SOLAR THERMAL HEATING OF A GLASSHOUSE USING PHASE CHANGE MATERIAL (PCM) THERMAL STORAGE TECHNIQUES

A thesis submitted for the degree Doctor of Philosophy

By

James Kwadwo Boampong

Supervised by Dr. Yunting Ge

School of Engineering and Design

Brunel University

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Brunel University London

Declaration of authenticity

The thesis is the author's original research work. It has been composed by the author and has not been submitted previously in any form for another degree at any university or other institute of tertiary education.

Signed:

Dated: 30 October 2015

Abstract

The Royal Botanic Gardens (RGB) is used as an umbrella name for the institution that runs Kew and Wakehurst Place gardens in Sussex

The RBG has a large number of glasshouses at Kew and Wakehurst sites that consume lots of heating energy which is a major concern and the group is looking for an alternative heating system that will be more efficient and sustainable to save energy, cost and reduce CO_2 emissions.

Glasshouse due to greenhouse effect trap solar energy in the space with the slightest solar gains but the energy trapped in the space most often is vented through the roof wasted to keep the space temperature to the required level.

An environmental measurement was carried out in twenty one zones of the glasshouse to establish the temperature and humidity profiles in the zones for at least three weeks. The investigation established that large amount of heat energy is vented to the atmosphere wasted and therefore need a heating system that could absorb and store the waste thermal energy. Phase change material (PCM) thermal energy storage technique was selected to be the best options compared to the others.

It has been established that active and passive solar systems could provide enough thermal energy to meet the glasshouse heating requirements. PCM filled heating pipes will be installed to absorb the heat energy trapped in the glasshouse and use it when needed. The research analysis established that 204 MWh of the trapped energy wasted could be saved.

The space temperature of the glasshouse could be maintained through melting and freezing of the PCM filled in the heating pipes. The site CHP waste heat could be useful. The research results have shown that nearly zero CO_2 emission heating system could be achieved and the project is technically, economically and environmentally viable.

Acknowledgement

This thesis originated as a necessity for Royal Botanical Gardens to reduce their CO_2 emissions to meet their corporate responsibility as an organization. My appreciation and gratitude go to Ben Monsoon, Sustainability and Energy Manager of Kew Gardens for their support and assistance for the success of the project.

I would like to thank my supervisor Dr. Yunting Ge and second supervisor Prof. Savvas Tassou for their guidance, encouragement and support. Dr. Yunting Ge has been very helpful for the several site visits he made with me to meet Kew Gardens Authorities for the project planning and the information needed from the organisation for the project.

I would also like to thank Zafer Ure of PCM Products Limited for supplying me all the information on their phase change materials (PCMs) products including PCM storage tanks cost.

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Nomenclature

η	Collector efficiency	[%]
η_0	Conversion factor	
a_1	Heat loss coefficient	[$Wm^{-2}K^{-1}$]
a_2	Temperature dependence heat loss coefficient	[$Wm^{-2}K^{-1}$]
T_m	Mean collector temperature	[^oC]
T_a	Ambient temperature	[°C]
G	Global solar irradiance	[Wm^{-2}]
G_a	Sum of solar and internal gains (W)	
Q_{coll}	Collected energy per collector area	[Wm^{-2}]
F_R	Collector heat removal factor	
τ	Transmittance of cover	
α	Shortwave absorptivity of the absorber	
UL	Overall heat loss coefficient of the collector	[$Wm^{-2}K^{-1}$]
ΔT	Temperature difference	[^oC]
d_m	Minimum diameter required	[m]
Re_{Ln}	Reynolds number	[Geometry characteristics length]
Re_D	Reynolds number	[Circular geometry]
C	Fluid mean velocity	[m/s]
V	Viscosity	[m^2/s]
m.f.r	Mass flow rate	[kg/s]
mk	Pipe material roughness	
LHS	Latent heat storage	[kJ/kg]
h_c	Convective heat transfer coefficient	[$Wm^{-2}K^{-1}$]
CV	Ventilation conductance	[WK^{-1}]
T_i	Inlet temperature	[^oC]
T_a	Ambient air temperature	[^oC]
f_r	Response factor	
Pv	Velocity pressure	[pa]
f	Friction factor	
A	Area	[m^2]
t_m	Time in minutes	[min]

Mh_{sf}	Stored energy	[kJ]
h_{sf}	Latent heat of fusion	[kJ/kg]
T_{hw}	Mean hot water temperature	[^oC]
T_{pcm}	PCM temperature	[^oC]
T_s	Surface temperature	[^oC]
T_{sur}	Surrounding temperature	[^oC]
T	Absolute temperature	[K]
T_∞	Moving fluid temperature	[^oC]
T_{mi}	The mean internal temperature per day [^oC]	
T_{mo}	External design temperature	[^oC]
q	The heating system output	[W]
H	The building heat transmission plus ventilation heat loss rate per degree (^oC) of internal to external temperature differential [W/K]	
q_{rad}	Radiation heat transfer	[W]
q_{conv}	Convection heat transfer	[W]
q_{tot}	Total heat transfer	[W]
h_r	Radiation heat transfer coefficient	[$Wm^{-2}K^{-1}$]
h_{ci}	Inner pipe surface heat transfer coefficient	[$Wm^{-2}K^{-1}$]
h_{cs}	Outer pipe surface heat transfer coefficient	[$Wm^{-2}K^{-1}$]
λ	thermal conductivity	[$Wm^{-1}K^{-1}$]
mL	Material thickness	[mm]
L_n	Length	[mm]
Re	Reynolds number	
Ra_{Ln}	Rayleigh number [L_n geometry characteristics length]	
Ra_D	Rayleigh number [D for circular geometry]	
T_f	Film temperature	[^oC]
ρ	Density	[kg/m^3]
L	Relative longwave sky irradiance	
ε	Longwave emissivity	
α	Absorptivity	
σ	Stefan Boltzman constant	
G_{ef}	Effective radiation on collector	[Wm^{-2}]

Q_{eff}	Effective heat transfer	[W]
$Q_{load, tot}$	Total collector load	[W]
f_{dirt}	loss of solar energy due to snow or dirt	[W]
f_{los}	Fraction loss of solar energy collected	[W]
Q_{did}	Energy delivered by the solar collector	[W]
Q_{act}	Energy collected by the solar collector	[W]
q''_s	Surface heat flux	[Wm^{-2}]
q''	Convection heat flux	[Wm^{-2}]
Q	Heat flow input or loss heat flows	[W]
P_r	Prandtl number	
ΔE_{st}	Change in stored energy	[kJ]
E_{in}	Energy in	[kJ]
E_{out}	Energy out	[kJ]
E^{tot}	Sum of thermal and mechanical energy	[kJ]
Eg	Thermal and mechanical energy generation	[kJ]
Δt	Change in time	[min]
U_{lat}	Latent heat	[kJ]

1 Introduction

1.1 Background

The Royal Botanic Gardens (RGB) is the institution that runs both the gardens at Kew and Wakehurst Place in Sussex and holds the world's largest collection of living plants and more than 30,000 different kinds of plants.

The herbarium has over 7,000,000 preserved plant specimens and the library contains more than 750,000 volumes, and the illustrations collection contains more than 175,000 prints and drawings of plants.

Kew Gardens is one hundred and eleven hectares of gardens and botanical glasshouses and it was created in 1759.

The greenhouses in both Kew Gardens and Wakehurst sites are heated using fossil fuels which are costly to run and the environmental impact is also significant.

RGB is seeking for an alternative heating system that is energy efficient, emit low or zero carbon dioxide emission and sustainable.

The European Commission is committed and seeking cost effective ways to make the European economy less energy consuming and make it more climate-friendly. They want to cut greenhouse gas emissions through the use of clean technologies. Thermal energy storage using phase change material (PCM) techniques could be one of such clean technologies where waste thermal or solar energy could be stored and use on demand.

Moving towards low-carbon society and economy will require increase innovation and investment in clean technologies and low or zero carbon emission systems.

This will need much greater attention and commitment to renewable sources of energy and this was the origin of the research aimed at achieving the following:

- Low or zero carbon dioxide emission heating system using phase change material (PCM) thermal energy storage techniques
- Space temperature controlled by melting and freezing of the PCM heating pipes
- Using space trapped solar energy to charge PCM filled heating pipes to maximise the use of solar energy
- Use of supplementary heat from CHP waste heat in winter season

1.1.1 Proposed systems

Two systems (Active and passive systems) were proposed to achieve the above aims by maximising the use of solar thermal energy through active and passive solar energy systems. Glasshouse gets warmer with the slightest solar gains but the heat energy trapped in the space most often is vented through the roof to keep the space temperature to the required level.

The active system will use solar collectors to absorb solar thermal energy through solar radiation whilst the passive system will use the PCM filled heating pipes to absorb the thermal energy trapped in the glasshouse due to greenhouse effect.

PCM thermal storage techniques have been used in the heating ventilation and air conditioning (HVAC) industry and other field applications such as heat recovery.

The selected salt hydrate PCM being used for the research project is produced by PCM Products Limited. They have identified large number of PCMs following an extensive research which suits majority of heating, air conditioning and refrigeration systems applications, electronics system enclosures and others.

The integrated environmental solutions (IES) software tool was used to design the glasshouse to determine zone hourly, daily and monthly heating demand, energy requirements, temperatures and relative humidity profiles.

This was to assist in knowing the periods when thermal energy needs to be generated to warm the glasshouse and periods where the solar gains are sufficient to warm the glasshouse. The excess heat gains trapped inside in the space will be stored using PCM heating pipes rather than venting it through the roof to the atmosphere.

The calculated glasshouse heating demand and energy requirement using the IES software tool for all the twenty one (21) zones under the research was 762 kW and 2497 MWh respectively.

Twenty per cent (20%) was added to the heating load of 762 kW to obtain 914 kW to account for heat loss in the system through hot water distribution pipework, storage tanks and other system components.

The calculated heating energy demand and solar gains in the glasshouse is 2497 MWh and 2359 MWh respectively. The difference between the heating energy demand and solar gains is 5.5 per cent (5.5%) which suggest that proper management of the solar gains could reduce the amount of thermal energy requirement from the active solar collector system.

Waste heat energy from a combine heat and power (CHP) plant close to the glasshouse under research shall be used to supplement heating energy from the solar collectors and stored the energy into the PCM storage tanks.

The system efficiency shall be achieved by systematic regulation of the hot water flow rates through the PCM heating pipes, the solar collector thermal energy hot water system and the CHP waste heat recovery system using Grundfos MAGNA3 variable speed circulation pumps to achieve heating system temperature set points to maintain system efficiency.

The glasshouse heating is by natural convection which is caused by buoyancy forces due to density differences caused by temperature variations in the fluid.

The convection heat transfer per unit surface area was first described by Newton in a relation known as the Newton's Law of Cooling. Each zone has a heat requirement and this has to be met by the PCM heating pipes.

Project appraisal was carried out to assess the proposed project's potential success, viability and risks using the discounted method of financial project appraisal.

The financial, economic and social benefits of the research project were assessed and analysed and found to be are technically sound, financially justified and environmentally acceptable.

1.2 Glasshouse brief

Controlled areas for plants growing have existed since Roman times [2]. Greenhouse system to grow plants is similar to the Roman gardener's artificial methods.

Greenhouse buildings are normally built with transparent materials that allow passage of sunlight and they are mostly glass or plastic.

The ground inside the greenhouse is warmed when there is sunshine and the space is heated up. The air inside the greenhouse is also warmed by the ground.

The air trapped inside the greenhouse continues to gain heat as the warm air near the ground rises and the cooler air aloft descend and the two mixes together. This was demonstrated experimentally by R. W. Wood, 1909 that a "greenhouse" with a cover of rock salt (which is transparent to infrared) heats up an enclosure similar to one with a glass cover [3].

The absorbed heat in a glasshouse is prevented from leaving the structure through convection. Figure 1:1 illustrates the sun energy transmission, reflection and absorption through a glsss. Some of the solar energy incident upon the glass is reflected, some absorbed and some transmitted.

The small amount of energy aborbed by the glass raises the temperature and by convection heat transfer transmit part of the absorbed heat to the outside and part to the room as demonstrated in Figure 1:1 below

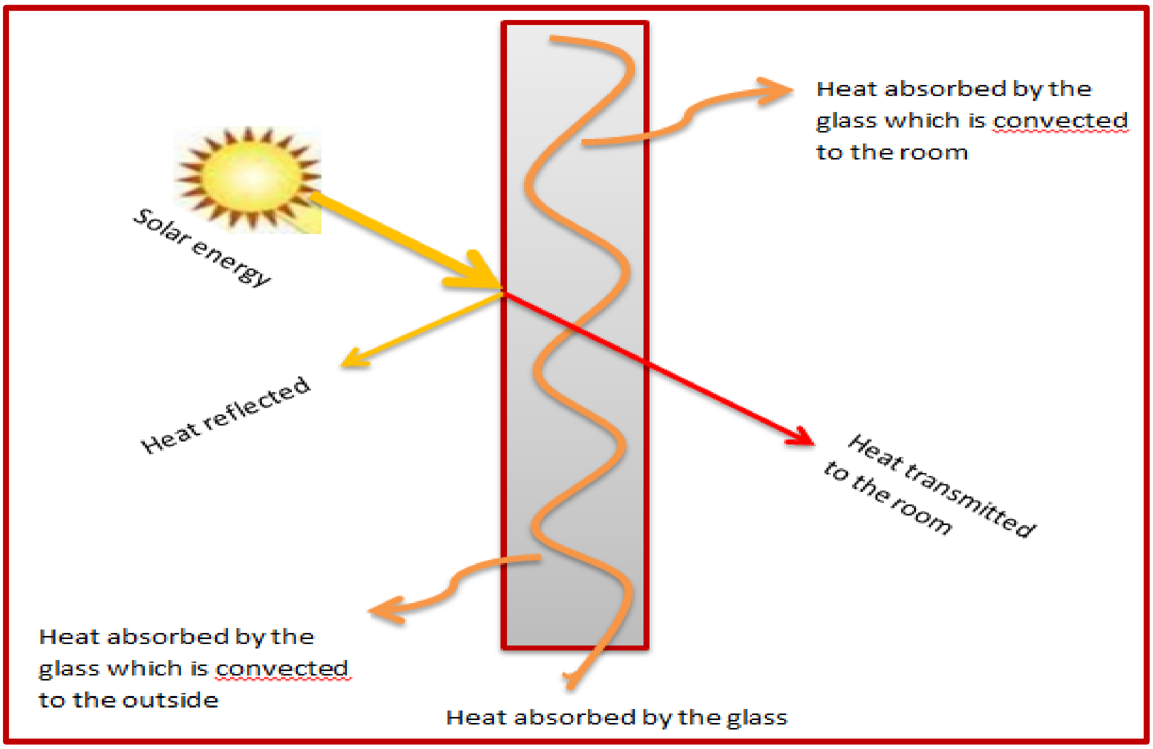

Figure 1:1The sun energy transmission, reflection and absorption through a glass

Plants in a glasshouse is protected and kept warmer than those outdoors. Solar radiation is often sufficient to keep the plants warmer during the day, but supplementary heating is needed during the night, overcast conditions and in winter.

The supplementary heating produced to warm the glasshouse is often by systems that use fossil fuels such as gas, coal, oil and electricity. Greenhouse gases are produced when fossil fuels are burnt and they are major contributors to climate change [4].

1.3 Solar energy

The most inexhaustible and cleanest source of energy is the solar energy [5].

Solar radiation is the radiation, heat and light emitted from the sun and contains huge amounts of energy which can be tapped for useful purposes. Some buildings are passively heated using solar thermal energy through architectural design and the use of certain building materials.

There are lots of building designs that the domestic hot water and space heating are produced through solar thermal energy. There are situations where absorption chillers for air conditioning system are powered by solar thermal technology.

1.3.1 Moving towards zero carbon economy

The European Commission is seeking cost effective ways to make the union less energy consuming economy [6]. They are committed to use of clean technologies to cut most of the greenhouse gas emissions by 2050.

One of these clean technologies could be thermal storage using phase change material (PCM) thermal energy storage techniques where waste thermal or solar energy could be stored and used on demand.

They have set out a clear pathway to achieve a deeper cut of emissions in the short, medium and long term and urged all major economies to do the same.

The EU commission adopted Directive 2012/27/EU on 25 October 2012 to promote energy efficiency. The Directive was established to promote energy efficiency within the Union. They want to achievement their 2020 targets by reducing greenhouse gas levels by 20%, increase share of renewables in the energy mix to 20% and reduce energy consumption by 20%.

The commission wants Europe to live and work in low-energy and low-emission buildings with intelligent heating and cooling systems.

1.4 Aims and objectives of the research

1.4.1 Aim of the research study

To design an efficient heating system for a glasshouse using low or zero carbon emission thermal energy through waste heat recovery and free solar energy with phase change material (PCM) thermal storage techniques.

1.4.2 Objectives of the research are:

- An energy efficient heating system that will reduce RBG CO_2 emissions by 20 % to meet their climate change commitment by 2020.
- Design a glasshouse that will meet plant requirements
- A heating system that the space set point temperature will be maintained by discharging and charging of the PCM filled heating pipes
- Carryout project cost analysis including economic life cycle cost analysis and the environmental impact of the heating system
- A heating system that will achieve 100% or nearly 100% zero carbon emissions
- A heating system that will eliminate the use fossil fuel
- A heating system that will improve operational performance of Kew Gardens to enhance their corporate social responsibilities within the community
- To improve resource efficiency

1.4.3 Scope of the research

- Design a heating system powered by solar thermal energy and supplemented by waste heat recovery from CHP plant
- To design both active and passive thermal energy storage systems using PCM to provide enough heating energy to meet glasshouse heating load
- To design heating pipes that will be filled with PCM solution to store thermal energy to heat the glasshouse
- The PCM thermal storage will be charged by LTHW generated by the active solar thermal energy hot water system and heating pipe system charged by both LTHW and the glasshouse space warm air

- Capture solar energy through solar collectors and PCM heating pipes
- Maximise the use of trapped solar energy in the glasshouse to reduce thermal energy generation in heating the glasshouse
- Improve temperature fluctuation in the glasshouse by regulating the opening and shutting of the vents to improve plants growth condition
- Store solar energy trapped inside the glasshouse due to greenhouse effect with PCM filled heating pipes.

1.4.4 The originality of the research

The research originated when the Royal Botanical Gardens (RGB) asked supplies to design efficient and cost effective alternative heating system that will emit low or zero carbon dioxide emission to replace the existing heating systems in Kew and Wakehurst glasshouses.

The current heating system in Kew Gardens and in Wakehurts use fossil fuel to power the boilers and not cost effective in their operations. The annual energy consumption and CO_2 emissions are large and need an innovative efficient heating system to assist reduce their energy consumption and CO_2 emission reduction targets by 2020 and beyond.

There are several heat generating systems that were considered such as biomass, heat pumps and others but the most innovative and effective system is to use solar thermal energy through active and passive systems. The CO_2 emission is zero, free source of energy and sustainable.

Glasshouse due to greenhouse effect trap solar energy in the space with the slightest solar gains but the energy trapped in the space most of the time is vented through the roof to keep the space temperature to the required level.

Considering the significant amount of thermal energy that is vented to the atmosphere wasted came the idea that if this thermal energy could be effectively stored without being wasted then fossil fuel use in glasshouse heating could be eliminated or reduced.

The benefits of the research apart from eliminating or reducing CO_2 emissions for using fossil fuel in heating the glasshouse will promote resource efficiency. The originality of the research is based on the three main principles stated below:

- A Zero or nearly zero carbon dioxide emission heating system using phase change material thermal energy storage techniques
- Space temperature controlled by melting and freezing of the PCM heating pipes
- Using space trapped solar energy to charge PCM filled heating pipes to maximise the use of the solar energy

The above goal when achieved will add knowledge to glasshouse heating systems and the design principles could be effectively applied to domestic and non-domestic buildings.

1.5 Solar energy

1.5.1 The sun as a source of energy

There is a belief that the sun energy will last for another 5 billion years and for every second the sun produces 400,000,000,000,000 terawatts of energy [7].

The most reliable and abundant source of energy is the sun as there is no prediction of when global oil production will peak.

1.5.2 Solar energy competing with fossil fuels

Solar energy systems have competed with fossil fuel in different fields for example, Horace de Saussure, a Swiss scientist in 1767 invented the first world solar energy collector hot box.

Charles Tellier (1885-1889) designed the first solar collector thermal heating system to heat domestic hot water.

The British astronomer, John Herschel in the 1830s used solar collector box to cook food in his expedition to Africa.

The 1970 energy crisis during the Organization of Petroleum Exporting Countries oil embargo was a lesson to the world to explore the potentials of renewable energy.

1.5.3 Solar water heating

Solar water heating has been used for more than one hundred years by simply using black painted tanks in many countries [8].

Solar hot water technology has improved and over thirty (30) million solar collectors installed in countries such as Germany, United Kingdom, Israel, Denmark, France, Japan and others. Figure 1:2 below is a sample of an integrated collector storage system.

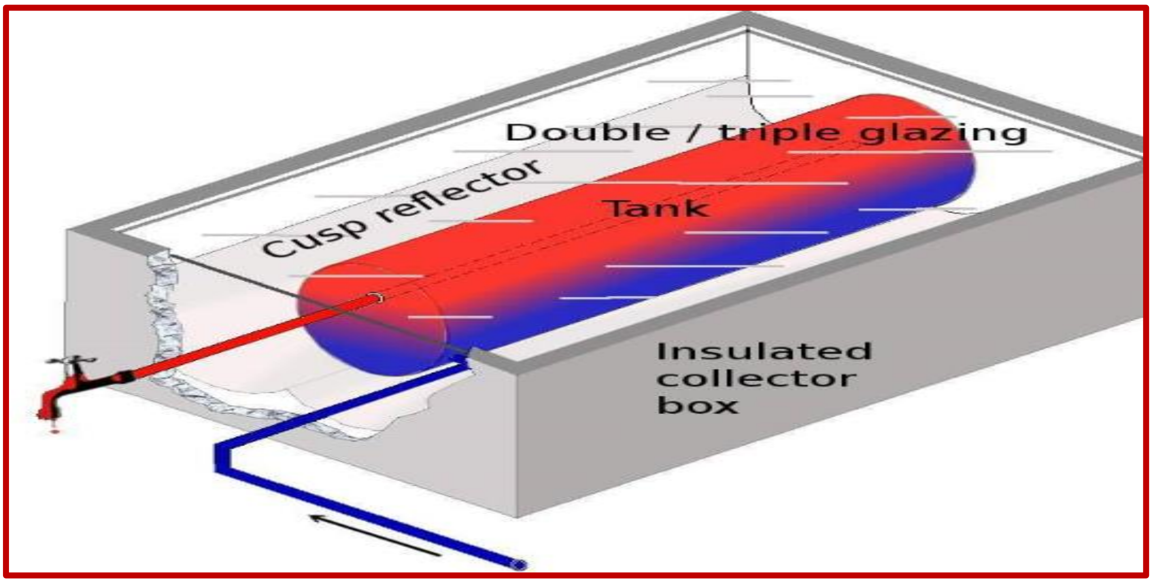

Figure 1:2 An integrated collector storage (ICS) system

Source: USA Department of Energy

Solar energy systems have been successfully used in the UK for the past thirty years. And the maximum solar radiation received in the UK is about 1000W/m^2 with seasonal variations.

In practical terms, it is unlikely that solar thermal energy system alone could meet building total thermal energy requirement in the UK at a viable cost. Most often convectional systems such as boilers or combine heat and power (CHP) is installed alongside it to supplement the heat provided by the solar collector if heating demand exceed the solar collector system capacity.

The research is focused in designing a heating system independent to heat a glasshouse without any convectional system installed alongside it but the heating system will take advantage of waste heat provided by CHP in Kew Gardens.

1.6 Research outline

The research development was structured to cover five mains areas of study. Chapters one gives the background and purpose of the research, scope, brief of glasshouse and heating methods.

It gives brief description of solar energy which is the most inexhaustible and cleanest energy and why it has not been harness until recently.

Research drivers and phase change material (PCM) storage methods and applications are briefly described.

Chapter two was the study and review of what has been achieved and covered by others and it was a study of what others have contributed towards thermal energy storage using phase change material.

It covers brief introduction to all areas of thermal energy storage but focus is on thermal energy storage using phase change material techniques.

It describes the thermodynamic properties of PCMs, type of PCMs, classification of PCMs and demonstrates melting and freezing curves of PCM samples.

Chapter three is the research methodology and covers the studies of plants behaviour and characteristics to assist effective design of the glasshouse.

Investigate the design requirements and parameters of glasshouse and studied the greenhouse effects of a glasshouse. Modelled the existing glasshouse and determined the predicted thermal space conditioning energy requirement per year using the integrated environmental solutions (IES) software tool.

It is under this chapter that the design of the heating system to serve all the zones including control mechanism, periods where solar thermal energy can be passively stored and effectively used in heating the space.

Detailed study of solar thermal collectors and their efficiencies at various ambient temperatures and inlet hot water temperatures were studied.

Chapter four dealt with the design results and analysis of the space heating requirements, temperature profiles, and periods where solar gains in the glasshouse can be stored for later use. Chapter five covered the glasshouse heat transfer analysis and chapter six was the detail study of the research project financial appraisal including economic life cycle cost analysis.

Summary of introduction

The world has come to a realisation that climate change is real and the risk is enormous if nothing is done control it.

The European Union is committed and wants to cut most of the greenhouse gas emissions by 2050 through the use of clean technologies.

The EU has put in place legislation to reduce its emissions to 20% in the short term below 1990 levels by 2020 and urges other major economies to do their bit in the global reduction effort.

The EU wants to be resource efficient and has put forward policy initiatives to use resources in a sustainable way. They continually put forward roadmap initiatives to cut CO2 emissions to 80% below 1990 levels by 2050.

This will cover the main sectors responsible for emissions such as power generation, industry, transport, buildings and construction, as well as agriculture.
Moving towards low-carbon society and economy will require increased innovation and investment in clean technologies.

Achieving low or zero carbon energy will need much greater attention and commitment to renewable sources of energy, energy-efficient building materials, smart grid equipment, low-carbon power generation, carbon capture and storage technologies.

There is a huge amount of energy in solar radiation needs to be harnessed. The research is aimed at maximising the use of this huge amount of energy from the solar radiation and the sun’s energy.

Glasshouses have existed since Roman times and they have been mostly heated using fossil fuel heating systems. Materials normally used in building glasshouses are transparent that allow the passage of sunlight such as glass or plastic.

Glasshouse during the day time gets overheated with sufficient sunlight and some of the heat has to be vented to the atmosphere wasted so the research is aimed at storing this wasted energy for use when needed.

The study maximises the use of solar energy in heating the glasshouse as this is the most in-exhaustive and cleanest of all known energy sources.

The research motivation is that Europe and other states in the world are seeking ways to become less energy consuming economies to make it more climate friendly.

All the heating energy requirement of the glasshouse will be stored using thermal energy storage techniques.

The space temperature will be controlled by melting and freezing of the PCM filled heating pipes. This will be supported by space temperature sensor which will control flow of low temperature hot water through the PCM heating pipes and also control opening of the vents to maintain space temperature requirements.

Reference

[1] www.kew.org accessed 31 August 2014

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[3] http://www.wmconnolley.org.uk/sci/wood_rw.1909.html

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[5] http://en.sunrain.com/ accessed 14 February 2013

[6] http://ec.europa.eu/clima/policies/g-gas/index_en.htm accessed 3 March 2013

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[8] www.retscreen.net/download.php/ang/120/0/textbook_swh.pdf accessed 14 October 2012

2 Literature review

2.1 Introduction

Thermal energy storage is an important technology in managing energy with emphasis on energy efficiency and conservation of waste heat and solar energy across industries and buildings.

An effective way of storing thermal energy is the use of latent heat storage techniques using phase change material (PCM). Phase change materials have the advantage of high energy storage density and have been used in many field applications in the heating, ventilation, air conditioning systems, solar systems and others.

There are a wide range of PCMs with different phase change temperatures making them suitable for different project applications. It is a thermal energy storage system that assists to take advantage to secure the benefits of renewable source energy, store waste energy that is not needed during production for later use

Capturing direct and indirect solar radiation as renewable source of energy and storing it for later use through thermal energy storage system is challenging but rewarding.

Thermal energy storage technique is important in industrial processes where huge amount of energy is wasted because it is not needed at the time of production and cannot be stored. Thermal energy storage reduces the gap between supply and demand, makes system reliable, conserves energy and improves performance efficiency [1]

There are many methods of energy storage such as mechanical, kinetic, chemical and thermal energy. All these forms of energy storage are important but for the purpose of this research thermal energy storage will be considered.

A material thermal energy could be stored by cooling, heating, melting, solidification or vaporisation [1] and the internal energy could be changed through sensible heat, latent heat and thermochemical or combination of both.

Figure 2:1 below shows sensible versus latent heat energy storage. A substance could exist in three states which are solid, liquid and gaseous depending on the pressure and temperature of the storage conditions.

Energy is stored in sensible heat storage by raising the substance temperature. This could be solid, liquid or gaseous. Figure 2:1 shows sensible only and PCM line.

It could be seen from the sensible only line that the amount of stored energy depends on the temperature change and the specific heat capacity which has been assumed to remain constant during the charging and discharging process.

Phase change materials (PCMs) are substances used for latent heat storage with the advantage of the temperature remaining constant during the phase change process.

The most commonly used phase change is the solid and liquid. However, solid / solid and liquid / gas phase changes could also be used.

In latent heat storage both sensible and latent storage may occur in the same material [2] and this is demonstrated in figure 2.1.

When a substance is heated, it increases the internal energy. In sensible heating, the temperature of the substance rises whilst in latent heating the phase changes [3]. PCM selection for heat storage plays an important role in thermal energy storage system [4].

The amount of heat energy that is absorbed or released during material phase change from solid to liquid state or vice-versa is large and called latent heat of melting or fusion [5].

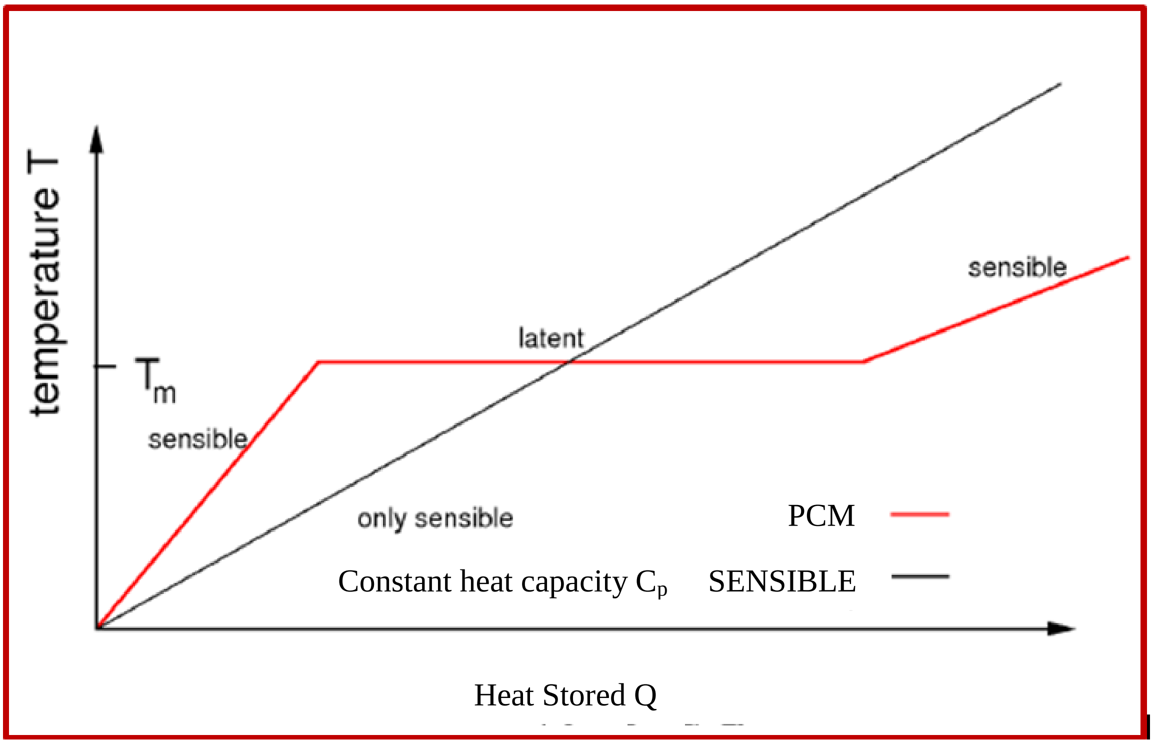

Figure 2:1 Illustrates sensible versus latent heat storage

Source: Institute of Solar Energiesystems

2.1.1 Sensible heat storage

In sensible heat storage, the amount of stored energy depends on change of the temperature and the specific heat capacity of the material during charging and discharging process [3]. Equations 2.1 and 2.2 show the relationship.

$$Q = \int_{T_i}^{T_f} mC_p dT \tag{2.1}$$

$$= mC_p(T_f - T_i) \tag{2.2}$$

2.1.2 Latent heat storage

Heat is absorbed or release in latent heat storage when the substance undergoes a phase change from solid to liquid, liquid to gas or vice-versa. The amount of PCM energy stored is given by

$$Q = \int_{T_i}^{T_f} mC_p dT + ma_m \Delta h_m + \int_{T_m}^{T_i} mC_p dT \quad (2.3)$$

$$Q = m\,[C_{sp}(T_m - T_i) + a_m \Delta h_m + C_{lp}(T_f - T_m)] \quad (2.4)$$

2.2 Thermal energy storage

Figure 2:2 [6] below gives an overview of thermal energy storage techniques. They are the major techniques or methods used in the thermal energy storage industry.

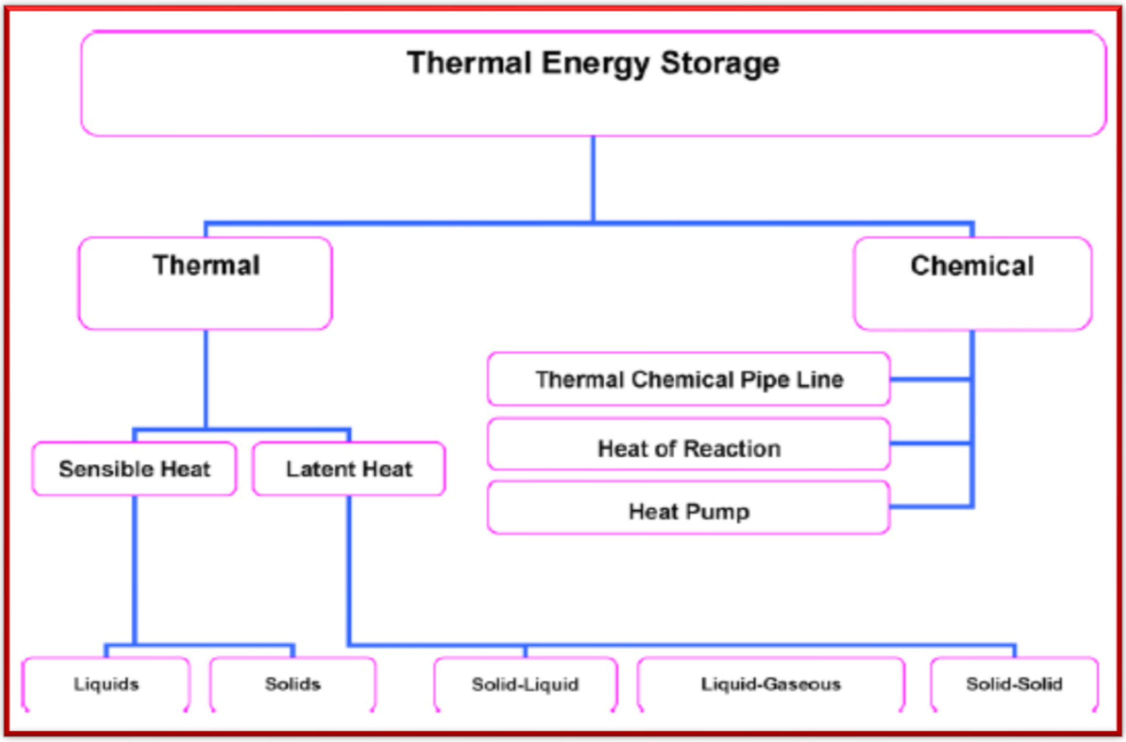

Figure 2:2 Different types of thermal energy storage

2.3 Phase change material

2.3.1 Background of PCMs

Phase change materials temperature during melting and solidification process remains almost constant and absorbs a lot of energy during the phase change.

Latent heat of fusion is measured in kilo-joules per kilogram (kJ / kg) and this is the amount of energy required to melt a kilogram (kg) of the substance.

Duration Index is a measure to determine the duration of which the PCM temperature will remain constant during the phase change. Calculated as: D.I. = $h_{sf}\ \rho\ /\ \Delta T$

Where h_{sf} is the latent heat of fusion of the PCM

ρ is the density of the PCM

ΔT is temperature difference between the ambient and the PCM melting point

Table 2.1 below demonstrates duration indexes of certain PCMs [7].

Table 2:1 Duration index based on temperature difference (ΔT) from melt point to ambient temperature of 25 ^{o}C

Material	Melt Point [^{o}C]	Heat of Fusion [kJ/kg]	Liquid Density [g/cm^{3}]	Duration Index [J/(cm^{3*} ^{o}C)]
Heptanone-4	-33	209	0.822	2.96
n-Undecane	-26	141	0.740	2.05
TEA-16	-16	289	1.100	7.75
Ethelene glycol	-11.5	179	1.109	5.44
n-Dodecane	-9.6	211	0.749	4.57
Water	0	333	1	13.32
Thermasorb 43	6	163	0.87	7.46
Thermasorb 65	18	173	0.88	21.75
Sodium hydrogen phosphate	36.1	280	1.45	-36.58
Thermasorb 175+	79	200	0.93	-3.44
Thermasorb 215+	101	193	0.93	-2.36

2.3.2 Consideration of PCMs

The following properties are important when selecting phase change materials for an application:

2.3.2.1 Thermodynamic properties

The phase change material should possess the following thermodynamic properties [8, 9]

- Phase change temperature within the required operating temperature range
- High latent heat of fusion per unit volume
- High specific heat capacity, high density, duration index and high thermal conductivity
- Small volume changes on phase transformation and low vapour pressure at operating temperatures
- Congruent melting (When the anhydrous salt is completely soluble in its water of hydration at the melting temperature)
- Favourable phase equilibrium

2.3.2.2 Thermal properties

The phase change material should possess the following thermal properties [8.9]

- Phase change temperature suitability
- High latent heat of fusion
- Good heat transfer

2.3.2.3 Economic properties

The phase change material should possess the following economic properties [8,9]

- Abundant
- Low cost
- Availability

2.3.2.4 Chemical properties

The phase change material should possess the following chemical properties [8,9]

- Long-term chemical stability
- Compatibility with materials of construction
- No toxicity
- No fire hazard
- Complete reversible freeze / melt cycle
- No degradation after a large number of freeze / melt cycle
- Non-corrosiveness, non-toxic, non-flammable and non-explosive materials

2.3.2.5 Kinetic properties

The phase change material should possess the following kinetic properties [8,9]

- High nucleation rate to avoid supercooling of the liquid phase
- High rate of crystal growth to meet demands of heat recovery from the storage system.

2.3.3 Classification of PCMs

Phase change materials can be classified as organic, inorganic and eutectic.
Figure 2:3 [10] below shows PCM classification.

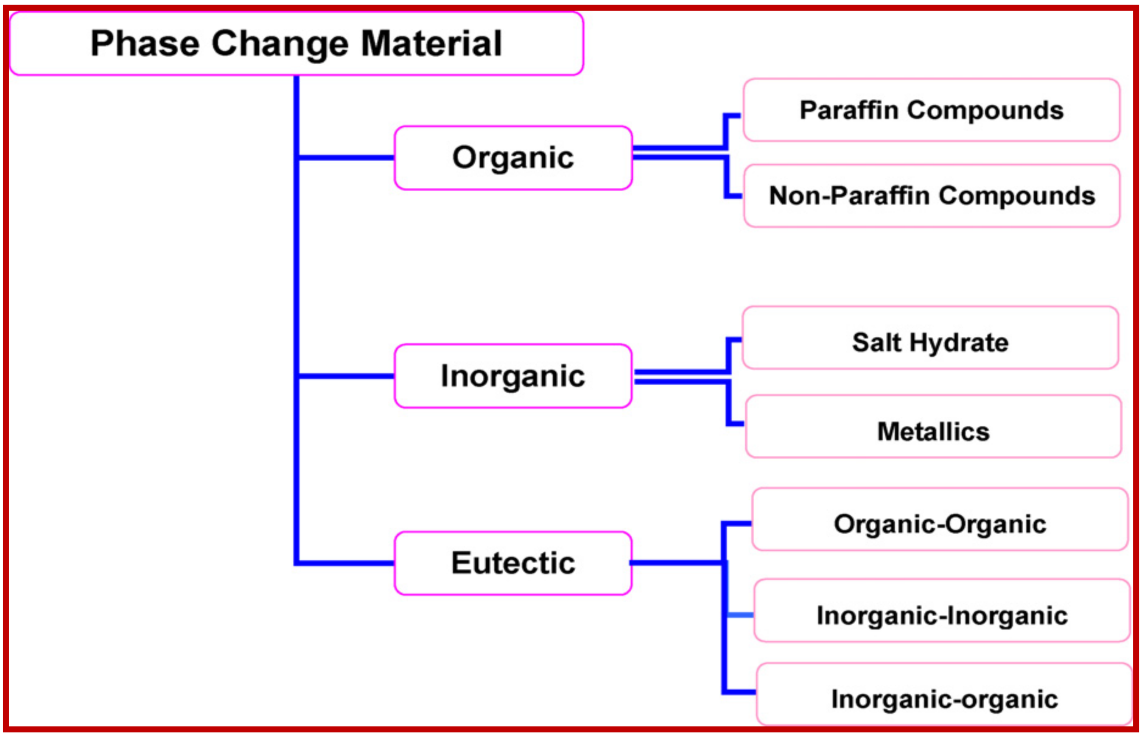

Figure 2:3 Classification of PCMs

Materials identified as PCMs from phase change temperature and latent heat of fusion point of view are organic and inorganic. The properties discussed above would have to be met by these materials but majority of them do not meet the criteria for an adequate storage.

As all the properties required for an ideal thermal storage media could not be satisfied by a material, the inadequate or the poor physical property would have to be met by efficient design [10].

Increasing thermal conductivity of a PCM for example, could be achieved by metallic fins or nucleating agent being introduced to suppress supercooling in the material [10]. The volumetric latent heat storage capacity of inorganic compounds (250-400 kg / dm^3) is almost twice that of organic compounds (128-200 kg / dm^3) [10].

The characteristics of each subgroup are different and therefore require detail study but this research is more to do with an application of PCMs rather the chemistry of it.

The most widely and commonly used PCMs in industry are water / ice phase change at zero degree (0 ^{0}C), salt hydrates, organics and clathrates.

2.3.3.1 Organic phase change materials

Paraffin and non-paraffin is described and classified as organic material. It melts and freezes repeatedly without phase segregation which means that it is congruent. Paraffins are usually non-corrosive and have little or no supercooling. The following are the features of organic materials [10]:

- Relatively expensive
- Not flammable
- Latent heat of fusion is high
- Flash points are low
- Thermal conductivity is low
- Not stable at high temperatures
- The toxicity level varies
- Density is low

2.3.3.1.1 Paraffins

Straight chain alkanes make the composition of paraffin wax. Paraffin releases large amount of latent heat. Paraffin has a large temperature ranges and that qualifies them to be used as storage materials but only technical grade paraffins are considered in thermal storage systems due to their cost. Table 2:2 demonstrates the properties of paraffin.

