following conclusions have been drawn from the experience of operating the 430m² horizontal collector for 293 days, nine individual tests, during which time 69,514kWh of thermal energy (250GJ) was delivered.

- COP ranged from 2.7 to 3.4 for temperature lifts of 46.7K and 35.0K respectively and a SPF of 2.90 was achieved while delivering an average sink temperature of +49.1°C.
- COPs were measured to an accuracy of ±3.3% and recorded values matched those published by the independent *Arsenal Research* test institute, under EN-14511 to within 2.2%. However, the difference reached 6.5% when collector pumping power, not considered by EN-14511 is included.
- Reflecting the mild climate, the minimum farfield ground temperature at the collector depth was +3.7°C.
- The average collector return temperature was +4.1°C and the lowest was +0.8°C. This allows the level of frost protection to be lowered from -15°C to -7°C, increasing collector fluid thermal properties and reducing pumping power.
- Collector drawdown was a function of duty and the greatest hourly average ground temperature drawdown was -5.2K with a 94% extract rate of 9.8kW (23W/m²).
- New methods of identifying and analysing the fluctuating Thermally Affected Zone (TAZ) around an active collector were presented.
- Ground thermal recovery is a function of the climate, ground type and surface cover.
- Ground cover type was shown to influence solar absorption with ground temperature at 300mm below a brick surface displaying a 4°C higher temperature during spring and summer than under grass. Negligible difference existed during winter.
- A new parameter called the Collector Performance Indicator (*CPI*) has been defined to indicate the climate sensitive performance of horizontal collectors. A *CPI* of 0.23K/(W/m²) was measured for the *HP-IRL* collector based on ground temperature drawdown of 0.23K per unit demand (W) of collector area (m²).

- A suite of 3-D graphical analysis tools were developed to assess the impact of climate/specific weather events and duty cycle on the collector ground temperature to be identified. This showed higher concentrations of thermal energy in the upper 500mm ground layer during spring and summer.
- An excel based simulation tool was developed to facilitate the evaluation of climate sensitive collector designs which allowed the optimum collector length and depth to be established and collector performance in a specific climate with source side management to be assessed.
- The modelling tool was used to assess the performance of a new split level collector positioned 0.9m beneath grass and 0.3m beneath brick. Using a simple source side management routine a SPF improvement of 8.1% over a VDI-4640 standard horizontal collector design was predicted and the *CPI* falls to 0.21K/(W/m²) indicating increased collector performance.

The horizontal collector simulation tool creates an opportunity to develop and further refine climate sensitive collector designs in conjunction with more effective source side management techniques. Initial results show the potential exists to increase heat pump SPF by 8-15%.

8.1.3 VERTICAL COLLECTOR GROUND SOURCE HEAT PUMP

Over the course of eight tests the vertical collector operated for 289 days, delivered 74,011kWh of energy (266GJ) and analysis of results yielded the following conclusions:

- COP ranged from 2.8 to 3.4 for temperature lifts of 44.8K and 38.8K respectively and a SPF of 2.95 was achieved while delivering a sink temperature of +49.5°C.
- COPs were measured to an accuracy of ±3.3%.
- The climate has a negative impact on COP during winter and spring as it reduced the farfield temperature on over 31% of the collector length. As a result the effective collector farfield temperature ranged from +10°C to +12.8°C, averaging +11.4°C overall.
- The average collector return temperature was +6.9°C, the lowest was +3.0°C.
- Collector drawdown is more sensitive to duty than the horizontal collector, the greatest hourly average ground temperature drawdown was -7.4K with an extract rate of 11.4kW (30W/m). The average drawdown was -4.1K compared with -3.5K for the horizontal collector.

- Ground thermal recovery rates vary along the entire length of the boreholes.
- A thermal response test yielded an effective ground thermal conductivity of between 5.9 and 6.3 W/m·K, some 10% higher than expected for limestone, possibly reflecting the influence of ground water movement.
- The resilience of the collector was reflected by a measured temperature penalty of just 0.25°C over three years of thermal extraction.

While the vertical collector performed consistently each year and delivered a similar SPF to the horizontal collector, the negative influence of the climate during winter could be mitigated using thermal insulation along the return line from a depth of 15m.

8.1.3 AIR SOURCE HEAT PUMP

The ASHP was operated for 165 days over the winter and spring of 2008/2009, delivering 24,997kWh of energy (90GJ) and the following conclusions were drawn:

- The heat pump delivered an average heat pump sink temperature of +47.8°C and COP ranged from 2.6 to 4.7 with temperature lifts of 45.5K and 20.6K respectively. The wide variation in COP was attributable to sensitivity to changes in ambient air temperature. This impressive SPF of 3.74 was achieved with an average external ambient air temperature of +8.9°C. This SPF is 29% and 27% higher than the horizontal collector and vertical collector sources heat pumps.
- The lowest recorded hourly averaged ambient air temperature was +1.7°C and the heat pump achieved an hourly averaged COP of 3.5, which highlights the ability of the unit to minimise the impact of defrosting.

While the Maritime climate did not produce a warmer ambient air source temperature (+8.9°C) than say the +11.4°C for the vertical collector that 27% higher SPF achieved demonstrates the high performance of the ASHP tested and its suitability for use in the Maritime climate.

8.1.4 TECHNO-ECONOMIC AND ENVIRONMENTAL STUDY

The technology comparison of collector performance showed:

- The average farfield ground and air temperatures for the $GSHP_{HC}$, $GSHP_{VC}$ and ASHP during the heating season was +7.9°C, +10.9°C and +8.9°C respectively.

- The horizontal and vertical collector show similar rates of ground temperature drawdown under steady-state conditions, but the horizontal collector is less reactive to changes in the rate of thermal extraction.
- The average SPF over the three winter heating seasons was 2.90, 2.95 and 3.74 for the $GSHP_{HC}$, $GSHP_{VC}$ and ASHP respectively.

The environmental review showed:

- A heat pump requires an SPF of 2.1 in Ireland to generate equal Primary Energy Ratio (PER) to that of an 85% efficient gas boiler.
- In terms of kg of CO_2 per kWh_{th} delivered, a heat pump requires an SPF of 2.7 to break even with an 85% efficient gas condensing boiler utilising natural gas. With improvements in electricity production, along with more renewable energy derived electricity, by 2015 this will potentially reduce to 1.9.

An economic study was undertaken to compare the cost of all three collector types, including associated heat pumps, and the results of the 16 responses obtained showed:

- For a 10kW design heat loss, an average capacity $GSHP_{HC}$, $GSHP_{VC}$ and ASHP of 11.7kW, 11.8kW and 13.8kW respectively was recommended.
- The average $GSHP_{HC}$, $GSHP_{VC}$ and ASHP cost €14,803, €19,747 and €16,330 respectively, yielding an average cost per kW of heat pump capacity of €1,265, €1,673 and €1,182.
- Of all the heat pump quotations received, system costs including installation, ranged from €11,781 to €25,893.

8.2 RECOMMENDATIONS FOR FUTURE WORK

The availability of both the test facility and test results now affords an excellent opportunity to conduct further work in the following areas:

Data analysis:

- Use data to refine source side management techniques for $GSHP_{HC/VC}$
- Refine numerical models and benchmark predictions against experimental data
- Explore potential for other collector performance parameters such as *CPI*

Enhance test facility:

- Horizontal Collector: add new ground covers; thermal storage; heat flux sensors

- Vertical Collector: sensor new profiles away from borehole to analyse groundwater and heat movement
- Air Source Heat Pump: evaluate the potential for preheating air through ground

New tests:

- Ground thermal storage in the vertical and horizontal collector
- Change flow directions in vertical and horizontal collector
- New Systems: hybrid renewable energy systems such as HP-SOLAR and HP-WIND

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APPENDIX A

The *Seasonal Performance Factor (SPF)* is established by summing the total useful energy output of a heat pump over a season and divided by the total electrical energy input during the same period. This can represent both heating and cooling.

$$SPF_{heating} = \frac{Heat\ Delivered\ (kWh)}{Electrical\ Energy\ Supplied\ (kWh)} \qquad \textit{Equation A.1}$$

The COP provides a micro analysis of the efficiency with which a heat pump uses the consumed electrical power. However, this figure does not represent the heat pump's ability to provide renewable energy and one must use another parameter to benchmark its output against other forms of heating systems.

The *Primary Energy Ratio (PER)* takes into account the COP along with the efficiency of the conversion process by which the primary fuel (oil, gas, peat, wind, or solar photovoltaic) generates the electricity, and the transmission losses accrued in delivering the electricity to the heat pump (Granryd, 2005).

$$PER = \frac{Useful\ heat\ delivered\ by\ the\ heat\ pump}{Primary\ energy\ consumed} \qquad \textit{Equation A.2}$$

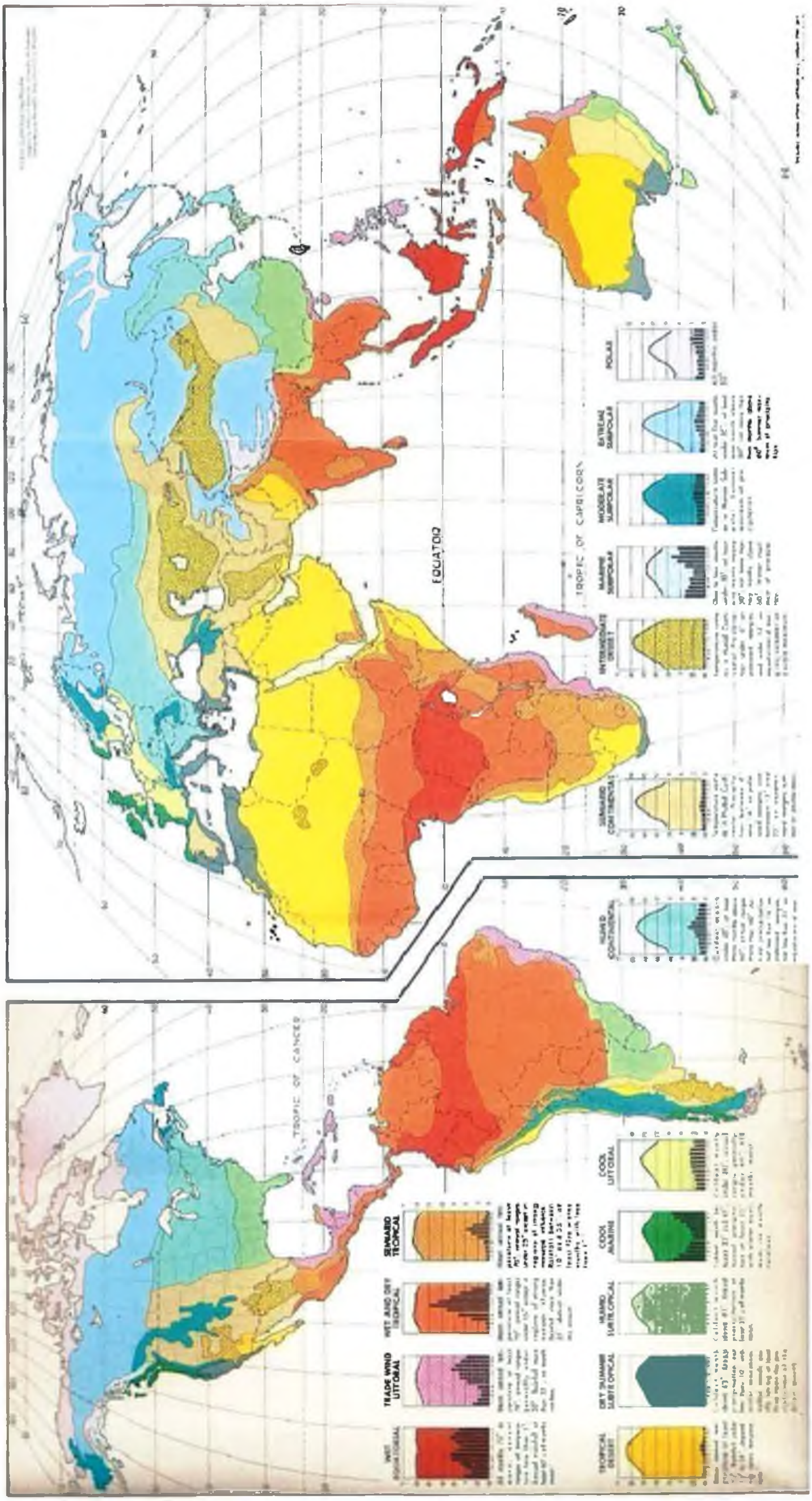
or

$$PER = \eta_{GEN}\ x\ COP \qquad \text{Equation } A.3$$

Where η_{GEN} is the efficiency in which the electrical power is generated and delivered to the heat pump. Once the heat pump PER exceeds 1 this indicates that the heat pump delivers more heat energy than the equivalent fossil fuel based heating system thereby reducing cost and CO_2 emissions.