Table 2:2 Properties of paraffin

Paraffin[a]	Freezing point / range (^{o}C)	Heat of Fusion (KJ/Kg)	Group[b]
6106	42-44	189	I
P116c	45-48	210	I
5838	48 – 50	189	I
6035	58- 60	189	I
6403	62 – 64	189	I
6499	66 – 68	189	I

Group I, most promising; group II, promising; group III, less promising; insufficient data.

Table 2:3 shows the melting point and latent heat of fusion of some technical grade paraffins. The mixtures of paraffins are not completely refined oil [10].

Undesirable properties of paraffins are:

- Low thermal conductivity
- Non-compatible with the plastic container
- Moderately flammable.

Table 2:3 Melting point and latent heat of fusion: paraffins

No. of carbon atoms	Melting point (^{o}C)	Latent heat of fusion (kJ/kg)	Group[a]
14	5.5	228	I
15	10	205	II
16	16.7	237.1	I
17	21.7	213	II
18	28.0	244	I
19	32.0	222	II
20	36.7	246	I
21	40.2	200	II
22	44.0	249	II
23	47.5	232	II
24	50.6	255	II
25	49.4	238	II
26	56.3	256	II
27	58.8	236	II
28	61.6	253	II
29	63.4	240	II
30	65.4	251	II
31	68.0	242	II
32	69.5	170	II
33	73.9	268	II
34	75.9	269	II

Some selected paraffins along-with their melting point, latent heat of fusion and groups. They are categorized as: Group I, most promising, group II promising; and group III less promising.

2.3.3.1.2 Non-paraffins

Table 2:4 shows melting point and latent heat of fusion of non paraffins.
Non-paraffin materials are the largest category for phase change storage [11] and Buddhi and Sawhney [9] conducted a survey about organic materials and esters, fatty acids, alcohols and glycols.

Table 2:4 Melting point and latent heat of fusion: non paraffins

Material	Melting point (°C)	Latent heat (kJ/kg)	Group[a]
Formic acid	7.8	247	III
Caprilic acid	16.3	149	-
Glycerin	17.9	198.7	III
D-Lattic acid	26	184	I
Methyl palmitate	29	205	II
Camphenilone	39	205	II
Docasyl bromide	40	201	II
Caprylone	40	259	II
Phenol	41	120	III
Heptadecanone	41	201	II
1-Cyclohexylooctadecane	41	218	II
4-Heptadacanone	41	197	II
p-Joluidine	43.3	167	-
Cyanamide	44	209	II
Methyl eicosanate	45	230	II
3- Heptadacanone	48	218	II
2-Heptadacanone	48	218	II
Hydrocinnamine acid	48.0	118	-
Cetyl alcohol	49.3	141	-
a-Nepthylamine	50.0	93	-
Camphene	50	238	III
O-Nitroaniline	50.0	93	-
9-Heptadecanone	51	213	II
Glautaric acid	97.5	156	-
p-Xylene dichloride	100	138.7	-
Catechol	104.3	207	III
Quinone	115	171	II
Acetanilide	118.9	222	II
Succinic anhydride	119	204	II
Benzoic acid	121.7	142.8	III
Stibene	124	167	-
Benzamide	127.2	169.4	III

Group I, most promising; group II, promising; group III, less promising; insufficient data.

2.3.3.1.3 Fatty acids

Fatty acids are organic materials and have high latent heat of fusion compare to paraffins and shows no supercooling in melting and freezing processes [12, 13].

Table 2:5 has a list of some fatty acids which are of interest to low temperature latent heat thermal energy storage applications.

Table 2:5 Melting point and latent heat of fusion: fatty acids

Material	Formula	Melting point (°C)	Latent heat (kJ/kg)	Group[a]
Accetic acid	CH_3COOH	16.7	184	I
Polyethylene glycol 600	$H(OC_2H_2)_{n*}OH$	20-25	146	I
Capric acid	$CH_3(CH_2)_{8*}COOH$	36	152	-
Eladic acid	$C_8H_7C_9H_{16*}COOH$	47	218	I
Lauric acid	$CH3(CH_2)_{10*}COOH$	49	178	II
Pentadecanoic acid	$CH_3(CH_2)_{13*}COOH$	52.5	178	-
Tritearin	$(C_{17}H_{35}COO)C_3H_5$	56	191	I
Myristic acid	$CH_3(CH_2)_{12*}COOH$	58	199	I
Palmatic acid	$CH_3(CH_2)_{14*}COOH$	55	163	I
Stearic acid	$CH_3(CH_2)_{16*}COOH$	69.4	199	I
Acetamide	CH_3CONH_2	81	241	I
Methyl fumarate	$(CHCO_2NH_3)_2$	102	242	I

Group I, most promising; group II, promising; group III, less promising; - insufficient data.

2.3.3.2 Inorganic phase change materials

2.3.3.2.1 Salt hydrates

The compounds of salt hydrates are salt and water with high latent heat of fusion due to the high water content. Salt hydrates major problem during the charging and discharging modes are phase segregation. It leaves salt settling at the solution bottom causing changes to the solution storage capacity. This is an irreversible process.

Salt hydrates have extensively been studied and used for thermal energy storage applications and below are the properties:

- Latent heat of fusion is high per unit volume
- Thermal conductivity is high
- Volume changes on melting is small
- They compatible with plastics, slightly toxic and not corrosive

Many salt hydrates are inexpensive for the use in thermal energy storage systems [14].

A list of salt hydrates showing melting point and latent heat of fusion is given in table 2:6 below.

Table 2:6 Salt hydrates melting point and latent heat of fusion

Melting point and latent heat of fusion: salt hydrates			
Material	Melting point (^oC)	Latent heat (kJ/kg)	Group[a]
$K_2HPO_4{*}6H_2O$	14.0	109	II
$FeBr_3{*}6H_2O$	21.0	105	II
$Mn(NO_3)_2{*}6H_2O$	25.5	148	II
$FeBr_3{*}6H_2O$	27.0	105	II
$CaCl_2{*}12H_2O$	29.8	174	I
$LiNO_3{*}2H_2O$	30.0	296	I
$LiNO_3{*}3H_2O$	30	189	I
$Na_2CO_3{*}10H_2O$	32	267	II
$Na_2SO_4{*}10H_2O$	32.4	241	II
$KFe(SO_4)_2{*}12H_2O$	33	173	I
$CaBr_2{*}6H_2O$	34	138	II
$LiBr_2{*}2H_2O$	34	124	I
$Zn(NO_3)_2{*}6H_2O$	36.1	134	III
$FeCl_3{*}6H_2O$	37.0	223	I
$Mn(NO_3)_2{*}4H_2O$	37.1	115	II
$Na_2HPO_4{*}12H_2O$	40.0	279	II
$CoSO_4{*}7H_2O$	40.7	170	I
$KF{*}2H_2O$	42	162	III
$MgI_2{*}8H_2O$	42	133	III
$CaI_2{*}6H_2O$	42	162	III
$K_2HPO_4{*}7H_2O$	45.0	145	II
$Zn(NO_3)_2{*}4H_2O$	45	110	III
$Mg(NO_3){*}4H_2O$	47.0	142	II

Group I, most promising; group II, promising; group III, less promising;-insufficient data.

2.3.3.2.2 Metallics

Low melting metals and metal eutectics include metallics. Because of their weight problem they have not been seriously considered for PCM thermal storage systems.

They are likely to be considered because of their high latent heat of fusion per unit volume and thermal conductivity. A list of some selected metallics is given in table 2:7 below.

Table 2:7 Metallics melting point and latent heat of fusion

Melting point and latent heat of fusion: metallic			
Material	Melting point (^oC)	Latent heat (kJ/kg)	Group[a]
Gallium-gallium antimony eutectic	29.8	-	-
Gallium	30.0	80.3	I
Cerrolow eutectic	58	90.9	-
Bi-Cd-In eutectic	61	25	-
Cerrobend eutectic	70	32.6	I
Bi-Pb-In eutectic	70	29	-
Bi-In eutectic	72	25	-
Bi-Pb-tin eutectic	96	-	-
Bi-Pb eutectic	12.5	-	-

Group I, most promising; group II, promising; group III, less promising;-insufficient data

2.3.3.3 Eutectics

Eutectic is mixing of two or more substances to provide the desire melting / freezing point. At the phase change temperature the mixtures melt completely and the overall composition will be in liquid phase [15].

However, very few of PCMs documented are unfortunately true Eutectics. For long term use many of them need modifications to be suitable.

Eutectics nearly always melt and freeze without segregation with little chance of components separating. The components liquefy simultaneously during melting with unlikely separation. Table 2.8 below shows a list of organic and inorganic eutectics.

Table 2:8 Organic and inorganic eutectics

List of organic and inorganic eutectics				
Material	Composition (wt.%)	Melting point (^oC)	Latent heat (kJ/kg)	Group[a]
$CaCl_2{*}6H_2O+CaBr_2{*}6H_2O$	45+55	14.7	140	-
Triethylolethane+Water+Urea	38.5+31.5+30	13.4	160	I
$C_{14}H_{28}O_2+C_{10}H_{20}O_2$	34+66	24	147.7	-
$CaCl_2+MgCl_2{*}6H_2O$	50+50	25	95	II
$CH_3CONH_2+NH_2CONH_2$	50+50	27	162	II
Triethylolethane + Urea	62.5+37.5	29.8	218	I
$Ca(NO_3){*}4H_2O+Mg(NO_3)_3{*}6H_2O$	47+53	30	136	-
$CH_3COONa{*}3H_2O+NH_2CONH_2$	40+60	30	200.5	I
$NH_2CONH_2+NH_4NO_3$	53+47	46	95	II
$Mg(NO_3)_3{*}6H_2O+NH_4NO_3$	61.5+38.5	52	125.5	I
$Mg(NO_3)_3{*}6H_2O+MgCl_2{*}6H_2O$	58.7+41.3	59	132.2	I
$Mg(NO_3)_3{*}6H_2O+MgCl_2{*}6H_2O$	50+50	59.1	144	-
$Mg(NO_3)_3{*}6H_2O+Al(NO_3)_2{*}9H_2O$	53+47	61	148	-
$CH_3CONH_2+C_{17}H_{35}COOH$	50+50	65	218	-
$Mg(NO_3)_2{*}6H_2O+MgBr_2{*}6H_2O$	59+41	66	168	I
Napthalene+benzoic acid	67.1+32.9	67	123.4	-
$NH_2CONH_2+NH_4Br$	66.6+33.4	76	151	II
$LiNO_3+NH_4NO_3+NaNO_3$	25+65+10	80.5	113	-
$LiNO_3+NH_4NO_3+KNO_3$	26.4+58.7+14.9	81.5	116	-
$LiNO_3+NH_4NO_3+NH_4Cl$	27+68+5	81.6	108	-

2.3.3.4 PCM products and applications

2.3.3.4.1 PCM Products Ltd

PCM Products Ltd in England produces PlusICE mixtures of non-toxic phase change materials (Positive temperature PCMs) solutions which the phase change temperatures are higher than that of water.

The solution is encapsulated in a unique design cylindrical beam used for large commercial, institutional and industrial applications. It is used in the heating, ventilation and air conditioning (HVAC) systems and processes.

2.3.3.4.2 PlusICE PCM solutions

A number of satisfactory PCMs such as eutectics have been identified by PCM Products Ltd through extensive research which suits the majority of heating, air conditioning and refrigeration applications.

The identified PlusICE eutectic PCMs are non-toxic, non-combustible and inorganic. Majority of the identified salt hydrates are not subject to contraction or expansion during the phase change process.

The composition of the salt hydrate PCM or the storage capacity do not degrade and performs reliably. PlusICE PCM relevant temperature range and technical details of salt hydrate are illustrated in table 2.9 below. Figure 2:4 below shows PlusICE PCM solutions temperatures.

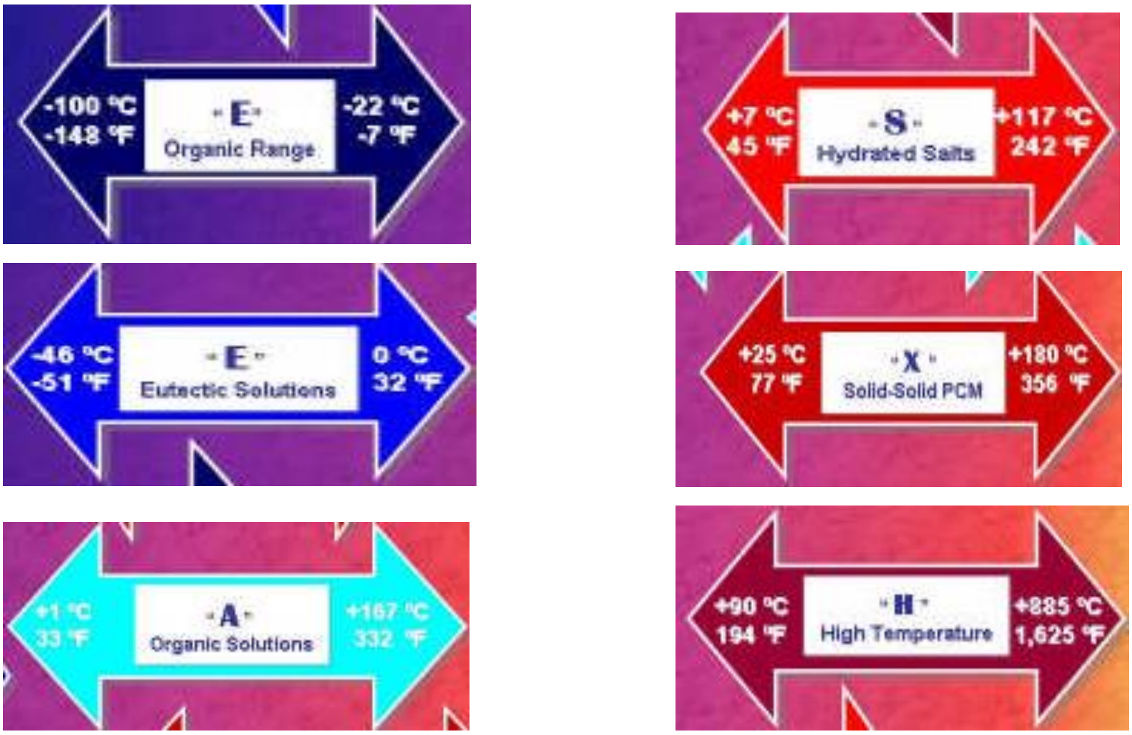

Figure 2:4 PlusICE PCM solutions temperature range
(Source: www.PCMproducts.net)

2.3.3.4.3 Comparison of typical physical size of thermal energy storage systems

Salt hydrate and some PCMs have high latent heat of fusion. As a result, large amount of thermal energy could be stored compared to its physical size and therefore reduces storage space.

Figure 2:5 below demonstrates thermal storage sizes of ICE, Eutectic and chilled water. It could be seen from figure 2:5 that the ICE and eutectic thermal energy storage sizes are 1/15 and 1/10 of that of the chilled water because the thermal energy capacity per volume of the ICE and eutectic are 306 MJ / m^3 and 206 MJ /m^3 respectively compare to the chilled water of 20 MJ / m^3.

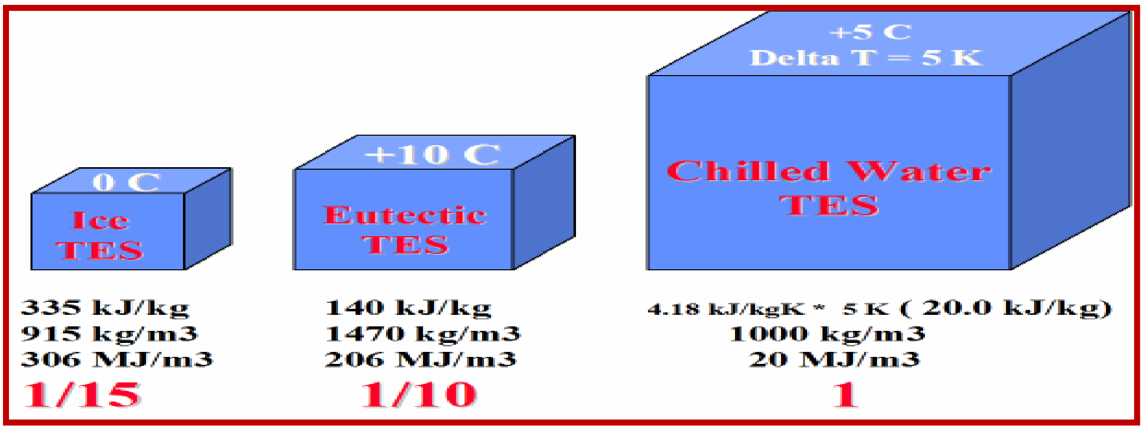

Figure 2:5 PCM thermal storage concept

Source: www.PCMproducts.net

2.3.3.4.4 PlusICE thermal energy storage (TES) concept

PCM products Ltd has a custom made plastic containers which the PCM solutions are filled in. They can be used for variety of thermal energy storage applications and can be stacked in cylindrical or rectangular tanks for pressured or atmospheric systems. Figure 2:6 shows TubeICE custom-made HDPE plastic container.

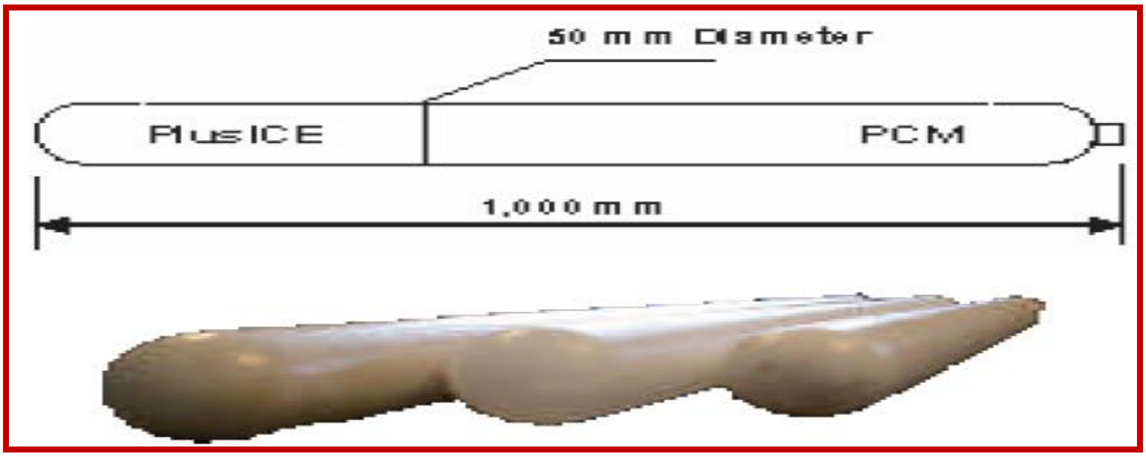

Figure 2:6 TubeICE custom-made HDPE plastic containers

Source: www.PCMproducts.net

The plastic containers have been designed in such a way that they can stack on top of each other in a large heat exchanger tank as demonstrated in figure 2:7. Figure 2:7 below shows plastic containers stacking on top of each other forming self-assembling heat exchanger.

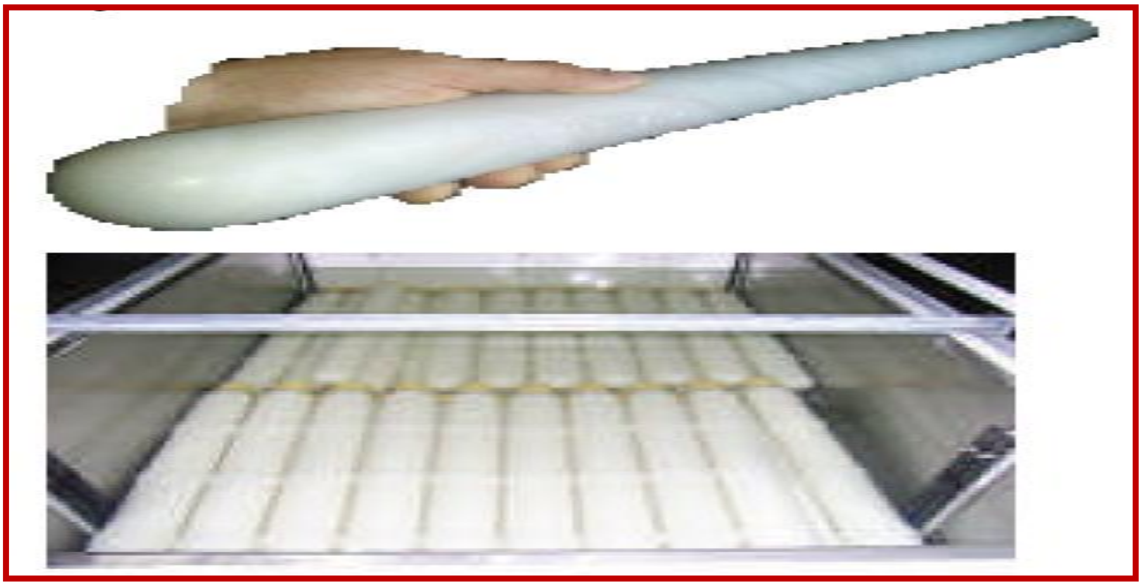

Figure 2:7 Plastic containers stacking on top of each other forming self-assembling heat exchanger.

Source: www.PCMproducts.net

The self-stacking arrangement is suitable for water and air circuits. The stacking of the containers provide gaps that makes ideal flow passage for air or water with large heat exchange surface.

The thermal energy storage (TES) capacity determines the tank size. The installation of the tanks can be under or above ground applications and could be plastic, steel or concrete constructed. Table 2:9 shows the TubeICE container capacity of salt hydrates solutions.

Table 2:9 TubeICE container capacity

PCM Type	Phase Change Temp (oC)	Phase Change Temp (oC)	Weight kg/TubeICE	Weight Lb/TubeICE	TubeICE (kWh/TubeICE)	TES Tank Capacity (kW/m3)	TubeICE (Ton-hr/TubeICE)	TES Tank Capacity (Ton-hr/USG)
S89	89	192	2.7	6.0	0.124	55	0.035	0.053
S83	83	181	2.8	6.2	0.119	52	0.034	0.051
S72	72	162	2.9	6.4	0.113	50	0.032	0.049
S58	58	136	2.7	5.9	0.124	55	0.035	0.053
S50	50	122	2.8	6.2	0.081	36	0.023	0.035
S46	46	115	2.8	6.2	0.148	65	0.042	0.064
S44	44	111	2.8	6.2	0.081	36	0.023	0.035
S34	34	93	3.6	7.9	0.114	50	0.032	0.049
S32	32	90	2.6	5.7	0.135	59	0.038	0.058
S30	30	86	2.4	5.2	0.132	58	0.038	0.057
S27	27	81	2.7	6.0	0.145	64	0.041	0.062
S25	25	77	2.7	6.0	0.143	63	0.041	0.062
S23	23	73	2.7	6.0	0.143	63	0.041	0.062
S21	22	72	2.7	6.0	0.143	63	0.041	0.062
S19	19	66	2.7	5.9	0.109	48	0.031	0.047
S17	17	63	2.7	6.0	0.107	47	0.030	0.046
S15	15	59	2.7	5.9	0.106	47	0.030	0.046
S13	13	55	2.7	5.9	0.105	46	0.030	0.45
S10	10	50	2.6	5.8	0.102	45	0.029	0.044
S8	8	46	2.6	5.8	0.102	45	0.029	0.044
S7	7	45	3.0	6.5	0.099	43	0.028	0.043
E-2	-2.0	28	2.0	4.5	0.114	67	0.032	0.049
E-3	-3.7	25	2.0	4.4	0.115	67	0.033	0.049
E-4	-3.9	25	2.0	4.4	0.104	61	0.029	0.045
E-6	-6.0	21	2.1	4.6	0.106	62	0.030	0.046
E-10	-10.0	14	2.1	4.7	0.113	66	0.032	0.049
E-11	-11.6	11	2.1	4.5	0.114	67	0.032	0.041
E-12	-12.3	10	2.1	4.6	0.096	57	0.027	0.044

Source: (www.PCMproducts.net)

2.3.3.4.5 Freezing and melting performance curves of PCM solution in TubeICE container

Freezing and melting performance curves of PCM solution in a typical TubeICE container against various temperature differences between the surrounding water and PCM solution container is demonstrated in figure 2:8 and figure 2: 9 respectively.

The performance curve demonstrates that the greater the temperature difference between the PCM phase change temperature and the surrounding hot water temperature, less the time used in melting or freezing.

The freezing and melting time apart from being influenced by the temperature difference between the surrounding water and the PCM could also be influenced by the hot water flow rate which is not considered in the performance curves.

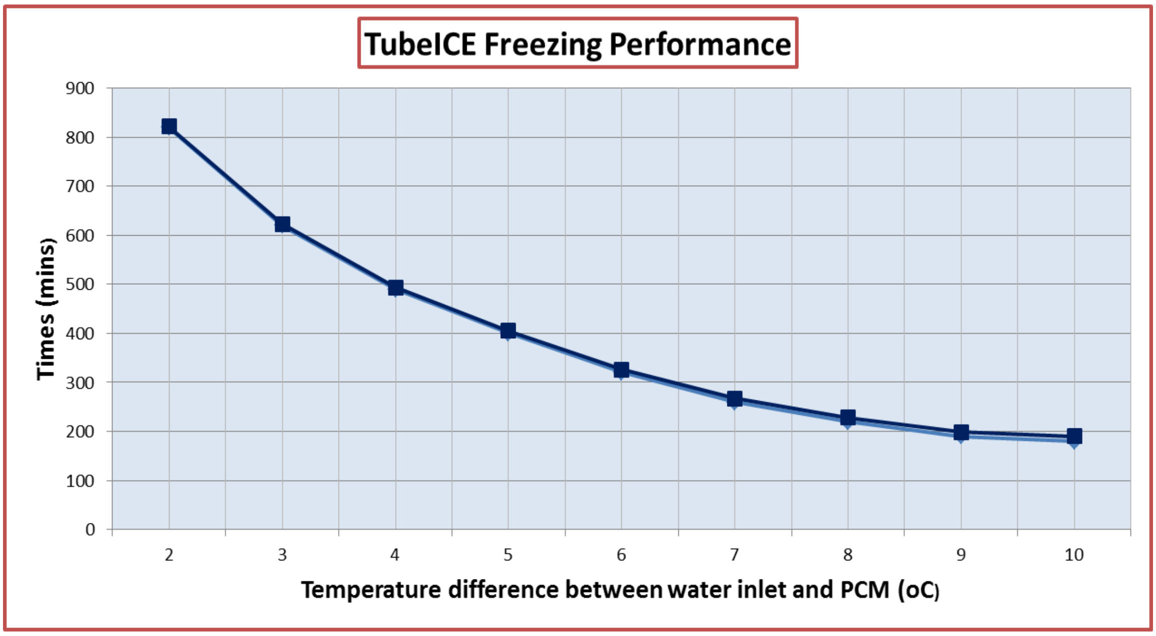

Figure 2:8 TubeICE freezing performance curve
Source: www.PCMproducts.net

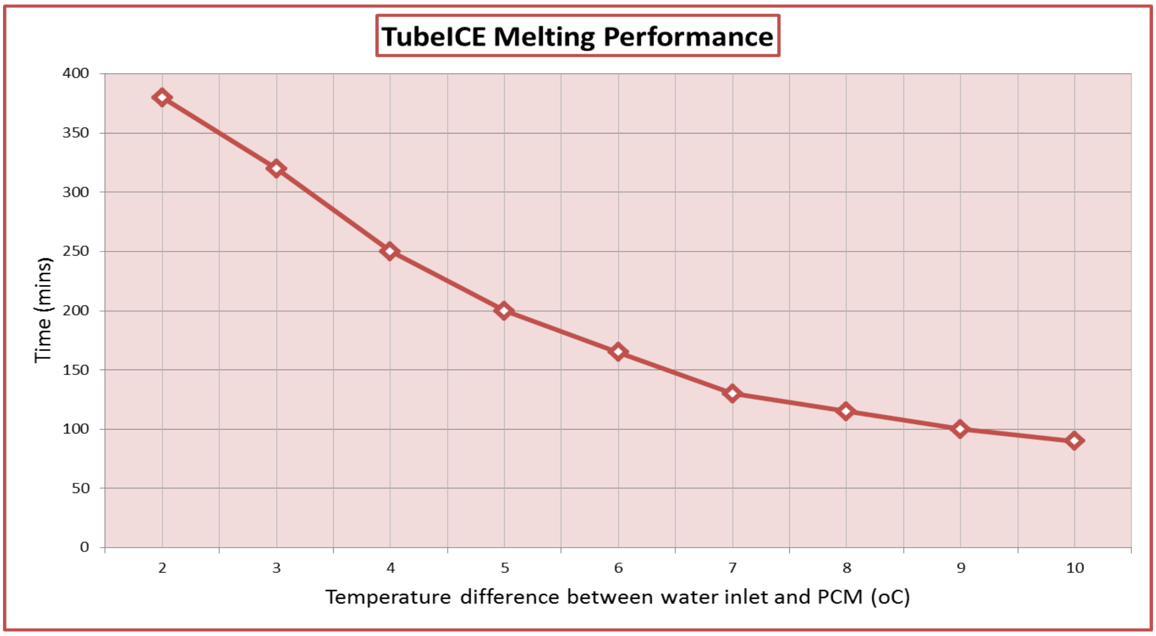

Figure 2:9 TubeICE melting performance curve

Source: www.PCMproducts.net

2.3.3.4.6 Freezing and melting performance curves of PCM solutions in FlatICE container

FlatICE containers are another type of containers designed by PCM Products Ltd used in certain situations in thermal energy storage (TES) solutions to make the project more economical. Figure 2:10 below shows PlusICE PCM filled FlatICE Containers.

FlatICE **FlatICE cross** **FlatICE self-stacking**

Figure 2:10 PlusICE PCM filled FlatICE Container Construction

(Source: www.PCMproducts.net)

FlatICE PCM containers are primarily designed for heavier PCM solutions and the average freezing and melting times are demonstrated below in figure 2:11and figure 2:12 respectively against various temperature differences between the surrounding water and the PCM phase change temperature.

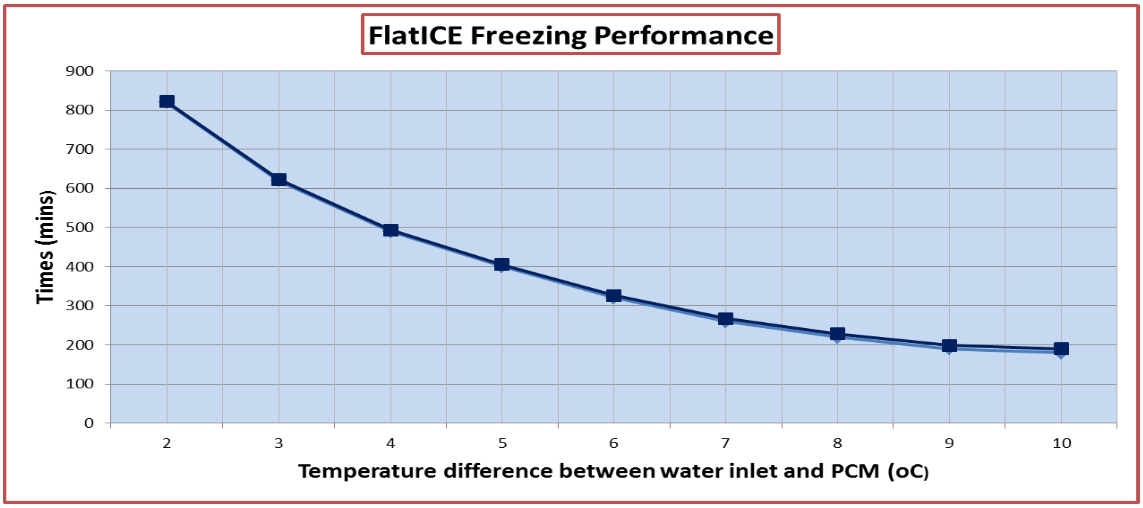

Figure 2:11 FlatICE freezing performance curve

Source: www.PCMproducts.net

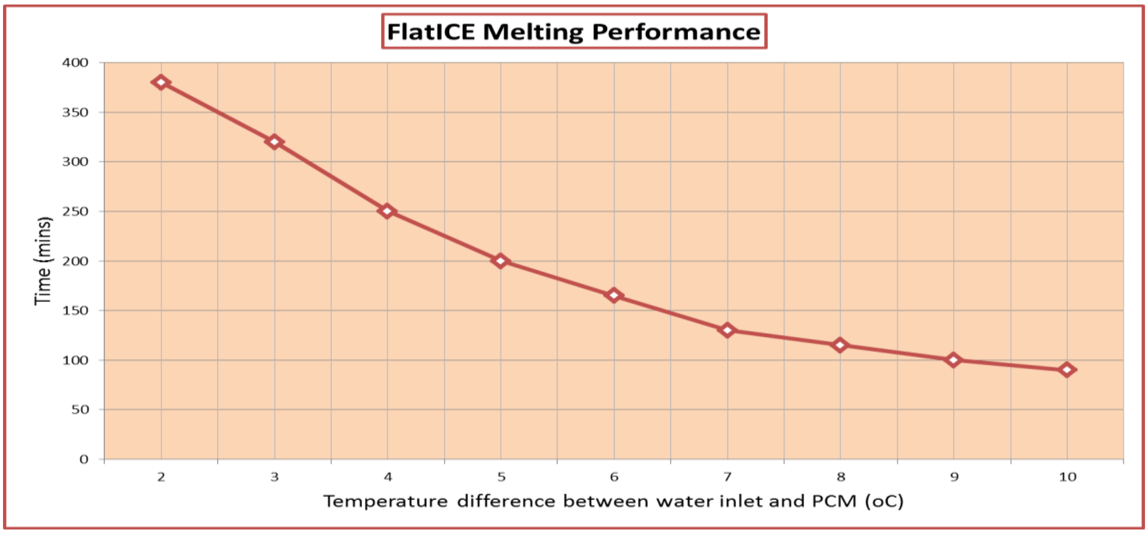

Figure 2:12 FlatICE melting performance curve

Source: www.PCMproducts.net

The PCM solution is permanently sealed inside the flat plastic containers which can be placed in any tank shape and allow water or air to pass around them providing heat exchange capability.

The standard FlatICE containers are designed in 500 mm (19.6”) x 250 mm (10”) x 32mm (1.25”) sizes but has the option to be produced to suit specific applications.

2.3.3.4.7 PCM selection for a particular application

Selecting PCM for an application should consider the following:

- The phase change temperature of the PCM should match the heating or cooling operating temperature.
- The latent heat of fusion should be high to minimise the physical size of the heat storage
- The thermal conductivity should be high to effectively enhance the charging and discharging of the energy storage.

2.3.3.5 Studies of using PCM in thermal energy storage with solar water-heating systems

Solar water heater is gaining popularity [16, 17] since they are relatively cheap and simple to fabricate and maintained.

Prakesh et al. [18] assessed and analysed hot water heater storage system filled with PCM layers at the bottom. The solar radiation during the day will heat the water and the heat is transferred to the PCM below and melts it. The stored energy in the PCM is used to heat cold water when hot water is needed.

Chaurasia et al. [19] studied sensible and latent heat storage systems all heated by solar collector system. The two storage systems were identical. The latent heat storage unit has 17.5 kg paraffin wax (Melting point about 54 ^{o}C) packed in a heat exchanger made of aluminium tubes and the other storage unit was simply hot water storage tank.

Flat plate solar collectors of the same size and absorbing area were used to charge each storage unit during the day. The study established that the latent heat storage system stored more hot water than the sensible heat storage system.

Summary of literature review

Thermal energy storage has become important in energy efficiency and conservation applications and methods in the energy industry.

Latent heat storage using PCM is an effective way of storing thermal energy and this is the research core objective. The research is aimed at reaping the benefits of renewable energy source, storing of waste heat for later use and maximising the use of trapped sun energy in the glasshouse.

The research discussed thermal and chemical energy storage and other storage techniques, their applications and suitability.

The research study focused on phase change materials with high latent heat of fusion that absorb and release significant amount of energy during melting and solidification process. The reason for using PCM in thermal energy storage system is that they have almost constant temperature thermal control for long duration compare to other installations. Large amount of energy could be stored with small mass and volume compare to other storage systems.

The research considered and studied the properties of PCMs which are significant in melting / freezing process for an application.

Phase change materials are classified into three main categories namely organic, inorganic and eutectic. These materials as discussed above have been identified as PCMs used for thermal energy storage systems but most of these materials do not satisfy the criteria required for an adequate storage media. However, they can be modified or improved to meet the energy storage criteria by efficient design.

PCM Products Ltd in England have tried and tested several PCMs for heating and cooling applications. The PCMs selected for this research are their products tested and used for heating applications.

The characteristics, behaviour and properties of these PCMs such as melting and freezing rate profiles are clearly demonstrated for various thermal energy storage systems. It shows the rate of melting and freezing of the material solution based on temperature difference between the PCM and hot water surrounding the PCM.

The greater the temperature difference between the surrounding hot water and the PCM solution, less the time it takes to melt or freeze the solution. The freezing of the PCM solution takes much longer time than melting and this was considered during the design of the heating system to ensure that enough energy can be transferred to heat the space to maintain the required temperature.

The research studied several solar water heating systems using PCM energy storage applications to heat glasshouses. The different designs were demonstration of the effectiveness of PCM thermal storage systems.

What was not demonstrated in all the studied examples were that the heat trapped in the space were not utilised and vented to the atmosphere and this trapped energy which go wasted is one of the focus areas of the research to maximise its use.

Prakesh et at. [18] assessed and analysed water heater storage system containing filled layer of PCM which was charged during sunshine periods but nothing was said about the system efficiency and the type of PCM used.

Chaurasia et al. [19] comparatively studied solar heated hot water using identical storage units. One was filled with 17kg paraffin PCM and the other storage tank without PCM. It was established that the PCM filled storage tank could store more hot water than the tank that holds simply hot water which is a demonstration of the effectiveness of PCM thermal storage system.

The literature review established that lot of studies have been conducted on PCM thermal storage systems and mostly charged using solar collectors and have proved successful in many ways. What have not been considered in the solar energy world is charging the PCM with environmental heat gains such as solar energy trapped in a glasshouse.

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3 Research Methodology

3.1 Introduction

The area of the research was new with little knowledge and skills and therefore had to study the design of the glasshouse, the nursery plants requirements and heating load requirements.

Thermal energy storage using phase change material was also challenging. The first step to get started was to have an in-depth knowledge through literature review. Below was the plan outline of the research:

- Study plants behaviour and characteristics to assist effective design of the glasshouse.
- Investigate the design requirements and parameters of glasshouse and study the greenhouse effects of a glasshouse
- Designed the glasshouse and determined the predicted space conditioning thermal energy requirement and zone temperature profiles per annum using IES software tool
- Detail study of solar thermal collector hot water system, the efficiencies at various ambient temperatures and inlet hot water temperatures to the solar collector system.
- Selection of the appropriate solar collectors that will suit the application
- Assessed periods where solar thermal energy can be passively stored and effectively used in heating the glasshouse space
- Design the heating system to serve all the zones including control mechanism
- Economic appraisal of the project

3.2 Background of the methodology

The research was aimed at using zero carbon emission heating system to heat the glasshouse by storing enough thermal solar energy through phase change material storage systems.

The research required knowledge and understanding of greenhouse design, the nursery plants requirements, behaviours and their characteristics. The study of solar thermal energy and how this source of energy could be easily and effectively source and stored was necessary.

Two main systems (Active and passive systems) were considered to source the thermal energy. The active system will produce hot water for heating through solar collectors. Two passive systems were considered.

Black painted pipes which will need further study could be used to absorb solar radiation to pre-heat or warm the heating circulating hot water prior to entering the solar collectors of the active solar system or divert it through the PCM thermal energy storage system tanks depending on the hot water exiting temperature from the black painted pipes. The black painted pipes were among the initial thoughts but have been recommended for further studies.

The other passive system is to use PCM filled heating pipes to absorb the heat trapped inside the glasshouse due to greenhouse effect and discharge it to condition the space when the space temperature falls below the PCM phase change temperature.

3.2.1 Glasshouse building

A greenhouse (Also called a glasshouse) [2] is a building in which plants are grown. Greenhouse structures vary in sizes depending on the requirements of the building. Traditionally, greenhouse has always been built with glass even though there are other options. This has been the case because of its beauty and permanence.
The overall U-value of a greenhouse with thin thermal curtain is 0.3 to 0.7 [2].

The Earth's atmosphere keeps the planet warm through transformation of light waves. Glasshouse traps heat energy in a similar manner and warm the space through convection heat transfer.

Solar radiation passes through the glass of the glasshouse, absorbed by the ground, plants and in some cases by the plants pot soil. The solar radiation is converted to heat energy which cannot escape through the glass easily.

The air inside the greenhouse is kept warm through convection heat transfer and keeps the greenhouse warm. The air near the ground gets warm and rises. The rising warm air mixes with the cooler air near the ceiling of the greenhouse as it falls.

The convection process allows the trapped air inside the glasshouse to absorb more and more heat each time it rises and mixes with the cooler air.

The convection process in the glasshouse is similar to convection cells also known as a Bénard cell [3] (Type of natural convection). The Bénard cell is a fluid flow pattern in many convection systems (Figure 3:1 below).

It explains that a body of fluid rising will lose heat when a cold surface is encountered. The rising fluid becomes denser than the fluid underneath as it encounters cold surface and loses heat.

During this process, global circulation cannot establish itself if the heating pattern is sufficiently uniform and the fluid breaks down into small convective cells forming the so called Rayleigh-Benard convection pattern. The fluid forms a closed circulation loop within each cell, rising at the center of the cell sinking at the edges. The cycle repeats itself as demonstrated in figure 3:1 below.

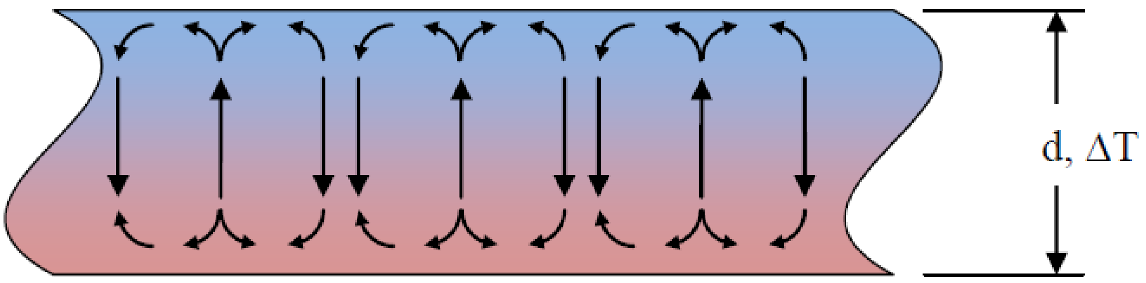

Figure 3:1 Bernard convection cells

Glasshouses are built of glasses as well as polycarbonate plastics. Glass and clear plastic functions are similar and both allowing light to pass through them.

The choice to use glass or plastic basically depends on the type of structure and the owner's desire. Plastics are generally lighter in weight and less expensive but cannot last longer compare to glass due to age and weather conditions

Most glasshouses are controlled by climate control logic boards which controls ventilation, shading, cooling (Natural ventilation) and heating systems in the glasshouse [4]. The glasshouse under research is also controlled by climate control logic board.

Greenhouse benefits from "greenhouse effect," as explained above to warm the space during day time, thus it captures the solar radiation from the sun and keeps the heat trapped inside the house [4].

This is most evident when the inside temperatures and relative humidity of the Kew Gardens nursery glasshouse under research were measured over three weeks period in different zones. The inside measured temperatures compared to the outside temperatures within the same measurement period were higher than the outside temperature.

3.2.2 Solar thermal heating of greenhouse

When a greenhouse receives solar thermal energy, it warms the structures, ground and plants inside the glasshouse and radiates some of the thermal energy which is also trapped inside the glasshouse.

The amount of energy that is radiated compare to convective heat transfer is minor. Thus, the primary heating mechanism of a greenhouse is by convection.

An auto-vent cooling system is used in cooling Kew Gardens Nursery glasshouse within a range of temperatures.

The basic principle of operation is that as the greenhouse heats above the space set point temperature, the vents open to allow some hot air out and draws cooler air in from outside.

The current Kew Gardens heating pipes are normal plain pipes with no fins to increase the surface heat transfer area. They are laid under the benches and suspended overhead.

3.2.2.1 Glasshouse heating using fossil fuels

Glasshouse heating systems almost use electrical power and fossil fuel to generate thermal energy to condition the space and these types of systems generate carbon dioxide emissions.

Figure 3:2 below demonstrates electric-charged air heater (Katsoulas & Kittas, 2009) that transfers heat to the space through force convection. Small proportion of the total heat is by radiation from the hot surface of the air heaters.

These types of systems achieve high efficiencies and easily controlled. They are quick to response to the load.

Figure 3:2 Electric-charged air heater (Katsoulas & Kittas, 2009)

Figure 3:3 also demonstrates hot water charged hot air heating system. This type of system is used mostly in greenhouses but the initial installation cost is higher compare to electric charge heating systems.

Figure 3:3 Hot water-charged air heater (Katsoulas & Kittas, 2009)

Figure 3:4 below also shows a typical hot water heating system. Hot water heating pipes are used in heating the space. The heating pipes are installed under and overhead the beds similar to Kew Gardens heating system.

Figure 3:4 Hot water heating system with pipes installed under the plants beds (Katsoulas & Kittas, 2009)

3.2.3 The fundamental knowledge of green plants

3.2.3.1 An introduction to photosynthesis

Photosynthesis is the process that plants produce their food [5]. In the process, they use the sun energy to produce sugar. When plants take in carbon dioxide and water, they produce sugar.

Figure 3:5 below shows the input and output process. The chemical equation below explains the process: Carbon dioxide + water produce → glucose + oxygen.
The photosynthesis analysis was to ascertain if the heat generated through this process is useful in heating the glasshouse.

Figure 3:5 Photosynthesis process

Photosynthesis and respiration occur in green plants. In bright light photosynthesis dominates and respiration dominates at night or in the absence of bright light. The heat energy produce during respiration is latent which is not useful in determining the glasshouse heating load. However, the design ensures that enough light is received by the glasshouse.

3.2.3.2 Estimation of heat gains in the glasshouse by the plants

The estimation of heat produced by plants is difficult but effort was being made at least to make certain assumptions that will assist to calculate the minimum heat gains in the greenhouse by plants.

The aerobic respiration process of plant may be expressed as a chemical equation as
$6CO_2 + 6H_2O$ + light energy $\rightarrow C6H_{12}O_2 + 6O_2$ (About 2800 kJ per mole of glucose)
Photosynthesis and respiration are the two main processes for green plants well-being and they are influenced by the environmental temperature, relative humidity, sunlight, bright light, night or absence of light and CO_2 concentration within the area.

The energy absorbed during the photosynthesis process is released to the environment and not all is heat and this heat is latent.

During respiration the estimated energy released per molecule of glucose is 2800 kJ [5], which is latent and not sensible, therefore not considered in estimating the space conditioning energy demand. Figure 3:6 shows a section of the tropical nursery glasshouse under research.

Figure 3:6 A section of the tropical nursery glasshouse in Kew Gardens

3.3 Building design, modelling and simulation

The IES software tool was used in the designing, modelling and simulation of the nursery glasshouse. To further explain how the building was designed, modelled and simulated, the software capabilities will be first discussed.

3.3.1 Introduction of the IES software tool

The IES programme has a suite of integrated applications linked by a common user interface (CUI) and integrated Data Model (IDM).

It has modules such as Apachesim for thermal simulation, Radiance for lighting simulation, Suncast for solar shading analysis and ModelIT for input 3D geometry to describe the model.

It does not effectively account for the seasonal and plant part load efficiencies well, as the outdoor-indoor temperature varies. Even though it gives the designer the option to select part load efficiency of the plant, the usefulness of this application is minimal in estimating plant energy consumption.

3.3.1.1 Weather editor

The APlocate model in the IES software tool is the editor of the weather and site location for CIBSE heat loss and gain method (Apache calc), ASHRAE heat balance method (Apache loads) and Apache simulation (Apachesim).

When the Aplocate is called from the virtual environment design software for a particular site location, default values of weather data will be set up but it gives the designer the option to edit the values to suit the project with the exception of the latitude, longitude and altitude. For example, the heating load outdoor winter design temperature can be altered to suit the particular application.

The cooling load maximum outside temperature can also be adjusted to suit the project application. It assists to obtain location and site data, weather data and simulation weather data for the design.

The site data contains values for the latitude, longitude and height above sea level. These are standard values drawn from tables published by CIBSE and ASHRAE

CIBSE and ASHRAE design default parameters for London / Heathrow were used for a set of load calculations and could be displayed to show the minimum, maximum dry-bulb and wet-bulb temperatures for each month throughout the year for that particular site location.

It has the function to set time such as hours ahead of Greenwich Meridian Time (GMT). The IES software tool has default design profiles that can be used but it gives the option to the design engineer to create convenient profile to suit the project application which is valuable depending upon the designer experience.

It gives the option to adjust the daylight saving time of the months for the location. The design ground reflectance, type of terrain and wind exposure are possible to set them up under this model for load calculations.

CIBSE Guide A table A2.31 has typical values of reflectance and absorbance listed for the sun hitting the earth surface and is demonstrated in table 3:1 below. The values are used in the APcalc and APachesim to calculate ground reflectance radiation on the building facades.

The general situation has typical values such as 0.2 for temperate and humid tropical localities and 0.5 for arid tropical localities.

Table 3:1 Ground reflectance typical values

Ground reflectance	Typical values are
Snow (fresh)	0.80 – 0.90
Snow (old)	0.45 – 0.70
Water (typical)	0.10 – 0.20
Ice	0.70
Grass	0.25
Crops and woodland (typical)	0.20
Concrete (average condition)	0.30
Brick	0.20 - 0.40
Asphalt	0.15

3.3.1.2 Terrain type

The type of terrain defines how wind speed varies with height. ASHRAE 2001 wind speed profiles define terrain wind speed with the varying height. The natural ventilation air exchange rates is affected by this data when Macroflo model in the IES programme is run in conjunction with APachesim and the external convection loss in Apachesim. Suburb was chosen as type of terrain for this building.

3.3.1.3 Design weather data

There is an assumption that the frequency level of a specific temperature will repeat in the future over a suitable time period and design temperatures are based on this assumption.

The temperatures at 1 and 99 % level may vary in order of 1 to 2 ^{o}C and this is evident by Metrological data at many locations from the previous fifteen year period for any 15 year period [6]. This is the percentile of reoccurrence.

In the UK the weather data records covers twenty year period from the previous period for example, 1982-2002 [7].

The percentage of annual design exceedence such as 0.4%, 1.0% and 2% was carefully selected to suit the building and the location as this has implication on plant size performance.

3.3.1.4 Weather Data with climate scenarios

Climatic data of various sets are available which contain hourly values for a complete year of relevant parameters [8]. Detailed descriptions of the types of data are available in the Charted Institution of Building Services Engineers (CIBSE) Guide J.

The Charted Institution of Building Services Engineers (CIBSE) has approved the following sets of data: CIBSE Example Weather Years (EWYs): They are complete years selected to be representative of long-term averages.

The purpose was for energy consumption calculations suitable for design or overheating assessment. CIBSE Test Reference Years (TRYs) now supersede EWYs. CIBSE Test Reference Years (TRYs): They have been selected as good statistical representation of the past.

London Heathrow location reference number with the World Metrological Organisation (WMO) is 37720 with latitude 51.48^{o}N, Longitude 0.45^{o}W and Altitude 25m.

Temperatures of near extreme values of dry-bulb are used as outside design temperatures and for this building an outside design temperature selected was -3.4^{o}C based on percentage of annual design exceedence of 0.4%.

In the IES simulation programme, the Apache simulation (Apachesim) file selected for the location has a weather file that cannot be edited.

The CIBSE Try Reference Years (TRYS) climate data file which was not built in the programme has now been built in IES 2012 version that can be used for Part L simulation trial of the Building Regulations compliance.

The building location (Kew Gardens) is closer to London / Heathrow therefore London / Heathrow weather data was used for the glasshouse design as it is more accurate compare to others.

3.3.1.5 Pre-conditioning days used in IES programme to improve calculation results

The building response factor determines whether the structure has high or low thermal mass. Response factor (fr) is defined as $\Sigma(AY) + CV / \Sigma(AU) + CV$ where $\Sigma(AY)$ is the sum of the products surface areas and their corresponding thermal admittances ($W.K^{-1}$), $\Sigma(AU)$ is the sum of the products of surface areas and corresponding thermal transmittance over surfaces through which heat flow occurs ($W.K^{-1}$) and CV is the ventilation conductance ($W.K^{-1}$) [8].

Structures with high thermal mass response factor greater than 4 are referred to as slow response building. Alternatively, structures with response factor less than or equal to 4 is classified as fast response building. Glasshouse for example, is a fast response building.

Weather in reality varies from day to day. On a selected design hot summer or cold winter day, the building will still have some residual heat stored from the preceding days which significantly can reduce predicted plant size.

The glasshouse structure is glass and has minimum storage facility if any.
The IES programme recommends that precondition period of 10 days are suitable for most light weight buildings and 30 days for heavy weight buildings. The building under consideration is a light weight and therefore selected 10 days preconditioning period for the simulation.

3.3.1.6 Problems with IES design tool

The problems associated with the tool is that it tries to solve every design problem by providing a whole lot of default design profiles, design parameters such as infiltration, temperature set point and others.

In trying to facilitate the ease of using the programme creates unreliable results or output in certain cases for inexperience users contributing to building operational inefficiencies.

The integrated environmental solutions (IES) tool was used to design the glasshouse to determine zone hourly heating demand, solar gains and temperature profiles of each zone. The space conditioning sensible energy for each zone was also calculated using the IES tool.

The determination of the hourly heating demand and spaces conditioning sensible energy are necessary to size the heat generation plant and energy requirements of the glasshouse throughout the year.

3.3.2 Building design and construction details

The building design and construction materials were based on1995 Part L building regulations (The year the glasshouse was constructed).

Table 3:2 below describes the construction details of the greenhouse building which was used in determining the space heating requirements whilst table 3:3 shows the design parameters for modelling.

The main frames of the building are steel beams and the external ground layer is made up of cast concrete, insulation, brickwork and clay. The external and internal wall partitions are 10mm single glazed material. The U-values of the building fabrics are demonstrated in table 3:2. The building gross internal floor area (GIA) and treated floor areas are 5607 and 4634 m^2 respectively.

Table 3:2 Building construction details

Item	Description	CIBSE U-values
Ground floor	Standard floor construction (insulated to 1995 building regulations)	0.4152
External wall	10mm single glass pane in unsealed openable frame with thin thermal curtain	0.35
Internal partition	10mm single glass pane in a sealed frame	6.2164
Door	10mm single glass pane in a sealed frame	6.2164
Roof	10mm single glass pane in a sealed frame	6.2164
Ceiling	Clear element ceiling screen	5.4723
Building floor area (m^2)	Gross internal area (GIA)	5607
Building floor area (m^2)	Treated internal floor area	4634

Table 3:3 Building heating system design parameters for modelling

Item	Value
External design condition (heating) ^oC	-3.4
External design condition (cooling) ^oC	31.40
Internal design condition (Degree Celsius) ^oC	Zone specific
Infiltration rate (m^3/m^2.h at 50pa)	0.6 (leaky building)
Heat given off by occupants (W/person)	150
Artificial lighting gains (W/m^2)	N/A

Table 3:4 Simulation weather data and site details

Design weather data source and application	Try reference year (TRY application)
Simulation weather data file	Kew Gardens FWT
Location and site data	London/Heathrow – UK
Design weather data source and application	Try reference year (TRY application)
Simulation weather data file	Kew Gardens FWT
Location:	Kew Garden, London
Latitude	51° 28’ 58.23” North
Longitude:	0° 17’ 45.97” West
Elevation	10 meters
Orientation	265° from North (clockwise)

3.3.3 Building modelling

Figure 3:7 and 3:8 are the glasshouse floor plan and 3D model respectively used for the simulation to determine the heating requirements and space conditioning energy demand throughout the design year. The site orientation angle is 265^{o} clockwise from north. The floor plan in figure 3:7 below is not dimensioned but detail size of each zone was measured and used when designing the building model.

Due to lack of information supply from Kew Gardens most of the data required for modelling were measured and therefore believe the accuracy of the model results.
The building fabric details such as glazing type for the external and internal partition walls were supplied by the Kew Gardens and the external ground details was based on Part L of the 1995 building regulations when the glasshouse was constructed.

Zone space conditions such as temperature, humidity and system type were populated based on each zone requirements. The heat gains in each zone such as people, lighting, equipment, miscellaneous and air exchanges were considered.

Table 3:5 below demonstrates zone requirements. The glasshouse under consideration constitutes twenty-one zones. Each zone requirements such as space minimum and maximum set point temperatures, humidity tolerance (Humidity operational range i.e. 65-80%) and maximum temperature limit that the vents open were populated into the IES software to calculate the glasshouse heating requirement and energy demand.

Table 3:5 Zones requirement

Zone description	Description of zone plants	Zone set point temperatures		Relative Humidity	Vent opening temperatures
		Min	Max		
Zone1:Temperate Production	Canary Island Plants, Bulbs, Passiflora	8°C	10°C	Ambient	15°C
Zone 2:Temperate Propagation		16°C	18°C	60-70%	24°C
Zone 3: Aquatic Collection	Victoria, Nymphaea	18°C	22°C	Ambient	27°C
Zone 4:Tropical Propagation		18°C	23°C	65 -80%	25°C
Zone 5: Temperate Carnivorous Plants	Sarracenia, Drosera, Pinguicula, Utricularia	5°C	8°C	Ambient	12°C
Zone 6: Cool /Arid	Agavaceae, Aloaceae, Succulent Propagation, Bromeliaceae, Tillandsia	10°C	12°C	: ambient	14°C
Zone 7: Cool / Arid Cacti	Cactaceae, Aloaceae, Crassulaceae, Aizoaceae, Geraniaceae	10°C	10°C	Ambient	12°C
Zone 8: Araceae Collection	Araceae, Epiphytic Cactaceae Amorphophallus	20°C	22°C	65 -80%	26°C
Zone 9: Moist Tropical	Passiflora, RhizophoraTropical Climbers, woody stock plants	20°C	22°C	65 -80%	27°C
Zone 10: Humid Tropical	Nepenthes, Utricularia, Hoya	18°C	22°C	78-100%	25°C
Zone 11: Weaning / Display Requests		18°C	21°C	65-75%	26°C
Zone 12: Tropical ferns	Aspleniaceae, Marattiaceae, Polyodiaceae, elaginellaceae	18°C	20°C	65-80%	25°C

Zone description	Description of zone plants	Zone set point temperatures		Relative Humidity	Vent opening temperatures
		Min	Max		
Zone13: Tropical Collections	Peperomia, Impatiens Marantaceae, Begoniaceae	18°C	22°C	60-90%	27°C
Zone 14: Warm/Arid	Euphorbiaceae, Dracanaceae, Asclepiadaceae, Cactaceae, conservation collections	14°C	16°C	Ambient	18°C
Zone 15: Temperate ferns	Adiantaceae, Dicksoniaceae, Pteridaceae	11°C	17°C	65 -80%	18°C
Zone 16: Orchidaceae - Cool	Cymbidium, Masdevallia, Dracula, Pleione Pleurothallis	12°C	15°C	60-75%	16°C
Zone 17: Orchidaceae - Cool Intermediate	Paphiopedilum, Phragmipedium, Coelogyne, Dendrobium, Gongora, , Stanhopea, Epidendrum, Maxillaria	16°C	21°C	60-70%	25°C
Zone 18: Orchidaceae - Warm Intermediate	Cattleya, Laelia, Encyclia, Polystachya, Miltonia, Lycaste, Zygopetalum, Dendrobium, Oncidium, Aerangis	18°C	23°C	60-75%	25°C
Zone 19: Moist Tropical	Bromeliaceae	18°C	18°C	60 -80%	24°C
Zone 20: Orchidaceae - Warm Tropical	Vanda, Angraecum, Dendrobium, Phalaenopis, Grammatophyllum, Vanilla, Grammangis	20°C	25°C	60-75%	28°C
Zone 21: Orchidaceae - Hot and Dry	Catasetum, Mormodes, Cycnoches, Thunias	18°C	23°C	60-75%	27°C

As described above, the Nursery Greenhouse consists of 21 separate zones or rooms and an unheated corridor.

Each one of the rooms host different species of plantation and thus having different heating requirements and this can be seen in figure 3:7 below. Figure 3:8 is the completed nursery building model generated using IES software.

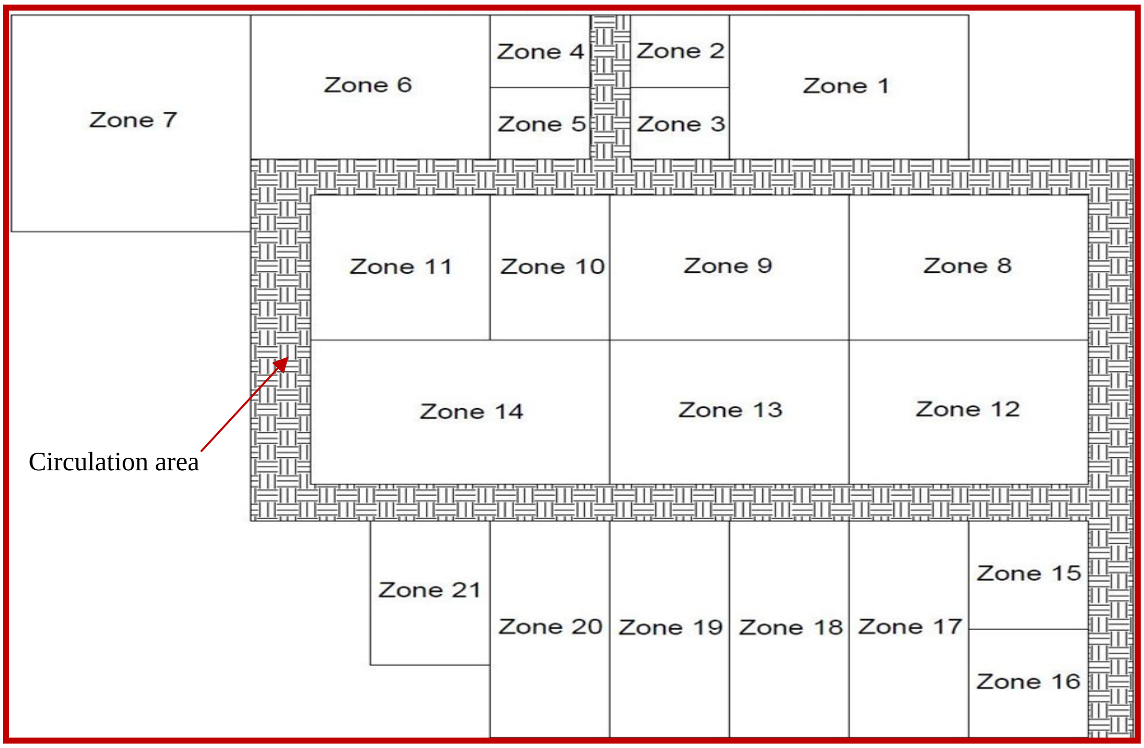

Figure 3:7 Glasshouse floor plan showing zone positions

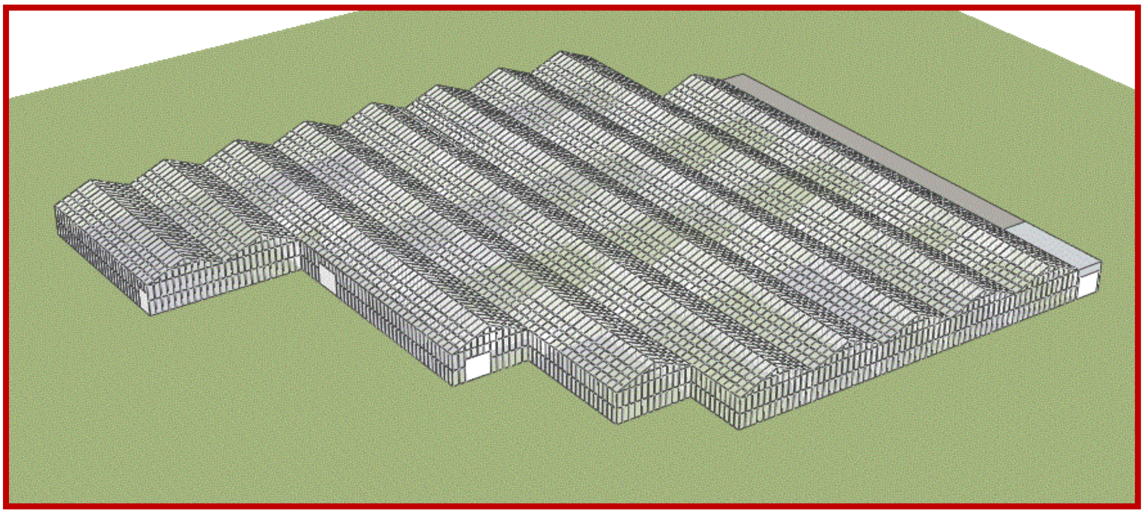

Figure 3:8 Glasshouse design model using IES software

3.3.4 Building simulation

The building simulation results could be effective and accurate provided all the design requirements are met and this could be achieved when the appropriate design parameters are populated into the software before the simulation is run.

3.3.4.1 External design conditions

The design required parameters will depend on the application. Sizing heat emitters for example, will require dry-bulb temperature data and in a heating system some or all of the following may be necessary to determine the heating plant capacity:

- Winter dry and wet bulb temperatures
- Summer dry and wet bulb temperatures
- Solar irradiation
- Long wave radiation loss
- Sol-air temperatures
- Wind speed and direction

All the above requirements are already built in the Integrated Environmental Solution (IES) software programme used in the calculation for a particular location but the design engineer has to enter the design temperatures.

3.3.4.2 Internal heat gains

Sensible and latent are internal heat gains emitted within the internal space from any source. This may increase the internal temperature and humidity. The sources include the following:

- Bodies (human)
- Lighting
- Equipment/machinery
- Electric motors

Designers normally choose to estimate the rate of internal heat gain either by knowing the type of building sufficiently or typical benchmark values [8]. Estimation or prediction of internal heat gains is based on current practice or benchmarks but benchmark values are only available for common buildings.

Internal heat gains are difficult to predict especially from glasshouse as heat stored by plants, ground and soil differ and their heat emission are not concurrent and do not produce heat at constant rate.

The rate of heat emission (respiration) from plants for example, depends on the stored energy that is influenced by so many factors such as sun energy, glucose and others. The application of diversity factor in estimating the heat gain adds more problems to the heat gain prediction. The heat emission from the plants is latent heat (respiration) and therefore not considered as heat gains in calculating the space conditioning load.

3.3.4.3 Internal design conditions

Design internal temperatures are assumed for calculating cooling and heating loads of a building. The assumed temperature may be a comfort temperature such as operative temperature, resultant temperature, air temperature or radiant temperature but glasshouse specifically requires air temperature.

Air speed is also an important function to achieve desire temperature. The IES programme has options of design internal temperatures to be selected by the design engineer to achieve the level of desired temperature.

The air speed is often assumed to lie within a range where it has little effect on the comfort index. Each zone in the glasshouse has different temperature so the heating system is designed to meet each zone’s temperature requirement.

3.3.4.4 Infiltration and ventilation

Intentional or unintentional outdoor air that flows through a building is important for two reasons. The intentional air normally refers to as ventilation often used to dilute indoor air contaminants. This is heated or cooled and the energy associated is significant as part of the space conditioning heating load.

Minimum air demand rates should be known to assure that contaminants are properly controlled to the required level but this building does not require ventilation as it is a glasshouse.

Infiltration which is uncontrolled airflow through cracks, interstices and other openings are unintentional. Natural, infiltration and exfiltration ventilation airflows are caused by the pressure differences due to wind, indoor -outdoor temperature difference and appliance operation. Part of space conditioning load is due to infiltration which needs to be calculated.

3.3.4.5 Empirical rates of infiltration

CIBSE Guide A chapter 4 table 4.21 gives empirical values for air infiltration in naturally ventilated buildings in winter. Empirical values [7] for air infiltration rate due to infiltration for rooms in buildings on normally-exposed sites in winter-dwellings partial exposure is illustrated in table 3:6 below.

Table 3:6 [7] Empirical values for air infiltration rate

Air permeability/ (m^3/m^{-2}.h at 50pa)	1 storey 10m x 8m x 2.75m Height to roof 5.5m		2 storey 10 x 8 x 2.75m* Height to roof 8.0m		Apartments 5 (storey 1 – 5) (10 x 8 x 2.75) (Floor spacing 3.0m)		Apartments (storeys 6 – 10) (10 x 8 x 2.75m)* Floor spacing 3.0m)	
 20.0 (leaky) 10.0 Part L (2002) 7.0 Part L (2005) 5.0 3.0	Peak 1.60 0.8 0.55 0.40 0.25	Ave 1.15 0.60 0.40 0.30 0.20	Peak 1.50 0.75 0.55 0.40 0.25	Ave 1.0 0.50 0.35 0.25 0.15	Peak 1.95 1.00 0.7 0.50 0.30	Ave 1.40 0.70 0.50 0.35 0.25	Peak 2.25 1.15 0.80 0.70 0.35	Ave 1.60 0.80 0.55 0.40 0.25
Air change rate		11.80		8.15		11.80		11.80

Note: Tabulated values should be adjusted for local conditions of exposure and this was done to improve simulation efficiency.

The building under consideration is one storey glasshouse and the infiltration rate selected for estimating space heating requirement was at air permeability of $20m^3 / m^2$.h at 50pa (Leaky building) with peak rate of 1.6 and 1.15 on average.

The infiltration default value used in the IES programme is 0.25 (BRE design default) and most design engineers use this value which eventually makes the predicted heating energy consumption less than the actual consumption.

3.3.4.6 Natural ventilation

CIBSE Guide A table 5.2 gives effective mean ventilation rates for openable windows as shown in Table 3:7 below. The vents opening of the glasshouse is similar to the window opening in a building.

Table 3:7 below demonstrates ventilation loss (W / m^2K) due to opening of windows which means that opening of the vents in the glasshouse can significantly affect the heating and energy demand of the glasshouse. As a result the vent opening in the glasshouse will be modified to regulate the vent opening using motorised motors to drive the vents.

Table 3:7 [7] Effective mean ventilation rates for openable windows

Location openable windows	Usage of windows		Effective mean ventilation rate	
One side of building only	Day	Night	Air changes per hour/h^{-1}	Ventilation loss w/m^2/k
	Closed	Closed	**1**	0.3
	Open	Closed	3	1.0
	Open	Open	10	3.3
More than one side of building	Closed	Closed	2	0.6
	Open	Closed	10	3.3
	Open	Open	30	10.0

Table 3:7 above shows ventilation energy loss rates for openable windows which is quite significant and this is the more reason why the current method of opening the vents should be modified to keep heat loss from the glasshouse to the minimum when the space temperature exceeds the required level especially in winter.

3.3.4.7 Heat given off by occupants

CIBSE Guide A table 6.2 (Appendix A2) and table 3 of AHSRAE Fundamentals Handbook 1993 page 16.8 states the rate of heat emission for a mixture of males and females as 115W total with 70W sensible and 45W latent. These typical values are for offices, hotels and apartments.

The rate of heat emission given off by a gardener was assumed to be 150W due to the nature of their work but the number of gardeners and heat given off compare to the size of the greenhouse was so little and therefore ignored. ASHRAE and CIBSE simplified heating calculation method ignores internal heat gains but the method used for this project accounted for the heat gains to size the heat generation system correctly.

Because internal heat gain is difficult to predict care was taken to use the minimum heat gains where appropriate by applying low diversity factors. Increase in internal heat gains above the minimum estimation during building operation will be an advantage to the heating system. Thus the actual heating energy requirement will not be more than predicted energy consumption.

3.3.4.8 Lighting heat gains

In general, lighting energy used by lamp is released as heat. The emitted energy is by means of conduction, convection and radiation [7].
The luminaire itself absorbs some of the heat emitted by the lamp when switched on. Some of the heat is transmitted to the surface of the building structure and this depends on the mounting method.

The radiant visible and invisible energy emitted from the lamp will generate heat to the space only after it has been absorbed by room surfaces. The absorbed energy has storage effect which will result in a time lag before the heat is delivered to the room space.

The following must be known in determining the internal gains due to artificial lighting:

- The total electrical input
- Fraction of heat emitted to the space
- Radiant, convection and conductive components

The distribution of heat output and total electrical input will vary with manufacturer. The power dissipated by the control is added to that of the lamp. The control gear power loss of electronic ballast and convectional ballast are likely to be 10% and 20% respectively.

The current survey of newer buildings found the lighting loads to be in the range 8 -18W / m^2 for maintained illuminance level of 350-500 lux [7].

The dissipation energy from fluorescent lamp is thirty per cent (30%) radiant and seventy per cent (70%) convective [7]. The nursery has just some few fluorescent lights installed at the corridors and in the zones and the heat given off by these lamps is very minimal and therefore ignored in the heating load calculation.

3.3.4.9 Equipment heat gains

It has been established that name plates power are overstated compare to actual operating power resulting to less heat gains. Hosni et al. [9] found that the ratio of heat gain to name plate power ranged from 25% to 50% and advised that the most accurate ratio is 25%.

3.3.5 Temperature, relative humidity and vent opening control profiles using IES simulation design software tool

The IES software tool gives the designer the option to set up temperature, relative humidity and vent opening control profiles for heating, cooling, ventilation and humidification.

The heating system of the glasshouse was designed to maintain each zone environmental requirements such as temperature and relative humidity. Daily temperature, humidification and vent opening temperatures were set up during the heating system design using the IES design tool simulation formula method which is demonstrated in figures below.

Two main routes or methods are available from the IES software tool to set up the control profiles of the heating system namely: graphical and formula methods. The formula method was used as this was appropriate to meet the glasshouse design requirements.

Figure 3:9 to figure 3:12 demonstrate temperature control profiles of zones 16, 14, 11, 12, 17and 19 with respective space temperature set points of 13, 15, 19 and 18 °C. All the zones temperature control profiles were established and they all look similar and for the purpose of this report demonstrated six zones temperature control profiles.

Figure 3:13 to figure 3:16 demonstrate vent opening temperatures and the minimum humidity level requirement in zones 2, 4, 8 and 10. All zones vent opening temperature and minimum relative humidity control profiles were developed but for the purpose of this report not all the zones profiles are shown as they all look similar.

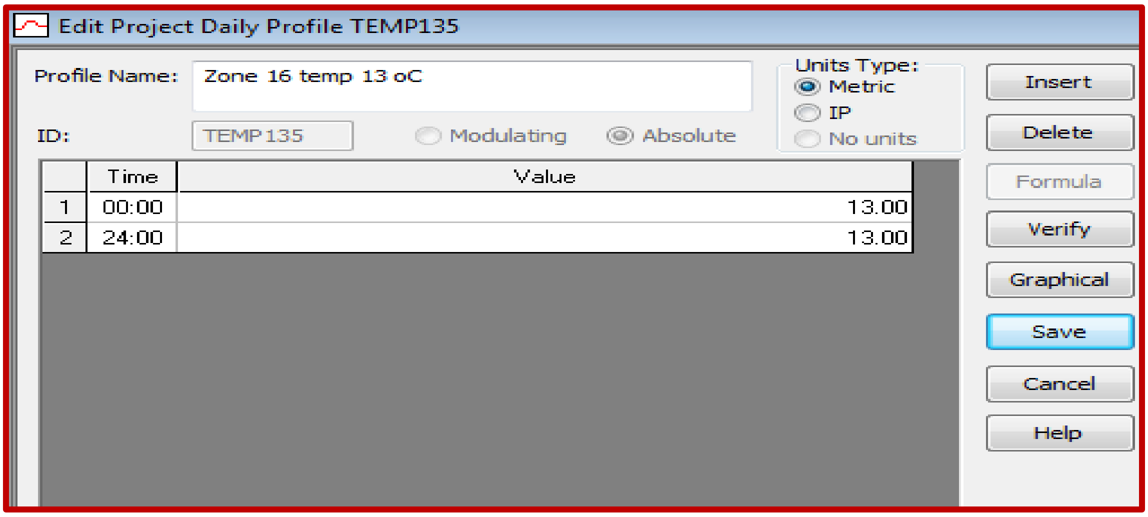

Figure 3:9 Zone 16 temperature profile (Constant temperature13 °C)

Figure 3:9 above illustrates zone 16 temperature profile with space set point temperature of 13 °C absolute. Thus the space temperature is set to be maintained at 13 °C constantly throughout the day and night.

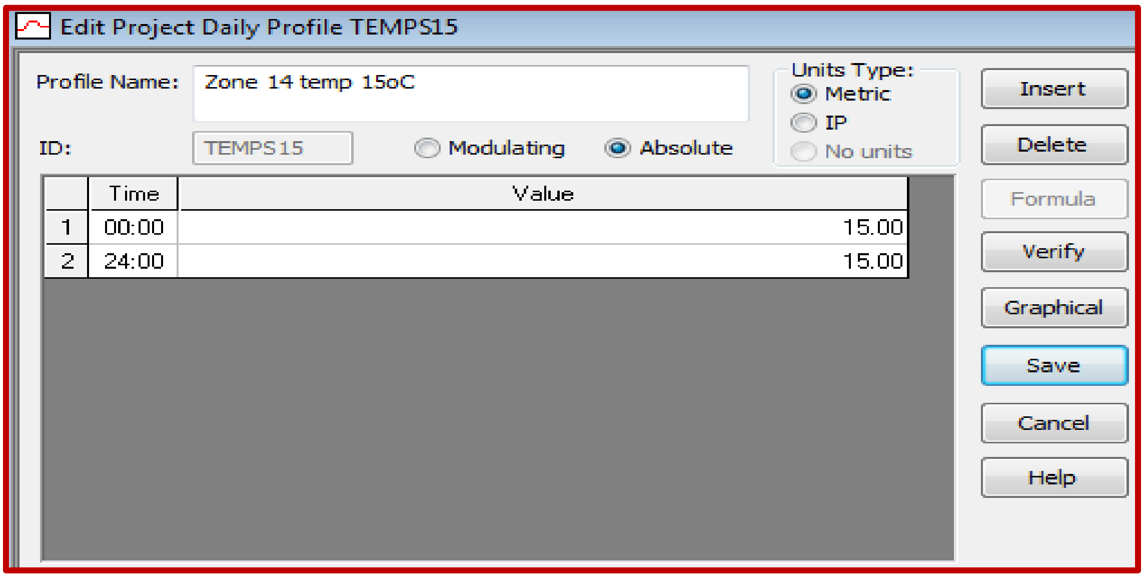

Figure 3:10 Zone 14 temperature profile (Constant temperature 15 ^{o}C)

Figure 3:10 above illustrates zone 14 temperature profile with space set point temperature of 15 ^{o}C absolute. Thus the space temperature is set to be maintained at 15 ^{o}C constantly throughout the day and night.

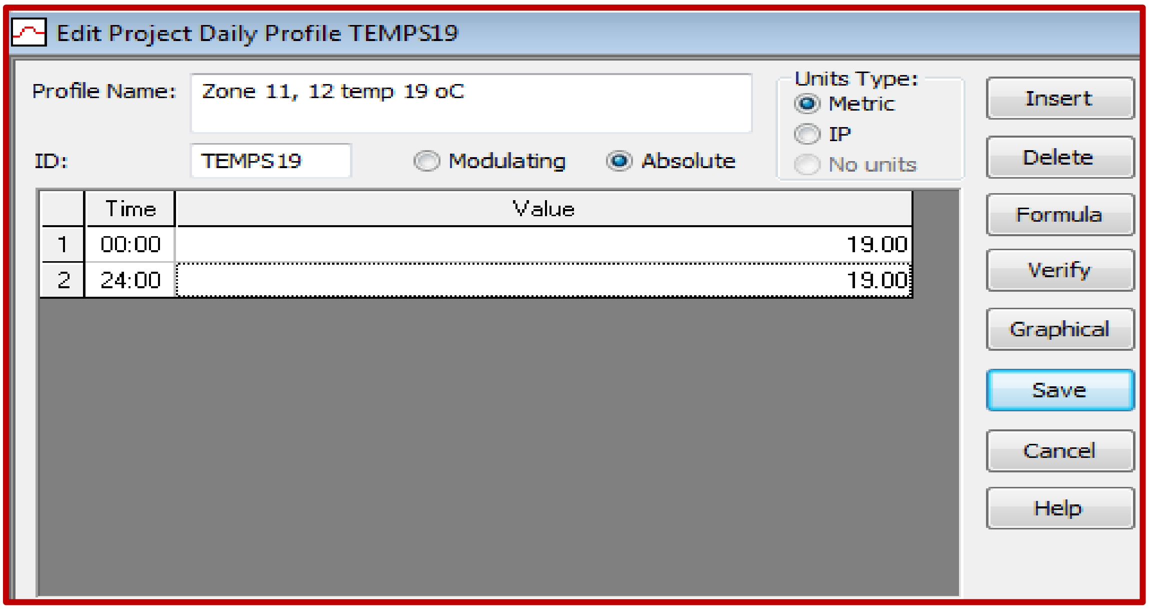

Figure 3:11 Zone 11 and 12 temperature profile (Constant temperature 19 ^{o}C)

Figure 3:11 above illustrates zone 11 and 12 temperature profile with space set point temperature of 19 ^{o}C absolute. Thus the space temperature is set to be maintained at 19 ^{o}C constantly throughout the day and night. Zone 11 and 12 space set point temperatures are the same.

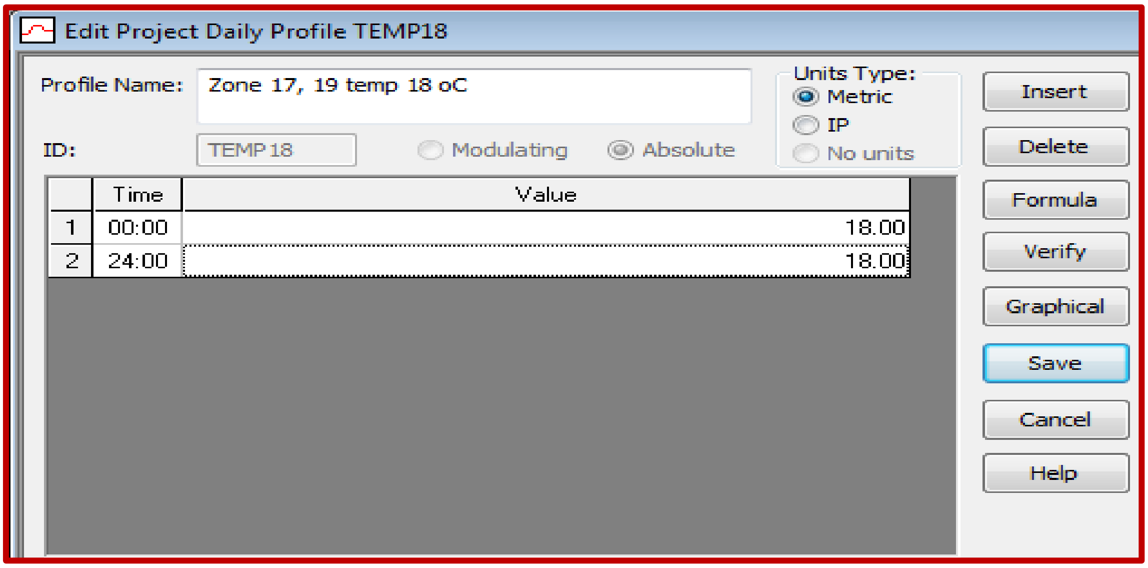

Figure 3:12 Zone 17 and 19 temperature profile (Constant temperature 18 ^{o}C)

Figure 3:12 above illustrates zone 17 and 19 temperature profile with space set point temperature of 18 ^{o}C absolute. Thus the space temperature is set to be maintained at 18 ^{o}C constantly throughout the day and night. Zone 17 and 19 space set point temperatures are the same.

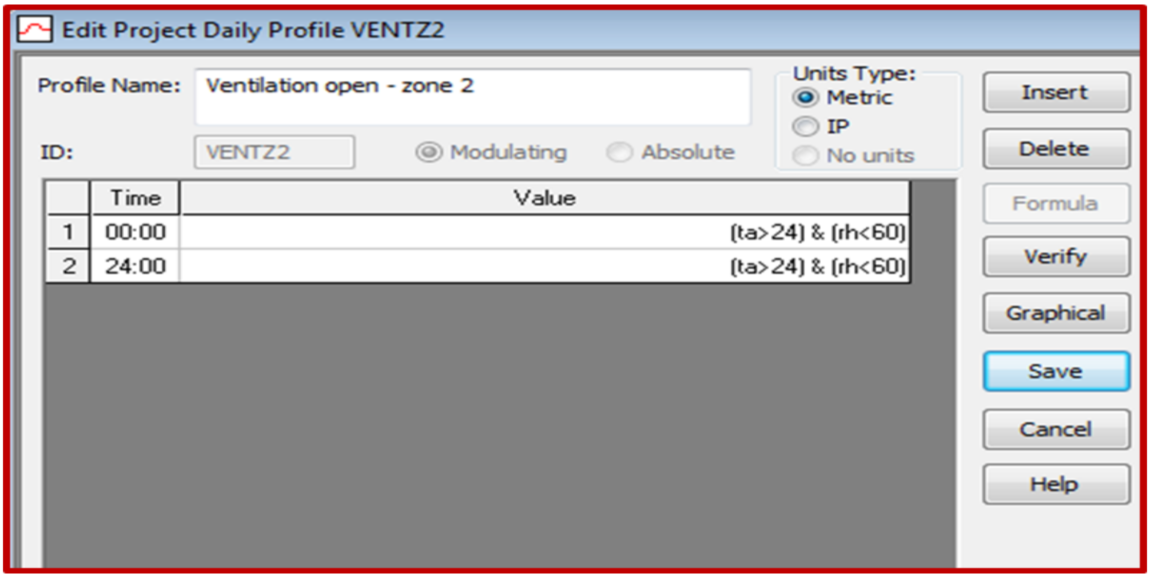

Figure 3:13 Zone 2 vent opening temperature and minimum relative humidity profile

Figure 3:13 above illustrates that vents in zone 2 will open when the space temperature exceeds 24 ^{o}C. The minimum acceptable zone humidity level is 60%.