The *operating duty* represents the output demanded and timeframe of a heat pump within a heating system. The operating duty of a heating system varies greatly from application to application and is a substantial factor in sizing a heat pump system.

APPENDIX B


TROPIC OF CANCER
EQUATOR
TROPIC OF CAPRICORN
WET EQUATORIAL
TRADE WIND LITTORAL
WET AND DRY TROPICAL
SEMIARID TROPICAL
TROPICAL DESERT
DRY SUMMER SUBTROPICAL
HUMID SUBTROPICAL
COOL MARINE
COOL LITTORAL
HUMID CONTINENTAL
SEMIARID CONTINENTAL
INTERMEDIATE DESERT
MARINE SUBPOLAR
MODERATE SUBPOLAR
EXTREME SUBPOLAR
POLAR

Koeppe Climate Classifications:

Wet Equatorial

All months 70°F (+21°C) or more, annual range of temperature less than 5°F (2.7°C). Annual rainfall at least 60" (1500mm), all months moist.

Trade Wind Littoral

Mean annual temperature at least 70°F (+21°C), annual ranges generally under 20°F (10.8°C). Rainfall more than 35" (890mm), no month rainless.

Wet and Dry Tropical

Mean annual temperature at least 70°F (+21°C), annual ranges generally under 15°F (7.1°C), except in strong monsoon influence. Rainfall more than 35" (890mm), distinct winter dry season.

Semiarid Tropical

Mean annual temperature at least 70°F (+21°C), annual ranges generally under 25°F (13.5°C), except in strong monsoon influence. Rainfall between 10" (250mm) and 35" (890mm), at least 5 winter months with less than 1" (25mm).

Tropical Desert

Mean annual temperature at least 70°F (+21°C). Rainfall under 10" (250mm) to 14" (350mm), depending upon temperature.

Dry Summer Subtropical

Coldest month above 45°F (+7.2°C). Annual precipitation not less than 10" (250mm) with winter maximum, wettest month usually having at least three times the precipitation of the driest month.

Humid Subtropical

Coldest month above 40°F (+4.4°C). Annual rainfall at least 35" (890mm), all months moist.

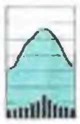
Cool Marine

Coldest month between 30°F (-1.1°C) and 45°F (+7.2°C). Annual precipitation at least 35" (890mm) with winter maximum, no month rainless.

Cool Littoral

Coldest month under 40°F (+4.4°C), annual ranges generally under 40°F (21.6°C). All months moist.

Humid Continental



Coldest month under 40°F (+4.4°C), at least three months above 60°F (+15.5°C), annual ranges more than 40°F (21.6°C). Annual precipitation not less than 16" (400mm) on poleward margins, nor less than 22" (560mm) on equatorial margins.

Semiarid Continental

Temperature same as *Humid Continental*. Precipitation between 6" (150mm) and 16" (400mm) on poleward margins, and between 12" (300mm) and 22" (560mm) on equatorial margins; summer or double maximum.

Intermediate Desert

Temperature same as *Humid Continental*. Precipitation under 6" (150mm) on poleward margins, and under 12" (300mm) on equatorial margins; summer or double maximum.

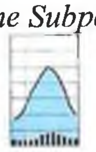
Marine Subpolar

One to four months under 30°F (-1.1°C), at least one month above 50°F (+10°C); not more than two months above 60°F (+15.5°C). Winter maximum of precipitation.

Moderate Subpolar

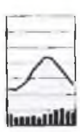
Temperature same as *marine subpolar*. Summer maximum of precipitation.

Extreme Subpolar



One to five months under 30°F (-1.1°C), at least one month above 50°F (+10°C); not more than two months above 60°F (+15.5°C). Summer maximum of precipitation.

Polar



All months under 50°F (+10°C).

APPENDIX C

HEAT PUMP STANDARDS

EN 1861:2008	Refrigeration systems and heat pumps. Flexible pipe elements, vibration isolators, expansion joints and non-metallic tubes. Requirements, design and installation.
EN 12693:2008	Refrigeration systems and heat pumps. Safety and environmental requirements.
EN 15450:2007	Heating systems in buildings – Design of heat pump heating systems
EN 14276:2006/2007	Pressure equipment for refrigerating systems and heat pumps - Vessels & Piping – General requirements
CEN/TS 14825:2006	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – Testing and rating at part load conditions
EN 15316-4-2:2005	Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4-2: Space heating generation systems, heat pump systems
EN 14511:2004	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling – requirements and testing
EN 12178:2003	Refrigerating systems and heat pumps – Liquid level indicating devices – Requirements, testing and marking
EN 255-2:2001	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors – Heating mode – Part 2: Testing and requirements for marking for space heating units (superseded by EN 14511)
EN 13136:2001/A1:2005	Refrigerating systems and heat pumps – Pressure relief devices and their associated piping – Method for calculation
EN 13313:2001	Refrigerating systems and heat pumps – Competence of personnel
EN 378:2000/2003/2008	Refrigerating systems and heat pumps – Safety and environmental requirements
EN 1736:2000	Refrigerating systems and heat pumps – Flexible pipe elements, vibration isolators and expansion joints – Requirements, design and installation
EN 12309:1999/2000	Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW – Safety and rational use of energy
EN 12263:1998	Refrigerating systems and heat pumps – Safety switching devices for limiting the pressure – Requirements and tests
EN 1861:1998	Refrigerating systems and heat pumps – System flow diagrams and piping and instrument diagrams – Layout and symbols
ISO 13256-1:1998	Water-source heat pumps – Testing and rating for performance – Part 1: Water-to-air and brine-to-air heat pumps
ISO 13261:1998	Sound power rating of air-conditioning and air-source heat pump equipment
ISO 13253:1995	Ducted air-conditioners and air-to-air heat pumps – Testing and rating for performance
ISO 5151:1994	Non-ducted air conditioners and heat pumps – Testing and rating for performance
ISO 5149:1993	Mechanical refrigerating systems used for cooling and heating - Safety requirements

Heat pump test institutes/standards

There are a number of European based heat pump test facilities that will carry out efficiency assessments of heat pumps in accordance with relevant standards. These test institutes are listed in Table C.1.

Table C.1 Heat pump test institutes

Country	Test Institute
Sweden	Swedish National Research and Testing Institute (P-label)
Austria	Arsenal Research
Holland	TNO-MEP Centre for development and Testing of Heat Pumps
France	Centre Technique des Industries Aérauliques et Thermiques (CETIAT)
Switzerland	Buchs Heat-pump checking and test centre
Germany	TÜV

Sweden uses the "*P-label*" as a quality labelling scheme which covers the heat pump efficiency over a range of operating conditions, noise levels, quality of literature with the system and conformance to CE-marking. Sweden also uses a label "*Swan*" to demonstrate the products good environmental attributes. However, the most renowned label in Europe is the "*DACH-label*". This label was initially set up in the German speaking countries, Germany (D) Austria (A) and Switzerland (CH) who agreed to establish a common set of criteria for the quality labelling of heat pumps. The DACH label requires satisfactory levels of quality under the headings: energy efficiency, operating range, manual, warranty, service capability and availability of spare parts.

The thermal efficiency of heat pumps operating in the heating mode are tested under steady-state operational conditions according to the European standard EN 14511. This standard was introduced in 2004 and supersedes the older standard 255 and the recommended test conditions for both standards are presented in Figure C.1.

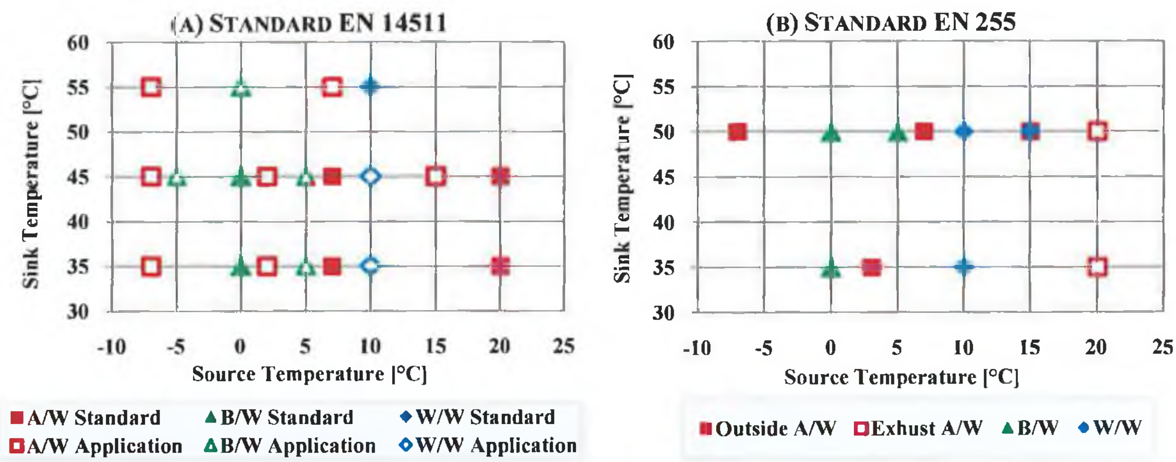
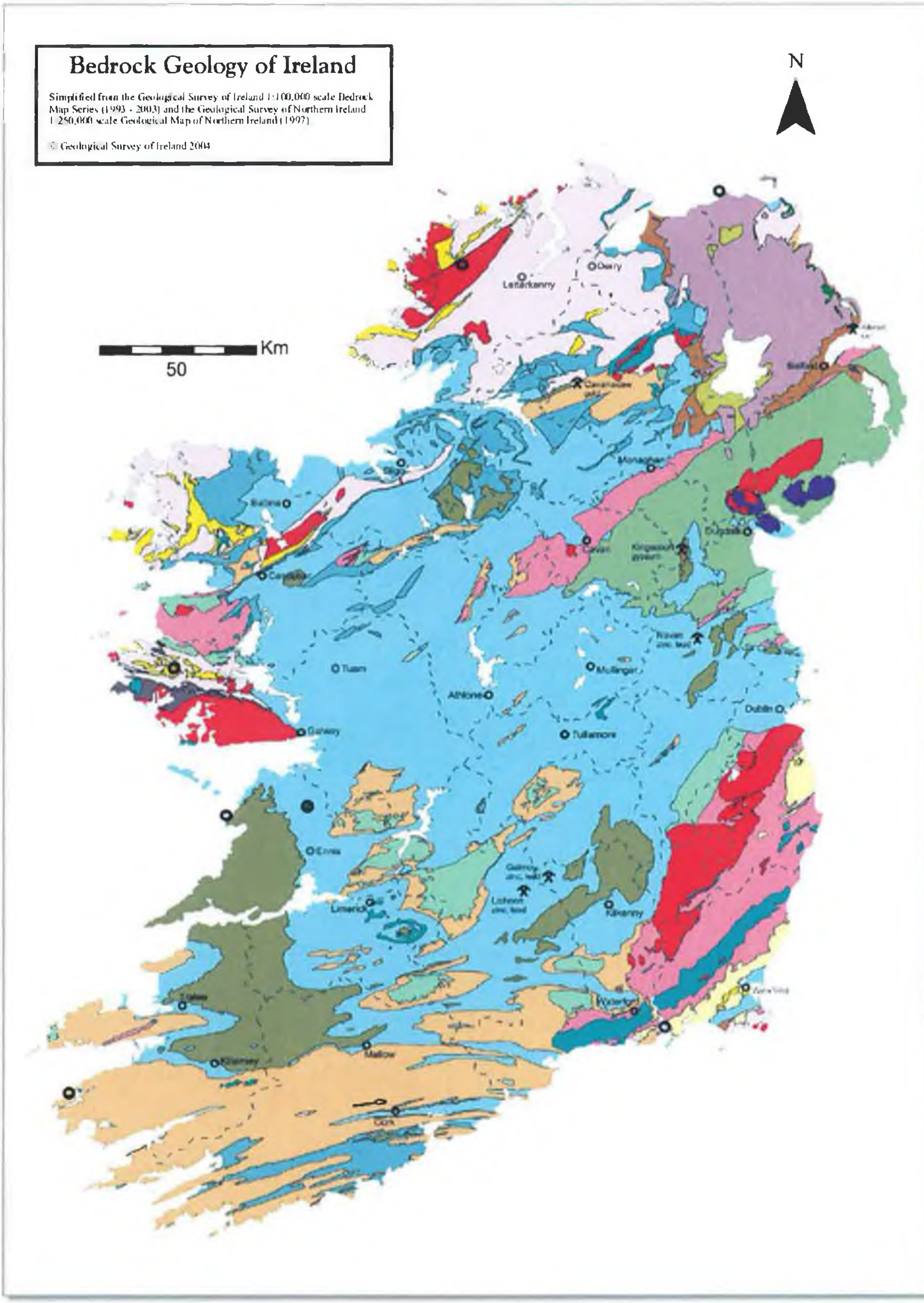
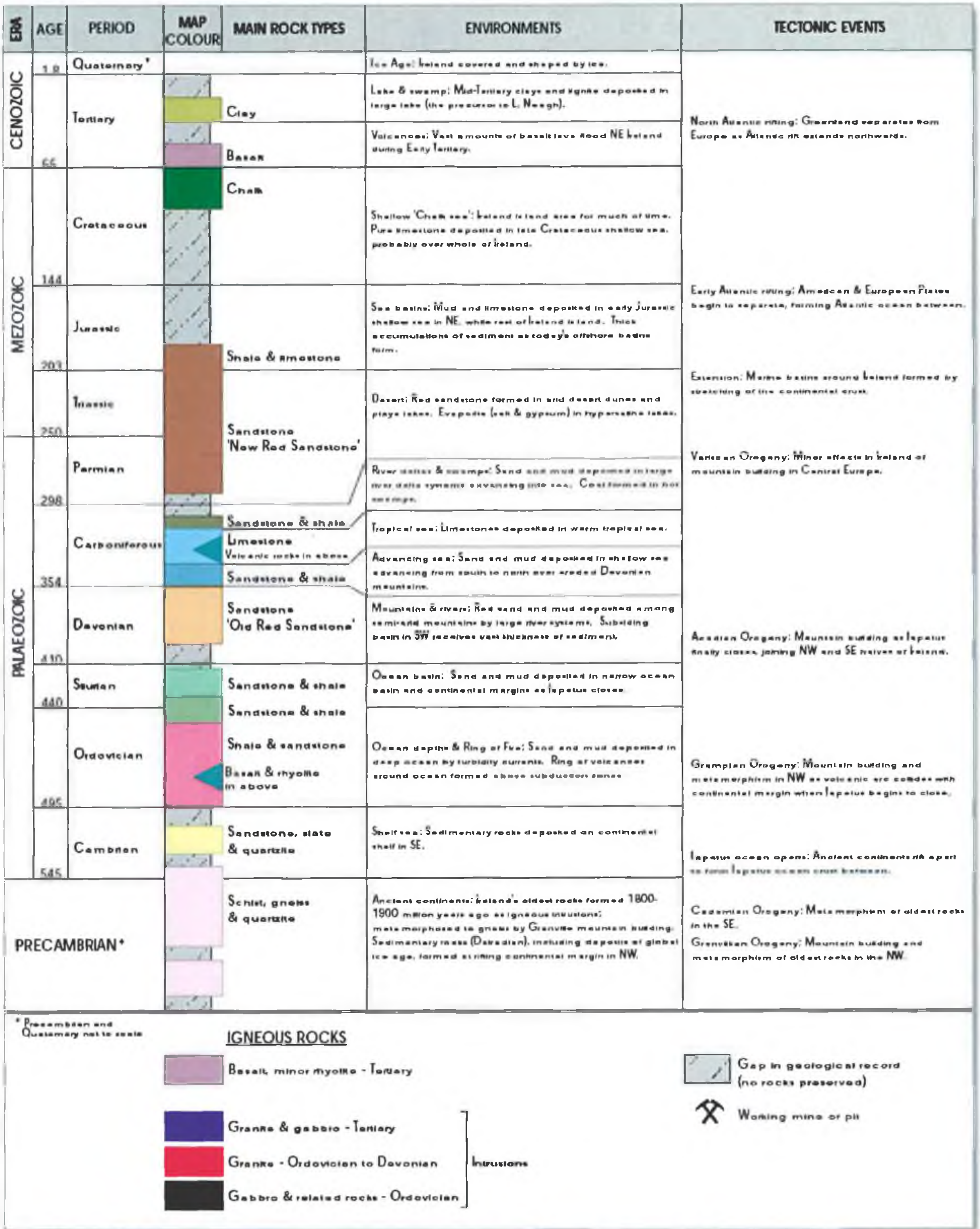


Figure C.1 Standard temperatures for heat pump performance analysis.

Appendix D


Bedrock Geology of Ireland
Simplified from the Geological Survey of Ireland 1:100,000 scale Bedrock Map Series (1993 - 2003) and the Geological Survey of Northern Ireland 1:250,000 scale Geological Map of Northern Ireland (1997)
© Geological Survey of Ireland 2004
N
Km
50
Letterkenny
Derry
Belfast
Sligo
Ballina
Monaghan
Cavan
Dundalk
Tuam
Mullingar
Athlone
Dublin
Tullamore
Galway
Ennis
Limerick
Kilkenny
Waterford
Mallow


ERA
AGE
PERIOD
MAP COLOUR
MAIN ROCK TYPES
ENVIRONMENTS
TECTONIC EVENTS
CENOZOIC
MEZOZOIC
PALAEOZOIC
PRECAMBRIAN*
1.8
65
144
203
250
298
354
410
440
495
545
Quaternary*
Tertiary
Cretaceous
Jurassic
Triassic
Permian
Carboniferous
Devonian
Silurian
Ordovician
Cambrian
Clay
Basalt
Chalk
Shale & limestone
Sandstone 'New Red Sandstone'
Sandstone & shale
Limestone
Volcanic rocks in above
Sandstone & shale
Sandstone 'Old Red Sandstone'
Sandstone & shale
Sandstone & shale
Shale & sandstone
Basalt & rhyolite in above
Sandstone, slate & quartzite
Schist, gneiss & quartzite
Ice Age: Ireland covered and shaped by ice.
Lake & swamp: Mid-Tertiary clays and lignite deposited in large lake (the precursor to L. Neagh).
Volcanoes: Vast amounts of basalt lava flood NE Ireland during Early Tertiary.
Shallow 'Chalk sea': Ireland island area for much of time. Pure limestone deposited in late Cretaceous shallow sea, probably over whole of Ireland.
Sea basins: Mud and limestone deposited in early Jurassic shallow sea in NE, while rest of Ireland is land. Thick accumulations of sediment as today's offshore basins form.
Desert: Red sandstone formed in arid desert dunes and playa lakes. Evaporite (salt & gypsum) in hypersaline lakes.
River deltas & swamps: Sand and mud deposited in large river delta systems advancing into sea. Coal formed in hot swamps.
Tropical sea: Limestones deposited in warm tropical sea.
Advancing sea: Sand and mud deposited in shallow sea advancing from south to north over eroded Devonian mountains.
Mountains & rivers: Red sand and mud deposited among semi-arid mountains by large river systems. Subsiding basin in SW receives vast thickness of sediment.
Ocean basin: Sand and mud deposited in narrow ocean basin and continental margins as Iapetus closes.
Ocean depths & Ring of Fire: Sand and mud deposited in deep ocean by turbidity currents. Ring of volcanoes around ocean formed above subduction zones.
Shelf sea: Sedimentary rocks deposited on continental shelf in SE.
Ancient continents: Ireland's oldest rocks formed 1800-1900 million years ago as igneous intrusions; metamorphosed to gneiss by Grenville mountain building. Sedimentary rocks (Dalradian), including deposits of global ice age, formed at rifting continental margin in NW.
North Atlantic rifting: Greenland separates from Europe as Atlantic rift extends northwards.
Early Atlantic rifting: American & European Plates begin to separate, forming Atlantic ocean between.
Extension: Marine basins around Ireland formed by stretching of the continental crust.
Variscan Orogeny: Minor effects in Ireland of mountain building in Central Europe.
Acadian Orogeny: Mountain building as Iapetus finally closes, joining NW and SE halves of Ireland.
Grampian Orogeny: Mountain building and metamorphism in NW as volcanic arc collides with continental margin when Iapetus begins to close.
Iapetus ocean opens: Ancient continents rift apart to form Iapetus ocean crust between.
Cadomian Orogeny: Metamorphism of oldest rocks in the SE.
Grenvillian Orogeny: Mountain building and metamorphism of oldest rocks in the NW.
* Precambrian and Quaternary not to scale
IGNEOUS ROCKS
Basalt, minor rhyolite - Tertiary
Granite & gabbro - Tertiary
Granite - Ordovician to Devonian
Gabbro & related rocks - Ordovician
Intrusions
Gap in geological record (no rocks preserved)
Working mine or pit

APPENDIX E

MEASUREMENT EXCELLENCE SINCE 1830

Kipp & Zonen 175 years

Kipp & Zonen B.V.
Röntgenweg 1 2624 BD Delft
P.O. Box 507 2600 AM Delft
The Netherlands
T +31 (0)15 269 8000
F +31 (0)15 262 0351
E info@kippzonen.com
www.kippzonen.com

CALIBRATION CERTIFICATE
PYRANOMETER

PYRANOMETER MODEL	:	CMP 3
SERIAL NUMBER	:	050093
SENSITIVITY at normal incidence on horizontal pyranometer	:	13.46 μV/W/m^2
IMPEDANCE	:	33 Ohm

CALIBRATION PROCEDURE : The indoor calibration procedure is based on a side-by-side comparison with a reference pyranometer under an artificial sun fed by an AC voltage stabiliser. It embodies a 150 W Metal-Halide high-pressure gas discharge lamp. Behind the lamp is a reflector with a diameter of 16.2 cm. The reflector is 110 cm above the pyranometers producing a vertical beam. The reference and test pyranometers are mounted horizontally on a table, which can rotate. The irradiance at the pyranometers is approximately 500 W/m². During the calibration procedure the reference and test pyranometer are interchanged to correct for any non-homogeneity of the beam. The dark offsets of both pyranometers are measured before and after the interchange and taken into account.