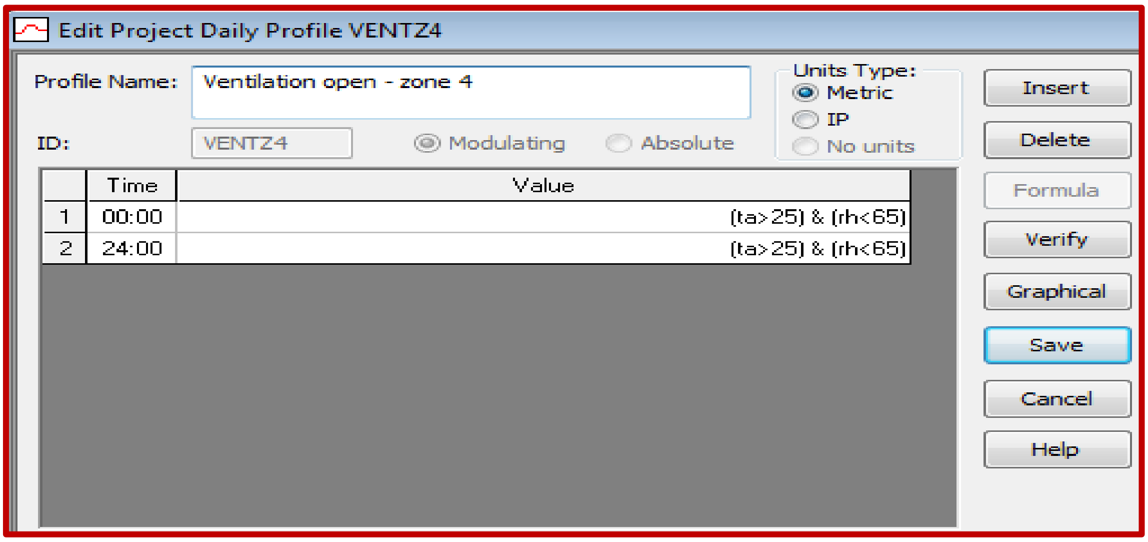

Figure 3:14 Zone 4 vent opening temperature and minimum relative humidity profile

Figure 3:14 above illustrates that vents in zone 4 will open when the space temperature exceeds 25 ^{o}C. The minimum acceptable zone humidity level is 65%.

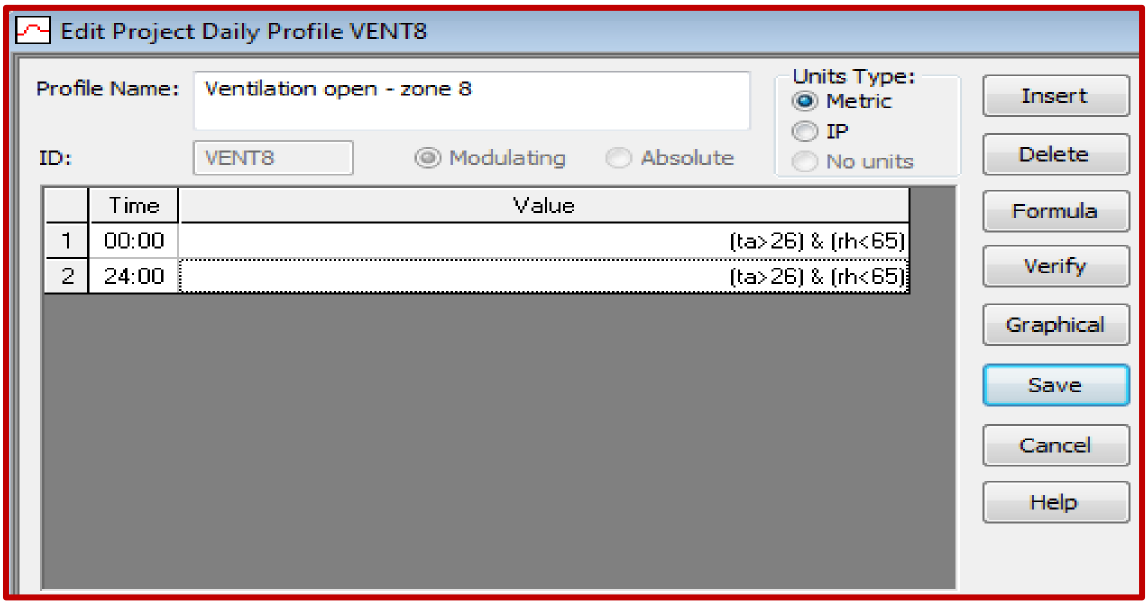

Figure 3:15 Zone 8 vent opening temperature and minimum relative humidity profile

Figure 3:15 above illustrates that vents in zone 8 will open when space the temperature exceeds 26 ^{o}C. The minimum acceptable zone humidity level is 65%.

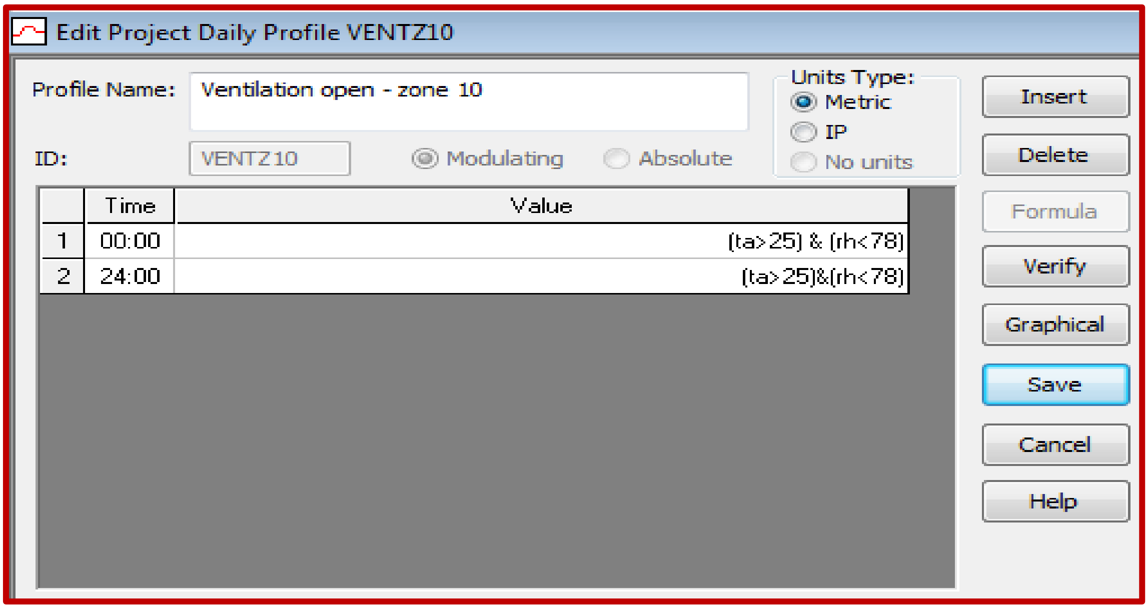

Figure 3:16 Zone 10 vent opening temperature and minimum relative humidity profile

Figure 3:16 above illustrates that vents in zone 10 will open when space temperature exceeds 25 ^{o}C. The minimum acceptable zone humidity level is 78%.

3.3.6 Determining the glasshouse space conditioning heating requirements

The IES software tool was used to calculate each zone's space conditioning heating demand and the overall heating demand of the glasshouse.
For the purpose of this report two zones and the overall glasshouse space conditioning heating demand is demonstrated in tables 3.8, 3.9 and 3.10 below.

3.3.6.1 **Building total heat loss**

The total heat loss of a building is not replaced wholly by the heating plant. Some of the loss heat is met by incidental heat gains arising from solar insolation, lights, equipment and people.

From building energy balance point of view, the sum of energy input equals the overall heat loss and this is demonstrated in figure 3:17 below.

The incidental gains contribute a lot in the building heating requirement.

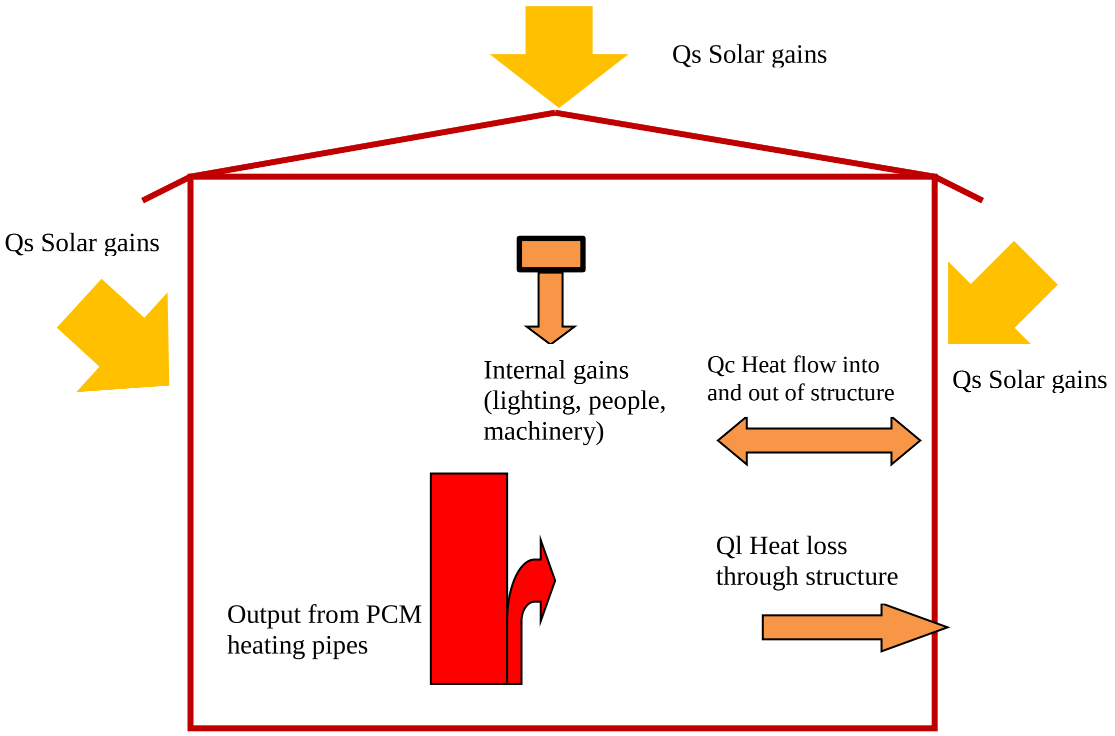

Figure 3:17 Energy balance of glasshouse heating

For this research, the Integrated Environmental Solutions (IES) software tool was used to calculate the building heat loss, the average space temperature for a day, internal heat and solar gains.

The internal gains and temperatures are not constant over the course of the day and they are normally averaged over the day and appropriate gain utilisation or diversity factors to account for the useful gains are applied.

The average daily solar gains are something of a problem as average useful gains data are not normally given in guidance literature. CIBSE Guide J (2002) gives measured monthly mean daily irradiation (in Watt hour / m^2) for three sites in the UK for different orientation and slope angles including vertical.

For this research project the average hourly solar gains used in calculating the space conditioning heating requirements were calculated using the IES software tool as part of the simulation process.

Equating rates of energy gain and rates of energy loss as demonstrated in figure 3:17 above will be:

Solar gains + internal gains + heating system output = Transmission heat loss + ventilation loss. Using symbols will be

$$G + q = H (T_{mi} - T_{mo}) \text{ [W]} \quad (3.1)$$

Where T_{mi} is the mean internal temperature per day (24 hours)
T_{mo} is the external design temperature (-3.4 ^oC)
G is the sum of solar and internal gains (W)
q is the heating system output (W)
H is the building heat transmission plus ventilation heat loss rate per degree (K) of the internal to external temperature differential (W/K)

H is the heat loss coefficient (W/K)

From the above formula $q = H (T_{mi}-T_{mo}) - G$ (3.2)

$$= H (T_b - T_{mo}) - G \text{ [W]}$$

$T_{mi} = T_b$, where T_b is the base temperature that requires no heating.

The base temperature $T_b = T_{mi} - G/H$ [K] (3.3)

This is the outdoor temperature above which heating will not be required from the heating plant. At this point the casual gains equal the heat loss and this outdoor temperature is the base temperature for the building.

This is sometimes called the balance point temperature by American Society of Heating Refrigeration and Air conditioning Engineers (ASHRAE) [6]

The IES software tool was run to calculate the zones and the overall glasshouse space condition heating requirements after all the design parameters such as external design temperature, space temperature and relative humidity set points, heat gains and others have been populated into the software tool.

Table 3:8, Table 3:9table 3:9 and table 3:10 demonstrate space conditioning heating demand of zone 3, zone 8 and the whole glasshouse building. The space conditioning heating demands of all the zones were calculated but for the purpose of this report zone 3, zone 8 and the whole glasshouse building are shown as examples.

The IES simulation results show the hourly maximum space heating demand of zone 3, zone 8 and all the 21 zones in January which is the coldest design month to be 17.4, 63.4 and 762 kW respectively.

Table 3:8 Zone 3 hourly space conditioning heating demand (kW) in January

	January (Days of the month)																														
Time	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
00:30	13.0	14.7	15.6	15.9	15.4	14.4	12.0	9.2	12.6	9.7	9.6	9.6	11.8	11.4	11.7	13.9	9.1	12.7	14.4	14.5	15.5	15.3	12.9	11.3	13.3	13.0	15.8	16.1	15.8	15.3	15.4
01:30	13.0	14.8	15.3	16.0	15.6	14.2	12.1	9.4	12.9	9.7	9.6	9.6	11.9	11.5	12.0	13.7	9.6	13.0	14.4	14.7	15.3	15.3	13.2	11.1	13.3	13.0	15.9	16.0	16.0	15.4	15.5
02:30	13.3	15.0	15.3	16.2	16.3	14.0	12.1	9.3	13.3	9.7	9.6	9.7	12.0	12.0	12.2	13.5	10.2	12.7	14.7	14.7	15.4	15.1	13.4	10.9	12.8	13.3	16.1	15.8	16.1	15.4	15.5
03:30	13.4	15.2	15.3	16.3	16.8	13.7	11.9	9.1	13.4	9.7	9.5	9.8	11.8	12.8	12.3	13.2	10.6	12.3	14.8	14.4	15.6	14.5	13.7	10.8	13.1	13.6	16.0	15.8	16.1	15.4	15.3
04:30	13.6	15.2	15.5	16.2	17.2	13.8	11.5	9.0	13.2	9.8	9.5	9.8	11.9	13.1	12.1	13.1	11.3	12.4	14.6	14.4	15.5	14.0	14.1	10.8	12.4	14.0	15.9	16.0	15.9	15.6	15.2
05:30	13.6	15.2	15.5	15.8	17.2	13.8	11.1	9.0	13.2	9.9	9.5	9.9	12.3	13.1	11.9	12.9	12.0	12.5	14.6	15.1	15.4	14.2	14.6	10.8	12.8	14.0	15.7	16.1	16.0	15.7	15.3
06:30	13.8	15.1	15.7	15.8	**17.4**	13.7	10.9	8.8	13.4	9.9	9.3	10.0	12.9	13.1	12.3	12.8	12.1	12.6	14.8	15.6	15.3	14.7	14.9	10.6	13.2	14.2	15.4	16.3	16.1	15.8	15.3
07:30	13.6	14.9	15.7	15.9	**17.4**	13.6	10.7	8.8	13.6	9.9	9.1	10.2	13.3	13.2	13.1	12.7	12.0	12.8	14.7	15.8	15.0	14.9	14.7	10.4	13.1	14.4	15.5	16.2	15.9	15.7	15.3
08:30	13.1	14.3	15.0	15.1	16.3	13.1	10.0	8.4	12.8	9.5	8.6	9.8	13.1	12.8	12.7	11.9	11.5	12.2	13.6	15.6	14.3	14.0	13.9	9.7	13.6	14.1	15.1	15.7	15.7	15.3	14.7
09:30	12.5	13.8	14.6	14.2	14.3	12.8	9.9	8.1	12.1	9.4	8.1	9.2	12.1	12.2	12.2	11.0	11.3	11.9	12.8	15.9	13.4	12.7	12.8	9.2	13.3	13.9	14.7	15.3	15.7	15.1	13.7
10:30	11.6	13.1	14.2	13.8	13.2	12.3	9.7	7.9	11.4	9.3	7.7	8.9	10.8	11.3	11.7	10.0	10.7	11.5	12.3	16.1	11.7	11.8	11.3	8.7	12.7	13.8	14.1	15.0	15.4	14.5	12.8
11:30	10.8	12.5	13.7	13.3	12.7	12.0	9.4	8.5	11.0	9.0	7.2	9.2	10.1	10.4	11.0	9.1	10.4	11.2	11.2	15.9	10.6	10.9	10.0	8.3	12.0	13.7	13.5	14.3	14.8	14.0	12.3
12:30	10.2	12.3	13.8	13.1	11.8	11.8	9.4	9.0	10.6	8.9	7.2	9.7	10.0	10.0	10.7	8.5	10.2	11.3	11.1	15.7	11.1	10.6	9.6	7.8	11.4	13.3	13.2	13.8	14.7	14.0	12.2
13:30	10.1	12.2	14.0	12.9	12.0	11.8	9.5	8.4	10.1	9.0	7.3	9.8	10.4	9.8	10.8	8.2	10.5	11.2	11.0	15.9	11.5	10.8	9.5	7.6	10.3	13.5	13.2	14.5	14.6	14.0	12.3
14:30	10.5	12.5	14.2	13.0	12.4	11.9	9.4	8.3	9.6	9.1	7.5	9.7	11.2	10.2	11.0	8.1	11.3	11.0	11.0	15.9	12.0	11.0	9.8	7.9	10.0	13.8	13.4	14.5	14.8	14.1	12.3
15:30	11.2	13.3	14.6	13.3	13.3	12.3	9.5	8.9	9.3	9.2	8.0	10.2	11.5	10.8	11.2	8.1	11.8	11.2	11.8	15.6	12.3	11.2	10.1	8.6	11.3	13.6	13.8	14.7	14.9	14.3	12.3
16:30	11.4	13.7	14.8	13.9	13.8	12.5	9.7	9.6	9.4	9.3	8.4	11.1	11.2	11.2	11.5	8.0	11.9	11.8	12.6	15.5	13.5	11.6	10.7	9.5	11.8	14.1	14.3	15.1	15.0	14.5	12.6
17:30	12.0	14.5	15.4	14.0	14.4	12.8	9.8	10.3	9.4	9.5	8.8	11.9	11.0	11.6	12.1	7.9	12.0	12.3	13.0	15.7	14.4	12.1	11.5	10.1	12.3	15.2	15.2	15.4	15.7	15.0	13.2
18:30	12.8	14.7	15.6	14.5	14.4	12.7	9.6	10.8	9.4	9.7	9.0	12.4	10.6	11.4	12.7	7.9	12.1	13.0	13.7	15.6	14.7	12.1	11.4	10.4	12.5	15.5	15.5	15.5	15.8	15.1	13.8
19:30	13.1	14.8	15.8	15.3	14.2	12.6	9.6	11.3	9.4	9.8	9.2	12.7	10.2	11.2	13.0	7.9	12.2	13.4	14.4	15.6	14.8	11.9	11.6	11.1	12.6	15.6	15.5	15.6	15.8	15.0	14.3
20:30	13.4	15.0	15.8	15.4	14.3	12.5	9.3	11.7	9.4	9.8	9.4	12.7	9.7	11.3	13.3	8.1	12.2	13.8	14.6	15.3	14.8	12.1	11.3	12.3	12.5	15.6	15.4	15.4	15.7	15.2	14.6
21:30	13.7	14.9	15.7	15.5	14.5	12.5	9.1	11.8	9.4	9.8	9.5	12.4	9.6	11.4	13.8	8.5	12.2	14.0	14.7	15.1	14.8	12.5	11.4	12.9	12.3	15.8	15.7	15.4	15.5	15.3	14.8
22:30	14.0	14.8	15.8	16.0	14.4	12.3	9.1	11.9	9.6	9.6	9.7	12.1	10.2	11.5	14.1	8.8	12.3	14.4	14.8	15.4	15.3	12.4	11.5	13.0	12.7	15.8	15.9	15.6	15.4	15.3	14.9
23:30	14.1	15.8	15.9	15.4	14.4	12.0	9.1	12.2	9.8	9.5	9.7	11.9	11.0	11.6	14.1	8.8	12.5	14.6	14.7	15.5	15.7	12.6	11.4	13.7	13.0	15.7	16.0	15.7	15.3	15.5	15.0

Table 3:9 Zone 8 hourly space conditioning heating demand (kW) in January

	January (Days of the month)																														
Time	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
00:30	43.1	51.6	56.1	54.6	56.4	48.4	37.4	26.7	41.2	28.6	28.2	29.7	37.5	35.0	37.3	47.1	26.1	40.4	49.8	49.8	52.3	52.8	43.2	35.1	47.1	42.7	53.6	55.8	53.5	50.3	52.0
01:30	43.4	52.0	55.8	55.1	56.6	47.9	37.9	27.2	42.5	28.8	28.2	29.7	38.1	35.5	38.6	46.3	28.4	41.3	49.2	49.5	51.7	52.8	44.5	34.2	46.2	42.5	54.1	55.2	53.8	50.6	52.1
02:30	44.2	52.7	55.8	55.5	58.7	47.0	37.8	27.1	44.2	28.9	28.0	29.9	38.4	37.3	39.8	45.1	30.6	40.0	49.9	49.0	52.0	52.5	45.8	32.9	44.4	42.9	55.0	54.3	53.8	50.8	51.8
03:30	45.1	53.7	55.3	55.8	60.3	45.8	37.0	26.3	44.6	28.9	27.9	29.9	37.9	40.7	40.4	43.8	32.2	38.3	50.3	47.6	52.6	51.0	47.8	32.3	45.0	44.2	54.7	54.2	53.3	50.9	51.2
04:30	45.8	53.6	55.0	55.5	62.0	45.8	35.5	25.7	44.0	29.4	27.9	29.8	37.7	41.9	39.8	43.0	35.3	38.7	50.0	47.2	52.8	49.9	49.7	32.3	43.0	45.8	54.1	54.3	53.7	51.6	51.0
05:30	46.6	53.7	54.9	54.4	62.4	45.5	34.3	25.7	43.5	29.7	27.8	29.9	39.2	41.8	38.6	42.0	38.2	39.1	50.0	49.8	53.2	50.3	51.5	32.3	44.0	45.9	53.4	54.7	54.0	51.9	51.7
06:30	47.5	53.2	55.7	54.6	**63.4**	45.3	34.2	25.4	44.2	29.5	27.1	30.4	41.8	41.8	40.0	41.3	38.4	39.9	50.9	51.9	53.5	51.6	52.8	32.0	45.3	46.4	52.7	55.2	53.4	52.0	52.3
07:30	47.6	52.6	55.9	55.5	**63.4**	44.9	33.4	25.3	44.5	29.4	26.2	31.1	43.5	41.9	43.2	40.7	38.0	40.6	51.5	52.3	52.8	50.2	51.0	30.7	45.3	46.8	52.3	54.4	53.0	51.6	51.6
08:30	45.1	49.4	52.0	52.8	57.4	43.4	31.3	24.0	41.8	28.6	23.9	29.0	43.2	41.1	40.7	36.2	36.3	38.4	46.1	52.7	48.8	44.9	46.8	29.0	47.0	46.2	50.2	53.1	52.1	50.8	48.9
09:30	38.5	44.8	47.4	45.2	45.0	41.1	29.4	21.5	37.4	27.5	19.6	24.3	37.6	37.1	35.1	31.7	30.9	34.1	38.2	53.5	41.9	33.3	37.0	26.2	44.3	44.4	45.8	50.3	49.4	48.2	38.8
10:30	29.4	37.9	41.7	39.2	35.7	38.1	27.2	19.5	33.6	26.3	14.4	22.2	28.9	29.0	29.0	28.1	22.7	30.5	30.4	53.6	32.2	22.6	24.8	21.9	39.1	42.5	40.8	47.2	46.5	42.0	30.4
11:30	24.3	33.3	37.0	36.4	31.6	36.1	25.1	21.6	31.9	24.4	11.7	23.6	23.8	21.1	23.8	24.8	19.9	29.6	23.6	52.2	26.6	17.5	16.7	17.3	33.8	40.4	35.0	43.3	45.6	36.3	28.2
12:30	22.1	31.6	36.4	35.5	28.0	35.7	25.8	23.5	30.3	23.7	13.1	25.4	23.3	17.6	21.6	23.0	20.4	29.7	21.8	51.5	27.7	18.4	15.4	13.9	30.4	37.9	32.2	40.6	45.0	35.6	29.0
13:30	22.9	30.5	38.1	34.6	28.0	36.2	27.7	20.0	28.7	24.6	14.9	25.1	26.9	17.8	22.9	22.3	25.6	28.4	20.7	52.3	30.3	23.2	18.7	13.0	24.4	39.2	33.1	42.6	45.9	37.5	31.3
14:30	27.6	34.4	42.5	35.6	32.0	37.5	28.0	18.8	26.9	25.7	16.9	24.6	31.7	23.2	27.4	22.4	32.0	27.8	23.8	52.9	34.6	27.5	24.1	16.6	23.3	42.2	34.9	43.9	47.7	40.3	33.0
15:30	34.3	42.7	47.7	41.5	40.2	39.3	28.6	23.4	26.6	26.7	20.5	28.4	34.2	30.5	33.0	22.8	35.1	31.2	33.0	52.1	38.9	31.9	28.7	23.1	31.9	43.5	38.9	46.6	49.7	43.3	35.5
16:30	38.0	47.9	50.7	49.3	46.6	40.6	29.4	29.4	27.3	27.5	23.7	34.4	34.0	35.3	37.0	22.5	37.0	36.7	42.0	51.9	45.3	36.7	33.4	29.2	37.2	46.7	46.1	49.8	51.7	46.7	38.9
17:30	39.9	50.1	52.1	50.5	49.7	40.7	29.0	32.7	26.5	27.8	25.6	38.1	32.7	36.8	39.2	21.6	37.5	39.9	45.8	51.5	48.1	38.7	36.1	32.2	39.3	50.0	50.7	50.6	52.2	48.7	41.2
18:30	42.7	51.5	52.9	52.2	49.9	40.3	28.3	34.6	26.6	28.4	26.6	39.9	31.1	36.0	41.8	21.4	37.6	42.7	48.4	51.2	49.0	39.4	36.0	34.4	40.7	51.6	52.2	51.0	52.1	49.1	43.8
19:30	44.6	52.4	53.6	54.7	49.1	40.0	28.0	36.6	26.7	29.1	27.4	41.0	29.4	35.4	43.9	21.3	38.2	45.0	50.7	51.7	49.4	39.1	36.2	37.6	41.3	52.3	53.1	51.4	51.8	49.4	46.3
20:30	46.0	53.2	53.7	55.8	48.7	39.7	27.1	38.3	26.8	29.1	28.2	40.8	27.6	35.4	45.0	22.2	38.2	46.8	51.0	50.5	49.3	39.3	35.7	42.0	41.4	52.5	53.1	51.1	51.2	50.0	47.7
21:30	47.2	53.7	53.5	56.5	49.1	39.4	26.3	38.7	27.1	28.9	28.9	39.6	27.4	35.6	46.4	23.9	38.4	48.2	50.9	49.9	49.4	40.7	35.8	44.8	41.0	53.5	53.9	51.2	50.6	50.7	48.5
22:30	48.8	53.5	53.9	58.3	49.0	38.4	26.1	39.1	28.1	28.2	29.7	38.6	30.1	36.2	47.8	24.9	38.7	49.7	50.9	51.2	51.5	40.9	36.0	45.8	41.8	53.3	54.9	51.9	50.3	51.1	49.2
23:30	49.7	56.1	54.1	56.9	48.9	37.6	26.3	40.1	28.7	27.9	29.9	37.8	33.4	36.6	47.8	25.1	39.2	50.4	51.0	51.6	53.8	41.6	35.6	48.2	42.8	53.1	55.4	52.3	50	51.9	50.1

Table 3:10 All zones (21 zones) hourly space conditioning heating demand (kW) in January

	January (Days of the month)																														
Time	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
00:30	468	595	657	632	662	551	400	252	447	277	269	293	397	361	390	532	242	432	568	569	600	610	476	366	541	473	622	654	610	574	594
01:30	472	601	655	638	666	544	405	258	466	281	269	294	406	369	408	521	271	444	559	563	594	610	493	354	529	471	630	647	620	579	594
02:30	484	610	655	644	693	532	404	258	490	281	267	295	410	393	424	505	299	427	567	557	597	606	511	336	506	476	642	635	625	581	590
03:30	497	624	648	647	716	517	393	248	498	281	265	294	404	438	433	489	320	404	572	537	607	588	540	327	514	492	638	633	625	584	581
04:30	507	623	643	643	738	516	373	241	489	288	266	292	402	456	426	477	362	409	568	531	608	573	566	327	490	513	630	634	619	592	579
05:30	518	624	642	629	746	513	358	241	482	292	264	293	422	454	410	461	401	415	568	565	614	577	592	327	502	516	622	638	625	596	587
06:30	530	617	654	633	760	510	357	236	490	289	256	300	458	454	429	451	403	425	581	594	621	594	609	323	519	522	612	645	628	598	596
07:30	532	609	656	646	**761**	505	346	235	494	287	245	309	482	456	472	442	399	434	591	600	612	611	620	315	526	532	617	639	619	597	607
08:30	546	616	658	643	**762**	497	337	258	490	288	259	324	480	456	506	448	414	461	598	608	631	616	624	304	560	542	632	630	615	599	621
09:30	571	621	666	667	725	498	347	277	476	292	289	345	461	475	538	416	491	496	625	628	642	652	635	310	567	561	652	640	633	621	667
10:30	599	641	692	693	723	490	360	287	436	297	356	333	454	524	567	348	552	494	669	644	603	691	643	346	584	579	665	657	640	663	658
11:30	571	643	698	665	697	474	350	308	397	297	388	324	449	558	583	290	550	470	645	639	556	666	618	380	579	589	680	643	617	688	609
12:30	519	630	697	634	640	456	325	341	363	287	324	350	414	560	574	260	495	472	629	633	556	599	564	388	555	583	680	624	609	678	568
13:30	472	600	663	609	612	442	314	349	332	276	270	373	389	510	535	233	413	482	606	633	538	514	465	375	515	572	651	640	600	638	529
14:30	418	552	609	584	571	436	302	343	300	269	249	375	392	428	460	214	395	457	526	625	516	453	387	327	459	546	629	609	597	600	488
15:30	394	519	573	553	544	439	294	316	277	267	227	365	382	379	405	208	410	418	469	601	495	409	353	292	422	510	586	577	589	567	458
16:30	402	536	576	557	548	443	290	311	264	265	223	376	351	378	399	201	401	408	484	593	523	403	362	304	411	528	561	576	579	549	441
17:30	432	569	600	579	568	445	284	334	251	265	237	404	330	385	420	191	395	432	515	590	546	415	384	331	428	573	580	581	593	551	449
18:30	469	589	610	602	571	440	275	358	251	272	250	427	309	374	456	187	396	469	551	586	558	426	381	363	449	594	600	587	600	557	484
19:30	497	602	619	635	560	435	270	385	253	281	260	442	286	365	487	185	404	501	581	593	563	422	383	408	456	604	614	592	599	559	518
20:30	516	614	620	651	555	431	259	408	254	282	270	441	264	365	503	197	404	526	585	576	561	423	377	466	459	608	616	588	596	566	537
21:30	533	624	618	661	559	426	249	414	258	278	280	424	260	367	521	217	407	545	582	568	561	441	377	507	454	622	628	590	588	576	547
22:30	556	624	622	686	558	412	246	421	269	270	290	411	296	376	540	228	411	566	583	585	590	445	379	523	463	619	642	599	579	582	557
23:30	570	654	625	669	557	401	248	433	277	266	295	400	340	381	541	231	417	576	585	592	622	455	374	554	474	617	649	604	574	592	571

Summary of glasshouse design

The research apart from seeking to develop a heating system that will use zero carbon emission energy sources to heat the glasshouse also tried to create an environment that plants can grow in healthier condition.

The existing glasshouse is currently heated using fossil fuel and the research seeks to heat the glasshouse using thermal energy storage system.

The space conditioning heating demand of the existing glasshouse could have been obtained easily if the thermal energy consumption of the glasshouse has been metered separately for 2011 and 2012 reference period of operation.

The thermal energy consumption metered covers other buildings on site making it difficult to know the energy consumed by the glasshouse. As a result the space heating energy consumption of the glasshouse was calculated from the total thermal energy consumed in 2011 and 2012 including the other buildings.

Initially three sources of heating energy (Solar hot water system, PCM filled heating pipe and the black painted pipes) were considered to ensure that enough thermal energy is obtained and stored to heat the glasshouse throughout the year but the black painted pipes idea was dropped for further studies due to research time limitation. The most interesting part of the heating system is that the stored energy is released to the space only when the space temperature falls below the PCM phase change temperature to avoid waste.

Buildings that the space temperatures are control by sensors, thermostats and TRVs waste lots of energy especially if the controls go faulty, but with the research design heating system energy will only be released to the space when there is a need for heating and will not be affected by any controls problems.

The existing climate control logic board which is currently used to control the space temperature and humidity will be modified to control space temperatures at the extremes.

For example, the climate control logic device will be used to control space temperatures when they exceed the maximum acceptable temperature to open the vents to allow some heat out to the atmosphere.

The control logic will also enable the motorised valves to allow LTHW flow through the PCM heating pipes when the space set point temperature is reached. The primary space set point temperatures will be controlled by melting and freezing of the PCM filled heating pipes.

During the preliminary study of the glasshouse environmental conditions, space temperatures and humidity levels were measured for three weeks and established that the space temperatures measured in most cases throughout the study were higher than the outside temperature within the same measurement period.

This could be supported by the study of Baille and Bouland of a glasshouse of 176m2 ground floor area [21]. In their study, the February and March outside temperatures were 3.8 and 6.6 ^{o}C respectively whilst the space temperature for the same months and periods were measured to be 10.9 and 13.5 ^{o}C respectively. This is a demonstration that solar energy trapped in the glasshouse due to greenhouse effects assist in heating a glasshouse.

For green plants to grow in healthier condition, the environmental temperature and relative humidity requirements should be adequately maintained. The glasshouse and the heating system were designed to meet these criteria.

The plants produce respiration heat but this heat is latent and therefore not considered when assessing the space heating requirement of the glasshouse. Latent heat occurs at a constant temperature and does not raise the space temperature.

The IES software tool was used in designing the glasshouse building, calculate the space conditioning heating demand for each zone and all the 21 zones, solar gains and temperature profiles of each zone.

3.4 Solar thermal energy

3.4.1 Solar thermal energy background

The most inexhaustible and cleanest among all energy sources is the solar energy [10]. Solar radiation is the heat, light and other radiation emitted from the sun. The amount of energy in solar radiation is huge and responsible for all the earth natural processes.

Buildings can be passively heated using solar energy through architectural design directly or indirectly.

Sun thermal energy is used in producing domestic hot water, heat space and even provides cooling through the use of absorption chillers.

3.4.2 Solar water heating

Water heating using the sun's energy has existed over one hundred years by simply black painted tanks in many countries [11].

Solar water heating (SWH) has improved in the last century and installed solar collectors around the globe exceeds thirty million square metres (30,000,000 m^2) [11].

In countries such as the UK, Denmark, Germany, USA, Japan, Australia and Greece have installed hundreds of thousands of modern solar water heaters such as the evacuated tube system shown in figure 3:18 below [11].

Figure 3:18 [11] Solar water heating system

In the UK solar energy systems have been successfully used and the maximum solar radiation received is about 1kW / m^2 [12] with seasonal variations.

Solar energy system alone in the UK in reality may not be able to meet the total building energy demand at economic cost. As a result solar energy systems for buildings in the UK are usually installed alongside conventional systems.

The research project is aim at using hundred per cent (100%) zero CO_2 emission heating system. The design system for the research project will use solar active and passive systems including waste heat from CHP to achieve the zero CO_2 emission target.

To achieve the project aim is to use zero carbon emission source of energy such as solar thermal systems. The disadvantage of the solar thermal system is that the solar radiation source is abundant during the spring and summer months whilst the demand for heating is greatest in winter. Upon all the disadvantages, the benefits derived from using the sun's energy are several and most importantly it is renewable.

3.4.3 The Basics of solar energy

Sunlight reaches every location on earth at least at some point within the year but the amount of solar radiation received varies according to the following:

- Geographic location
- Time of day
- Season
- Local landscape
- Local weather

The sun strikes the surface at different angles because the earth is round and the sun strikes the surface at different angles ranging from 0° (Just above the horizon) to 90° (directly overhead) [13]

The earth revolves around the sun in an elliptical orbit and get closer to the sun at certain part of the year. When the sun is nearer the earth that is where the earth's surface receives a little more solar energy.

In summer the earth is nearer the sun in the southern hemisphere and winter in the northern hemisphere. The greatest amount of solar energy reaches a solar collector around solar noon on a clear day.

Since solar energy system will be used to generate the heating energy requirement of the glasshouse, the concepts of solar energy need to be understood.

3.4.3.1 Solar radiation in the UK

The radiation from the sun is absorbed, scattered and reflected by water vapour, clouds, dust and others as it passes through the earth's atmosphere.

Direct radiation is the rays that come straight from the sun and diffuse radiation is the proportion of scattered radiation that reaches the earth's surface and appears to come from all over the sky [12].

The UK total annual radiation received is 50% diffuse [12]. Solar system manufacturers that produce solar system products for the UK and Western Europe design systems that can absorb as much diffuse radiation as possible and not just direct radiation to capture solar energy on cloudy days.

Average annual solar irradiation variation over the UK is shown in figure 3:19 and figure 3:20 shows direct and diffuse irradiation available in Western Europe.

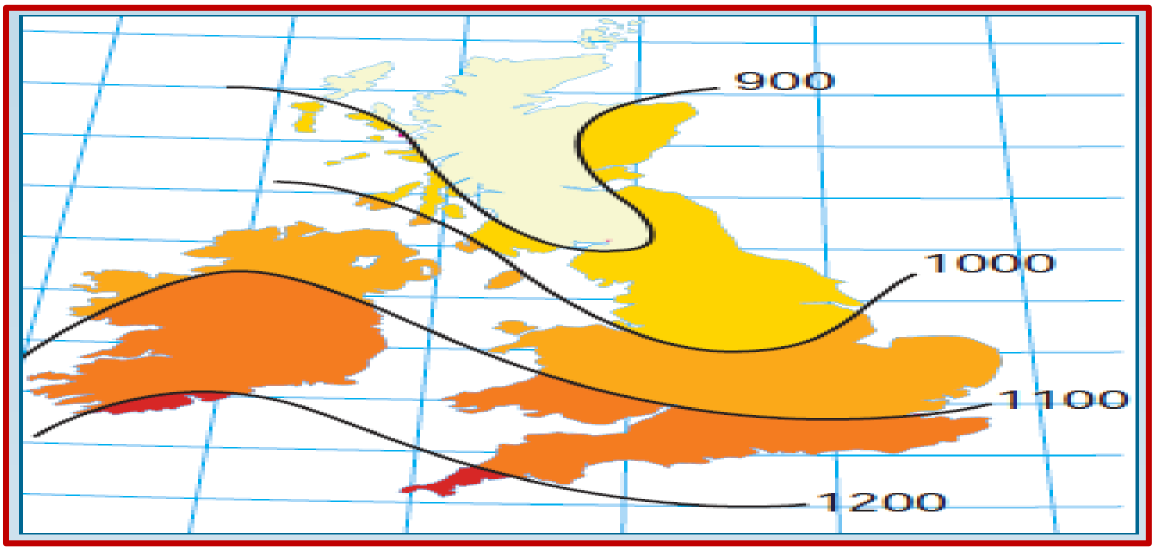

Figure 3:19 [12] UK annual average solar irradiation (KWh / m^2) on a 30° incline facing due south

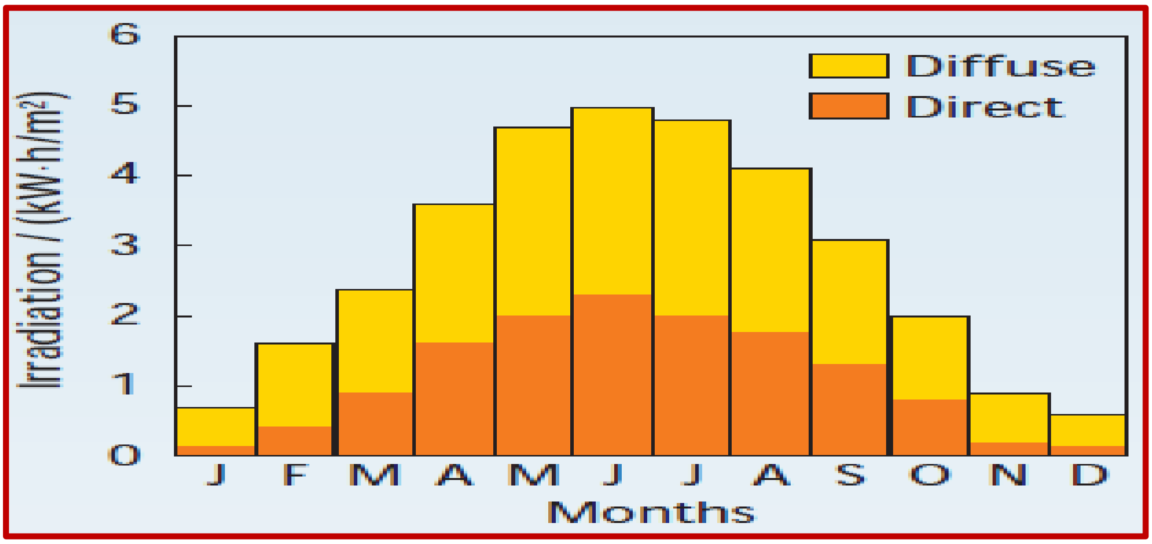

Figure 3:20 [12] Daily amounts of direct and diffuse radiation (KWh / m^2) available in Europe

The geographical location determines the amount of solar radiation received and varies over the year. The monthly solar radiation available in December on a horizontal surface in the UK compared to that of June is only about 10% with slightly improvement ratio for inclined surfaces.

Table 3:11 below shows monthly mean daily average irradiation in London per month on a south-facing plane inclined at 30° to the horizontal.

Table 3:11 [7] Monthly mean daily total irradiation (direct plus diffuse) on south-facing plane inclined at 30° to horizontal for London

Location	Daily mean irradiation (kWh / m^2) for stated month											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
London	1.11	1.89	2.74	4.03	4.78	5.03	4.98	4.68	3.39	2.45	1.14	0.93

(source: CIBSE Guide A Tables 2.27-2.29).

Maximum amount of solar radiation could be received annually from south orientation at an angle equal to the site latitude minus approximately 20° from the horizontal [14].

For example, the approximate tilt angle from horizontal is 30° in Southern England increasing to almost 40° in northern Scotland. Figure 3:21 below illustrates the effect of tilt and orientation on solar energy generation.