REFERENCE PYRANOMETER : Kipp & Zonen CM 3 sn950512 active from 01/01/2005.

hierarchy of traceability : This pyranometer was compared with the sun and sky radiation as source under mainly clear sky conditions using the "continuous sun-and-shade method". The readings are referred to the World Radiometric Reference (WRR) as stated in the WMO Technical Regulations. The measurements were performed in Davos (latitude: 46.8143°, longitude: -9.8458°, altitude: 1588m above sea level).

The inclination of the receiver surfaces versus their horizontal position were set to 0.0 degrees, the instrument signal wire to the north. During the comparisons, the instrument received global radiation intensities from 653 to 1005 with a mean of 820 W/m^2. The angle between the solar beam and the normal of the receiver surface varied from 24 to 50 with a mean of 39 degrees. The instrument temperature ranged from +11.7 to +20.7 with a mean of +17.7°C. The sensitivity calculation and the single measurements deviation (σ) are based on 1001 individual measurements. The obtained sensitivity value is valid for similar conditions and is: 16.04 ± 0.11 μV/W/m^2 (but is corrected by Kipp & Zonen to 16.33 μV/W/m². See "correction applied" below.)
The testing was done June 7, July 28, 29, September 2, 5, 6, 7 and 9, 2004.

Global radiation data were calculated from the direct solar radiation as measured with the absolute cavity pyrheliometer HF18748 (member of the WSG, WRR-Factor: 0.99568, based on the last International Pyrheliometer Comparison IPC-2000) and from the diffuse radiation as measured with a continuous disk shaded pyranometer Kipp & Zonen CM 22 sn020059 with sensitivity 8.91 (ventilated with heated air, instrument-wire to the north).

correction applied : +1.8 %
This correction was necessary to correct for the mean directional errors of the reference CM 3 in Davos. This error is estimated at Kipp & Zonen measuring the cosine error for the mean angle of incidence at azimuth S-30° and S+30°. The reference CM 3 now measures the vertical directed beam of the indoor calibration facility more correctly.

IN CHARGE OF TEST : F. de Wit Date:August 29, 2005 Kipp & Zonen, Delft, Holland

Notice
The calibration certificate supplied with the instrument is valid from the date of shipment to the customer. Even though the calibration certificate is dated relative to manufacture or recalibration the instrument does not undergo any sensitivity changes when kept in the original packing. From the moment the instrument is taken from its packaging and exposed to irradiance the sensitivity will deviate with time. See also the 'non-stability performance (max. sensitivity change / year) given in the radiometer specification list.

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CALIBRATION CERTIFICATE

PYRGEOMETER

PYRGEOMETER MODEL : CGR 3

SERIAL NUMBER : 060070

SENSITIVITY : 10.97 μV/W/m²

IMPEDANCE : 91 Ohm

CALIBRATION PROCEDURE :

The reference and test pyrgeometer are mounted horizontally on a table under an extended warm plate (67°). The table can rotate to exchange the positions of both instruments. The net irradiance at the pyrgeometers is approximately 150 W/m². The indoor procedure is based on a sequence of simultaneous readings.
After 30 s exposure to the warm plate, the output voltages of both pyrgeometer are integrated 30 s.
Next both pyrgeometers are covered by a blackened "shutter" with stable "room temperature".
After 30 s both signals are integrated again. The resulting two 'zero' signals are subtracted from the former signals to get comparable responses. In this way is compensated for temperature differences between both pyrgeometers.
Next the pyrgeometer positions are interchanged by rotation of the table and the procedure is repeated. The mean of former and latter responses is compared to derive the sensitivity figure of the test pyrgeometer. In this way asymmetry in the warm plate configuration and IR environment is cancelled out.

REFERENCE PYRGEOMETER : Kipp & Zonen CG 3 sn030003 active from 21/07/2006

hierarchy of traceability :

The reference CG 3 has been compared against a reference pyrgeometer CG 4 under mainly clear sky conditions during nighttime at Kipp & Zonen, Delft Holland. (On his turn the CG 4 was calibrated outdoors October to December, 2005, at the IR-centre of the World Radiation Center Davos against their pyrgeometer reference group.)
The reference CG 3 and CG 4 were placed horizontally side by side. During the calibration period from 13 June 2006 to 14 June 2006 the (outgoing) radiation signal (U_{emf} / S) ranged from -00 to -40 W/m². The instrument temperatures ranged from +24.3° to 21.3°C. The pyrgeometer thermopile outputs (U_{emf}, U_i) and body temperatures (T_b) were measured every second by a COMBILOG 1020 data logger and averages of 60 measurements have been logged as 1 min values. Later on the downward radiation (L_d) can be determined with the formula:

$$L_d = \frac{Uemf}{S} + 5.67 \cdot 10^{-8} \cdot T_b^{\,4}$$

For the (modified) reference CG 4 sn010536 a sensitivity of 8.98 μV/W/m² has been applied and with its voltage U_{emf} and temperature T_b data the reference L_d curve is calculated.
For the reference CG 3 a one minute average sensitivity S_i is calculated with the formula:

$$S_i = U_i \cdot \left(L_d - 5.67 \cdot 10^{-8} \cdot T_b^{\,4}\right)^{-1}$$

The final S_i is the average of one minute S_i's determined in periods with a net IR signal < -40 W/m² (Clear sky). The sum of all periods must be at least 6 hours.
The derived CG 3 sn030003 sensitivity and its expanded uncertainty are: 12.65 ± 0.07 μV/W/m².

IN CHARGE OF TEST : G. van der Wilt Date:Wednesday, August 02, 2006 Kipp & Zonen, Delft Holland

Notice
The calibration certificate supplied with the instrument is valid from the date of shipment to the customer. Even though the calibration certificate is dated relative to manufacture or recalibration the instrument does not undergo any sensitivity changes when kept in the original packing. From the moment the instrument is taken from its packaging and exposed to irradiance the sensitivity will deviate with time. See also the 'non-stability' performance (max. sensitivity change / year) given in the radiometer specification list.

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CALIBRATION CERTIFICATE

PYRGEOMETER

PYRGEOMETER MODEL	:	CGR 3
SERIAL NUMBER	:	060069
SENSITIVITY	:	5.56 μV/W/m^2
IMPEDANCE	:	38 Ohm

CALIBRATION PROCEDURE :

The reference and test pyrgeometer are mounted horizontally on a table under an extended warm plate (67°). The table can rotate to exchange the positions of both instruments. The net irradiance at the pyrgeometers is approximately 150 W/m². The indoor procedure is based on a sequence of simultaneous readings.
After 30 s exposure to the warm plate, the output voltages of both pyrgeometer are integrated 30 s. Next both pyrgeometers are covered by a blackened "shutter" with stable "room temperature". After 30 s both signals are integrated again. The resulting two "zero" signals are subtracted from the former signals to get comparable responses. In this way is compensated for temperature differences between both pyrgeometers.
Next the pyrgeometer positions are interchanged by rotation of the table and the procedure is repeated. The mean of former and latter responses is compared to derive the sensitivity figure of the test pyrgeometer. In this way asymmetry in the warm plate configuration and IR environment is cancelled out.

REFERENCE PYRGEOMETER : Kipp & Zonen CG 3 sn030003 active from 21/07/2006.

hierarchy of traceability :

The reference CG 3 has been compared against a reference pyrgeometer CG 4 under mainly clear sky conditions during nighttime at Kipp & Zonen, Delft Holland. (On his turn the CG 4 was calibrated outdoors October to December, 2005, at the IR-centre of the World Radiation Center Davos against their pyrgeometer reference group.)
The reference CG 3 and CG 4 were placed horizontally side by side. During the calibration period from 13 June 2006 to 14 June 2006 the (outgoing) radiation signal (U_{emf} / S) ranged from - 80 to - 40 W/m². The instrument temperatures ranged from +24.3° to 21.3°C. The pyrgeometer thermopile outputs (U_{emf}, U_t) and body temperatures (T_t) were measured every second by a COMBILOG 1020 data logger and averages of 60 measurements have been logged as 1 min. values. Later on the downward radiation (L_d) can be determined with the formula:

$$L_d = \frac{Uemf}{S} + 5.67 \cdot 10^{-8} \cdot T_b{}^4$$

For the (modified) reference CG 4 sn010536 a sensitivity of 8.98 μV/W/m² has been applied and with its voltage U_{emf} and temperature T_b data the reference L_d curve is calculated.
For the reference CG 3 a one minute average sensitivity S_t is calculated with the formula:

$$S_t = U_t \cdot \left(L_d - 5.67 \cdot 10^{-8} \cdot T_b{}^4\right)^{-1}$$

The final S_t is the average of one minute S_ts determined in periods with a net IR signal < -40 W/m² (Clear sky). The sum of all periods must be at least 6 hours.
The derived CG 3 sn030003 sensitivity and its expanded uncertainty are: 12.65 ± 0.07 μV/W/m².

IN CHARGE OF TEST : G. van der Wilt Date:Wednesday, August 02, 2006 Kipp & Zonen, Delft, Holland

Notice
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MEASUREMENT EXCELLENCE SINCE 1830

Kipp & Zonen 175 YEARS

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CALIBRATION CERTIFICATE

PYRANOMETER

PYRANOMETER MODEL	:	CMP 3
SERIAL NUMBER	:	050094
SENSITIVITY at normal incidence on horizontal pyranometer	:	13.23 $\mu V/W/m^2$
IMPEDANCE	:	34 Ohm
CALIBRATION PROCEDURE	:	The indoor calibration procedure is based on a side-by-side comparison with a reference pyranometer under an artificial sun fed by an AC voltage stabiliser. It embodies a 150 W Metal-Halide high-pressure gas discharge lamp. Behind the lamp is a reflector with a diameter of 16.2 cm. The reflector is 110 cm above the pyranometers producing a vertical beam. The reference and test pyranometers are mounted horizontally on a table, which can rotate. The irradiance at the pyranometers is approximately 500 W/m^2. During the calibration procedure the reference and test pyranometer are interchanged to correct for any non-homogeneity of the beam. The dark offsets of both pyranometers are measured before and after the interchange and taken into account.
REFERENCE PYRANOMETER	:	Kipp & Zonen CM 3 sn950512 active from 01/01/2005
hierarchy of traceability	:	This pyranometer was compared with the sun and sky radiation as source under mainly clear sky conditions using the "continuous sun-and-shade method". The readings are referred to the World Radiometric Reference (WRR) as stated in the WMO Technical Regulations. The measurements were performed in Davos (latitude: 46.8143°, longitude: -9.8458°, altitude: 1588m above sea level). The inclination of the receiver surfaces versus their horizontal position were set to 0.0 degrees, the instrument signal wire to the north. During the comparisons, the instrument received global radiation intensities from 653 to 1005 with a mean of 820 W/m^2. The angle between the solar beam and the normal of the receiver surface varied from 24 to 50 with a mean of 39 degrees. The instrument temperature ranged from +11.7 to +20.7 with a mean of +17.7°C. The sensitivity calculation and the single measurements deviation (σ) are based on 1001 individual measurements. The obtained sensitivity value is valid for similar conditions and is: 16.04 ± 0.11 $\mu V/W/m^2$ (but is corrected by Kipp & Zonen to 16.33 $\mu V/W/m^2$. See "correction applied" below.) The testing was done June 7, July 28, 29, September 2, 5, 6, 7 and 9, 2004. Global radiation data were calculated from the direct solar radiation as measured with the absolute cavity pyrheliometer HF18748 (member of the WSG, WRR-Factor: 0.99568, based on the last International Pyrheliometer Comparison IPC-2000) and from the diffuse radiation as measured with a continuous disk shaded pyranometer Kipp & Zonen CM 22 sn020059 with sensitivity 8.91 (ventilated with heated air, instrument-wire to the north).
correction applied	:	+1.8 % This correction was necessary to correct for the mean directional errors of the reference CM 3 in Davos. This error is estimated at Kipp & Zonen measuring the cosine error for the mean angle of incidence at azimuth S-30° and S+30°. The reference CM 3 now measures the vertical directed beam of the indoor calibration facility more correctly.
IN CHARGE OF TEST	:	F. de Wit Date:August 29, 2005 Kipp & Zonen, Delft, Holland

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APPENDIX F

TEMPERATURE SENSOR CALIBRATION

The temperature calibration was carried out at fixed points for the ground temperature sensors and calibration by fixed point and by comparison (to one another) for the high accuracy fluid flow temperature sensors. The fixed temperatures for the ground sensors were 0°C, +10°C, and +20°C representing the range of temperatures that the sensors are likely to inhabit. The high accuracy fluid temperature sensors were calibrated up to +60°C. All the temperature sensors were calibrated using a calibration machine outlined in Chapter 3. They were further calibrated using an ice bath. The ice bath calibration at 0°C used distilled and de-ionised water and crushed ice, with continuous stirring. The high accuracy fluid temperature sensors were all calibrated in a single ice bath where the ice bath was allowed to warm over time under continuous stirring. This allowed the sensors to be validated against each other for comparative accuracy, with is most important for accuracy of the temperature difference between the flow and return fluid temperatures. Figure F.1 contains the test comparison results for four of the high accuracy temperature sensors, showing conformity of output across all sensors.