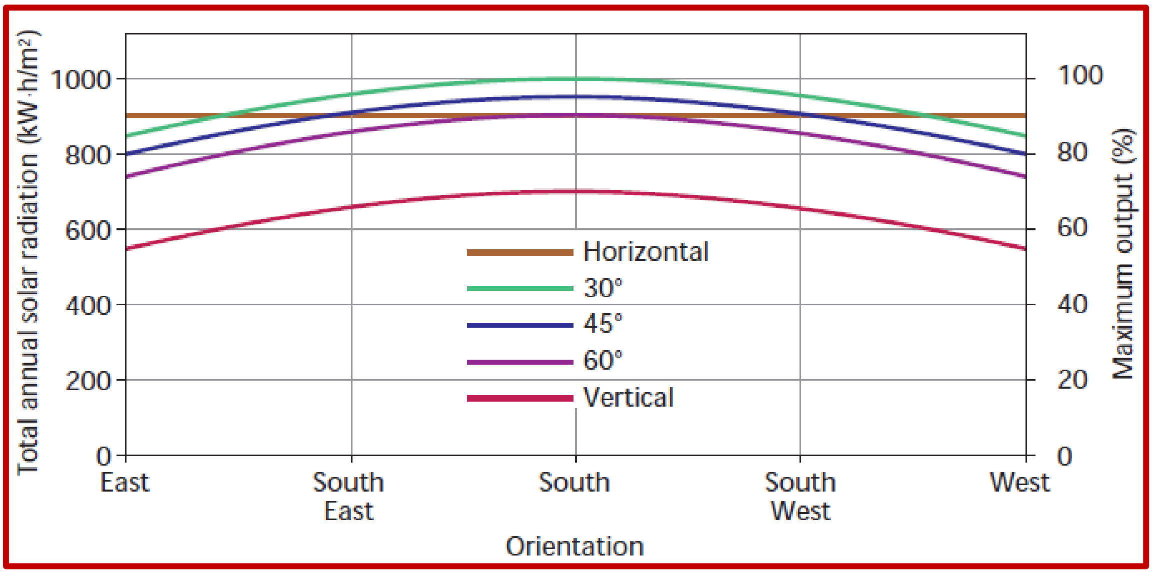

Figure 3:21 [14] Effect of tilt and orientation on energy generation (Based on data for SE England)

The angle from the horizontal should be decreased and increased by 10° to receive maximum solar irradiation in summer and in winter respectively. Across the UK total annual solar irradiation between southwest and southeast will receive over 90% of the maximum annual energy with small variation. The maximum total annual solar irradiation for London per year for example, is 1045 kWh / m^2 [14].

Solar energy system performance is mainly dependent on the amount of solar radiation available at local climate and could be affected by fog or mist and exposure to wind. Solar energy system closer to shaded site or building may not be suitable.

3.4.3.2 Cold water temperature in the UK

The energy required to heat up the water to the desired temperature is calculated from the temperature of the cold water supplied by the public.

The cold water temperature can be calculated from two options. One of the options to calculate the cold water temperature is using monthly ambient temperature values and the other is using minimum and maximum values [15]. The glasshouse heating system is a close loop circuit and may not be affected by cold water temperature unless there is a need for make-up water.

3.4.3.2.1 Cold water inlet temperatures case study in UK

The research tried to establish the average cold water temperature in the UK as this may affect the operational efficiency of the active solar system if additional make-up water is needed. The reason for assessing the mean cold water temperature in the UK is to establish the effect that cold water make-up could have on the solar hot water heating system if there is a need to add water to the system.

A case study carried out on behalf of Energy Saving Trust in UK on 112 dwellings measured the inlet cold water temperatures to the boilers serving 112 dwellings. The case study established that the cold water feed temperatures of all the 112 boilers have a mean value of 15.2 °C with 95% confidence interval ± 0.5 °C [15].

The distribution of the measured cold water inlet temperatures data is shown in Figure 3:22.

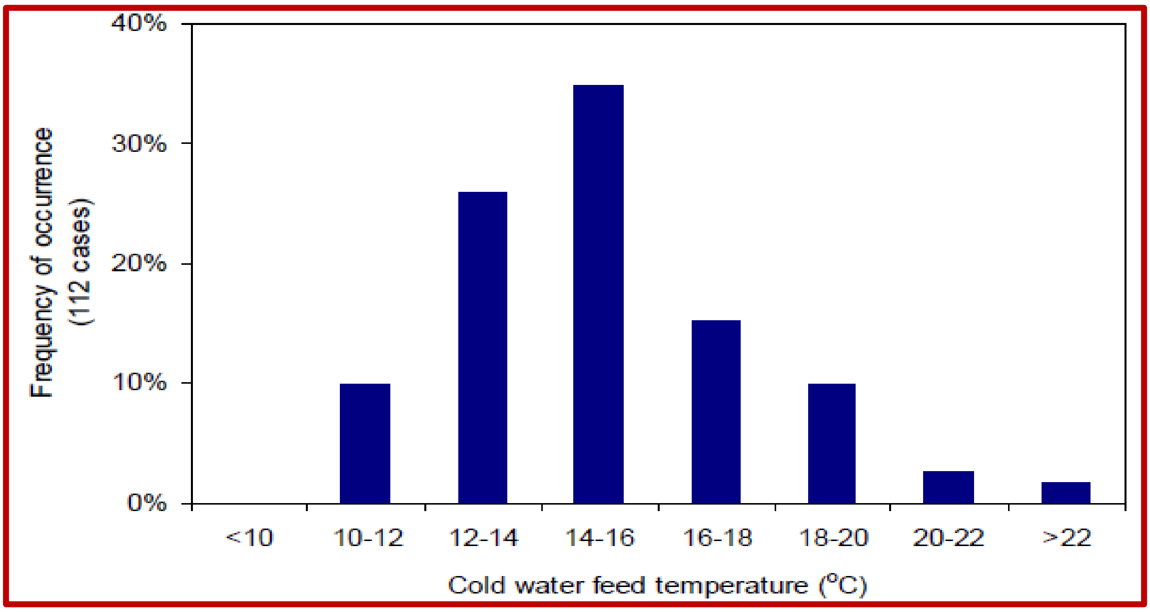

Figure 3:22 [15] Cold water feed temperatures distribution for the whole sample of 112 case studies

3.4.3.3 Solar collectors

The key element of solar collectors is the absorber on which the solar radiation falls. Absorber plates of solar collectors collect solar energy. Selective coating can be used to improve the absorber plate overall collector efficiency. The absorbed energy by the absorber is collected by the thermal fluid.

The application temperature being considered in the system design will determine the type of solar collector needed. Unglazed liquid flat-plate, glazed liquid flat-plate and evacuated tube solar collectors are the types mostly used as they can be easily installed by technicians, prices have fallen over the years, they are available, easily maintained and qualify for renewable heat incentive in the UK.

3.4.3.3.1 Unglazed liquid flat-plate collectors

Figure 3:23 illustrates unglazed liquid flat-plate collectors usually made of a black polymer. They do not have a selective coating, no insulation at the back and no frame. They are usually laid on a wooden support or on the roof.

They are good in capturing the sun energy, low cost but thermal losses increases as water temperature rises especially in windy locations.

Unglazed collectors are commonly used for low temperature applications such as pool heating, process heating and make-up water in fish farms. They are also used in colder climates in summer season due to collector high thermal losses.

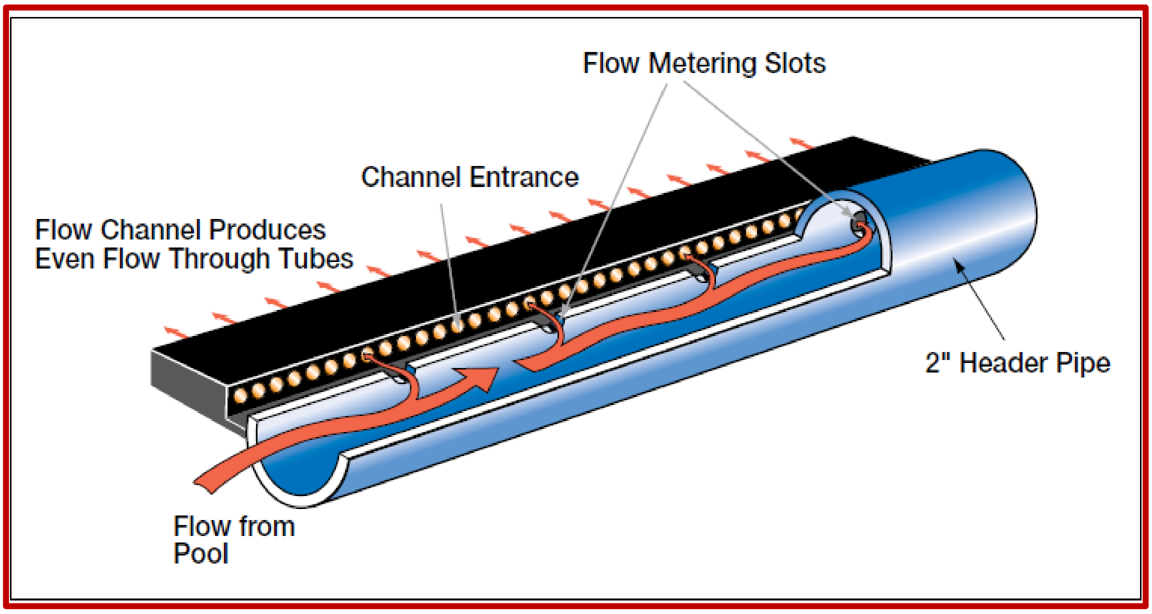

Figure 3:23 [11] Unglazed flat-pate solar collector

3.4.3.3.2 Glazed liquid flat-plate collectors

In glazed liquid flat-plate collectors, as shown in figure 3:24 below most often has selective coating and the flat-plate absorber is fixed in a frame between a single or double layers of glass and the back is insulated.

They are commonly used in moderate temperature applications such as domestic hot water heating, process heating, indoor pools and space heating.
The absorber plate collects the sun radiation through the transparent cover made of plastic or glass and could be either single or double-glazed.

The absorber plate transfers the absorbed heat to the liquid which is useful energy gain and some are lost by convection to the surroundings and by conduction through the back and edges. The solar energy loss is prevented or reduced from escaping due to the greenhouse effect.

The liquid most commonly used is water, although oil can be used. Figure 3:24 and figure 3:25 below demonstrate samples of glazed liquid flat plate collector and liquid flow schematic of glazed collector respectively.

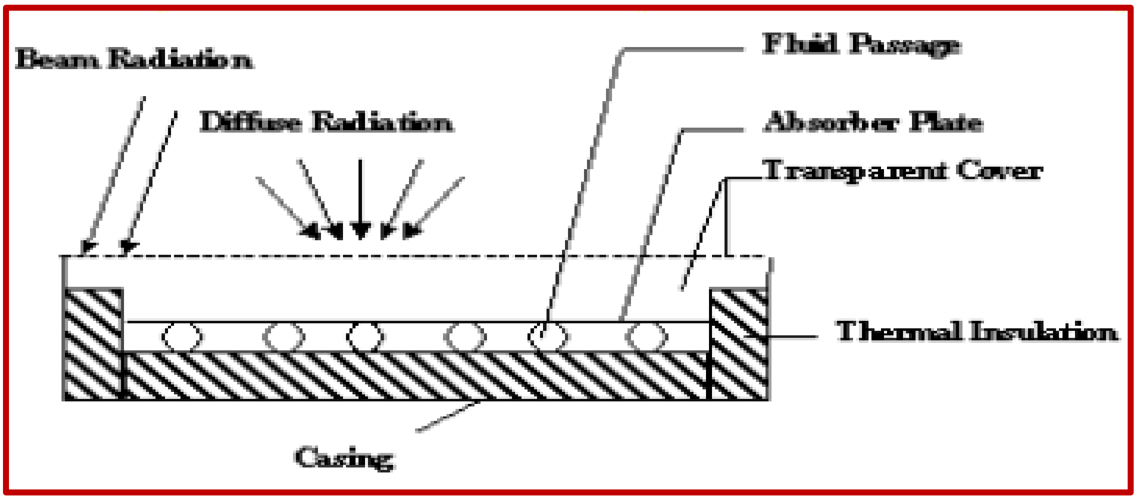

Figure 3:24 [16] Glazed liquid flat plate solar collector

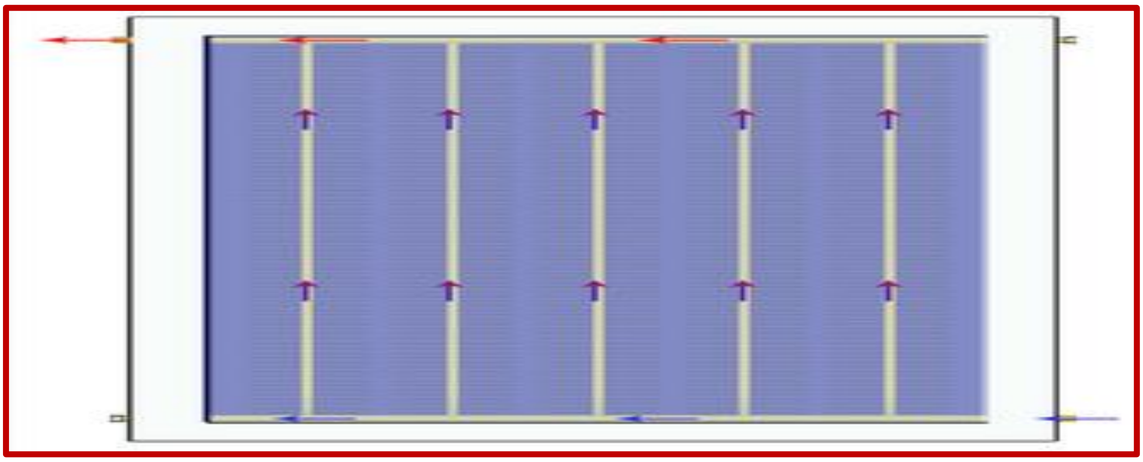

Figure 3:25 Flat plate collector liquid flow schematic [16]

Figure 3:26 below illustrates how manufacturers use different definitions for the collector area. The collector area has effect on the energy generation so care was taken when selecting the flat plate collector for the research project.

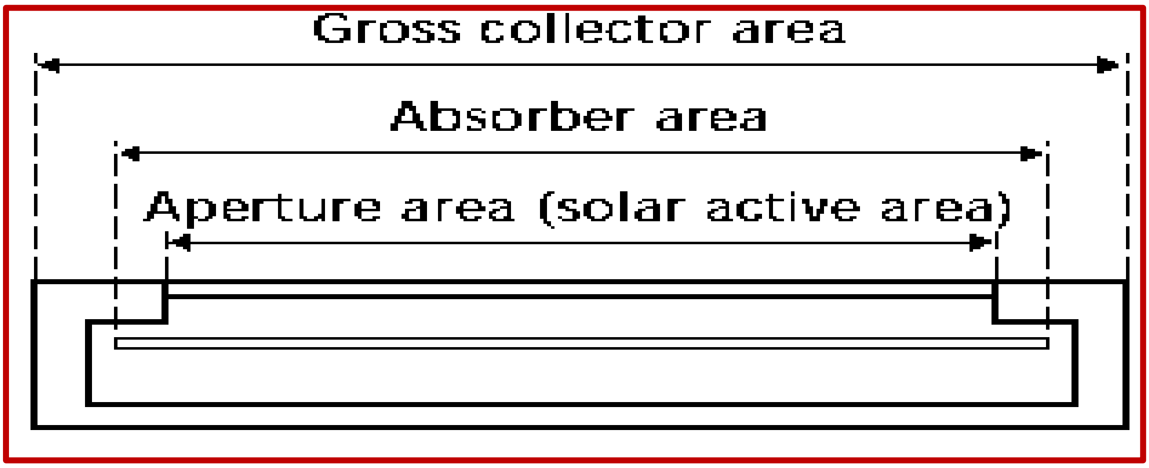

Figure 3:26 [12] Area measurements used for flat plate collectors

Source: CIBSE K15

3.4.3.3.3 Evacuated tube solar collectors

At high temperature applications evacuated tube collectors are more efficient but it can still operate under cloudy conditions and generate energy all year round compare to the others.

In an evacuated-tube collector, the sunlight enters through the outer glass tube and strikes the absorber where the energy is converted to heat. The heat is then transferred to the liquid flowing through the absorber. Each transparent parallel glass tube has an absorber covered with a selective coating.

The thermal losses are extremely low and they are good in capturing the sun's energy at low and high radiation levels. There are two main types of evacuated tube collectors, namely direct flow and heat pipe collectors as illustrated in figure 3:27 and figure 3:28 below respectively.

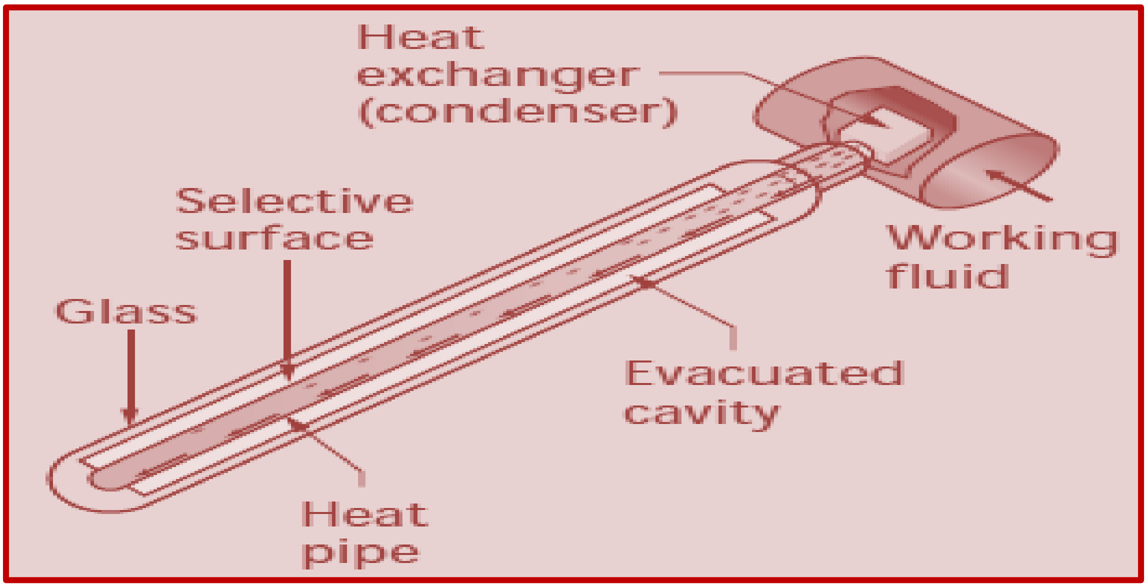

Figure 3:27 [12] Direct flow evacuated tube
Source: CIBSE K15

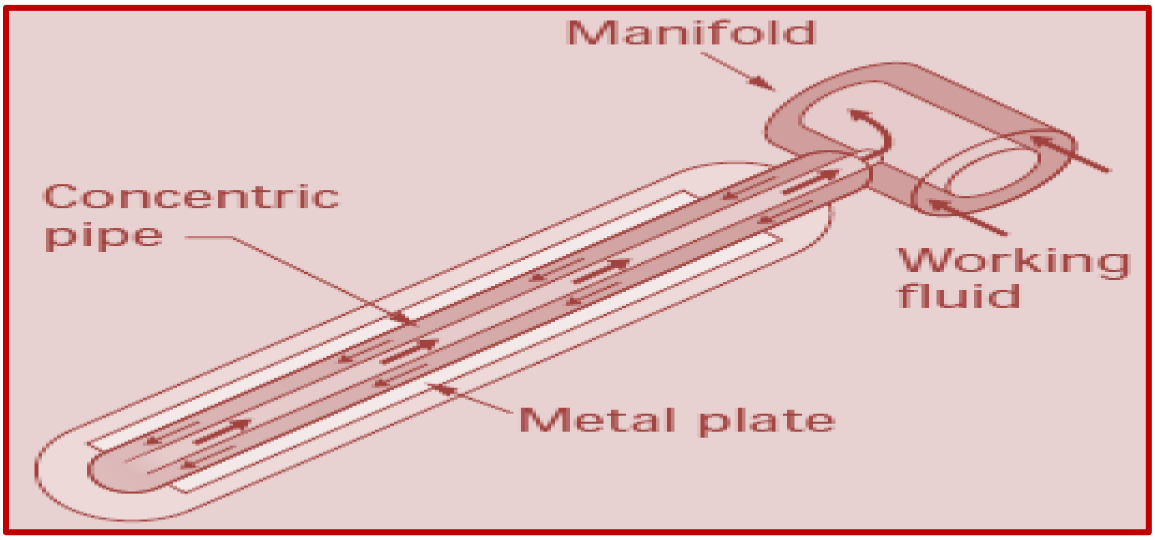

Figure 3:28 [12] Heat pipe evacuated tube

Source: CIBSE K15

Figure 3:29 and figure 3:30 below demonstrate samples of evacuated tube collectors and liquid flow schematic of evacuated tube Split U Pipe collector and evacuated tube Heat Pipe collector respectively.

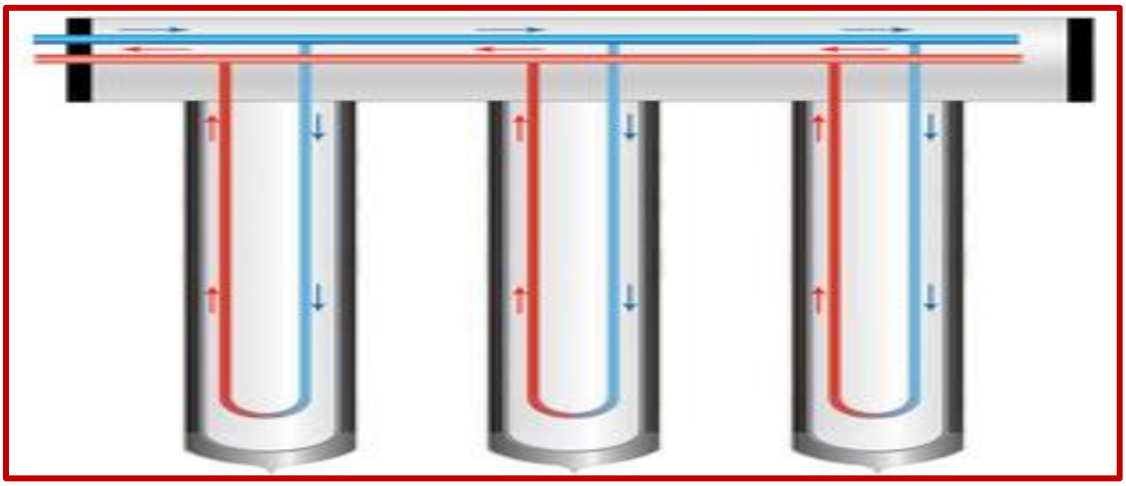

Figure 3:29 [17] Evacuated tube Split U Pipe collector

Figure 3:30 [17] Evacuated tube Heat Pipe collector

3.4.3.3.4 Solar collector efficiency

Solar collectors are currently rated in five categories below by the Solar Rating and Certification Corporation (SRCC) [18] according to the collector inlet fluid temperature (Ti) and the ambient air temperature (Ta) difference.

Table 3:12 [18] Solar rating categories

Category		Heating application
	T inlet – T ambient (^{o}C)	
A	-5	Pool Heating in Warm climate
B	5	Pool Heating in Cool Climate
C	20	Water Heating in Warm Climate
D	50	Water Heating in Cool Climate
E	90	Industrial Process Water Heating

The Solar Rating and Certification Corporation (SRCC) is a third-party independent certification organization that certification and rating of solar energy equipment are administered.

SRCC is a non-profit corporation incorporated in October 1980. The two major solar programs they run are collector certification (OG-100) and heating system certification (OG-300). The part of the solar collector system exposed to the sun to collect the sun's heat energy is the collector certification (OG-100) program.

The program that assess the whole system performance which includes collector test and the entire system to determine whether the complete system meet minimum standards for durability, reliability, safety and operation is the OG-300 rating and certification.
The factors that affect the total system design, installation, maintenance and service are also evaluated.

The solar collector heating system will not qualify for the UK government renewable heat incentive unless the system could meet the minimum standards or requirements as set out above.

3.4.3.3.4.1 Solar collector efficiency equations

All the SRCC data is calculated from efficiency equations acquired from testing collectors at certified test laboratories. The efficiency is defined as:
Efficiency = output / input.

The output of the solar system will be low at low solar radiation level inputs no matter the efficiency of the collector. Collector performance can quickly and easily be compared using the efficiency curves.

The useful thermal energy divided by the solar energy received by the solar thermal collector is the efficiency and depends on factors such as collector type, the absorbing surface spectral response, the collector insulation and difference in temperature between the collector and the ambient air.

BS EN 12975 [19] test standard is used to measure collector performance by means of plotting efficiency against temperature difference between collector and the ambient temperature.

The efficiency is given by the equation [12]:

$$\eta = \eta_0 - a_1(T_m - T_a)/G - a_2(T_m - T_a)^2/G \qquad (1)$$

and hence the power:

$$\text{Power} = A*(\eta_0*G - a_1*(Tm-Ta) - a_2*(Tm-Ta)^2) \qquad (2)$$

where η_0 is conversion factor (The efficiency when the collector and surrounding temperature is the same).

η is the collector efficiency
a_1 is the heat loss coefficient ($Wm^{-2}K^{-1}$)
a_2 is the temperature dependence of the heat loss coefficient ($Wm^{-2}K^{-1}$)
T_m is the mean collector temperature (°C)
T_a is the ambient temperature (°C)
G is the global solar irradiance (Wm^{-2}).

The manufacturers as a standard provide the values for η_0 and the heat loss coefficients a_1 and a_2 from the test carried out on the solar collectors.

The collector efficiency decreases as the collector heats up and heat loss increases through infrared radiation and convection.
Solar collectors are described by their efficiency equations and the three main types of collectors that were considered in the research project are:

- Glazed collectors
- Evacuated collectors
- Unglazed collectors

The unglazed collectors use a wind-dependent efficiency equation whilst glazed and evacuated tube collectors share wind-independent efficiency equation.

The angle of incidence effects, losses due to snow and dirt, and loss of heat through the piping and the solar tank factors are accounted for separately.

3.4.3.3.4.2 Glazed or evacuated collectors

The following equation describes glazed or evacuated tube collectors (Duffie and Beckman, 1991, eq. 6.17.2):

$$\dot{Q}_{coll} = F_R\,(\tau\alpha)\,G - F_R U_L\,\Delta T \qquad (3)$$

Where the collected energy per unit collector area per unit time is $\dot{Q}_{coll}$

F_R is the collector's heat removal factor

τ is the transmittance of the cover

α is the shortwave absorptivity of the absorber

G is the global incident solar radiation on the collector

UL is the overall heat loss coefficient of the collector

ΔT is the temperature difference between the working fluid entering the collectors and the ambient.

F_R $(\tau\alpha)$ and $F_R U_L$ for glazed and evacuated collectors are independent of the wind. Generic values are provided with glazed and evacuated collectors.

Glazed collectors are provided with F_R $(\tau\alpha)$ = 0.68 and $F_R U_L$ = 4.90 (W/m^2)/ºC [12]. These values correspond to test results for Thermo Dynamics collectors (Chandrashekar and Thevenard,1995).

FR $(\tau\alpha)$ = 0.58 and F_R U_L = 0.7 (Wm^{-2}) / ºC is the generic value provided with evacuated collectors. These values correspond to a Fournelle evacuated tube collector (Philips technology; Hosatte, 1998).

3.4.3.3.4.3 Unglazed collectors

The following equation (Soltau, 1992) describes unglazed collectors:

$$\dot{Q}_{coll} = (F_R\alpha)\,[G + \{\varepsilon/\alpha\}L] - (F_RU_L)\,\Delta T \quad (4)$$

where ε is the longwave emissivity of the absorber, and L is the relative longwave sky irradiance. L is defined as:

$$L = T_{sky} - \sigma\,(T_a + 273.2)^4 \quad (5)$$

Where L_{sky} is the longwave sky irradiance

T_a the ambient temperature expressed in °C.

$F_R\alpha$ and F_RU_L are a function of the wind speed

G is the solar radiation incident upon the collector.

The wind speed incident upon the collector is set to be 20% of the free stream air velocity [11]

The ratio ε /α is set to 0.96.

Unglazed collector generic is defined as:

$$F_R\alpha = 0.85 - 0.04V \quad (6)$$

$$F_RU_L - 11.56 + 4.37V \quad (7)$$

The values were obtained by averaging the performance of several collectors (NRCan, 1998)

3.4.3.3.4.4 Incidence angle modifiers

Part of the solar radiation incident upon the collector may bounce off if the sun rays hit the surface of the collector with a high angle incidence. The effect due to incident angle is approximately 5 % [11]. As a result, a constant factor equal to 0.95 is multiplied by F_R (τα).

3.4.3.3.4.5 Piping and solar tank losses

In a solar collector system heat will be lost to the environment through distribution pipes and storage tanks as they are not perfectly insulated.

Systems with storage account for pipework and tank losses but those without storage tanks only account for pipework losses.

System without storage tank solar collector delivered energy, Q_{dld} is equal to the energy collected Q_{act} minus piping losses, expressed as a fraction f_{los} of energy collected:

$$Q_{dld} = Q_{act} - (1-f_{los}) \quad (8)$$

For systems with storage tank, the pipework and tank losses will need extra energy to compensate the loss. As a result, the total load $Q_{load,\ tot}$ needs to be increased to include pipework and tank losses:

$$Q_{load,\ tot} = Q_{load}\ (1+f_{los}) \quad (9)$$

3.4.3.3.4.6 Snow and dirt losses

The level of irradiance experienced by solar collector is affected by snow and dirt. Therefore, $F_R\ (\tau\alpha)$ is multiplied by $(1 - f_{dirt})$
where f_{dirt} are the dirt and snow losses expressed as a fraction of the collected energy.

3.4.3.3.4.7 Collector efficiency comparison curves

Collector efficiency curve as shown in figure 3:31 was achieved by plotting collector mean temperature and ambient temperature difference against collector efficiency.
The ‘conversion factor’ is the efficiency when the collector and ambient temperatures are the same. The conversion factor described in figure 3:31 below of the solar collector is 80% [12].

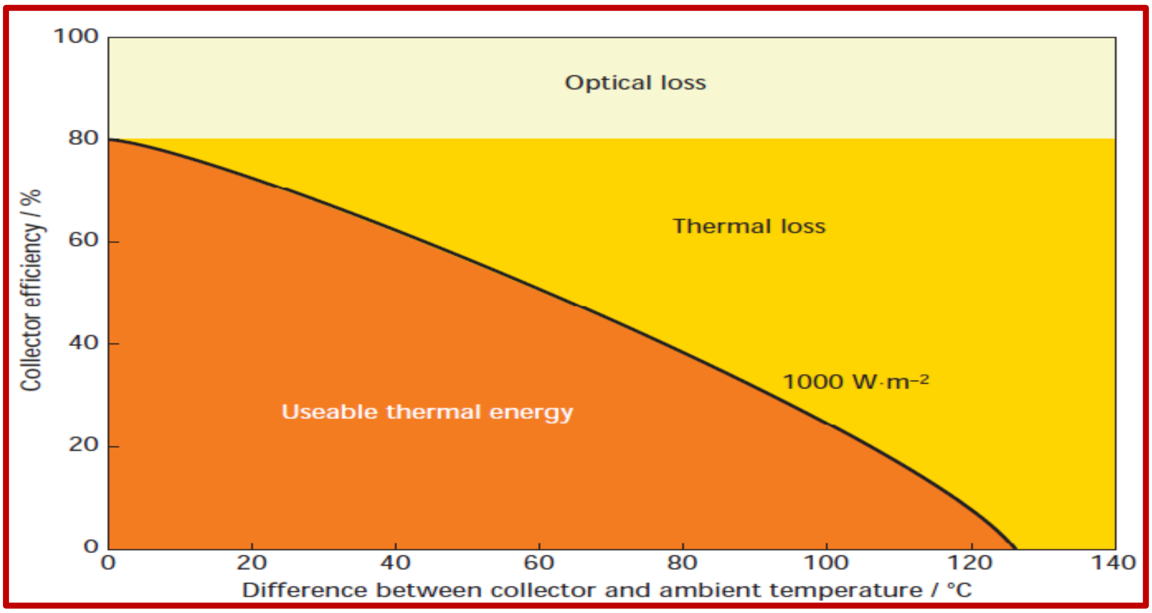

Figure 3:31 [12] Solar collector efficiency curve

Source: CIBSE KS15

The conversion factor or the optical efficiency indicates the received proportion of solar energy available after transmission and absorption losses. The heat loss and the temperature difference between the absorber and its surroundings are indicated by the heat loss coefficient in $Wm^{-2}K^{-1}$.

For the solar collector described above no energy is delivered when the temperature difference between the collector and the ambient is above 127 °C as the heat loss is equal to the useful heat yield by the collector.

The choice of solar collector for an application is determined by the resultant temperature range needed. Other factors to consider when selecting solar collector for an application is the amount of radiation expected at the locality, weather conditions and installation space available.

Evacuated tube collectors compared to flat plate collectors are expensive. Flat plate collectors are commonly used than evacuated tube collectors but the efficiency is lower at higher temperature ranges. The efficiency and temperature ranges of unglazed, flat plate and evacuated collectors are shown in figure 3:32 below.

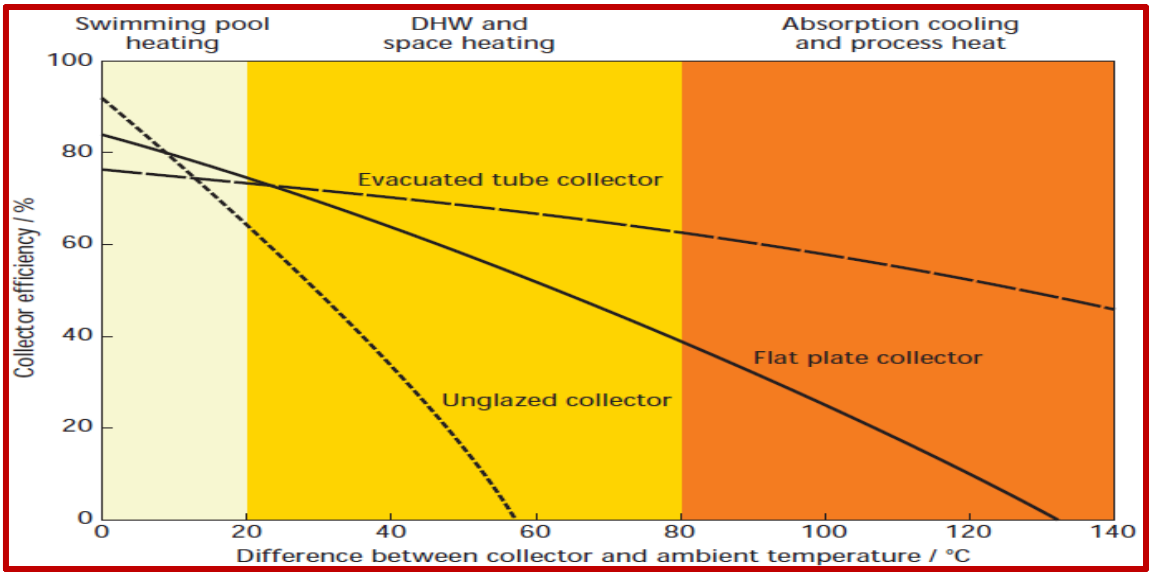

Figure 3:32 [12] Efficiency and temperature ranges for various types of collectors
Source: CIBSE KS15

3.4.4 Description of solar water heating systems

Solar collectors and a liquid handling unit are used to transfer heat to the load in solar water heating systems most commonly via a storage tank.

In active systems pump(s) as part of the liquid handling unit is used to circulate the working fluid from the collectors to the storage tank which includes control and safety equipment.

Solar water heating system is protected from overheating on hot sunny days and can effectively work well if properly designed when outside temperature is below freezing.

Many systems have back-up facility to ensure that hot water requirements are met during periods when the sunshine is insufficient. Three basic operations are performed in solar water heating system as shown dramatically in figure 3:33 below. The three basic operations are:

Solar radiation collection: The solar collector captures the solar radiation

Transfer: Energy is transferred to a storage tank or medium (For example PCM) by the circulating fluids. Circulation can be natural (thermosiphon systems) or forced using a pump.

Storage: Hot water is stored in a storage tank or on the roof in the case of a thermosiphon system until it is needed at a later time.

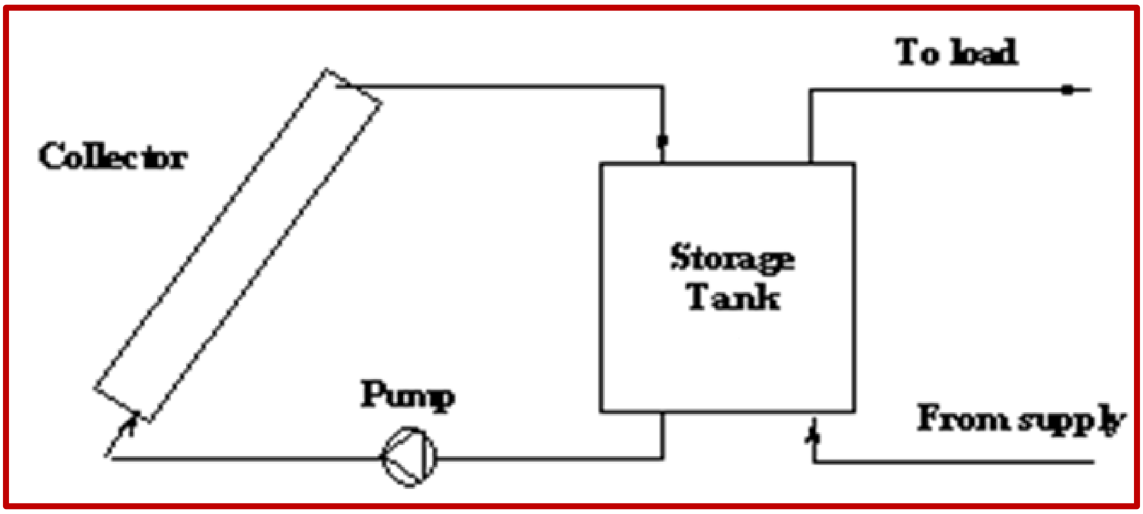

Figure 3:33 [12] Basic solar thermal storage system

Active systems are normally used for industrial and commercial applications and some domestic applications as well. In temperate climates such as central and northern Europe the active systems are predominant (Keith & Goswami, 2007), (DGS, 2010). Typical active and passive systems are shown in figure 3:34 below. The passive (Thermosyphon) system does not require pump and hot water circulation is by gravitational force.

They are usually used for domestic hot water applications rather than space heating. They are common in Southern Europe and sunny countries (Keith & Goswami, 2007), (DGS, 2010).

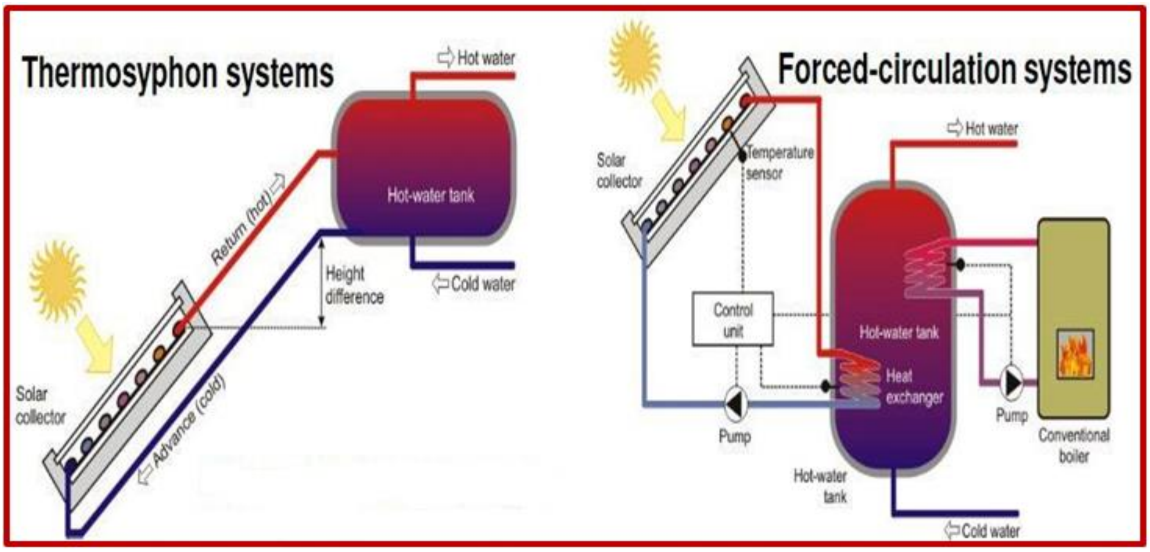

Figure 3:34 A typical thermosyphon and active systems

3.4.4.1 Closed-loop and open-loop systems

A closed-loop (Twin-circuit or indirect) system is the one which the solar collector circuit is separate from the hot water circuit and operates within the pressures of 1.5 to 10 bars.

The operational performance of the solar collector system directly depends on the storage temperature which means that the system running is proportional to the storage temperature.

The operating pressure influences the physical properties such as the evaporation temperature of the working fluid and care should be taken when designing such a system. The solar collector fluid circulating medium does not mix with the fluid supplying thermal energy to the load. Heat exchanger is used to transfer heat to the load hot water loop.

Antifreeze and anti-corrosion mixtures are added to the solar collector system circulating fluid for protection. This prevents contaminants from the incoming cold mains into the system. A typical twin-circuit system described above is shown in figure 3:35 below.

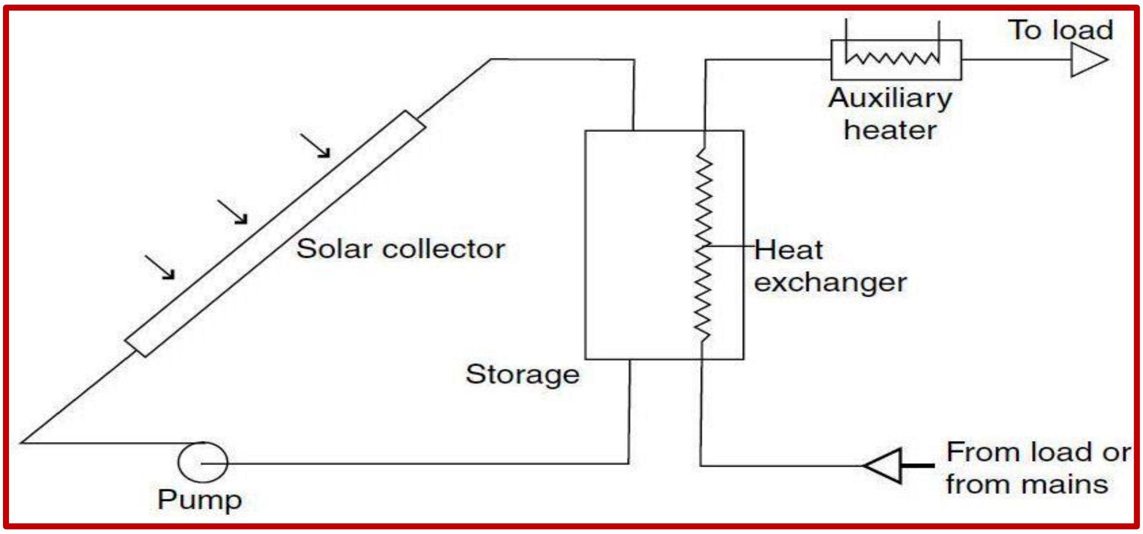

Figure 3:35 Schematic of a closed-loop (twin-circuit type) solar thermal system (Keith & Goswami, 2007)

There are two main types of this system namely topping-up and all-or-nothing system. The "topping-up" type has auxiliary heater placed in parallel (Figure 3:36) whilst in the "all-or-nothing" system the auxiliary heater is placed in series (Figure 3:37).