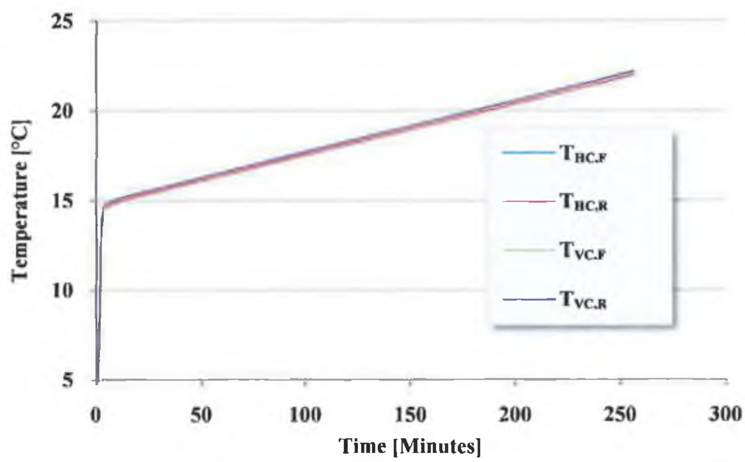


FIGURE F.1 TEMPERATURE SENSOR COMPARATIVE TEST RESULTS.

Figure F.2 presents pictures of the heat pump collector flow and return temperature sensors in situ with insulation.

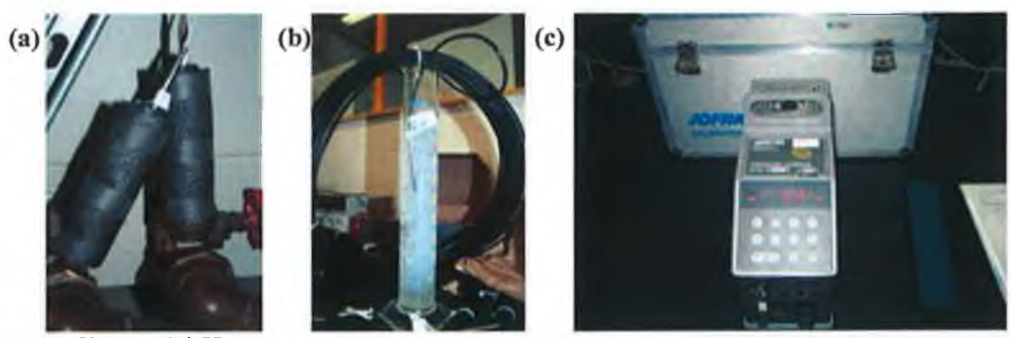


FIGURE F.2 (A) HEAT PUMP COLLECTOR FLOW AND RETURN TEMPERATURE SENSORS IN SITU, (B) TEMPERATURE SENSOR ICE BATH CALIBRATION AND (C) *ZOFRA* TEMPERATURE SENSOR CALIBRATOR.

HP-IRL COEFFICIENT OF PERFORMANCE (COP) CALCULATION ACCURACY

In determining the validity of the experimental measurements certain verifications are required. The accuracy is determined by means of analysing the heat balance equation which is:

$$\dot{q}_{in} = \dot{m} \cdot C_p \cdot (T_{out} - T_{in})$$

Where $\dot{q}_{in}$ [W] is the heat input, $\dot{m}$ [kg/second] is the flow rate, C_p is the specific heat of the liquid, T_{in} and T_{out} are the liquid flow and return temperatures. The temperature probes have an accuracy of ±0.015°C (factory accuracy, validated as part of the study) and the data acquisition system receiving and recording the analogue signal has an accuracy of ±0.15°C. However, through on-site calibration the inaccuracy of the data acquisition system was eliminated giving a total uncertainty for the temperature measurement as:

$$\Delta T = (\pm 0.015°\mathrm{C})_{\mathrm{in}} + (\pm 0.015°\mathrm{C})_{\mathrm{out}}$$

$$\Delta T \approx \pm 0.03\mathrm{K}$$

The typical ΔT for the collector across the heat pump evaporator is approximately 2.65K, and therefore the uncertainty of the temperate measurement is:

$$Temperature\ Error = \frac{\pm 0.03\mathrm{K}}{2.65\ \mathrm{K}} \approx \pm 1.1\%$$

Taking an error of ±2% of reading for the flowmeters (factory calibrated), along with the data acquisition system accuracy of ±0.04% of reading, the total uncertainty for the flowrate measurement is:

$$Flowrate\ Error = \sqrt{(\pm 0.02)^2 + (\pm 0.0004)^2}$$

$$Flowrate\ Error = \pm 2\%$$

Then the total thermal energy error, $Q_{T,in}$, measurement is:

$$Total\ Q_{T,in}\ Error\ = \sqrt{(\pm 0.02)^2 + (\pm 0.011)^2}$$

$$Total\ Q_{T,in}\ Error\ = \pm 2.28\%$$

Taking the supplier calibrated error of ±1% of reading for the electrical power monitor, along with a data acquisition system accuracy of ±0.04% of reading, the total uncertainty for the electrical power, $Q_{E,in}$, measurement is:

$$Q_{E,in}, Electrical\ Power\ Error = \sqrt{(\pm 0.01)^2 + (\pm 0.0004)^2}$$

$$Electrical\ Power\ Error = \pm 1\%$$

Then the accuracy of the Coefficients Of Performance (COP) can be shown as:

$$\boldsymbol{COP\ Error = 2.28\% + 1\%}$$

$$\boldsymbol{COP\ Error \approx \pm 3.28\%}$$

APPENDIX G

PE3408 SDR-11 HDPE Pipe

Nominal Pipe Size (NPS)	Nominal Diameter	Outside Diameter (OD)	Inside Diameter (ID)	Min. Wall (Thickness)
3/4	DN 25	26.7mm	21.8mm	2.4mm
1	DN 32	33.4mm	27.4mm	3.0mm
1 1/4	DN 40	42.2mm	34.5mm	3.8mm
1 1/2	DN 50	48.3mm	39.5mm	4.4mm
2	DN 63	60.3mm	49.4mm	5.5mm

PE100 SDR-11 HDPE Pipe

-	Nominal Diameter	Outside Diameter (OD)	Inside Diameter (ID)	Min. Wall (Thickness)
-	DN 25	25mm	20.4mm	2.3mm
-	DN 32	32mm	26.0mm	3.0mm
-	DN 40	40mm	32.4mm	3.87mm
-	DN 50	50mm	40.8mm	4.6mm
-	DN 63	63mm	51.4mm	5.8mm

PE308 SDR-11 HDPE Pipe is manufactured as per ASTM D 3035 and ASTM 2447 and is applicable to North America. PE100 SDR-11 HDPE Pipe is manufactured to ISO 4427 or EN12201 and is applicable to the EU, Australia and New Zealand.

SDR is the Standard Dimension Ratio and SDR-11 characterises a pipe capable of working in conditions of up to 1.6 MPa (232 psi) water at +20°C (68°F).

APPENDIX H

DETERMINING COLLECTOR PUMPING POWER

To ensure movement of fluid within the collector pipes, all hydraulic losses (friction, bends, fittings, changes in elevation) in the pipe must be overcome. Resistance to flow is affected by the cross sectional area of the pipe, the flow rate and the surface roughness of the collector pipe inner wall. Minimising unnecessary pumping power for closed loop systems will help improve heat pump performance and therefore circulating pump sizes must be calculated correctly.

The volumetric flow rate of the *HP-IRL* horizontal collector is 3.6 m^3/h, which corresponds to 0.24 m^3/h per kW of heating capacity. However, Kavanaugh and Rafferty (1997) suggested that the optimum volumetric flow rates for the circulating pump should range from 0.162 - 0.192 m^3/h per kW of heating capacity indicating some potential pump oversizing for the *HP-IRL* collector. The circulation pump power demand can be established using the following method.

From the continuity equation $V = Q/A$, the flow velocity within each of the ten collector loops is:

$$V_{pipe1} = \frac{\left(\frac{Q_{pipe2}}{No.\,of\ Loops}\right)}{\pi \cdot r_{pipe1}^2} = \frac{\left(\frac{0.001\,m^3/s}{10\ Loops}\right)}{\pi(0.013\,m)^2} = 0.22\ m/s\ (per\ collector\ loop)$$

Where Q_{pipe2}, (m^3/s), is the volumetric flow rate of the whole collector (volumetric flow rate through the $GSHP_{HC}$ evaporator).

The *Reynolds number, Re,* is:

$$Re = \frac{Dynamic\ Pressure}{Kinematic\ Viscosity} = \frac{V_{pipe1} \cdot D_{pipe1}}{v}$$

Where D_{pipe1} is the diameter of the collector loop (m) and v is the kinematic viscosity of the fluid (m^2/s).

The horizontal collector used in this *HP-IRL* study utilises a 70/30 mix of water and ethylene glycol in order to lower the fluid freezing point to -16.1°C. The thermal characteristics of the fluid are therefore determined by the volumetric fraction of each constituent contained in the overall volume, details of which are contained in Table H.1.

As the fluid is a blend between water and ethylene glycol, the kinematic viscosity of the brine solution is a representation of this mixture. Displaying a non-linear dependence on temperature, the kinematic viscosity is determined using the *Refutas* equation:

$$VBN = 14.534 \times \ln[\ln(v + 0.8)] + 10.975$$

Where *VBN* is the *Viscosity Blending Number* and v is the kinematic viscosity of the fluid in centistokes.

$$VBN_{Blend} = [\varphi_{Water} \times VBN_{Water}] + [\varphi_{EG} \times VBN_{EG}]$$

Where φ_{Water} and φ_{EG} is the mass fraction of the two brine constituents.

$$v_{Blend} = e^{\frac{VBN_{Blend} - 10.975}{14.534}} - 0.8$$

TABLE H.1 PROPERTIES OF WATER/ETHYLENE GLYCOL MIXTURE

	Thermal Conductivity, λ_f (W/m·K)	Density, ρ (kg/m³)	Kinematic Viscosity, v (m²/s)	Dynamic Viscosity, μ (kg/m·s)	Specific Heat Capacity, Cp (kJ/kg·K)
Water (@ 4°C)	0.577	1000	1.14E-06	1.14E-03	4.204
Ethylene Glycol (@ 4°C)	0.243	1128	4.60E-05	5.19E-02	2.611
Volume: $Water_{70\%}$, $EG_{30\%}$	0.477	1038	2.27E-06	3.03E-03	3.726

This gives a Reynolds number of:

$$Re = \frac{Dynamic\ Pressure}{Kinematic\ Viscosity} = \frac{V_{pipe1} \cdot D_{pipe1}}{v} = \frac{(0.22 m/s)(0.026\ m)}{2.27\ x10^{-6} m^2/s} = 2476$$

The critical Reynolds number identifying transition of flow to turbulence is typically $Re_{crit} \approx 2300$.