These systems have been widely used for domestic hot water and space heating applications and they are predominantly type of solar system use in the UK and Ireland (Keith & Goswami, 2007), (DGS, 2010), (CIBSE et al, 2007).

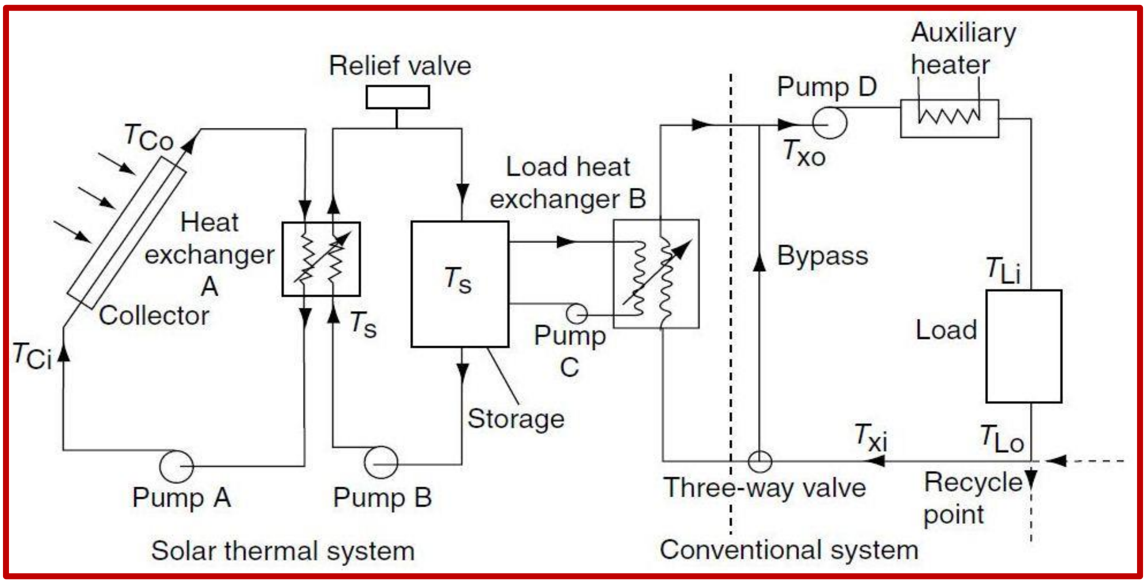

Figure 3:36 Schematic diagram of a typical topping-up type solar thermal system (Keith & Goswami, 2007)

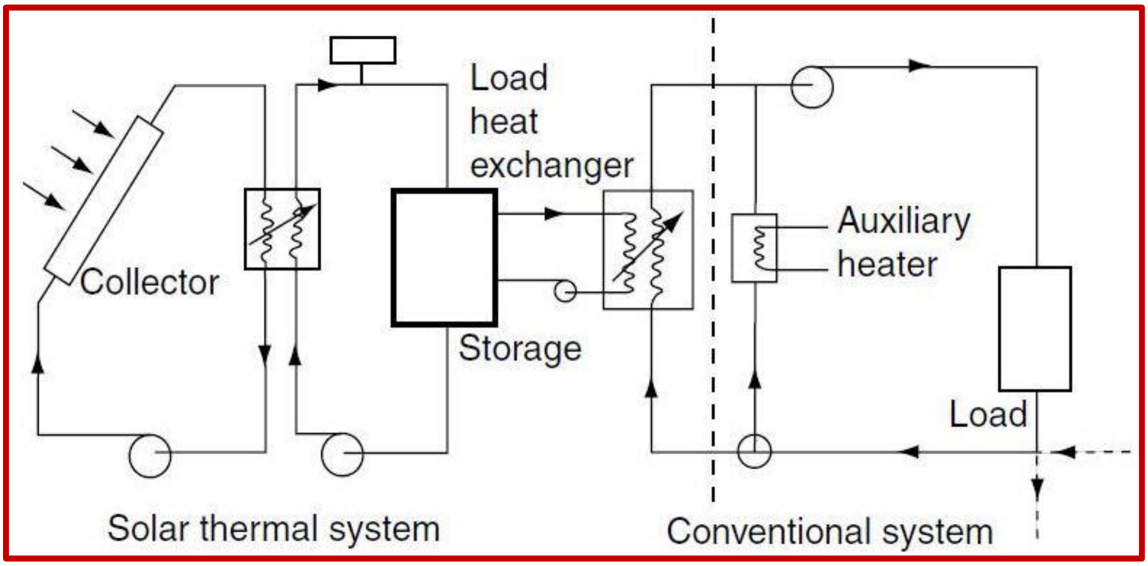

Figure 3:37 Schematic of a typical all-or-nothing type solar thermal system (Keith & Goswami, 2007)

The solar collector system designed for the glasshouse heating is different from these systems. The hot water generated by the solar collectors will be used to charge phase change material in storage tanks and use the stored energy when needed to heat the glasshouse. This is demonstrated in chapter four.

3.4.4.2 System design considerations

Solar systems are subject to extreme temperatures. Flat and evacuated solar collectors may experience temperatures ranging from - 20 °C on clear winter nights to over 200 °C and 350 °C respectively. This is possible if high solar radiation falling on the solar collector is continues without heat consumption. For example, stagnation temperatures without fluid flow but the active solar system designed for the glasshouse will avoid such a situation.

A high performance solar collector under significant pressure can capture enough heat and convert the circulating fluid into steam. Below are the issues that were considered when designing the solar water heating system for the research project:

- High temperatures situations
- Control of expansion and overpressure as a result of high temperatures
- Prevention of steam
- Protection from freezing of fluid in the collector or other parts of the solar primary circuit
- Control of legionella bacteria

3.4.4.2.1 High temperatures

Materials used in the solar collector system will experience whole range of temperatures and therefore should be able to withstand high temperatures especially during stagnation.

3.4.4.2.2 Control of expansion and overpressure as a result of high temperature

Expansion occurs within the fluid during normal and abnormal conditions. As a result, section 4.1.4 of BS EN 12976 [19] requires the solar system to be hydraulically secured.

The design of the solar system should be such that:

- Vapour or liquid at high temperature is released to the atmosphere under any circumstances.
- The end-user should check the system after stagnation before normal operation is resumed. Automatic resumption of normal operation should be avoided.

Expansion vessel with an internal membrane could be used to accommodate fluid expansion in fully pressurised sealed indirect system.

Alternatively, the fluid could be allowed to drain from the collectors into a drain-back vessel when the pump is switched off to isolate the heat source. This is the method to avoid expansion and overpressure for the solar active system.

At least one pressure relief valve will be needed in a fully-filled pressurised system to act as a safety mechanism. The solar collector system was designed to meet the requirements of the Pressure Equipment Regulations 2001[20].

Freezing

In an indirect solar primary circuit an anti-freeze along with corrosion inhibitors is needed to prevent freezing. In some systems expansion caused by freezing uses freeze-tolerant materials to accommodate it but the research design system is direct.

Control of legionella bacteria

There is risk of legionella bacteria for hot water held between 20 and 46 ^{o}C but the designed heating system for the glasshouse will be a close loop system to charge the phase change material (PCM) therefore the risk of legionella bacterial will be nil or minimal.

System sizing

The object of this project is to store enough solar thermal energy to meet 100 % heating demand of the glasshouse but it will be supported by waste heat from combined heat and power (CHP) plant currently installed to support the solar thermal system should the unpredicted happens.

Summary of solar thermal energy

The most inexhaustible and cleanest of all known energy sources is solar energy. Buildings can be passively heated using solar thermal energy through PCM and architectural design.

Solar water heating has been used in the past century and countries such as Denmark, Germany, UK, Japan and many more have installed hundreds of thousands of solar thermal energy systems.

Unglazed, glazed and evacuated tubes were the collectors considered when selecting solar collectors for the research project. The selection assessment was based on performance efficiency, cost, life expectancy and system application.

Flat plate collectors were selected because their efficiency at low to moderate temperature applications compare to evacuated tubes collectors are almost the same and will be convenient for this research building.

The most important factors in solar water heating using solar collectors are the temperature difference between water inlet to the collector system and the ambient.

In the UK it is almost uncertain to use solar thermal energy alone to heat a building. The research seeks to study the effectiveness of using PCM thermal energy storage techniques in heating a glasshouse but have made provision to use waste heat from CHP.

The maximum total annual solar irradiation for London which is close to Kew Gardens is 1045 kWh / m^2. The mean value of cold water temperature measurement case study of 112 dwellings in the UK observed the temperature to be15.2 ^oC with 95 % confidence interval at +- 0.5 ^oC.

This suggests that the initial cold water supply temperature to the hot water generation system will have a mean temperature of 15.2 ^oC.

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4 System design assessment and analysis

4.1 Introduction

The heating system was designed with the main focus to achieve zero carbon emission energy generating system. The integrated environmental solutions (IES) software tool was used to design the glasshouse to determine zone hourly, daily and monthly heating demand, energy requirement, temperature and relative humidity profiles.

This will assist to know the periods when thermal energy trapped in the glasshouse is not sufficient to warm the glasshouse and periods when the solar gains are sufficient to warm the glasshouse and store the excess heat gains trapped inside in the glasshouse into PCM heating pipes rather than venting it into the atmosphere.

4.2 Design simulation results

The IES software tool was used to determine zone monthly energy demand and the overall energy demand of the glasshouse.

Table 4:1, figure 4:1 and figure 4:2 demonstrate zones monthly space conditioning energy requirements and energy demand profiles throughout the design year.

Table 4:1 Zones monthly energy requirements of the glasshouse

Months	ZONE 1	ZONE 2	ZONE 3	ZONE 4	ZONE 5	ZONE 6	Zone 7	ZONE 8	ZONE 9	ZONE 10	ZONE 11	ZONE 12	ZONE 13	ZONE 14	Zone 15	ZONE 16	ZONE 17	ZONE 18	ZONE 19	ZONE 20	ZONE 21	**Zone total**
Jan 01-31	16.1	4.6	9.4	4.9	0.5	19.5	23.7	29.7	38.8	17.9	23.7	23.9	34.5	24.2	10.1	9.7	19.0	30.2	29.1	25.2	17.1	**411.7**
Feb 01-28	14.3	4.2	8.5	4.4	0.4	17.2	20.9	25.9	34.2	15.7	20.6	21.0	30.5	21.1	8.9	8.5	16.7	26.7	25.9	22.1	15.0	**362.8**
Mar 01-31	11.2	3.7	8.0	4.1	0.4	13.3	16.4	21.5	29.4	13.3	17.1	16.8	26.0	16.2	7.2	6.9	13.9	23.0	22.5	19.1	12.8	**302.4**
Apr 01-30	5.1	2.6	5.9	3.2	0.0	6.4	7.9	12.9	18.7	8.1	10.2	9.0	16.0	7.5	3.9	3.7	8.2	14.8	14.6	12.4	7.9	**179.0**
May 01-31	1.8	1.6	4.1	2.3	0.0	2.3	3.0	7.8	11.9	4.5	5.8	4.3	9.7	2.5	1.8	1.7	4.6	9.3	8.7	8.0	4.8	**100.3**
Jun 01-30	0.6	1.0	2.7	1.6	0.0	0.7	1.0	4.8	7.5	2.5	3.4	1.9	5.9	0.7	0.8	0.7	2.5	5.9	5.1	5.2	2.9	**57.3**
Jul 01-31	0.3	1.0	2.9	1.8	0.0	0.4	0.7	5.1	7.9	2.8	3.5	1.8	6.1	0.3	0.7	0.5	2.6	6.2	5.2	5.5	3.0	**58.4**
Aug 01-31	0.3	0.8	2.6	1.5	0.0	0.4	0.6	5.0	7.5	2.4	3.3	1.7	5.8	0.3	0.6	0.5	2.5	5.6	4.6	5.2	2.9	**54.3**
Sep 01-30	1.8	1.5	4.0	2.3	0.0	2.4	3.0	8.9	12.8	5.2	6.6	4.8	10.4	2.6	1.9	1.8	5.1	9.8	8.7	8.6	5.1	**107.1**
Oct 01-31	6.3	2.7	6.1	3.3	0.0	7.9	9.8	16.1	22.2	9.8	12.6	11.1	19.1	9.7	4.7	4.4	10.0	17.0	16.0	14.6	9.5	**212.9**
Nov 01-30	7.3	3.1	6.6	3.7	0.1	9.4	11.5	19.2	25.7	11.6	14.8	13.3	22.2	12.0	5.5	5.2	11.9	19.7	18.1	16.9	11.1	**248.7**
Dec 01-31	15.7	4.4	9.1	4.8	0.5	19.0	23.1	29.2	37.9	17.5	23.4	23.4	33.6	23.7	9.8	9.4	18.6	29.4	28.2	24.8	16.7	**402.1**
Summed total (MWh)	**80.7**	**31.1**	**69.8**	**37.7**	**1.9**	**99.0**	**121.6**	**186.0**	**254.4**	**111.3**	**144.8**	**133.0**	**219.8**	**120.9**	**55.9**	**52.9**	**115.5**	**197.5**	**186.6**	**167.6**	**108.8**	**2497.0**

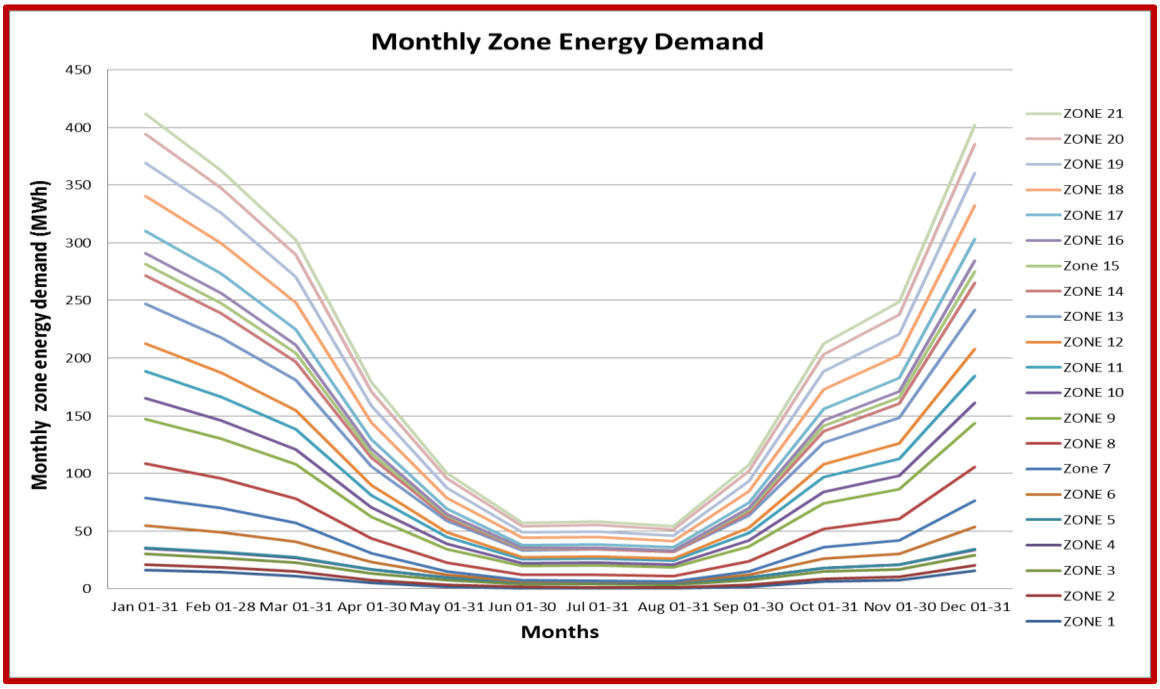

Figure 4:1 Illustrates monthly zones space conditioning energy demand profile

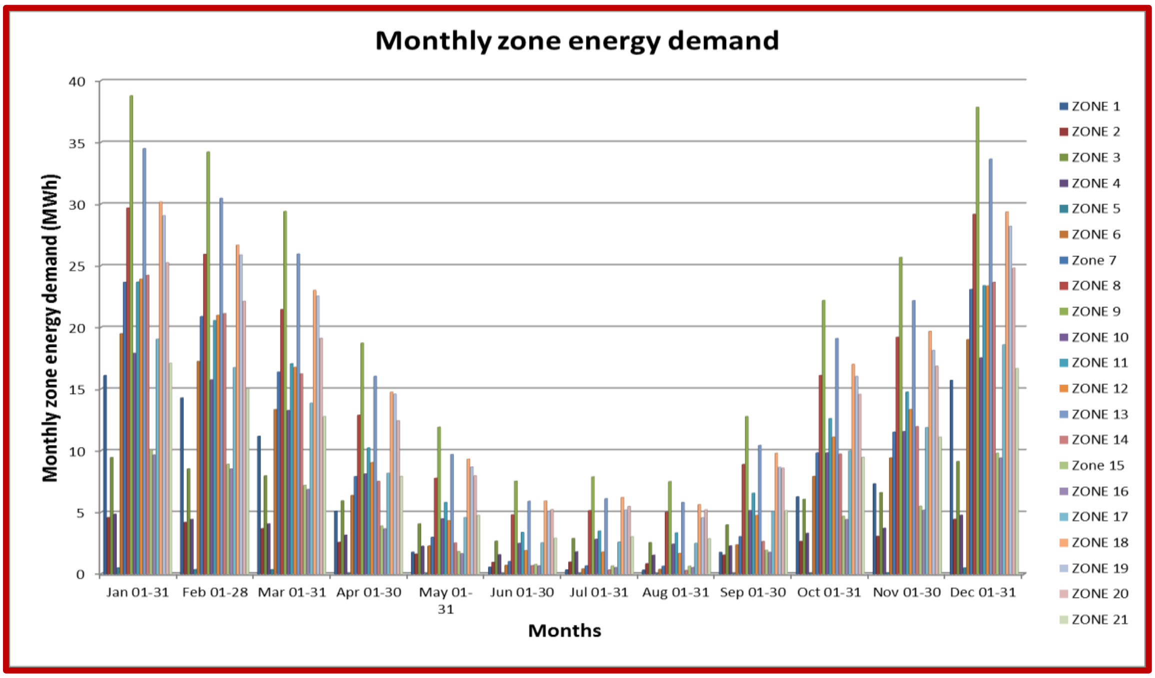

Figure 4:2 Illustrates monthly zones space conditioning energy demand

It could be seen from figure 4:1 and figure 4:2 that the heating energy demand from the middle of March starts to decrease as a result of solar gains therefore if the solar energy can be effectively stored then the demand for thermal energy generation using fossil fuel will be reduced. But the research heating system design will not consider the use of fossil fuel. One of the effective ways to achieve this is by using phase change material thermal storage techniques.

Table 4.2 below illustrates the space conditioning energy requirement and solar gains for all the zones throughout the year. The difference between the space conditioning energy demand and the solar gains is just five and half per cent (5.5%), therefore maximising the use of the solar gains in the glasshouse could reduce the amount of thermal energy generation to heat the glasshouse.

This is what the project seek to achieve by installing PCM filled heating pipes in the zone spaces so that excessive energy trapped inside the glasshouse could be absorbed by the PCM filled heating pipes and store for later use rather than to vent it out to the atmosphere.

Figure 4:3 below compares space conditioning heating energy requirement and solar gains. The two profiles is an illustration of how much energy is vented to the atmosphere from April to the end of September to maintain space temperature requirement.

Table 4:2 Space conditioning sensible energy and solar gains

Months	Space conditioning heating energy (MWh)	Solar gains (MWh)	Percentage difference between space conditioning heating energy and solar gains (%)
Jan 01-31	411.7	46.2	88.8
Feb 01-28	362.8	63.9	82.4
Mar 01-31	302.4	183.1	39.5
Apr 01-30	179.0	263.3	-47.1
May 01-31	100.3	376.9	-275.8
Jun 01-30	57.3	389.5	-579.9
Jul 01-31	58.4	303.9	-420.9
Aug 01-31	54.3	312.0	-475.1
Sep 01-30	107.1	204.4	-90.8
Oct 01-31	212.9	135.0	36.6
Nov 01-30	248.7	48.7	80.4
Dec 01-31	402.1	32.2	92.0
Summed total (MWh)	2497.0	2359.2	5.5

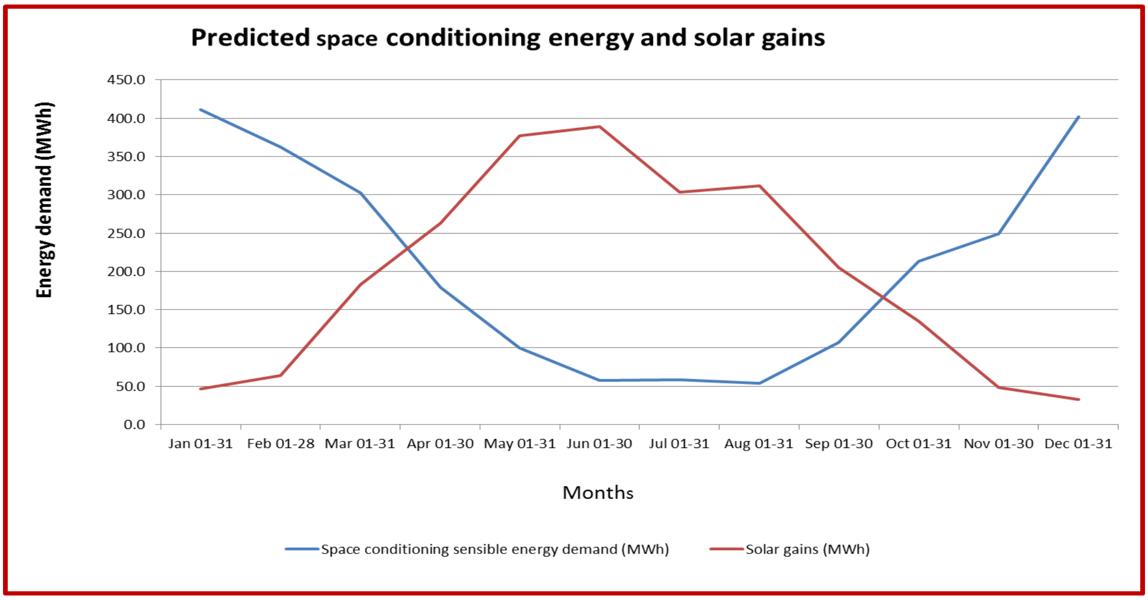

Figure 4:3 Comparison of predicted space conditioning heating energy and solar gains

During the preliminary study of the glasshouse, zone space temperatures and relative humidity were measured over three weeks and established that the space temperatures measured in most cases were higher than the outside temperature within the same measurement period. This could be supported by the study of Baille and Bouland of a glasshouse of 176m^2 ground floor area [1].

In their study, the February and March outside temperatures were 3.8 and 6.6 ^{o}C respectively whilst the space temperature for the same months and periods in the glasshouse were measured to be 10.9 and 13.5^{o}C respectively. Figure 4:4 below support the above study.

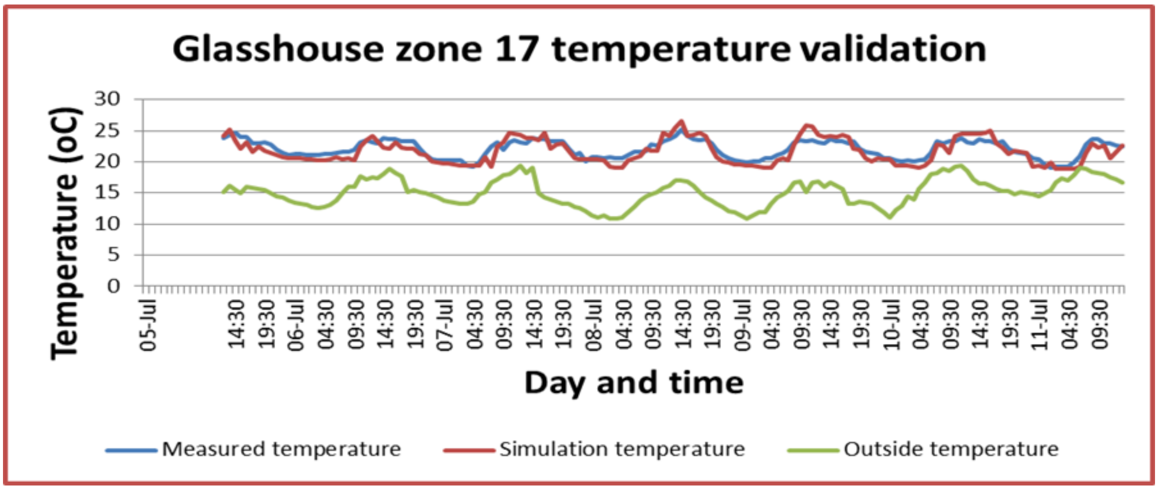

Figure 4:4 Zone 17 temperature validation (Measured values from 05 Jul to12 Jul 2012)

The temperature of zone 17 was measured from 05 July to the 12July of 2012 and it could be seen from figure 4:4 that the measured and simulation space temperatures are higher than the outside temperatures throughout the measurement period. This is a demonstration of how the space temperature of a glasshouse could be warmed from solar radiation due to greenhouse effect. There were no other internal gains during the measured period.

For green plants to grow in healthier condition, the environmental temperature and humidity requirement should be adequately maintained. The glasshouse and the heating system were designed to meet these criteria.

The plants produce respiration heat but this heat is latent and therefore not considered when assessing the space heating requirement of the glasshouse.

4.2.1 Kew Gardens current heating configuration

A site survey was carried out after the initial meeting with RBG to discuss the research aim, objectives and scope to establish the current heating system configuration and performance.

The current heating system in Kew Gardens that conditions the nursery glasshouse is a centralised system with primary and secondary circuits. The heat source is from eleven modular gas boilers that generate low temperature hot water (LTHW) to a common header and distributes to buildings and the nursery glasshouse. Figure 4:5 below shows the current heat system configuration.

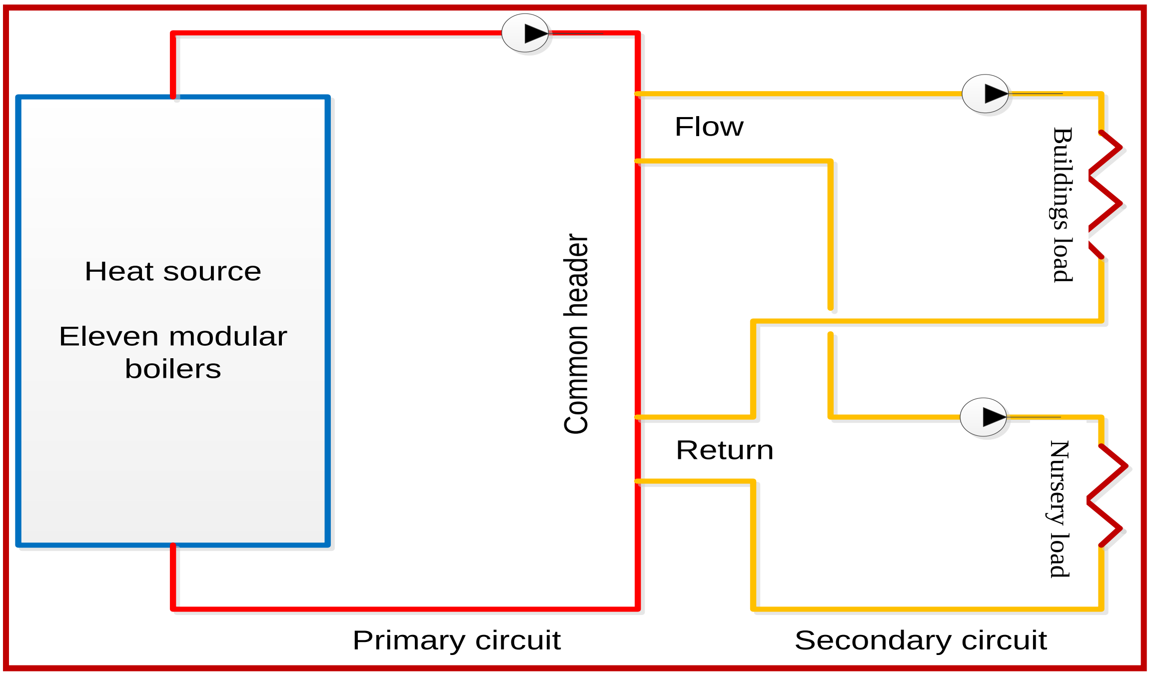

Figure 4:5 Kew Gardens sector heating system

The eleven modular gas fired boiler system consists of a series of Hamworthy boilers and the design operational principles are ranked in three categories as summarised and explained in

Table 4:3 below shows the details of the current boiler models and duty.

Table 4:3 Kew Gardens gas fired boiler plants details and assigned operational periods

Ranking	Model name	Quantity	Heating capacity	Total heating Capacity	Assigned operational periods
1	Warmwell W 140	5	140 kW	700 kW	Operate all year around
2	Purewell P 120	4	120 kW	480 kW	Operate in normal winter weather conditions (not extreme cold)
3	3Wimborne HE 650	2	650 kW	1300 kW	Operate in extreme cold weather conditions
Operating Temperatures Supply: 85 °C, Return: 75 °C					Seasonal efficiency: 80-85%

The heating system has three operating rankings and the first ranking system consist of five Warmwell W140 models of heating capacity 140 kW each which operates throughout the year.

The second ranking system consist of four (4) Purewell P 120 models of heating capacity 120 kW each and the third ranking system consist of two (2) Wimborne HE 650 of heating capacity 650 kW each. The heating system is sequence operated by starting the first ranking system followed by the second and third rankings respectively depending on the sector heating demand.

The nursery greenhouse operates throughout the year and modular gas boiler systems are used to satisfy the heating requirements. The type of heating system used is a hot water heating system with double piping network, both overhead and under the flowerbeds.

4.3 Glasshouse design space conditioning energy demand, temperature and relative humidity validation

Kew Gardens had insufficient data regarding the building floor plans, measured energy consumption and operational documents such logbooks. We had to carryout measurements and calculations to find out the actual dimensions of the glasshouse and its equivalent energy consumption for the design year.

Some of the boilers had sub-meters to measure gas energy consumption but thermal energy consumption of the buildings and the nursery are not metered so a calculation was needed to establish the consumption of the nursery glasshouse. The energy data available was from September 2011 to Jan 2012 which was calculated to be 1352 MWh for the glasshouse.

The space conditioning heating energy demand calculation from the IES simulation within the same period (September-January) was 1382 MWh, thus the difference between the measured energy consumption and the simulation results was 2.3 % as demonstrated in table 4.4 below.

To establish that the IES design model and simulation result is a representative of the glasshouse, the calculated or predicted energy consumption results, zones temperature and relative humidity profiles had to be validated against the measured values obtained from the glasshouse.

Table 4:4 compares Kew Gardens supplied measured energy consumption of the glasshouse with the simulation results and the percentage difference was 2.3 % which suggests that the modelling and simulation results of the glasshouse are accurate. This provided confident and certainty to proceed the research analysis using the calculated values. The annual energy consumption of the glasshouse was calculated using degree days and again compared to the simulation results.

Table 4:4 Space conditioning heating energy demand validation

Months	Simulation energy (MWh)	Energy consumption data supplied by Kew Gardens (MWh)	Percentage difference (%)
Sep	107.1369		
Oct	212.872		
Nov	248.7308		
Dec	402.0895		
Jan	411.6601		
Total	1382.4893	1352	- 2.3

4.3.1 Establishing the existing glasshouse annual and monthly energy consumption using degree days method

Degree days are a measure to determine the severity and period of the cold weather. It determines the rate at which heat is transferred through the building fabric.

The base temperature of degree days is the outside temperature above which would not require the heating system to operate and this is the central point to understand degree days. For most buildings in UK, 15.5 °C is used to calculate the degree days.

Heat is lost to the external environment in heated buildings in cold weather. Some of the heat loss is replaced by casual heat gains to the space from people, lights, equipment and solar gains whilst the remainder is supplied by the heating plant.

Because the casual gains contribute in heating the building causes rise in internal temperature. The outdoor temperature below which heating will not be required from the heating plant is the point where the casual gains equal the heat loss. This outdoor temperature will be the base temperature for the building and this is sometimes called the balance point temperature [2].

Heating will be required from the heating system when the outdoor temperature falls below the base temperature of the building. The degree-days are measured by the amount of times the outdoor temperature falls below the base temperature. These are the difference of sums between the base and outdoor temperatures.

4.3.1.1 Base temperature problems

Majority of UK energy professionals basically use 15.5 ^{o}C base temperature degree days in analysing building heating energy performance. This is because 15.5 ^{o}C base temperature historical degree days data are readily and freely available compare to other base temperatures.

The justification and argument of using 15.5 ^{o}C base temperature is that:

- Typically, buildings are heated around temperatures of 19 ^{o}C
- Internal heat gains contribute to raise space temperature on the average around 3.5 ^{o}C
- By subtracting the internal gains temperature of 3.5 ^{o}C from the typical building heated temperature of 19 ^{o}C results to a base temperature of 15.5 ^{o}C. This is the temperature that the heating system effectively requires to heat the building to and the 3.5 ^{o}C supplemented by internal heat gains.

The approximation method cannot be completely rule out but the point is that different buildings are heated to different temperatures. Offices for example are recommended to be heated to 19 ^{o}C but are heated several degrees warmer in practice.

Buildings average internal heat gains varies from building to the other depending on the occupancy density, equipment used and number, lighting levels and other factors.

This suggests that building base temperature of 15.5 ^{o}C in practice will not be the appropriate base temperature to use for degree days calculation.

Analysing building energy performance or calculating consumption based on degree day method can greatly be affected using inappropriate base temperature to calculate the degree days.

Appropriate base temperature of a building can be established by using linear regression analysis to find the best fit correlation graph of the energy consumption and the degree days produced by the estimated base temperature.

This is appropriate where the energy consumption is the actual and could be validated. Where the measured energy consumption data is actual and validated, then the root square (R^2) method can be used in determining the building heating base temperature.

The importance of the root square (R^2) value in building energy consumption analysis is basically a measure to determine the relationship or correlation between the energy consumption and the degree days.

For any particular building, the base temperature of the degree days makes a big difference to how well they correlate to the energy consumption.

Calculating or estimating the correct base temperature accurately is difficult for a particular building using logic alone. At times it is best to make a rough estimate for example, using 15.5 ^{o}C base temperature and try correlating the energy consumption (kWh) with calculated degree days to the various base temperatures around 15.5 ^{o}C.

This assist to decide which base temperature is appropriate for the building to use in calculating the degree days to analysis the energy consumption.

The simulation and calculated energy consumption based on supplied energy consumption data by Kew Gardens of the designed glasshouse using the IES software tool from September 2011 to January was 1382 and1352 MWh respectively.

The 1352 MWh was the five months (September 2011 to January 2012) energy consumption supplied by Kew Gardens with no monthly energy consumption break down data.

The annual energy consumption of the glasshouse was not known and it was important to know and compare with the annual IES simulated glasshouse heating energy demand.

There are several ways that this could be done but the best and the most efficient method is using heating degree days as the energy consumption is weather dependant and five months energy consumption data is known.

London Heathrow heating degree days (HDD) data with base temperatures ranging from 12.5 to 18.5 ^{o}C was used in the assessment and analysis. The distance from Heathrow to Kew Gardens in a straight line is seven miles calculated using Google map which suggests that Kew Gardens is within the same location as Heathrow.

Table 4.5 below shows London Heathrow Airport (0.46W,51.48N) degree days data from the period September 2011 to August 2012 ranging from 12.5 to 18.5 base temperatures. The degree days data was obtained from www.degreeday.net [3]

Table 4:5 London Heathrow Airport degree days data from September 2011 to August 2012

	Base temperatures (oC)												
Month	**12.5**	**13**	**13.5**	**14**	**14.5**	**15**	**15.5**	**16**	**16.5**	**17**	**17.5**	**18**	**18.5**
Sep-11	8	9	13	16	21	26	33	39	48	56	66	77	88
Oct-11	37	42	50	57	66	74	85	95	108	120	134	147	162
Nov-11	78	89	102	116	130	144	159	174	188	203	218	233	248
Dec-11	170	185	200	216	231	247	262	278	293	309	324	340	355
Jan-12	184	199	215	230	246	261	277	292	308	323	339	354	370
Feb-12	225	239	254	268	282	297	311	326	340	355	369	384	398
Mar-12	121	134	147	160	174	187	202	216	230	245	260	274	290
Apr-12	122	135	149	164	178	193	208	222	237	252	267	282	297
May-12	49	56	64	72	82	91	102	112	124	136	148	161	173
Jun-12	11	15	21	26	34	41	50	58	68	78	90	101	114
Jul-12	2	3	5	7	10	14	20	26	34	42	53	63	75
Aug-12	2	3	4	4	7	9	12	16	21	27	33	41	49

Let us assume that the glasshouse base temperature is 15.5 ^{o}C and the 1352 MWh energy consumption was based on 15.5 ^{o}C degree days and try correlating the energy consumption with the degree days to the various base temperatures around 15.5 ^{o}C. This will assist to determine which base temperature is appropriate for the glasshouse to use in calculating the degree days to calculate the annual and monthly energy consumption.

Table 4.6 below demonstrates energy consumption from September 2011 to January 2012 of the glasshouse using the various base temperatures and degree days around 15.5 ^{o}C base temperature.

The total heating degree days of 15.5 ^{o}C base temperature is 816 with the assumed energy consumption of 1352 MWh.

Using the same degree days principle calculation for example, resulted energy consumption of 790 MWh using 12.5 ^{o}C base temperature with 477 heating degree days. Similarly, the energy consumption of 18 ^{o}C base temperature with 1151 heating degree days within the same period from September 2011 to January 2012 is 1791 MWh.

Table 4:6 Calculated energy consumption using various base temperatures around 15.5 ^{o}C base temperature and the degree days

	Base temperatures (oC)												
Month	**12.5**	**13**	**13.5**	**14**	**14.5**	**15**	**15.5**	**16**	**16.5**	**17**	**17.5**	**18**	**18.5**
Sep-11	8	9	13	16	21	26	33	39	48	56	66	77	88
Oct-11	37	42	50	57	66	74	85	95	108	120	134	147	162
Nov-11	78	89	102	116	130	144	159	174	188	203	218	233	248
Dec-11	170	185	200	216	231	247	262	278	293	309	324	340	355
Jan-12	184	199	215	230	246	261	277	292	308	323	339	354	370
Total degree days	477	524	580	635	694	752	816	878	945	1011	1081	1151	1223
Sep 2011-Jan 2012 energy used (MWh)	790	868	961	1052	1150	1246	**1352**	1467	1455	1566	1675	1791	1907

The annual energy consumption of the glasshouse will be calculated based on the five months calculated results from September 2011 to Jauary 2012 in table 4.6.

The five months energy consumption using 15.5 ^{o}C base temperature of 816 degree days is 1352 MWh. The total anuual heating degree days based on 15.5 ^{o}C base temperature is 1721 and its equivalent energy consumption is 2851MWh. Similarly the total annual heating degreedays of 12.5 ^{o}C base temperature is 1009 and its equivalent energy consumption is 1672 MWh.

Table 4:7 demonstrates annual total degree days and energy consumption of the base temperatures from 12.5 to 18.5 ^{o}C.

Table 4:7 Total annual heating degree days and energy consumption of base temperatures from 12.5 to 18.5 ^{o}C

	Base temperatures (oC)													
Month	**12.5**	**13**	**13.5**	**14**	**14.5**	**15**	**15.5**	**16**	**16.5**	**17**	**17.5**	**18**	**18.5**	Monthly energy used (MWh)
Sep-11	8	9	13	16	21	26	33	39	48	56	66	77	88	35
Oct-11	37	42	50	57	66	74	85	95	108	120	134	147	162	109
Nov-11	78	89	102	116	130	144	159	174	188	203	218	233	248	215
Dec-11	170	185	200	216	231	247	262	278	293	309	324	340	355	383
Jan-12	184	199	215	230	246	261	277	292	308	323	339	354	370	408
Feb-12	225	239	254	268	282	297	311	326	340	355	369	384	398	467
Mar-12	121	134	147	160	174	187	202	216	230	245	260	274	290	288
Apr-12	122	135	149	164	178	193	208	222	237	252	267	282	297	295
May-12	49	56	64	72	82	91	102	112	124	136	148	161	173	136
Jun-12	11	15	21	26	34	41	50	58	68	78	90	101	114	56
Jul-12	2	3	5	7	10	14	20	26	34	42	53	63	75	17
Aug-12	2	3	4	4	7	9	12	16	21	27	33	41	49	12
Total annual degree days	1009	1109	1224	1336	1461	1584	1721	1854	1999	2146	2301	2457	2619	
Sep 2011-Aug 2012 energy used (MWh)	1672	1837	2028	2214	**2421**	2624	2851	3098	3072	3312	3556	3812	4071	
Percentage difference (%)	33	26	19	11	**3**	-5	-14	-24	-23	-33	-42	-53	-63	

The difference in energy consumption between the IES simulation results and the glasshouse supplied energy consumption of the glasshouse from September 2011 to January 2012 was 2.3%. The difference in annual energy consumption of the simulated result and the actual energy consumption of the glasshouse should be 2.3% or very close to 2.3%.

From table 4:7 results, the base tempertaure with the closet annual energy consumption is 14.5 ^{o}C with equivalent energy consumption of 2421 MWh which is 3% different compare to the simulated annual energy consumption of 2497 MWh. This suggests that the glasshouse heating base temperature is 14.5 ^{o}C.

Fingures 4:6 and 4:7 below show monthly energy consumption versus degree days and monthly energy consumption of the glasshouse from September 2011 to August 2012 respectively.