The roughness value is an index of a surface roughness, waviness and form and is a function of the machining or extrusion process. The recommended roughness value for drawn pipe (HDPE) is $\epsilon = 0.0015$mm, giving a roughness ratio of:

$$\frac{\epsilon}{D_i} = \frac{0.0015\ mm}{26\ mm} = 5.77 \times 10^{-5}$$

The friction factor can be found from equation for laminar/transition flow:

$$f = \frac{64}{Re} = 0.039$$

Knowing the roughness ratio, the friction factor can be identified from the Moody Chart as $f \approx 0.036$.

The head loss per metre on the straight pipe (in-line) section of the horizontal collector, h_s, is calculated using the *Darcy Weisbach* equation as follows:

$$h_s = f\frac{4 \cdot L}{D_{pipe1}}\frac{V_{pipe1}^2}{2 \cdot g} = (0.039)\frac{4 \cdot 1\,m}{0.026\,m}\frac{\left(0.22\frac{m}{s}\right)^2}{2(9.81\,m/s^2)} = 0.01314\,m$$

Where L is the pipe length (m), this represents a head pressure of *15.8m* for the 1200m in-line sections of the entire collector. Another method of calculating the pressure drop is:

$$\Delta P_{pipe1} = \frac{f \cdot \dot{m}^2 \cdot L}{4 \cdot A^2 \cdot D_{pipe1} \cdot \rho \cdot g_c} = \frac{(0.039)(0.119\,kg/s)^2(1m)}{4(0.00053\,m^2)^2(0.026m)(1039\,kg/m^3)(1kg \cdot m/N \cdot s^2)} = 143\ \text{Pa} \cdot m^{-1}$$

This gives a pressure drop of 143 *Pa/m* (ASHRAE, 2003 - 2006) across one metre of in-line horizontal collector, and therefore 172 *kPa* for the 1200m in-line portion of the horizontal collector, generating an alternative total in-line collector (1200m) heat loss pressure of:

$$Head\ (m) = \frac{Head\ Pressure\ (Pa)}{\text{Gravitational Pressure}\ (Pa)} = \frac{172{,}000\ Pa}{9800\ Pa} = 17.5\ m$$

The portion of the horizontal collector spiral loops that can be deemed straight is approximately half the loop length (75m +75m), giving a head loss pressure for the two spiral loops as 21.5 kPa. The larger diameter flow and return pipes (Pipe 2 in Figure 4.43) from the $GHSP_{HC}$ evaporator to the collector has a head loss of:

$$\Delta P_{pipe2} = \frac{f \cdot \dot{m}_{pipe2}^2 \cdot L}{4 \cdot A^2 \cdot D_{pipe2} \cdot \rho \cdot g_c} = \frac{(0.039)(1.19\,kg/s)^2(60m)}{4(0.00207\,m^2)^2(0.0514m)(1039kg/m^3)(1kg \cdot m/N \cdot s^2)} = 2844\ Pa$$

The head loss associated with a large radius 90° bend, h_b, is calculated as follows:

$$h_b = K\left[\frac{V_{pipe1}^2}{2g}\right] = (0.6)\frac{\left(0.22\frac{m}{s}\right)^2}{2(9.81m/s^2)} = 1.35\ mm\ (1.35mm\ Head\ Pressure = 13.2\ Pa)$$

Where *K* represents the head loss factor associated with a large radius 90° bend. Therefore, each 90° bend requires a bend pressure, ΔP_b, of 13.2 *Pa.* An estimate of the total amount of bends on the horizontal collector is 92 for the eight in-line collector loops (1.2 *kPa*) and 276 bends for the two spiral collector loops (3.6 *kPa*), which means that the total head loss pressure for the bends is 4.8 *kPa.*

The head pressure across the heat pump evaporator heat exchanger, ΔP_{HE}, has been recorded as 33 *kPa*, as per the performance testing of the *Solterra* heat pump by Arsenal Research (AR, 2003). Thus, the total horizontal collector system head loss is 234 *kPa.*

The power of the circulation pump required can be calculated using the equation:

$$Q_{pump} = \frac{\Delta P \cdot \dot{m}}{\rho \cdot \eta_{circ}} = \frac{(234\ kPa)(1.19\ kg/s)}{(1039\ kg/m^3)(0.6)} = 447\ W$$

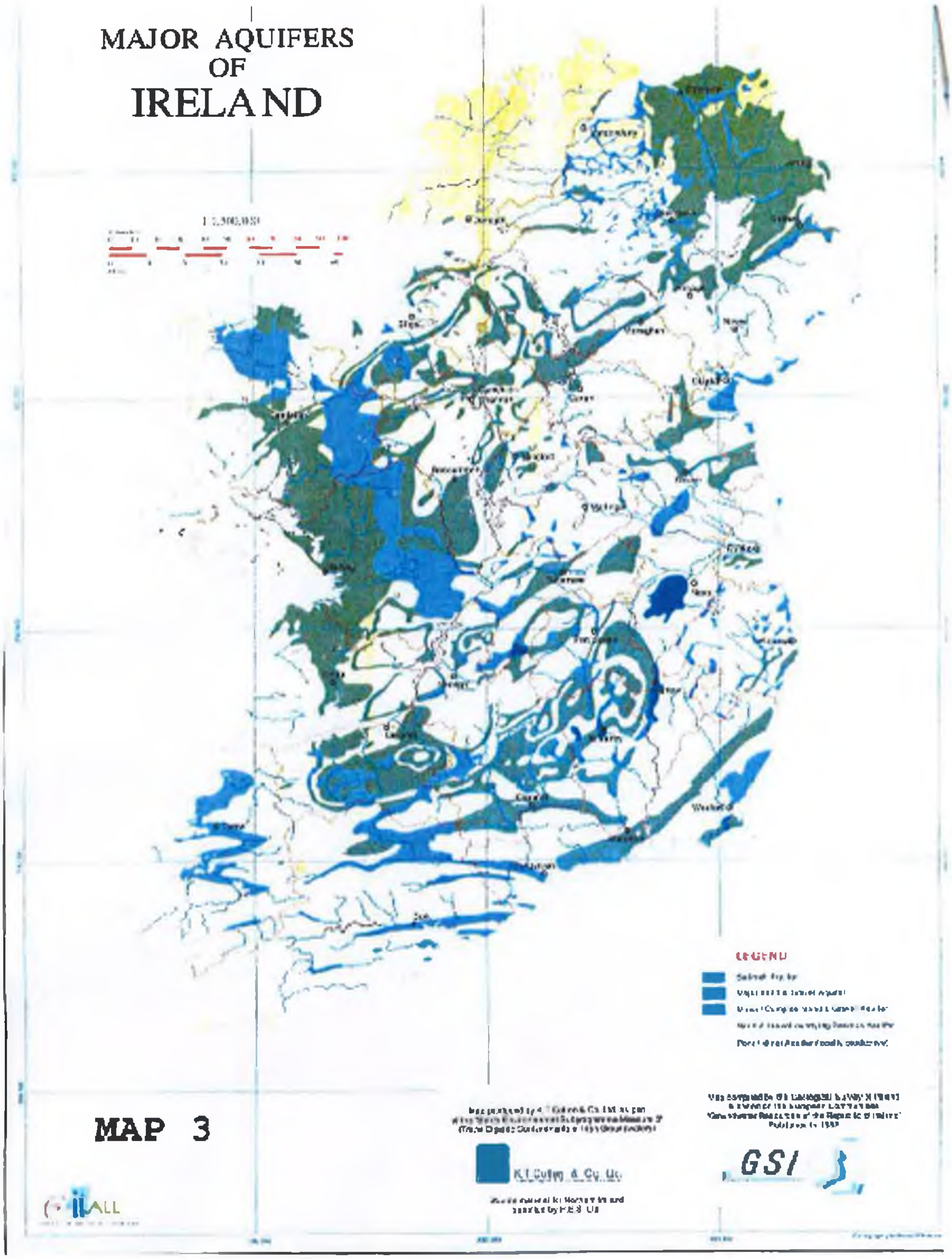
Where Q_{pump} is the power consumption of the circulating pump (W), $\dot{m}$ is the mass flow rate across the pump (kg/s), ΔP is the head pressure across the pump (kPa), η_{circ} is the efficiency of the circulation pump (-) and ρ is the density of the fluid (kg/m^3). This gives us a circulation pump size of 30 Watts per kW of heat pump capacity, or 3% of the overall capacity (15 kW_{th}). A general rule of thumb for horizontal collector sizing is that for between 20 and 30 Watts of heat pump capacity there should be 1m of collector piping. This rule of thumb is subject to suitable pipe and fluid flow dynamics and pipe spacing conditions. Another rule of thumb is that the pumping power for the collector should be between 14 and 40 Watts for every kW of heat pump capacity. Kavanaugh (1998) suggested a third method, where the power consumption when running should be between 6 and 7% of the overall heat output. In the *HP-IRL* study two 220W circulation pumps were installed in the horizontal collector $GSHP_{HC}$. This delivers an actual circulation pump demand of 29 Watts per kW of heat pump capacity which is within the 14 - 40W rule of thumb and is only 4% of the overall heat pump capacity, which is below the levels for the third rule of thumb. In the collector system all of the energy used to run the circulation pump is eventually dissipated into the collector fluid and into the air surrounding the circulation pump. Thus the contributor of the circulation pump to the thermal energy in the collector fluid is calculated as:

$$T_{HC,R} = T_{HC,F} + \Delta P_{\text{Total}} \left[\frac{\frac{1}{\eta_{circ}} - 1}{\rho \cdot C_p} \right] = 4°\text{C} + 234kPa \left[\frac{\frac{1}{0.6} - 1}{(1039\ kg/m^3)(3.73kJ/kg \cdot K)} \right] = 4.04°\text{C}$$

Where $T_{HC,F}$ is the inlet fluid temperature (°C) and C_p is the fluid specific heat capacity (kJ/kg·K).

This equates to approximately 158 Watts, which is a thermal recovery of 37% of the circulation pump power or 1.4% of the total thermal energy delivered to the heat pump. This is negligible and can be ignored.

APPENDIX I



APPENDIX J

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(54) **A heat pump**

(57) The invention relates to a heat pump consisting of a pair of heat exchangers (31, 32) air/coolant connected to a coolant feed piping (2) into a compressor (1) and further connected to a coolant return piping (5) from a heat exchanger (4) coolant/water. The invention consists in that the coolant return piping (5) from the heat exchanger (4) coolant/water is before entering exchangers (31,32) air/coolant split in two branches (51, 52), where each of them is connected to an evaporating inlet of one heat exchanger (31, 32) air/coolant, while each of the branches (51, 52) before entering one of the pair of heat exchangers (31, 32) air/coolant forms a heating piping (81, 82) of the second from the pair of heat exchangers (31, 32) air/coolant and each of the branches (51, 52) of the coolant return piping (5) is closable.