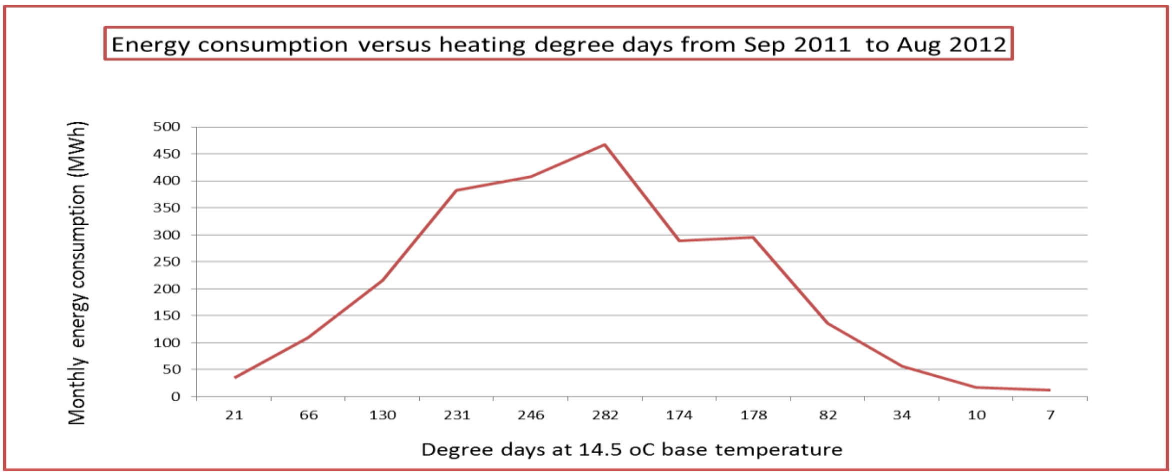

Figure 4:6 Glasshouse energy monthly consumption versus degree days

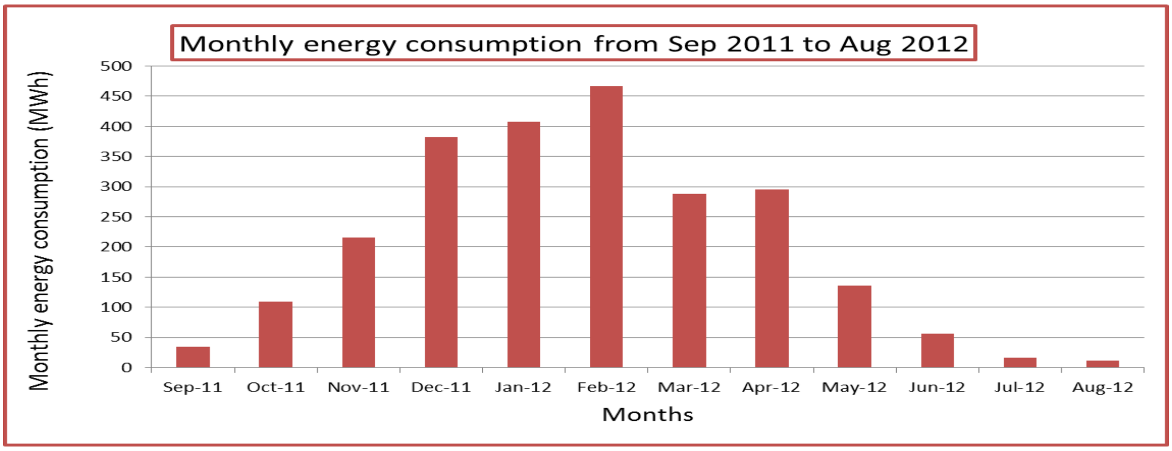

Figure 4:7 Glasshouse monthly energy consumption from Sep 2011 to Aug 2012

The above analysis stresses the importance of degree days application in assessing and analysing building heating energy consumption performace.

4.3.2 Nearly zero carbon emission of the designed heating system calculation

CIBSE Guide F, Energy Efficiency in Buildings section 11.2 motor sizing and selection suggests that installed motor sizes are oversized and always operates at two thirds or less of its rated output.

Hosni et al [4] found that the ratio of heat gain to name plate power ranged from 25% to 50% and advised that the most accurate ratio was 25% which also suggest that the actual operating power of a motor is less compare to the size on the name plate.

Building services equipment such as fans, pumps operate most of the time at part load but they are sized based on peak load and makes them inefficient.

The relationship between flow and the speed of the pump is such that the input power reduces in a cube law relationship with the speed reduction and typically less than 20% of full volume input energy is required at 50% flow [5].

Figure 11.5 of the CIBSE Guide F illustrates the savings of a variable speed drive heating pump in a typical office operation of ten hours a day of an average energy savings of 66.9 %.

This suggests that more savings is possible for a pump operating at longer hours more than ten hours.

The selected pump motor size for the research project is rated at 1,002 W (1KW). If we assume that the maximum operating power is 70 % of the rated manufacturer's power then the input power of the pump will be 0.7 kW. The energy consumption of the pump per annum operating at 8,760 hours will be 6,132 kWh.

If we assume that the energy savings of the variable speed drive pump is 60%, then the actual energy consumption will be 2,453 kWh (0.4 x 6,132 kWh) per annum.

Three pumps are required for the project so the total energy consumption of the pumps per annum will be 6,133 kWh (2,453 x 0.5 + 2,453 x 2). The pump for the CHP circuit will operate for six months in a year.

The annual energy consumption of the glasshouse is calculated to be 2,421 MWh. The CO_2 emission of the pumps compare to the heating energy consumption is 0.7 % [(6,133 x 0.494 / 2,421,000 x 0.186) x100] which is insignificant. The CO_2 conversion factors of electricity and natural gas were taken as 0.494 and 0.186 kgCO_2 / kWh respectively.

The designed research heating system could be said to nearing zero CO_2 emission.

4.3.3 Existing and proposed research heating system energy efficiency comparison

Heating energy efficiency could be assessed or determined based on several factors or criteria but for the purpose of this report the factors detailed in table 4:8 will be used in assessing the existing and the proposed heating system energy efficiency performance.

Table 4:8 Existing and proposed research heating system energy efficiency comparison

Factors or criteria	Description of the factors	Existing heating system	Research heating system
Heat source	Boilers	Multiple boilers generate hot water	Solar energy
Fuel	Fuel used by boilers	Natural gas	Sun energy
CO2 emission	Fuel combustion generate CO_2 emission	Annual CO_2 emission of 450.4 tonnes CO_2	0.000086 tonnes CO_2 for pump energy used
Supply temperature	High hot water supply and return temperature	Supply and return temperature of 82 and 71 ^{o}C respectively	Supply and return temperature of 32 and 25 ^{o}C respectively
Heat losses	Distribution and standing losses	High distribution and boiler standing losses	Low distribution and PCM storage tank losses
Maintenance	Maintenance of boilers, burners, pressurisation units and others	The cost of maintenance and parts replacement is high	Very minimum maintenance required
Seasonal efficiency	Seasonal efficiency is achieved through actual operations at different loading	The seasonal efficiency is greatly affected by boiler part load efficiency and heat losses when boiler is off.	The proposed heating system is not affected by part load efficiency
Sequence control	The multiple boilers are controlled sequentially	The efficiency of the multiple boilers operation is greatly affected by the effectiveness of the sequence control	The operational performance is not affected by sequence control and no energy losses

Life expectancy	The expected life expectancy of the boiler is between 15 to 30 years	The boilers used in generating the hot water could last between 15 to 30 years	The solar collectors could last over 25 years
Boiler cycling	System running and boiler cycling unnecessarily	Boiler cycling with inappropriate operating temperature	No boiler cycling to waste energy
Zones temperature control	Each zone has different temperature requirement	Zones temperature control is difficult to achieve and waste energy	Zone temperature could be easily achieved as the PCM heating pipes only discharges heat when the space temperature falls below PCM phase change temperature
Hot water distribution pumps	The pumps distribute hot water to the zones	The existing heating system uses constant flow rate pumps irrespective of building heating load wasting energy	The pump will be variable speed pumps saving 20-50 % of pump energy (CIBSE guide H)
Space temperature control	The space temperature should be maintained to space requirement	The existing heating system uses space temperature sensor to maintain zone temperature is affected by positioning, re-calibration and others	The space temperature is maintained by melting and freezing of the PCM heating pipes
Environmental impact	The pollution as a result of burning fossil fuel affects the environment	The annual pollution impact to the environment is high	The environmental impact is negligible

The reasearch heating system is the best and efficient alternative system comapared to the current system.

4.3.4 Zones temperature validation

Seven Hobo data loggers were supplied from Brunel University to assist measure the temperature and humidity profiles in the glasshouse. The technical specifications of the data logger are summarised in table 4.9 below and figure 4.8 shows the Hobo data logger.

Table 4:9 HOBO U12-012 Technical Specifications

Name/Model	HOBO U12-012 Temperature/RH/Light/External Data logger
Measurement range	• Temperature: -20° to 70 °C • Relative Humidity: 5% to 95% RH
Accuracy	• Temperature: ± 0.35°C from 0° to 50°C • Relative Humidity: ±2.5% from 10% to 90% RH (typical), to a maximum of ±3.5%
Resolution	• Temperature: 0.03°C at 25°C • Relative Humidity: 0.03% RH
Drift	• Temperature: 0.1°C/year • Relative Humidity: <1% per year (typical)
Response time (in airflow of 1 m/s)	• Temperature: 6 minutes , typical to 90% • Relative Humidity: 1 minute, typical to 90%

Figure 4:8 HOBO U12-012 Data Logger

The glasshouse has twenty-one zones and three sets of weekly measurements were carried out to obtain a weekly indoor temperature and relative humidity (RH) profile for each zone. The data loggers were setup to log values at 10 minute intervals and the logged data were exported into excel spread sheet to create corresponding graphs and compare with the simulated results from the IES software tool. Every data logger was located approximately in the middle of the room for each zone. This was the appropriate position to measure the temperature and the humidity in the zone.

This was a good practice to protect the data loggers from receiving direct sunlight to minimise measurement errors.

The three weeks measurement period was found to be satisfactory and representative for summer, winter and other transitional seasons as the profiles were proportionately influenced by the space heat gains and outside weather conditions.

The investigation established that large amount of heat energy is vented to the atmosphere wasted. The heating system was therefore designed to absorb and store the waste thermal energy using phase change material (PCM) thermal energy storage techniques.

The timetable of the measurements are summarised in table 4:10 below.

Table 4:10 Schedule of the measurements

Zones	Duration of the measurements
1-7	21-June – 28-June
8-14	28-June – 05-July
15-21	05-July – 12-July

For the purpose of this report 18 zones out of the 21 zones temperature validation are illustrated as they all look similar. Zone 1 and zone 18 is shown below in figure 4:9 and figure 4:10 respectively and all the other sixteen zones are demonstrated in Appendix A. The validation period was measured for one week for example, from 21- 28 June 2012 as shown in figure 4:9 below.

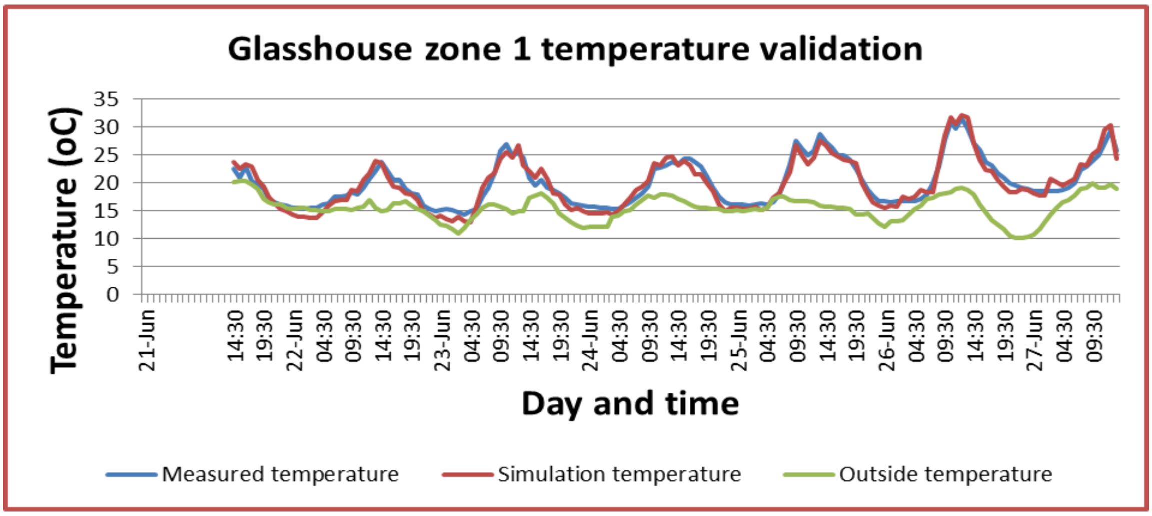

Figure 4:9 Zone 1 temperature validation from 21- 28 June

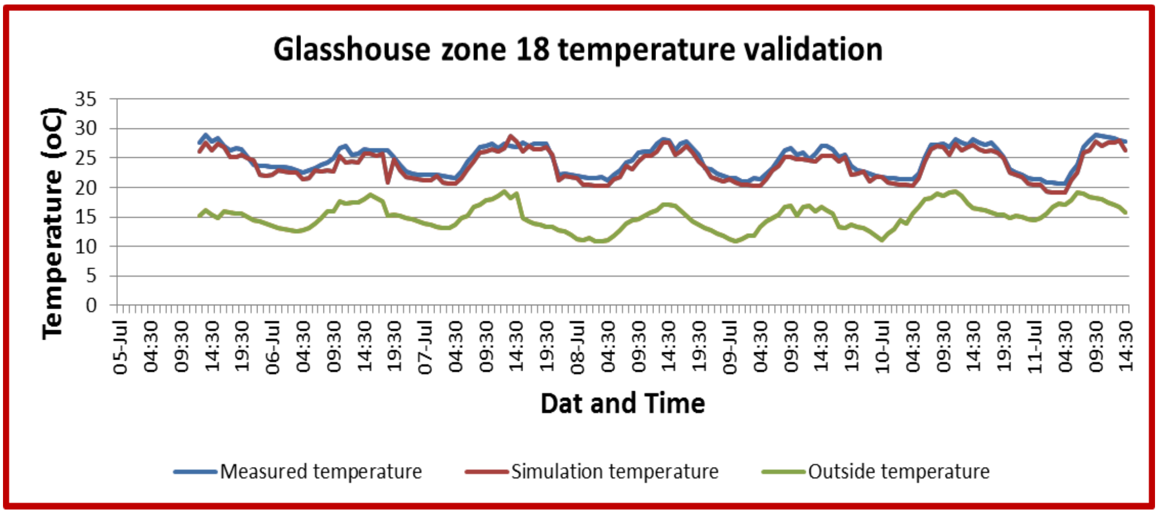

Figure 4:10 Zone 18 temperature validation from 05-12 July

4.3.5 Relative humidity validation

Relative humidity validation graphs were plotted to validate the results obtained from the IES simulation calculation and the measured values obtained from Kew Gardens.

The graphs also demonstrate the correlation or the relationship of the measured values, simulation results and the outside relative humidity obtained from the IES software results. The measured and the simulated relative humidity correlate well.

The outside relative humidity follows similar pattern of the measured and simulated values but inside the glasshouse humidity is dictated by the plants requirements.

For the purpose of this report four validated zones are demonstrated. Zone 1 and 2 are demonstrated in figure 4:11 and figure 4:12 below and zone 3 and 4 are shown in appendix B.

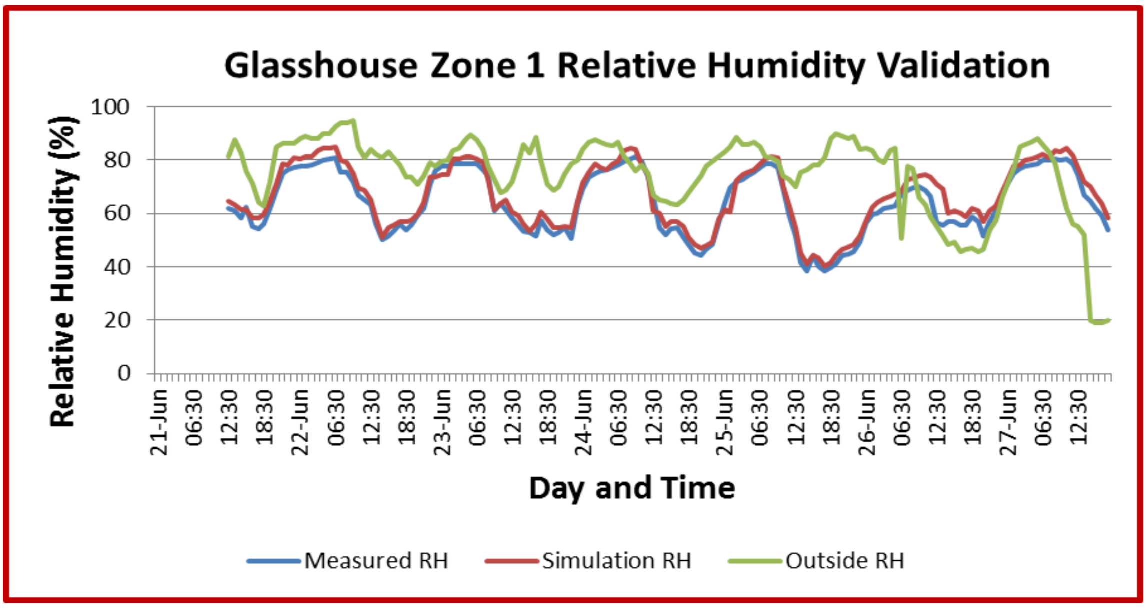

Figure 4:11 Zone 1 Relative humidity validation from 21-28 June

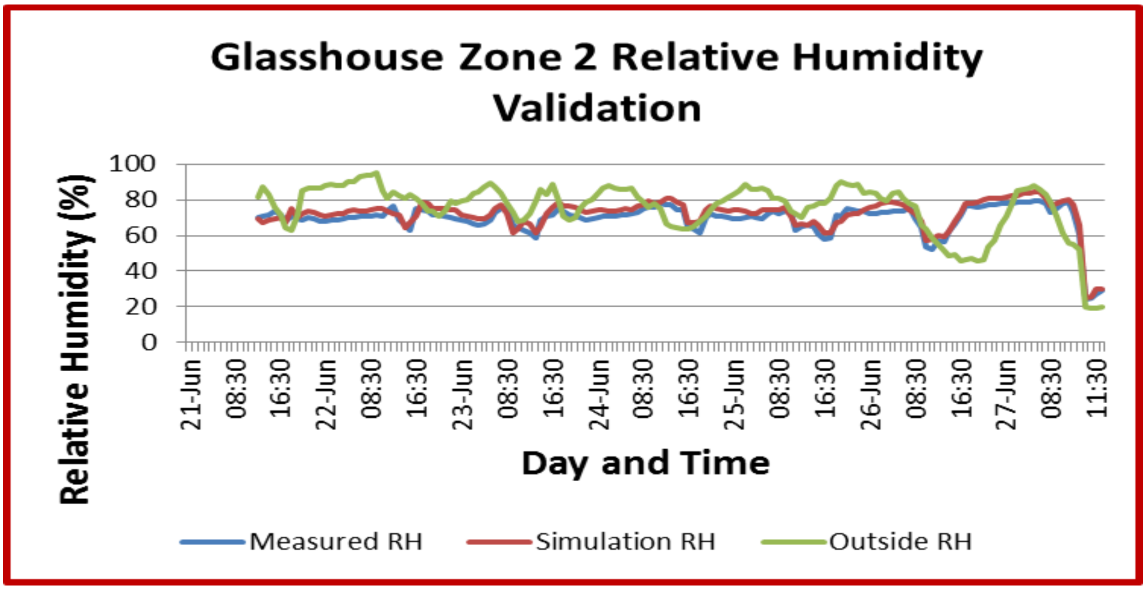

Figure 4:12 Zone 2 Relative humidity validation from 21-28 June

4.4 Configuration of the proposed glasshouse heating system

Figure 4:13 is the proposed heating system of the glasshouse using both active and passive systems. A single zone is used to illustrate the design principles of the heating system. The active system will be using solar collectors to generate thermal energy to produce low temperature hot water (LTHW), store the thermal energy in the PCM storage tanks and use it when there is demand for heating from the glasshouse.

4.4.1 Active system description

Two types of PCM storage tanks have been selected with phase change temperatures of 44 and 34 ^{o}C. The reason for selecting the two different storage thanks of 44 and 34 ^{o}C phase change temperatures is to extract as much energy as possible from the solar collector system.

Hot water will be generated between 50 ^{o}C and 35 ^{o}C by regulating the water flow rate through the solar collectors to match the solar radiation input to the system. The working fluid in the solar collector system will be drained into the drain back vessel when the return hot water temperature from the solar collectors is 30 ^{o}C to prevent the water from freezing in cold weather conditions or when insufficient solar radiation is falling on the collectors.

The system has been designed such that the little thermal energy generated by the solar collector system could be useful.

Table 4:11 below is used to demonstrate more practically why 44 and 34 ^{o}C PCM are selected for the project. For example, if the return hot water temperature from the solar collectors is 45 ^{o}C, it will first enter the storage tank with PCM melting point temperature of 44 ^{o}C to charge it a bit and leave around 44 or 43 ^{o}C and enter the 34 ^{o}C melting point temperature and exit from 34 ^{o}C PCM storage tank and supply to the zones at 32 ^{o}C to heat the glasshouse.

Table 4:11 illustrates the predicted power to be generated by the 803 solar collectors in 15 May 2010 to be 737 kW. The output water temperature with a nominal flow rate of 9 kilograms per second (9 kg/s) will be 45 ^{o}C.

The manufacturer of the selected solar collector (Agena SA Energies) gives the maximum, nominal and minimum flow rates of a collector as 200, 40 and 25 litres per hour respectively. The maximum, nominal and minimum flow rates of the 803 collectors selected is calculated to be 45, 9 and 6 litres per second respectively which are approximately 45, 9 and 6 kilograms per seconds (kg/s).

Table 4:11 shows predicted solar power generation, hot water flow rates and output hot water temperatures under nominal and minimum flow rates. The design inlet hot water temperature to the solar hot water system of 803 solar collectors is 25 ^{o}C with expected output temperature of 50 ^{o}C. The output temperature depends on the solar radiation input into the solar collector system and the hot water flow rates and this is demonstrated in table 4:11 below.

In another example, the hot water return temperature from the collectors is 36 ^{o}C, with the minimum flow rate of 6 kg/s, the thermal energy could still be absorbed by the 34 ^{o}C PCM storage thank maximising the use of the thermal energy generated by the solar collector system.

This method will assist to extract maximum energy as possible from the heat absorbed by the solar collector system. Without the 34 ^{o}C PCM storage tank not enough energy could have been extracted for the hot water return temperatures of 45 and 36 ^{o}C from the solar collectors.

The hot water supply temperature to the glasshouse has been kept low to increase the solar hot water system efficiency and again lower distribution losses through the pipework.

Table 4:11 Predicted solar power generation, hot water flow rates and output hot water temperatures under nominal and minimum flow rates.

Date	Daily Mean temp (oC) (0900-0900)	Daily Total global radiation (KJ/m2)	FR(τα)	FR UL	Ti (oC)	Gcoll MJ/m2	Collector aperture area (m2)	Collector energy delivery (kWh/day)	No of collectors	Collectors power generated per day (KW/day)	Hot water mass flow rate (kg/s)	output temp (oC)
01/05/2010	11.4	13879	0.85	4.12	25	11.73	2.5	8.02	803	537	9	39
02/05/2010	7.1	5192	0.85	4.12	25	4.33	2.5	2.97	803	198	9	30
03/05/2010	6.9	16182	0.85	4.12	25	13.66	2.5	9.35	803	626	9	42
04/05/2010	6.9	14426	0.85	4.12	25	12.17	2.5	8.33	803	557	9	40
05/05/2010	11.0	13971	0.85	4.12	25	11.80	2.5	8.08	803	540	9	39
06/05/2010	9.9	17119	0.85	4.12	25	14.47	2.5	9.90	803	663	9	43
07/05/2010	9.0	11479	0.85	4.12	25	9.68	2.5	6.62	803	443	9	37
08/05/2010	8.2	4316	0.85	4.12	25	3.60	2.5	2.46	803	165	9	29
09/05/2010	8.6	7195	0.85	4.12	25	6.04	2.5	4.13	803	277	9	32
10/05/2010	8.4	18311	0.85	4.12	25	15.48	2.5	10.59	803	709	9	44
11/05/2010	5.7	10906	0.85	4.12	25	9.18	2.5	6.28	803	420	9	36
12/05/2010	6.4	16899	0.85	4.12	25	14.27	2.5	9.76	803	653	9	42
13/05/2010	8.5	17328	0.85	4.12	25	14.64	2.5	10.02	803	670	9	43
14/05/2010	11.1	17770	0.85	4.12	25	15.03	2.5	10.28	803	688	9	43
15/05/2010	**11.0**	**19028**	**0.85**	**4.12**	**25**	**16.10**	**2.5**	**11.01**	**803**	**737**	**9**	**45**
16/05/2010	10.1	11586	0.85	4.12	25	9.78	2.5	6.69	803	448	9	37
17/05/2010	11.4	21374	0.85	4.12	25	18.09	2.5	12.38	803	828	9	47
18/05/2010	12.2	16448	0.85	4.12	25	13.91	2.5	9.52	803	637	9	42
19/05/2010	14.6	17850	0.85	4.12	25	15.11	2.5	10.34	803	692	9	43
20/05/2010	16.9	11974	0.85	4.12	25	10.13	2.5	6.93	803	464	9	37
21/05/2010	17.1	21671	0.85	4.12	25	18.37	2.5	12.57	803	841	9	47
22/05/2010	16.8	26785	0.85	4.12	25	22.71	2.5	15.54	803	1040	9	53
23/05/2010	19.7	27075	0.85	4.12	25	22.96	2.5	15.71	803	1051	9	53
24/05/2010	19.1	27380	0.85	4.12	25	23.22	2.5	15.89	803	1063	9	53
25/05/2010	13.9	24997	0.85	4.12	25	21.18	2.5	14.49	803	970	9	51
26/05/2010	12.4	17216	0.85	4.12	25	14.56	2.5	9.96	803	667	9	43
27/05/2010	11.8	14026	0.85	4.12	25	11.85	2.5	8.11	803	543	9	39
28/05/2010	14.0	21473	0.85	4.12	25	18.19	2.5	12.44	803	833	9	47
29/05/2010	**13.9**	**7088**	**0.85**	**4.12**	**25**	**5.97**	**2.5**	**4.09**	**803**	**273**	**6**	**36**
30/05/2010	14.0	21700	0.85	4.12	25	18.38	2.5	12.57	803	841	9	47
31/05/2010	**13.2**	**7962**	**0.85**	**4.12**	**25**	**6.71**	**2.5**	**4.59**	**803**	**307**	**6**	**37**

4.4.2 Passive system description

The passive system includes PCM filled heating pipes and black painted pipes as illustrated in figure 4:13 below.

Two PCM filled heating pipes of phase change temperatures of 25 and 22 ^{o}C have been selected based on the space temperature set point. The zone illustrated in figure 4:13 has a space set point temperature of 20 ^{o}C. The supply hot water temperature of 32 ^{o}C will enter the 25 and 22 ^{o}C melting point temperature PCMs and leave around 25 ^{o}C.

The exit temperature of 25 ^{o}C will be met by regulating the flow through the motorised valves and exit temperature sensors installed at the end of the PCM heating pipes.
In the initial design plan, the exit water from the PCM heating pipes will enter the black painted pipes which will preheated the water prior to the entry of the solar collector

systems at around 27 ^{o}C but this has been excluded from the design for further research study as detail study is required and limited time of the research will not allow it.

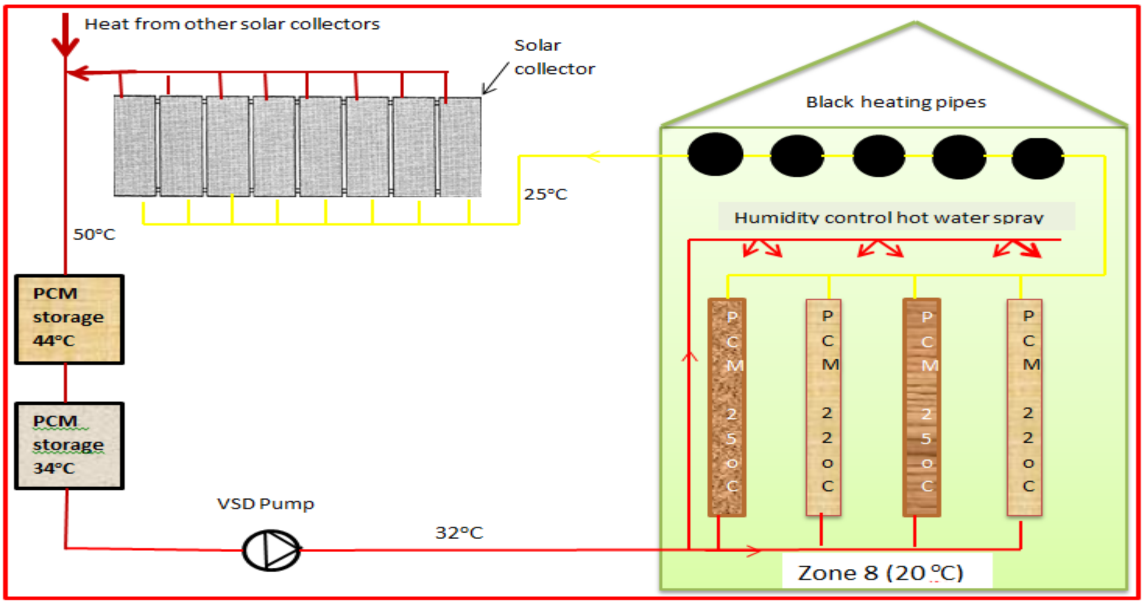

Figure 4:13 The proposed heating system of the glasshouse using both active and passive systems

Summary of system configuration and operational principles:

- The active solar energy system will use solar collectors to generate thermal energy and store it in the 44 ^{o}C and 34 ^{o}C melting points temperature PCM storage tanks
- The passive solar energy system will comprise of PCM filled heating pipes with 25 ^{o}C, 22 ^{o}C and 15 ^{o}C melting points. The PCMs will discharge thermal energy at temperatures below 25 ^{o}C, 22 ^{o}C and 15 ^{o}C and will be charged at temperatures above 25 ^{o}C, 22 ^{o}C and 15 ^{o}C.

Figure 4:14 demonstrates the schematic diagram of the glasshouse heating system. The zones have been categorised or grouped into two sectors to simplify the selection of the phase change material of the heating pipes based on space set point temperatures, PCM melting point temperatures and system operational effectiveness.

Zones with space set point temperatures from 6.5 to 15 °C are categorised into one sector and zones with space set point temperatures above 15 to 21 °C are also categorised into another sector.

The following are how the zones are categorised into sectors:

Sector 1	Zone 1	Zone 5	Zone 6	Zone 7	Zone 14	Zone 15	Zone 16
Set point temperature (°C)	9	6.5	11	10	15	14	13

Sector 2	Zone 2	Zone 3	Zone 4	Zone 8	Zone 9	Zone 10	Zone 11
Set point temperature (°C)	17	20	20	21	21	20	19

Sector 2 continuation	Zone 12	Zone 13	Zone 17	Zone 18	Zone 19	Zone 20	Zone 21
Set point temperature (°C)	19	20	18	20	20	21	21

In sector one (1) the supply hot water temperature of 32 °C will enter the 22 °C melting point temperature PCM and leaves around 25°C and at the same time enter the 15 °C melting point temperature PCM and also leaves around 25 °C to the solar collector system.

The operational principle in sector two is similar to sector 1. Hot water temperature sensors will be installed at strategic points to measure the flow temperatures and regulate the flow to achieve the entry and exit temperatures described above. In figure 4:14 below, the first row is sector one zones and the second and third rows are sector two zones.

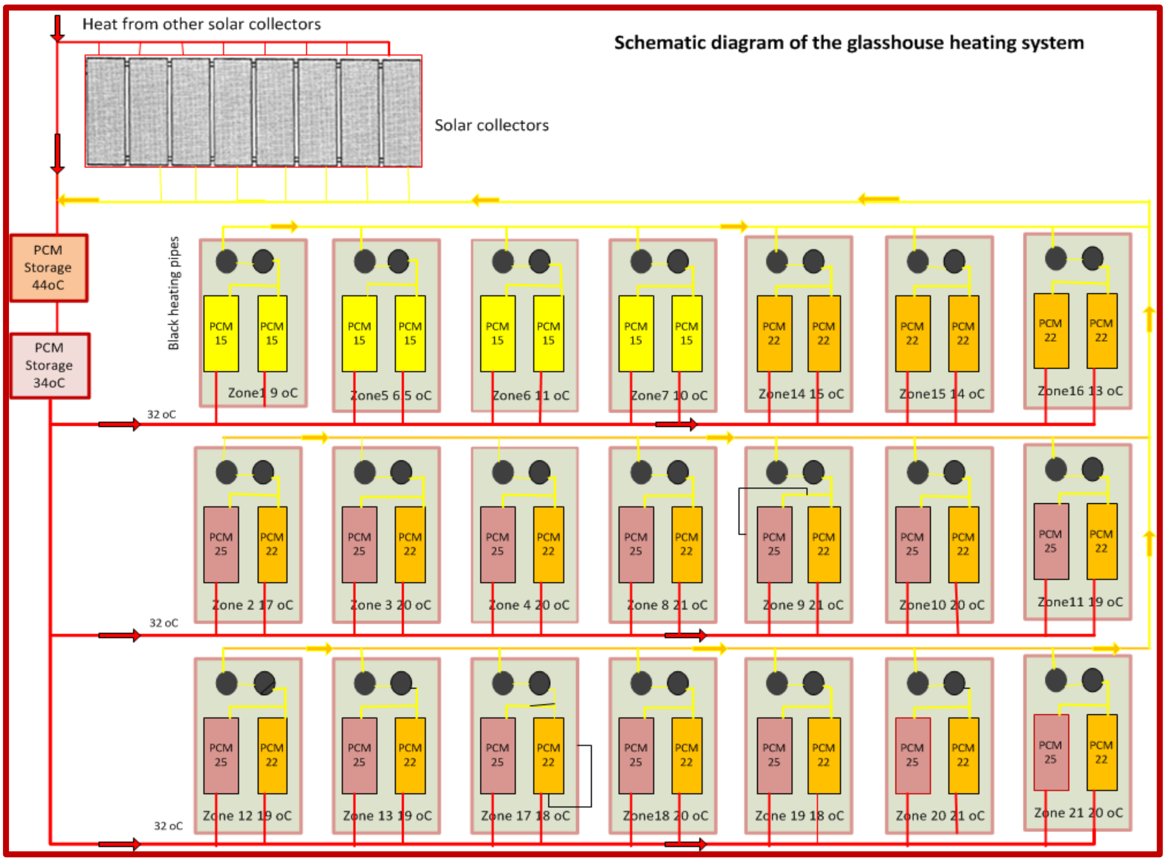

Figure 4:14 Schematic diagram of the glasshouse heating system

4.4.3 Auxiliary heat supply from CHP system in winter

The site has CHP plant close to the glasshouse under research which primarily used to generate power even though the thermal energy is used in heating buildings in winter. Currently the CHP is not run in summer as there is no demand for the thermal energy in summer. The thermal energy would have to be dumped if run in summer as a result runs in winter.

The CHP unit consist of six main components namely the engine, engine heat exchanger, exhaust heat exchanger, generator, control panel and the engine exhaust. There are also three main common heat exchangers used in the CHP system.

They are:

- Water-to-water for the engine cooling jacket
- Gas-to-water for the hot exhaust gases
- Condensing for the cool exhaust gas

The exhaust system in a CHP Plant carries away the products of combustion in a save and quiet manner, avoiding the components to corrode.

The system is incorporated with acoustic materials to dampen the noise generated to make its operation quieter. Silencers are also installed at the intake and exhaust system of the engine to make it quieter.

The packaged CHP unit installed is based on a gas engine and fuelled by natural gas. The scheme has a rated electrical output of 500 kWe and thermal output of 611 kW_{th}. The unit does not have a facility to recover heat from the engine cooling systems and rejects heat into the atmosphere wasted.

The CHP produces a high grade heat in the form of LTHW at temperature of 85-100 ^{O}C and low grade heat at 40 to 60 ^{O}C. Figure 4:15 shows the schematic diagram of the CHP system.

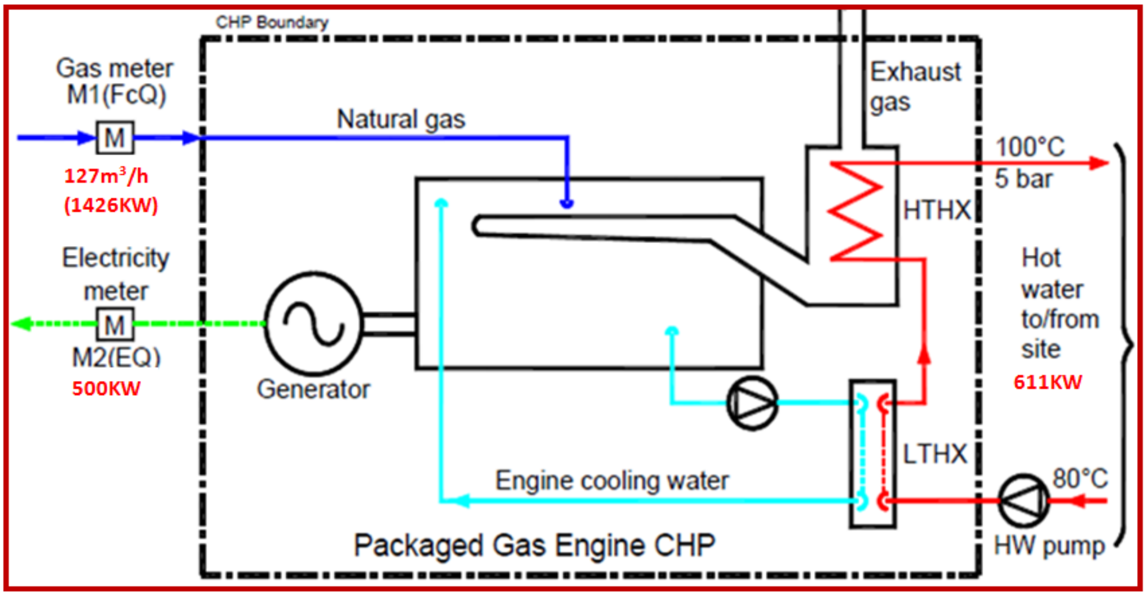

Figure 4:15 Kew Gardens CHP system

Below is the glasshouse heating system including auxiliary CHP plant schematic. The primary heat source in winter will still come from the solar collectors and use the CHP recovered heat energy as a backup or supplementary heat source.

The PCM storage tanks will have full capacity of heat energy stored before the winter season sets in as the solar gains and heat energy stored in the PCM heating pipes will be sufficient to meet the glasshouse heating demand in the summer period. Figure 4:16 illustrates glasshouse heating system including the auxiliary CHP plant.

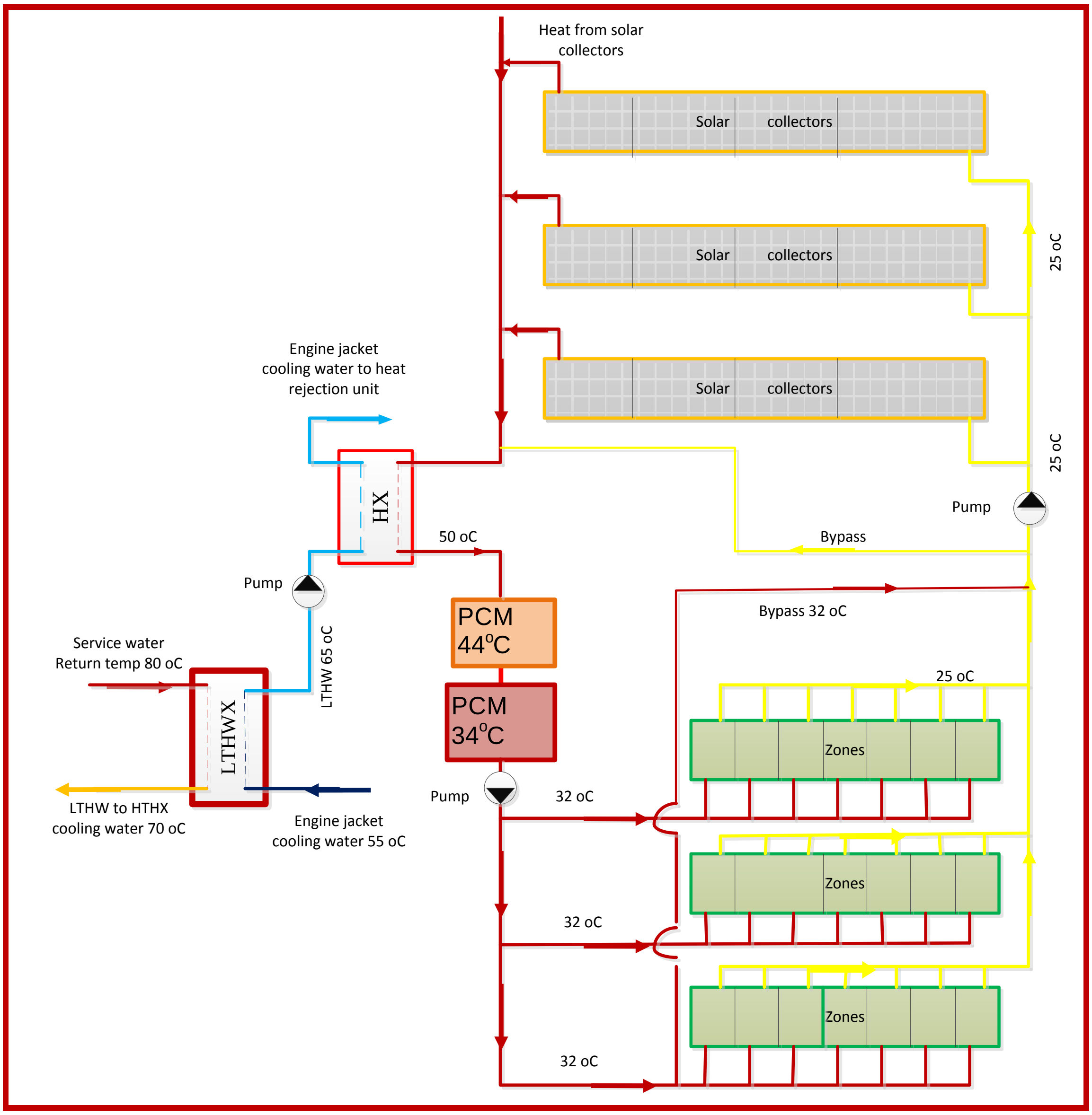

Figure 4:16 Glasshouse heating system including auxiliary CHP plant schematic

4.4.4 Control methodology

The flow of hot water through the PCM heating pipes will be controlled by motorised valves. The motorised valves will open to allow hot water to flow through the PCM filled heating pipes when the space temperature reaches the set point.

For example, the space set point temperature of zone 9 is 20 ^{o}C with a minimum set point of 19 ^{o}C therefore the motorised valve will open to allow the hot water to flow through the PCM heating pipes when the space temperature reaches 20 ^{o}C to charge the PCM heating pipes. This will safeguard the room temperature from falling below the 19 ^{o}C space minimum set point temperature.

The system will assume that more thermal energy is needed once the space temperature reaches the space set point temperature and therefore opens the motorised valves to allow hot water to follow through the PCM heating pipes.

The hot water flow rate will be modulated to achieve exit hot water temperature of 25 ^{o}C from each zone or all the zones prior to entry to the solar collector hot water systems. This will be achieved by installing temperature sensors at exit points at strategic positions. Low temperature water entry to the solar collector system will enhance maximum heat transfer from the solar collectors to the hot water.

The motorised valves will be disabled when the space temperature set point reaches 2 ^{o}C above the space temperature set point. For example the motorised valve will be disabled when the space temperature reaches 22 ^{o}C in zone 9 with space set point temperature of 20 ^{o}C.

Figure 4:17 shows the control devices with sensors installed to measure space set point temperatures, exit water temperatures from the PCM heating pipes and entry water temperature prior to the solar collector systems. The return water temperature from the solar collectors is constantly measured to avoid the return water temperature falling below 30 ^{o}C.

The hot water returning from the PCM storage tank at 32 ^{o}C will pass through the by-pass line (Figure 4:17) when the motorised valve is disabled to join the common hot water return pipe header before entry to the solar collectors.

In this situation the entry hot water temperature to the solar collector system may rise above 25 ^{o}C but the effect to the solar collector efficiency may not be significant.