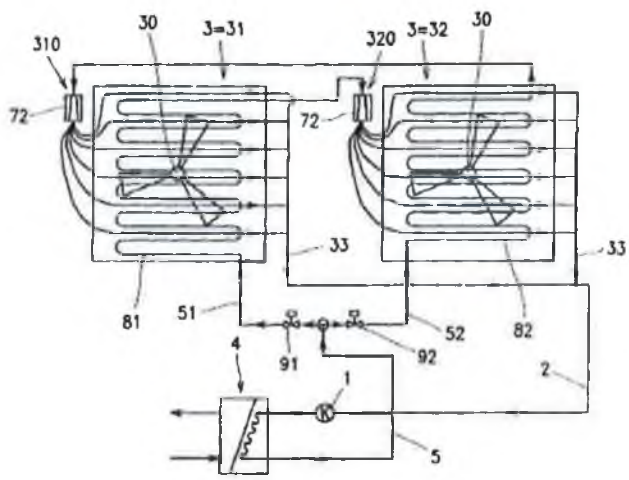


Fig. 1

EP 1 577 624 A2

TC MACH ASHP Patent Schematic.

APPENDIX K

HORIZONTAL COLLECTOR HEAT TRANSFER CHARACTERISTICS

After the brine solution is delivered to the collector manifold, it is subdivided into each of the ten collector circuits (Figure 3.22). Traditional fluid mechanics enables the flow characteristics to be represented as shown in Figure K.1. It sub-divided the collector entrance and fully developed hydrodynamic regions, each with their characteristic heat transfer regimes.

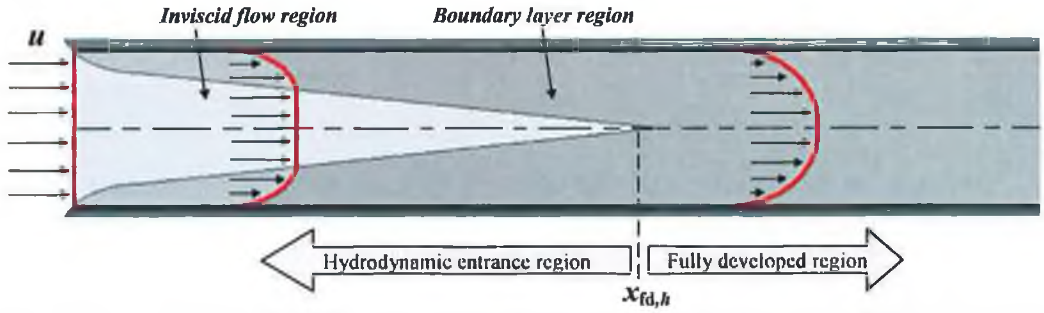


FIGURE K.1 LAMINAR, HYDRODYNAMIC BOUNDARY LAYER DEVELOPMENT IN THE COLLECTOR PIPE.

The length of the pipe in which is known as the hydrodynamic entrance region is termed the hydrodynamic entry length, $x_{\mathrm{fd},h}$.

For laminar flow ($Re \lesssim 2300$), the hydrodynamic entry length is calculated using the following equation:

$$\left(\frac{x_{\mathrm{fd},h}}{D_{pipe1}}\right)_{lam} \approx 0.05Re$$

$$x_{\mathrm{fd},h} = D_{pipe1}(0.05Re) = (0.026m)\big(0.05(2476)\big) = 3.2m$$

The length of the pipe in which is known as the thermal entrance region is termed the thermal entry length, $x_{\mathrm{fd},t}$.

For laminar flow the thermal entry length is calculated using as follows:

$$\left(\frac{x_{\mathrm{fd},t}}{D_{pipe1}}\right)_{lam} \approx 0.05RePr$$

$$x_{\mathrm{fd},t} = D_{pipe1}(0.05Re)Pr = (0.026m)\big(0.05(2476)\big)25.85 = 83.2m$$

The thermal resistance network between the ground and the horizontal collector fluid is shown in Figure K.2.

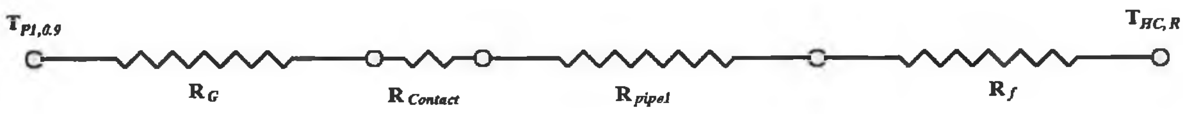


FIGURE K.2 GROUND-COLLECTOR THERMAL RESISTANCE CIRCUIT.

Then both the conductive (R_{CD}) and the convective resistance (R_{CV}) can be summed to produce the total collector thermal resistance as follows:

$$R_{Total} = R_{CD} + R_{CV} \qquad \textit{Equation K.1}$$

As highlighted by O'Connell and Cassidy (2004), the contact resistance between the collector pipe and the ground ($R_{Contact}$) is negligible and can be discounted, giving the overall resistance as follows:

$$R_{Total} = \left(R_G + R_{pipe1}\right) + R_f \qquad \textit{Equation K.2}$$

The resistance to thermal flow across the collector pipe wall (R_{pipe1}) and between the inner pipe wall and the fluid (R_f) are constant, and the calculation of these resistances are presented as follows:

The thermal resistance of the fluid, R_f, can be determined using Newton's law of cooling which states:

$$R_f = \frac{1}{h_{pipe1} \cdot A_{pipe1}}$$

Where h_{pipe1} is the pipe heat transfer coefficient (W/m²·K) and A_{pipe1} is the inner surface area of the collector per 1 meter length (m²).

The heat transfer coefficient is calculated through the following laminar flow formula, where a constant surface temperature is assumed, λ_f is the collector fluid thermal conductivity:

$$h_{pipe1} = \frac{Nu \cdot \lambda_f}{D_{pipe1}} \qquad \textit{Equation K.3}$$

or transition from laminar to turbulent:

$$h_{pipe1} = \frac{Nu \cdot \lambda_f}{D_{pipe1}} = \left[\frac{\left(\left(\frac{f}{2}\right)(Re - 1000) \cdot Pr\right)}{1 + 1.27\left(\frac{f}{2}\right)^{0.5}(Pr^{0.667} - 1)}\right]\left[\frac{\lambda_f}{D_{pipe1}}\right]$$

Where:

$$Pr = Prandtl\ No. = \frac{C_P \cdot \mu}{\lambda_f} = 23.7$$

Nu is the Nusselt number for a fully developed flow, and is calculated as follows:

$$Nu = 3.66 + \left[\frac{\left(0.0668\left(D_{pipe1}/L_{HC\ Loop}\right) \cdot Re \cdot Pr\right)}{\left(1 + \left(0.04\left[\left(D_{pipe1}/L_{HC\ Loop}\right) \cdot Re \cdot Pr\right]^{(2/3)}\right)\right)} \right] = 4.23$$

where μ is the fluid dynamic viscosity (kg/m·s), ν is the fluid kinematic viscosity (m²/s), D_{pipe1} is the inner diameter of the collector pipe (m) and λ_f is the fluid thermal conductivity (W/m·K).

The laminar flow heat transfer coefficient of the collector fluid can be calculated using Equation K.3:

$$h_{pipe1} = 4.23 \left[\frac{0.48\, W/m \cdot K}{0.026\, m} \right] = 77.6\, W/m^2 \cdot K$$

Going by the laminar flow heat transfer coefficient we get a convective fluid thermal resistance of:

$$R_f = \frac{1}{2 \cdot \pi \cdot r_i \cdot h_{pipe1}} = \frac{1}{2 \cdot \pi \cdot (0.013m)(77.6\, W/m^2 \cdot K)} = 0.162\ m \cdot K/W$$

Where r_i is the internal radius of the collector pipe.

The pipe internal thermal convective resistance, R_{pipe1}, is derived as:

$$R_{pipe1} = \frac{\ln\left[\frac{r_o}{r_i}\right]}{2 \cdot \pi \cdot \lambda_{pipe1}} = \frac{\ln\left[\frac{0.016\, m}{0.013\, m}\right]}{2 \cdot \pi \cdot (0.46\, W/m \cdot K)} = 0.072\, m \cdot K/W$$

Where r_o is the collector pipe outer radius (m), r_i is the collector pipe inner radius and λ_{pipe1} is the thermal conductivity of the collector pipe (W/m·K).

Using the experimental data, to estimate the total thermal resistance is as follows:

If a nominal duty cycle in a winter season operation is 90% of the heat pump capacity (12kW), corresponding to a horizontal collector extract rate (Q_{HC}) of 9kW on average (COP_{AVG} 3.0), then the nominal $\Delta T_{HC,G}$ can be obtained as follows.

$$q'_{HC,PIPE} = \frac{\Delta T_{HC,G}}{R_{Total}} = \frac{Q_{HC}}{L_{HC}} = \frac{9000\, W}{1500\, m} = 6.0\, W/m \qquad \textit{Equation K.4}$$

Where , $q'_{HC,PIPE}$ (W/m) is the nominal heat transfer rate per unit length along the collector.

The nominal heat transfer rate per unit area of collector ground, $q'_{HC,Coll}$ (W/m²) is:

$$q'_{HC,Coll} = \frac{Q_{HC}}{A_{HC}} = \frac{9000\ W}{430\ m^2} = 20.9\ W/m^2 \qquad \textit{Equation K.5}$$

Where A_{HC} is the area of the collector. If the nominal load of 90% of capacity (6.0 W/m) is applied to the $GSHP_{HC}$ it can be seen from the test period HC9 in Figure 4.28 that the steady-state $\Delta T_{HC,G}$ will be -4.7K.

Therefore rearranging Equation K.4:

$$R_{Total} = \frac{\Delta T_{HC,G}}{q'_{HC,PIPE}} = \frac{4.7\ K}{6.0\ W/m} = 0.78\, m \cdot K/W$$

Therefore:

$$R_{Total} = R_G + R_{pipe1} + R_f = 0.78\ m \cdot K/W$$

Then:

$$R_G = R_{Total} - (R_{pipe1} + R_f) = 0.78 - (0.072 + 0.184) = 0.53\ \ m \cdot K/W$$

If:

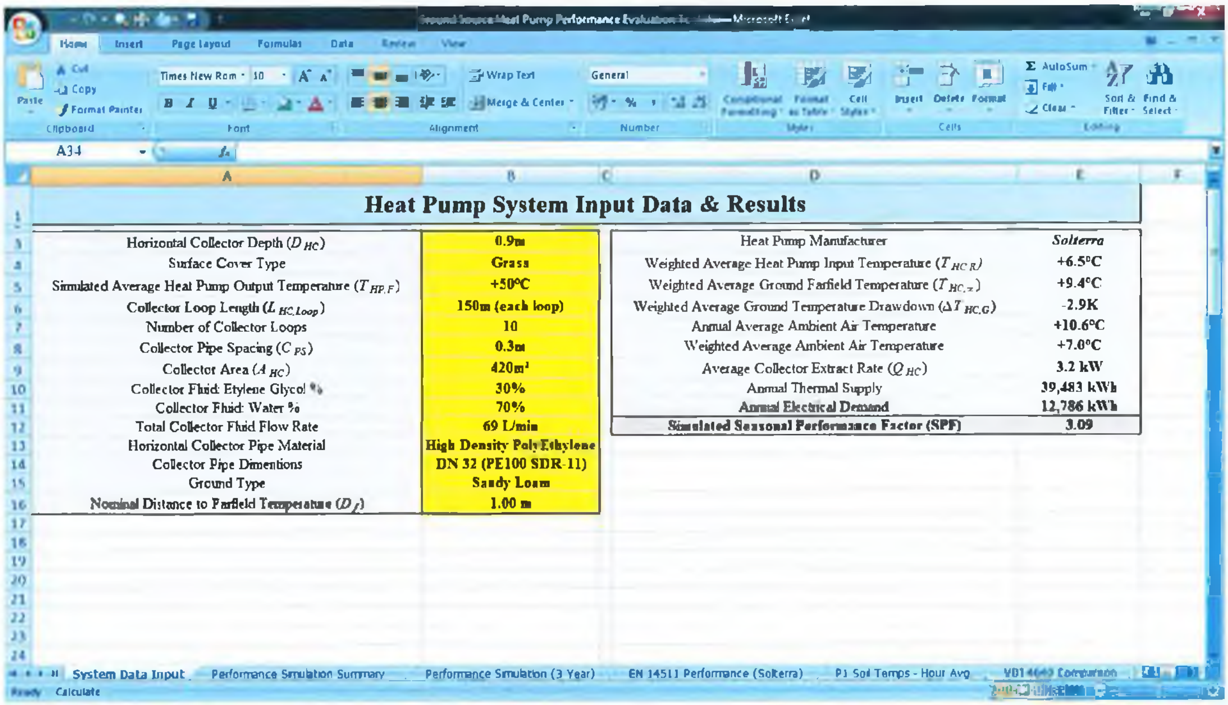
$$R_G = \frac{\ln\left[\frac{r_f}{r_o}\right]}{2 \cdot \pi \cdot \lambda_G} = \frac{\ln\left[\frac{6.0\ m}{0.016\ m}\right]}{2 \cdot \pi \cdot (2.3\ W/m \cdot K)} = 0.41\, m \cdot K/W$$

The discrepancy in the thermal resistance is a characteristic of thermal interference between the collector pipes. This interaction induces a linear resistance in both the upward and downward directions once the farfield radii meet. Once interaction starts the total ground resistance becomes the following:

$$R_G = \frac{\ln\left[\frac{PS}{D_o}\right]}{2 \cdot \pi \cdot \lambda_G} + \frac{1}{\lambda_G \cdot PS/_{\Delta Z_{above}} + \lambda_G \cdot PS/_{\Delta Z_{below}}}$$

Where PS is the collector pipe spacing, D_O is the pipe outer diameter, and ΔZ is the distance to farfield both above and below the collector.

APPENDIX L

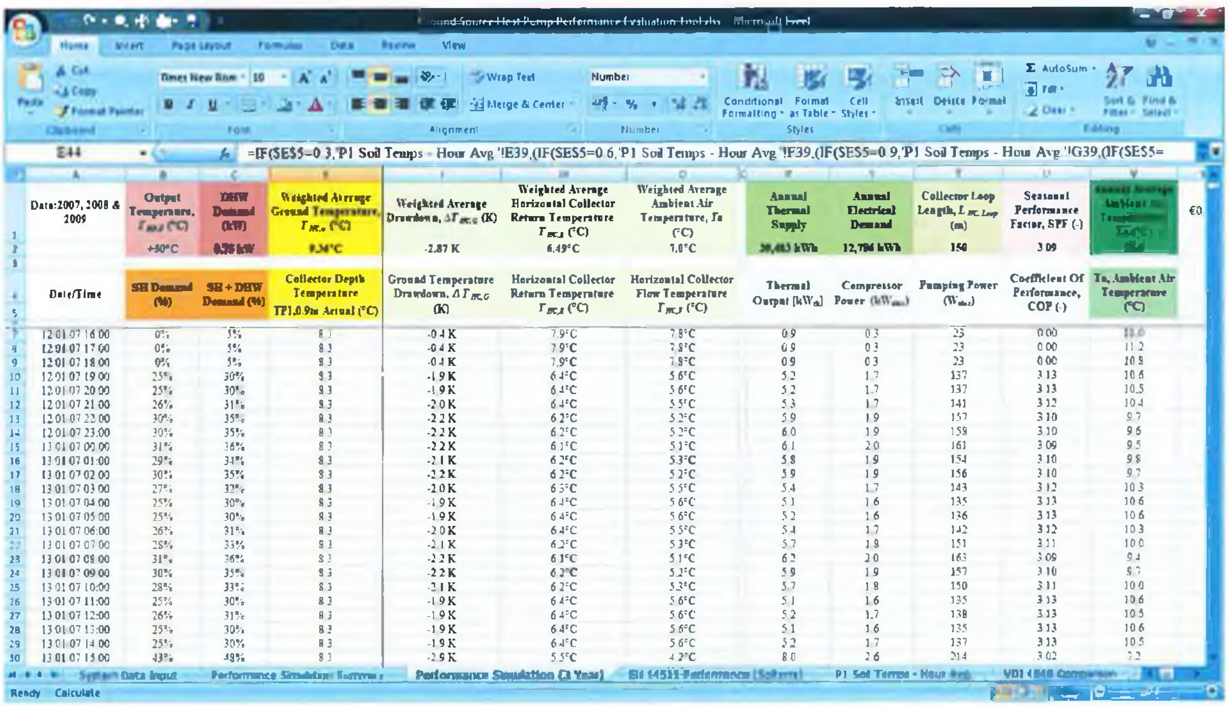


Heat Pump System Input Data & Results

Horizontal Collector Depth (D_{HC})	0.9m
Surface Cover Type	Grass
Simulated Average Heat Pump Output Temperature ($T_{HP,F}$)	+50°C
Collector Loop Length ($L_{HC,Loop}$)	150m (each loop)
Number of Collector Loops	10
Collector Pipe Spacing (C_{PS})	0.3m
Collector Area (A_{HC})	420m²
Collector Fluid: Etylene Glycol %	30%
Collector Fluid: Water %	70%
Total Collector Fluid Flow Rate	69 L/min
Horizontal Collector Pipe Material	High Density PolyEthylene
Collector Pipe Dimentions	DN 32 (PE100 SDR-11)
Ground Type	Sandy Loam
Nominal Distance to Farfield Temperature (D_f)	1.00 m

Heat Pump Manufacturer	*Solterra*
Weighted Average Heat Pump Input Temperature ($T_{HC,R}$)	+6.5°C
Weighted Average Ground Farfield Temperature ($T_{HC,\infty}$)	+9.4°C
Weighted Average Ground Temperature Drawdown ($\Delta T_{HC,G}$)	-2.9K
Annual Average Ambient Air Temperature	+10.6°C
Weighted Average Ambient Air Temperature	+7.0°C
Average Collector Extract Rate (Q_{HC})	3.2 kW
Annual Thermal Supply	39,483 kWh
Annual Electrical Demand	12,786 kWh
Simulated Seasonal Performance Factor (SPF)	**3.09**

FIGURE L.1 SCREEN-GRAB OF GSHP PERFORMANCE EVALUATION TOOL USING THREE YEARS OF MARITIME CLIMATE DATA AND *SOLTERRA* HEAT PUMP PERFORMANCE CHARACTERISTICS, AS PER TEST STANDARD EN 14511.



=IF(E5=0.3,'P1 Soil Temps - Hour Avg'!E39,(IF(E5=0.6,'P1 Soil Temps - Hour Avg'!F39,(IF(E5=0.9,'P1 Soil Temps - Hour Avg'!G39,(IF(E5=

Data:2007, 2008 & 2009	Output Temperature, $T_{HP,F}$ (°C)	DHW Demand (kW)	Weighted Average Ground Temperature, $T_{HC,\infty}$ (°C)	Weighted Average Drawdown, $\Delta T_{HC,G}$ (K)	Weighted Average Horizontal Collector Return Temperature $T_{HC,R}$ (°C)	Weighted Average Ambient Air Temperature, T_a (°C)	Annual Thermal Supply	Annual Electrical Demand	Collector Loop Length, $L_{HC,Loop}$ (m)	Seasonal Performance Factor, SPF (-)	[illegible]
	+50°C	0.36 kW	9.34°C	-2.87 K	6.49°C	7.0°C	39,483 kWh	12,786 kWh	150	3.09	[illegible]
Date/Time	**SH Demand (%)**	**SH + DHW Demand (%)**	**Collector Depth Temperature TP1,0.9m Actual (°C)**	**Ground Temperature Drawdown, $\Delta T_{HC,G}$ (K)**	**Horizontal Collector Return Temperature $T_{HC,R}$ (°C)**	**Horizontal Collector Flow Temperature $T_{HC,F}$ (°C)**	**Thermal Output [kW$_{th}$]**	**Compressor Power (kW$_{elec}$)**	**Pumping Power (W$_{elec}$)**	**Coefficient Of Performance, COP (-)**	**T_a, Ambient Air Temperature (°C)**
12.01.07 16:00	0%	5%	8.3	-0.4 K	7.9°C	7.8°C	0.9	0.3	25	0.00	11.0
12.01.07 17:00	0%	5%	8.3	-0.4 K	7.9°C	7.8°C	0.9	0.3	23	0.00	11.2
12.01.07 18:00	0%	5%	8.3	-0.4 K	7.9°C	7.8°C	0.9	0.3	23	0.00	10.8
12.01.07 19:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.2	1.7	137	3.13	10.6
12.01.07 20:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.2	1.7	137	3.13	10.5
12.01.07 21:00	26%	31%	8.3	-2.0 K	6.4°C	5.5°C	5.3	1.7	141	3.12	10.4
12.01.07 22:00	30%	35%	8.3	-2.2 K	6.2°C	5.2°C	5.9	1.9	157	3.10	9.7
12.01.07 23:00	30%	35%	8.3	-2.2 K	6.2°C	5.2°C	6.0	1.9	158	3.10	9.6
13.01.07 00:00	31%	36%	8.3	-2.2 K	6.1°C	5.1°C	6.1	2.0	161	3.09	9.5
13.01.07 01:00	29%	34%	8.3	-2.1 K	6.2°C	5.3°C	5.8	1.9	154	3.10	9.8
13.01.07 02:00	30%	35%	8.3	-2.2 K	6.2°C	5.2°C	5.9	1.9	156	3.10	9.7
13.01.07 03:00	27%	32%	8.3	-2.0 K	6.3°C	5.5°C	5.4	1.7	143	3.12	10.3
13.01.07 04:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.1	1.6	135	3.13	10.6
13.01.07 05:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.2	1.6	136	3.13	10.6
13.01.07 06:00	26%	31%	8.3	-2.0 K	6.4°C	5.5°C	5.4	1.7	142	3.12	10.3
13.01.07 07:00	28%	33%	8.3	-2.1 K	6.2°C	5.3°C	5.7	1.8	151	3.11	10.0
13.01.07 08:00	31%	36%	8.3	-2.2 K	6.1°C	5.1°C	6.2	2.0	163	3.09	9.4
13.01.07 09:00	30%	35%	8.3	-2.2 K	6.2°C	5.2°C	5.9	1.9	157	3.10	9.7
13.01.07 10:00	28%	33%	8.3	-2.1 K	6.2°C	5.3°C	5.7	1.8	150	3.11	10.0
13.01.07 11:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.1	1.6	135	3.13	10.6
13.01.07 12:00	26%	31%	8.3	-1.9 K	6.4°C	5.6°C	5.2	1.7	138	3.13	10.5
13.01.07 13:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.1	1.6	135	3.13	10.6
13.01.07 14:00	25%	30%	8.3	-1.9 K	6.4°C	5.6°C	5.2	1.7	137	3.13	10.5
13.01.07 15:00	43%	48%	8.3	-2.9 K	5.5°C	4.2°C	8.0	2.6	214	3.02	7.2

FIGURE L.2 SCREEN-GRAB OF GSHP PERFORMANCE EVALUATION TEST DATA.