The flow of water through the solar active system will be regulated by the variable speed drive pump to ensure that return water temperature from the solar active system is not below 35 ^{o}C and not exceeding 50 ^{o}C to ensure system consistency during the charging period.

The system is designed such that exit water from the storage tanks is 32 ^{o}C and this is achievable by regulating the flow rate through the PCM storage tanks.

Grundfos MAGNA3 variable speed drive (Figure 4:18 below) will be installed to ensure that the system water flow is properly regulated to achieve system efficiency.

If for any reason the return hot water from the storage tank is above 32 ^{o}C then a mixing valve will be activated to reduce the hot water temperature by mixing it with return hot water from the zones to achieve the 32 ^{o}C supply temperature to the zones to ensure system consistency.

When the hot water returning from the solar collectors temperature is below 30 ^{o}C then the hot water entry valve to the collectors will be disabled and flow from the collectors will be drained to the drain back vessel until sufficient heat can be generated by the solar collector.

Figure 4:17 below illustrates the control mechanism as explained above. The same operational control principle applies to all the zones.

The space set point temperature is designed to be achieved by discharging and charging of the PCM filled heating pipes but a supplementary space temperature sensor will be installed to open the vents to allow some heat out to the atmosphere when the vent opening temperature is reached.

There will also be a sensor that will enable the motorised valve when the space set point temperature falls below the zone set point temperature.
The opening of the vents will be modulated to regulate the heat that is allowed to the atmosphere to avoid excessive temperature drop in the space which may require heating the glasshouse again. The current system only opens and closes which causes unnecessary heat to be vented out resulting to unstable temperature in the zones.

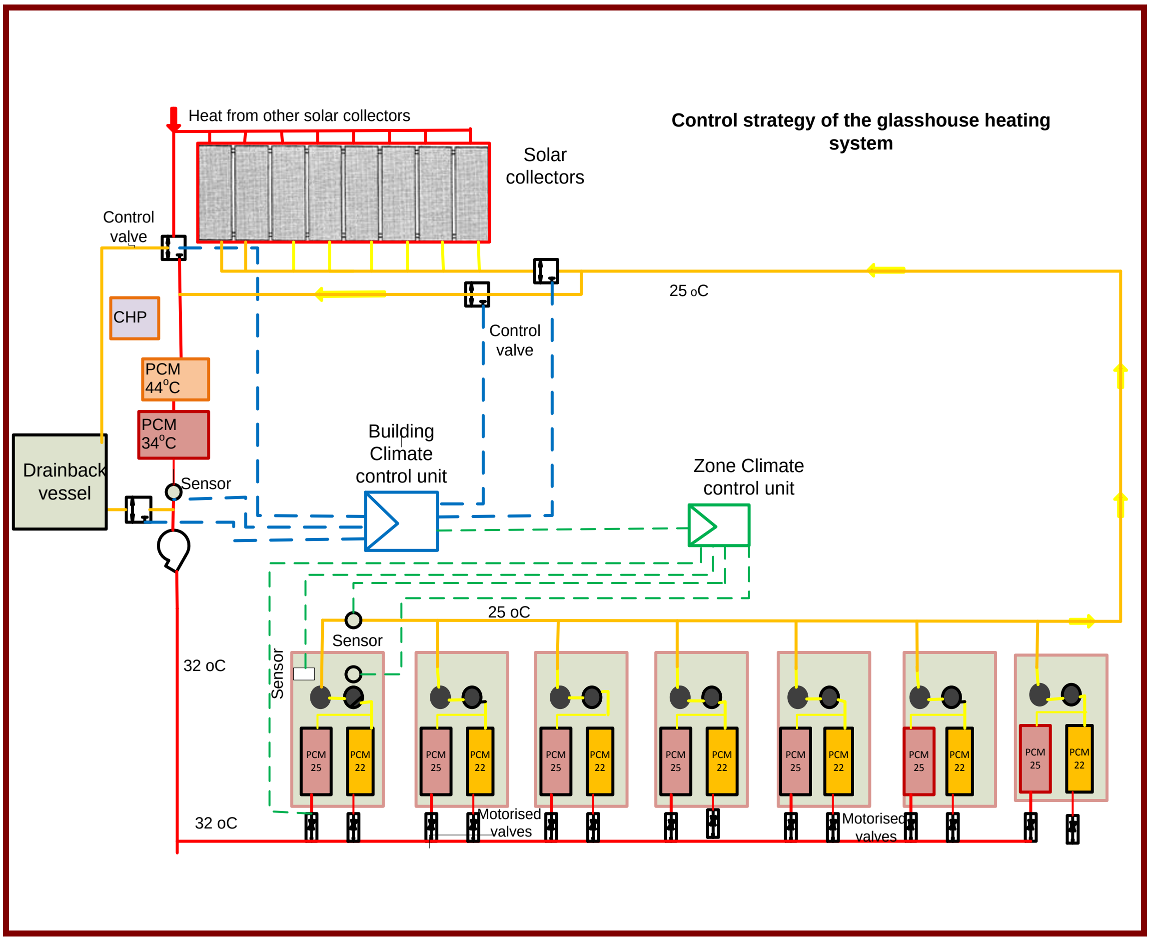

Figure 4:17 Zone control method schematic

4.4.5 System water flow rate regulation

The system efficiency could be effectively achieved by systematic regulation of the hot water flow rates through the PCM heating pipes, the solar collector thermal energy hot water system and the CHP waste heat recovery system. These three systems water flow rates will be regulated by Grundfos MAGNA3 circulation pumps to achieve the heating system temperature set points to maintain the system efficiency.

The Grundfos MAGNA3 is a range of medium and large circulator pumps complete with electronically controlled motors.

The MAGNA3 range is designed for circulating liquids in:

- Heating systems
- Air conditioning and cooling systems
- Domestic water systems
- Ground-source heat pump systems
- Solar heating systems.

The pump has the following unique features:

- The MAGNA3 has a facility to limit maximum flow removing the need to use a balancing valve on the main flow and return pipe.
- FLOWADAPT combines the benefits of both the FLOWLIMIT and AUTOADAPT (Continuous water flow regulation) functions together.
- The MAGNA3 has a built-in heat energy meter. By fitting a separate temperature sensor in the return pipe the energy consumption in the system can be monitored and logged in the MAGNA3. This allows end users to monitor their energy consumption in order to avoid excessive energy bills caused by poor system balance and potential problems with system controls
- The MAGNA3 is the most efficient circulator available, with an Energy Efficiency Index (EEI) that already complies with the EuP 2015 requirements and can achieve savings of up to 75 % compared to a typical installed circulator.

- Once the AUTOADAPT has been set up, the pump will analyse the heating system demands and find the optimum setting. It will then continuously adjust its operation to deal with changing system demands. This is an efficient pump that regulates systems requirements effectively.

Figure 4:18 Grundfos MAGNA3 D 80-100F (360) Variable Speed Twin Head Circulator 240V (97924504)

Benefits of the Grundfos MAGNA3 water pump

Below are the benefits of the MAGNA3 pumps

- Low energy consumption and complies with the EuP 2015 requirements
- The AUTOADAPT function ensures energy savings
- FLOWADAPT which is a combination of the well-known AUTOADAPT control mode and a new FLOWLIMIT function.
- Built-in Grundfos differential-pressure and temperature sensor.
- No maintenance and long life.
- Extended user interface with TFT display
- Control panel with self-explanatory push-buttons made of high-quality silicone.

- Work log history
- Easy system optimisation
- Heat energy meter
- Multipump function
- External control and monitoring enabled via add-on modules

4.4.6 Maximising the use of solar heat gains in the zones

Table 4:2 above demonstrates that the amount of solar gains in the zones is almost equal to the heating demand for the design year and this can be significantly beneficial if effort can be made to store the heat and use it when needed rather than venting it out to the atmosphere.

Figure 4:19 illustrates the hourly heating demand, solar gains and temperature profile of zone 9 in 2 May of the design year. It could be seen from figure 4:19 that solar gains or energy trapped in the glasshouse for the day is far greater than the heating requirement which suggest that no thermal energy generation will be needed if the energy trapped in the glasshouse as a result of the solar gains can be stored.

Figure 4:19 demonstrates that from 0730 hours no heating is required till 2030 hours so the trapped energy instead of venting it out to the atmosphere could be stored and used when needed without thermal energy generation from fossil fuel or other heat generating source.

The demonstration of figure 4:19 is an example of how solar energy can be stored across the whole zones to use zero or minimal fossil fuel to generate thermal energy to heat the glasshouse.

The hourly heating demand, solar gains and temperature profiles from January to December analysis is carried out across the whole zones and figure 4:19 below is an example.

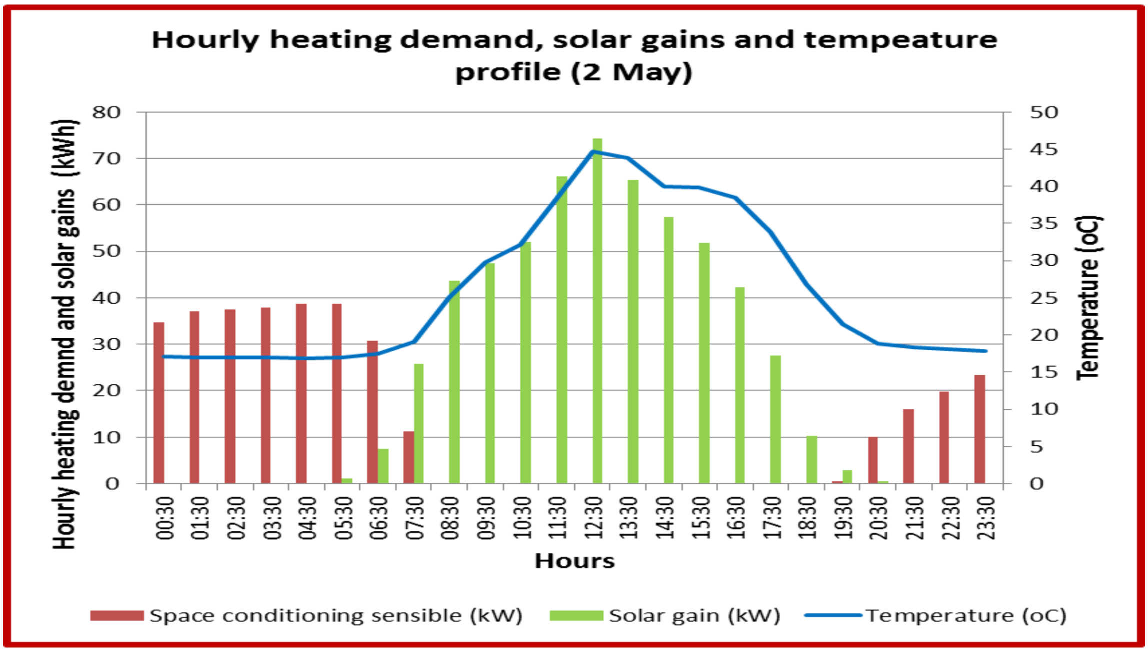

Figure 4:19 Hourly heating demand, solar gains and temperature profile of zone 9 in a design day of 2 May.

4.4.7 Space temperature control through melting and freezing of the PCM filled in the heating pipes

As it has already been explained above the space temperature will be controlled by discharging and charging of the PCM filled heating pipes. Thus the PCM will discharge heat to the space when the temperature drops below its phase change temperature and will charge or absorb heat when the space temperature exceeds the phase change temperature.

4.4.8 Heating pipe design

Annulus pipes will be used as heating pipes that will be filled with PCM solutions. Figure 4:20 shows an example of an annulus heating pipe filled with PCM solutions. The heating pipe is formed by two concentric pipes. Water flows through the smaller inner pipe and the outer pipe contains the phase change material as demonstrated in Figure 4:20below.

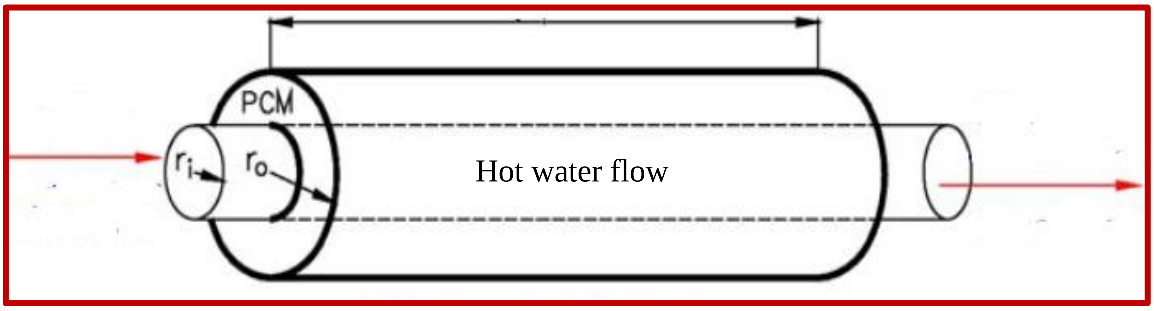

Figure 4:20 Annulus heating pipe configuration

4.4.9 Zone space temperature control

The space set point temperature will be controlled by melting and freezing of the PCM filled heating pipes. The space heating set point of zone 9 is 20 ^{o}C and will be primarily be maintained by melting and freezing of the PCM filled heating pipes. The heating pipes in zone 9 for example, are filled with PCMs of melting points of 25 and 22 ^{o}C.

The heating pipes will discharge energy to warm the space when space temperature falls below 25 and 22 ^{o}C as demonstrated in figure 4:21 below and will absorb heat energy when the space temperature rises above the melting points.

The vents will open when the space temperature reaches 27 ^{o}C to allow some heat out to maintain the space temperature to the required level. The key to figure 4:21 illustrates the periods where the described processes occur.

It could be seen from Figure 4:21 below that the 22 and 25 ^{o}C heating pipes discharge heat energy to the space when the space temperature was below 22 ^{o}C until 0730 hours. From 0730 hours the space temperature rose to 22 ^{o}C and as a result the 22 ^{o}C melting point PCM starts to melt and absorbs heat energy from the space till approximately 2030 hours when it starts to discharge energy when the space temperature reaches 20 ^{o}C. The 25 ^{o}C melting point PCM will start to melt when the space temperature reaches 25 ^{o}C at 0830 hours. It will continue to absorb heat energy as it is melting till around 1700 hours when the space temperature falls to 25 ^{o}C.

The vents will open when the space temperature reaches 27 ^{o}C and for the purpose of explaining the operational principle of the heating system the vents will open at about 1130 hours as demonstrated in figure 4:21 to allow some heat out.

In actual operation the vents may not even open as the heat trapped inside the space will be absorbed by the PCM filled heating pipes. Figure 4:22 illustrates how the current vent system operates.

The current vent system opens and closes fully with no modulation or regulation and this causes temperature fluctuation in the space. The new design vent opening will be regulated using motorised control method to avoid temperature fluctuation and unnecessary loss of heat from the space.

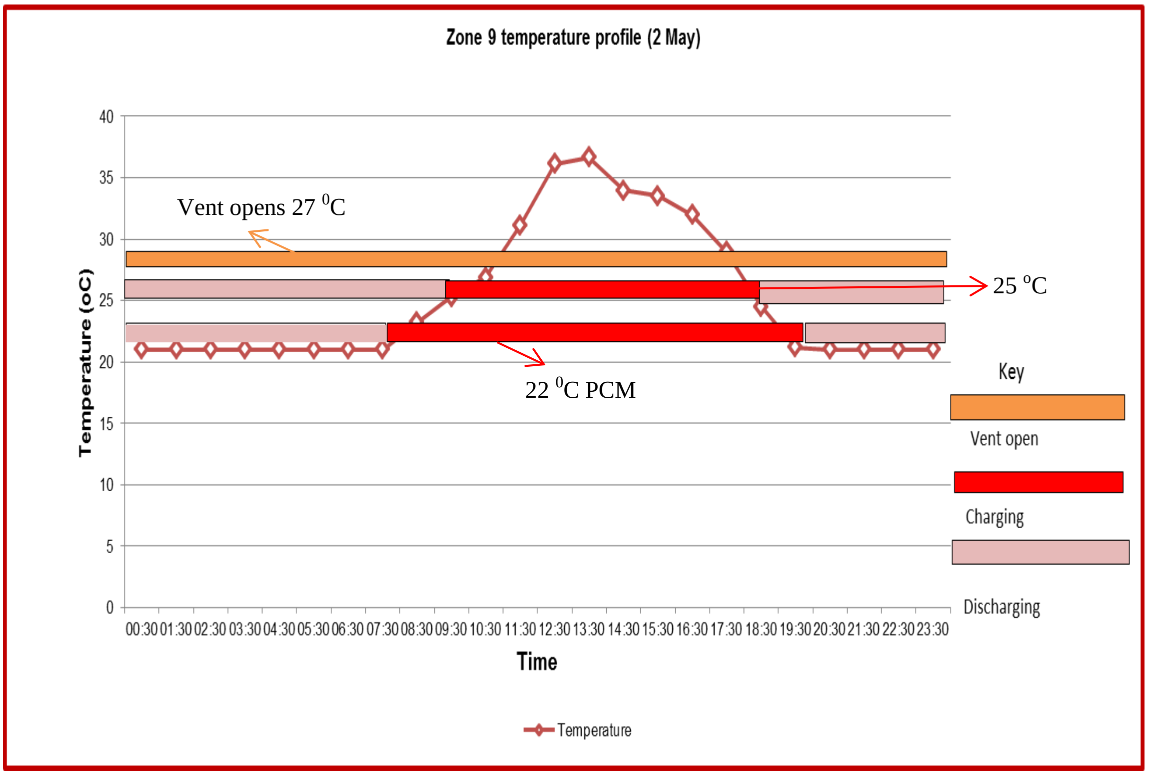

Figure 4:21 Space temperature control through charging and discharging of the PCM heating pipes

Figure 4:22 Glasshouse vents opening

4.5 Analysing zones hourly heating demand, solar gains and temperature profiles of the simulation results

The analysis of the zones hourly heating demand, solar gains and the temperature profiles was assessed from January 1 to December 31of each zone to understand the heating characteristics and behaviour throughout a complete year cycle.

This was necessary to know the periods where active and passive systems can be effectively utilised.

In all, three hundred and sixty-five graphs were plotted for each zone. For example, 31 graphs were plotted in January for zone 1 to establish the hourly heating energy demand, solar gains and temperature profile.

For the purpose of this report zone sampling is used to illustrate the above parameters and they are representatives of all the zones. The selection of the sampling is based on the zones temperature set points and weather conditions throughout a complete year cycle.

The selected zones were zone 1 with space set point temperature of 9 ^oC which represents zones of temperature set points between 6.5 to 9 ^oC.

Zone 6 with space set point temperature of 11 ^oC representing zones of set point temperatures between 10-12, Zone 9 with space set point temperature of 21 ^oC representing zones of set point temperatures of 20 and 21, Zone 15 with space set point temperature of 14 ^oC representing zones of set point temperatures between 13-16, Zone 17 with space set point temperature of 18 ^oC representing zones of set point temperatures between 17-19 ^oC.

The assessment period of the above parameters was based on the weather conditions throughout the design year and the selected months are January, February, March, May and July which are representative months appropriate to assess heating system performance and requirements.

Figure 4:23 to Figure 4:27 show the hourly heating demand, solar gains and temperature profiles of sampling zones selected as representatives to all the other zones from January to July of the design year. Zone 1 (Figure 4:23) was used as a sampling zone for January, zone 6 for March, zone 9 for May, zone 15 for July and zone 17 again for March.

The analysis was to ensure that enough heat energy is stored and at the same time assists in determining the number of solar collectors that will be required to generate sufficient heat energy at all times to warm the glasshouse.

Appendix C illustrates more hourly heating demand, solar gains and temperature profiles of the sampling zones.

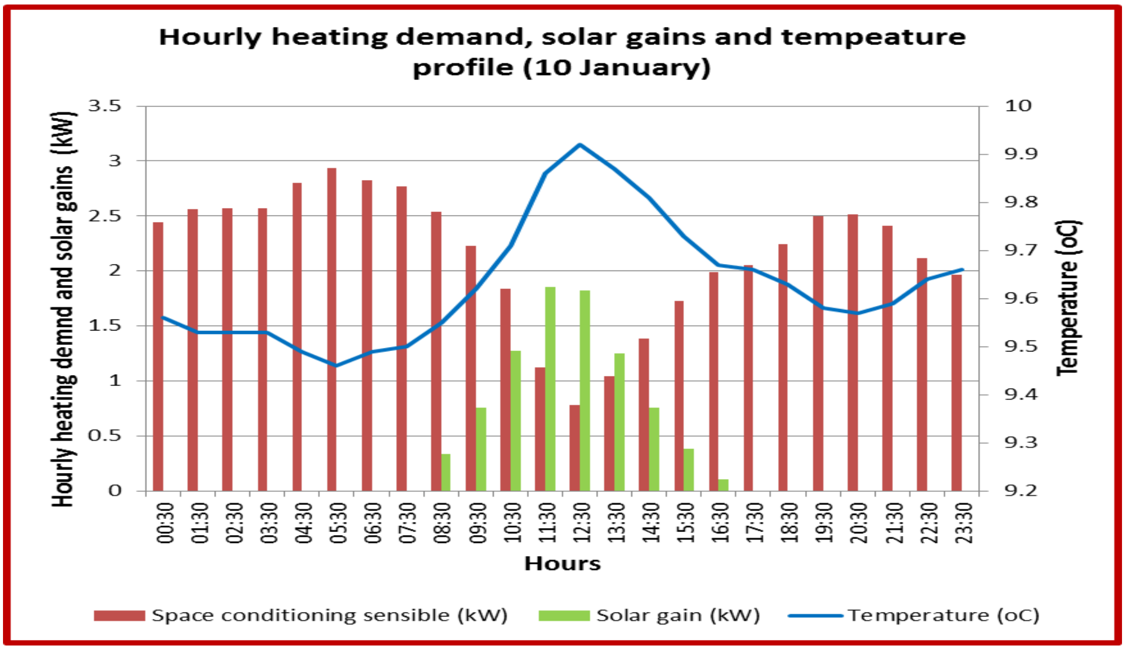

Figure 4:23 Zone 1 hourly heating demand, solar gains and temperature profile with space temperature set point of 9 ^{o}C 10 January

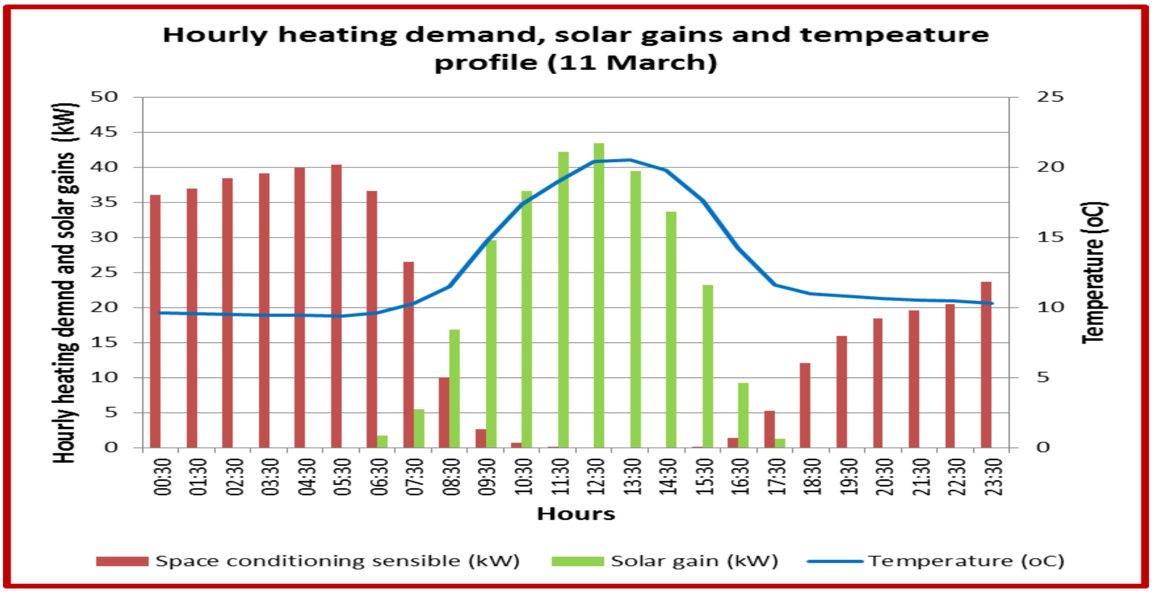

Figure 4:24 Zone 6 hourly heating demand, solar gains and temperature profile with space temperature set point of 11 ^{o}C in 11 March

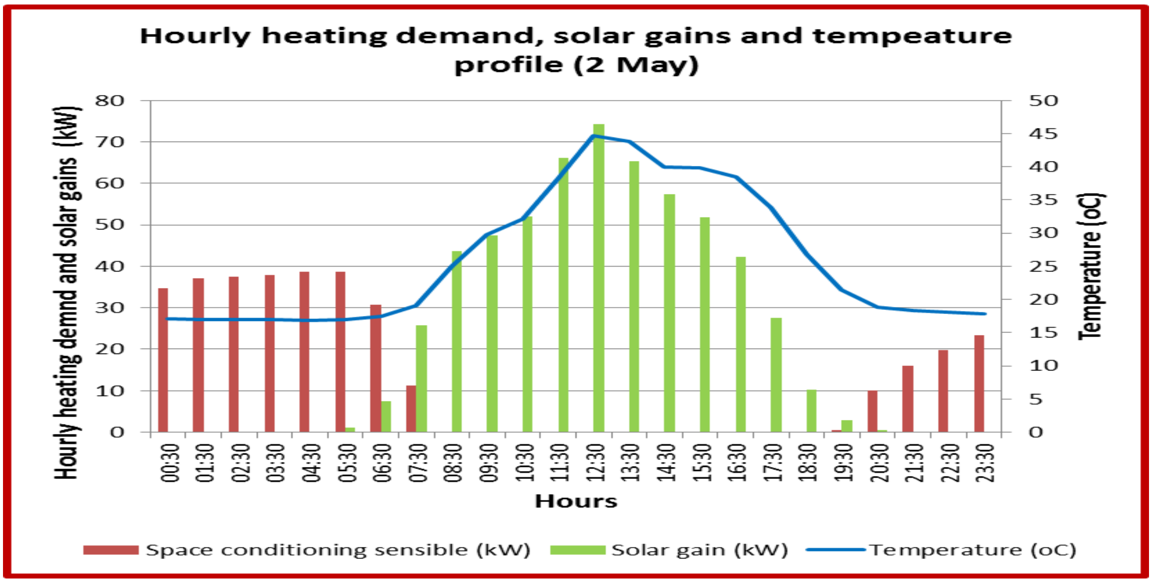

Figure 4:25 Zone 9 hourly heating demand, solar gains and temperature profile with space temperature set point of 21 ^{o}C in 2 May

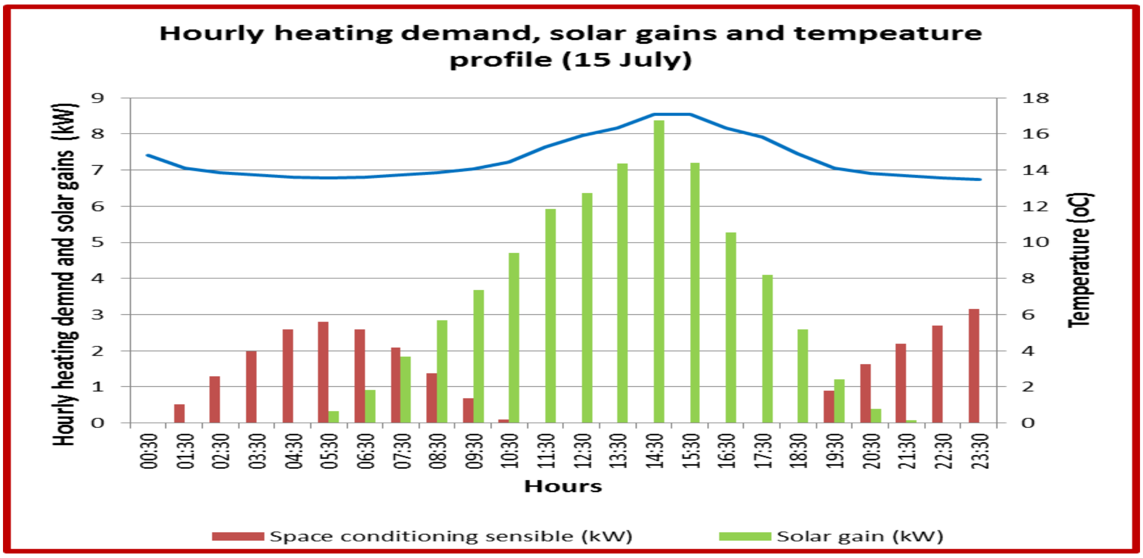

Figure 4:26 Zone 15 hourly heating demand, solar gains and temperature profile with space temperature set point of 14 ^{o}C in 15 July

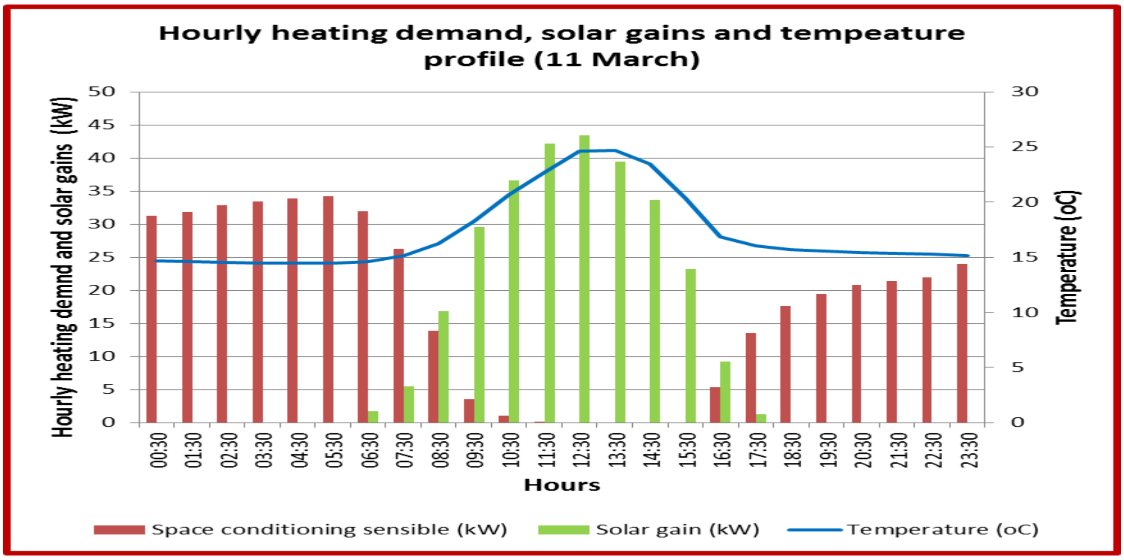

Figure 4:27 Zone 17 hourly heating demand, solar gains and temperature profile with space temperature set point of 18 ^{o}C in 11 March

4.6 PCM selection for the proposed Kew Gardens heating system

The selection of PCMs that will be suitable for the research project will be based on tested PCMs used for field applications for heating and heat recovery systems. They are products of PCM Products Ltd in England.

The PCM solutions have been encapsulated in a unique cylindrical beam and flat container designs that can be used for industrial, large commercial or institutional applications. The composition or the storage capacity do not degrade and performs reliably [6].

4.6.1 Freezing and melting rates of a typical PCMs in TubeICE and FlatICE containers

PCM products Ltd freezing and melting rates of their PCM products encapsulated in TubeICE container are as demonstrated in figure 2:8 and figure 2:9 performance curves respectively in chapter 2. Similarly, freezing and melting rates of PCM encapsulated in FlatICE container are also as demonstrated in figure 2:11 and figure 2:12 performance curves respectively in chapter 2.

The freezing and melting rates of PCM solutions in TubeICE container was obtained from the performance curves in figure 2.8 and 2.9 respectively. For example, the freezing rate of PCM solution in TubeICE container with temperature difference of 3 ^{o}C between the PCM and the surrounding water will take 620 minutes to freeze whilst the melting rate is 310 minutes.

Similarly, freezing and melting rates of PCM solutions in FlatICE container was obtained from the performance curve in figure 2.11 and 2.12 respectively. For example, the freezing rate of PCM solution in FlatICE container with temperature difference of 5 ^{o}C between the PCM and the surrounding water will take 400 minutes to freeze whilst the melting rate is 200 minutes.

The freezing and melting rates of other temperature differences between the TubeICE and FlatICE containers were also obtained from the performance curves as explained above to assess and analysis freezing and melting rates of the selected PCMs for the research and other PCMs.

Table 4:12 and table 4:13 below show the technical details and freezing rates of Organic (A) Range and Hydrated Salt PCMs respectively in TubeICE container based on the values obtained from the freezing performance curve of figure 2:8.

The freezing rates are influenced by the PCM characteristics such as latent heat capacity, phase change temperature and others.

Table 4:12 Organic (A) Range technical details and freezing times (Time in minutes) in TubeICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	TubeICE Container volume	TubeICE energy (kWh/TubeICE)	ΔT b/n PCM & hot water	Freezing time at 3 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 4 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 5 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 7 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 10 oC temp diff
A40	40	230	186	51.7	0.007855	0.41	3	620	4	500	5	400	7	260	10	180
A36	36	217	171	47.5	0.007855	0.37	3	585	5	472	5	377	7	245	10	170
A32	32	130	110	30.6	0.007855	0.24	3	350	5	283	5	226	7	147	10	102
A29	29	226	183	50.8	0.007855	0.40	3	609	5	491	5	393	7	255	10	177
A26	26	150	119	33.1	0.007855	0.26	3	404	5	326	5	261	7	170	10	117
A25H	25	226	183	50.8	0.007855	0.40	3	609	5	491	5	393	7	255	10	177
A24	24	145	115	31.9	0.007855	0.25	3	391	5	315	5	252	7	164	10	113
A23	23	145	114	31.7	0.007855	0.25	3	391	5	315	5	252	7	164	10	113
A22H	22	216	177	49.2	0.007855	0.39	3	582	5	470	5	376	7	244	10	169
A17	17	150	118	32.8	0.007855	0.26	3	404	5	326	5	261	7	170	10	117
A16	16	213	162	45.0	0.007855	0.35	3	574	5	463	5	370	7	241	10	167

Table 4:13 Hydrated Salt based PCM solution technical details and freezing rates (Time in minutes) in TubeICE container

PCM type	Phase change temp (^{o}C)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m^{3})	PCM energy capacity per volume (kWh/m3	TubeICE Container volume	TubeICE energy (kWh/Tube ICE)	**ΔT** b/n PCM & hot water	Freezing time at 3 oC temp diff	**ΔT** b/n PCM & hot water	Freezing time at 4 oC temp diff	**ΔT** b/n PCM & hot water	Freezing time at 5 oC temp diff	**ΔT** b/n PCM & hot water	Freezing time at 7 oC temp diff	**ΔT** b/n PCM & hot water	Freezing time at 10 oC temp diff
S117	117	160	232	64.4	0.007855	0.506	3	551	4	444	5	356	7	231	10	142
S89	89	151	234	65.0	0.007855	0.511	3	520	4	419	5	336	7	218	10	134
S83	83	141	226	62.8	0.007855	0.493	3	486	4	392	5	313	7	204	10	125
S72	72	127	212	58.9	0.007855	0.463	3	437	4	353	5	282	7	183	10	113
S70	70	110	185	51.4	0.007855	0.404	3	379	4	306	5	244	7	159	10	98
S58	58	145	218	60.6	0.007855	0.476	3	499	4	403	5	322	7	209	10	129
S50	50	100	160	44.4	0.007855	0.349	3	344	4	278	5	222	7	144	10	89
S46	46	210	333	92.5	0.007855	0.727	3	723	4	583	5	467	7	303	10	187
S44	44	100	158	43.9	0.007855	0.345	3	344	4	278	5	222	7	144	10	89
S34	34	115	242	67.2	0.007855	0.528	3	396	4	319	5	256	7	166	10	102
S32	32	200	292	81.1	0.007855	0.637	3	689	4	556	5	444	7	289	10	178
S30	30	190	248	68.9	0.007855	0.541	3	654	4	528	5	422	7	274	10	169
S27	27	183	280	77.8	0.007855	0.611	3	630	4	508	5	407	7	264	10	163
S25	25	180	275	76.4	0.007855	0.600	3	620	4	500	5	400	7	260	10	160
S23	23	175	268	74.4	0.007855	0.585	3	603	4	486	5	389	7	253	10	156
S21	22	170	260	72.2	0.007855	0.567	3	586	4	472	5	378	7	246	10	151
S19	19	160	243	67.5	0.007855	0.530	3	551	4	444	5	356	7	231	10	142
S17	17	160	244	67.8	0.007855	0.532	3	551	4	444	5	356	7	231	10	142
S15	15	160	242	67.2	0.007855	0.528	3	551	4	444	5	356	7	231	10	142
S13	13	160	242	67.2	0.007855	0.528	3	551	4	444	5	356	7	231	10	142

Table 4:14 and table 4:15 below show the technical details and melting rates of Organic (A) Range and Hydrated Salt PCMs respectively in TubeICE container based on the values obtained from the melting performance curve of figure 2:9

The melting rates are influenced by the PCM characteristics such as latent heat capacity, phase change temperature and others.

Table 4:14 Organic (A) Range technical details and melting rates (Time in minutes) in TubeICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	TubeICE Container volume	TubeICE energy (kWh/TubeICE)	ΔT b/n PCM & hot water	Melting time at 3 oC temp diff	ΔT b/n PCM & hot water	Melting time at 4 oC temp diff	ΔT b/n PCM & hot water	Melting time at 5 oC temp diff	ΔT b/n PCM & hot water	Melting time at 7 oC temp diff	ΔT b/n PCM & hot water	Melting time at 10 oC temp diff
A40	40	230	186	51.7	0.007855	0.41	3	310	4	250	5	200	7	120	10	90
A36	36	217	171	47.5	0.007855	0.37	3	292	4	236	5	189	7	113	10	85
A32	32	130	110	30.6	0.007855	0.24	3	175	4	141	5	113	7	68	10	51
A29	29	226	183	50.8	0.007855	0.40	3	305	4	246	5	197	7	118	10	88
A26	26	150	119	33.1	0.007855	0.26	3	202	4	163	5	130	7	78	10	59
A25H	25	226	183	50.8	0.007855	0.40	3	305	4	246	5	197	7	118	10	88
A24	24	145	115	31.9	0.007855	0.25	3	195	4	158	5	126	7	76	10	57
A23	23	145	114	31.7	0.007855	0.25	3	195	4	158	5	126	7	76	10	57
A22H	22	216	177	49.2	0.007855	0.39	3	291	4	235	5	188	7	113	10	85
A17	17	150	118	32.8	0.007855	0.26	3	202	4	163	5	130	7	78	10	59
A16	16	213	162	45.0	0.007855	0.35	3	287	4	232	5	185	7	111	10	83

Table 4:15 Hydrated Salt PCM solutions technical details and melting rates (Time in minutes) in TubeICE container

PCM type	Phase change temp (^{o}C)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m^3)	PCM energy capacity per volume (kWh/m3	TubeICE Container volume	TubeICE energy (kWh/Tube ICE)	ΔT b/n PCM & hot water	Melting time at 3 oC temp diff	ΔT b/n PCM & hot water	Melting time at 4 oC temp diff	ΔT b/n PCM & hot water	Melting time at 5 oC temp diff	ΔT b/n PCM & hot water	Melting time at 7 oC temp diff	ΔT b/n PCM & hot water	Melting time at 10 oC temp diff
S117	117	160	232	64.4	0.007855	0.506	3	284	4	222	5	178	7	116	10	80
S89	89	151	234	65.0	0.007855	0.511	3	268	4	210	5	168	7	109	10	76
S83	83	141	226	62.8	0.007855	0.493	3	251	4	196	5	157	7	102	10	71
S72	72	127	212	58.9	0.007855	0.463	3	226	4	176	5	141	7	92	10	64
S70	70	110	185	51.4	0.007855	0.404	3	196	4	153	5	122	7	79	10	55
S58	58	145	218	60.6	0.007855	0.476	3	258	4	201	5	161	7	105	10	73
S50	50	100	160	44.4	0.007855	0.349	3	178	4	139	5	111	7	72	10	50
S46	46	210	333	92.5	0.007855	0.727	3	373	4	292	5	233	7	152	10	105
S44	44	100	158	43.9	0.007855	0.345	3	178	4	139	5	111	7	72	10	50
S34	34	115	242	67.2	0.007855	0.528	3	204	4	160	5	128	7	83	10	58
S32	32	200	292	81.1	0.007855	0.637	3	356	4	278	5	222	7	144	10	100
S30	30	190	248	68.9	0.007855	0.541	3	338	4	264	5	211	7	137	10	95
S27	27	183	280	77.8	0.007855	0.611	3	325	4	254	5	203	7	132	10	92
S25	25	180	275	76.4	0.007855	0.600	3	320	4	250	5	200	7	130	10	90
S23	23	175	268	74.4	0.007855	0.585	3	311	4	243	5	194	7	126	10	88
S21	22	170	260	72.2	0.007855	0.567	3	302	4	236	5	189	7	123	10	85
S19	19	160	243	67.5	0.007855	0.530	3	284	4	222	5	178	7	116	10	80
S17	17	160	244	67.8	0.007855	0.532	3	284	4	222	5	178	7	116	10	80
S15	15	160	242	67.2	0.007855	0.528	3	284	4	222	5	178	7	116	10	80
S13	13	160	242	67.2	0.007855	0.528	3	284	4	222	5	178	7	116	10	80

4.6.2 Freezing and melting rates of PCM solutions in FlatICE containers

Using plastic FlatICE containers in certain applications can be economical TES solution. Figure 2:10 in chapter 2 demonstrates FlatICE containers.

Table 4:16 and table 4:17 below show the technical details and freezing rates of Organic (A) Range and Hydrated Salt PCMs in FlatICE container based on the values obtained from the freezing performance curve of figure 2:11.

The freezing rates are influenced by the PCM characteristics such as latent heat capacity, phase change temperature and others.

Table 4:16 Organic (A) Range technical details and freezing rates (Time in minutes) in FlatICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	TubeICE Container volume	TubeICE energy (kWh/TubeICE)	ΔT b/n PCM & hot water	Freezing time at 3 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 5 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 7 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 8 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 10 oC temp diff
A40	40	230	186	51.7	0.007855	0.41	3	630	4	500	5	400	7	260	10	160
A36	36	217	171	47.5	0.007855	0.37	3	594	4	460	5	368	7	245	10	147
A32	32	130	110	30.6	0.007855	0.24	3	356	4	296	5	237	7	147	10	95
A29	29	226	183	50.8	0.007855	0.40	3	619	4	492	5	394	7	255	10	157
A26	26	150	119	33.1	0.007855	0.26	3	411	4	320	5	256	7	170	10	102
A25H	25	226	183	50.8	0.007855	0.40	3	619	4	492	5	394	7	255	10	157
A24	24	145	115	31.9	0.007855	0.25	3	397	4	309	5	247	7	164	10	99
A23	23	145	114	31.7	0.007855	0.25	3	397	4	306	5	245	7	164	10	98
A22H	22	216	177	49.2	0.007855	0.39	3	592	4	476	5	381	7	244	10	152
A17	17	150	118	32.8	0.007855	0.26	3	411	4	317	5	254	7	170	10	102
A16	16	213	162	45.0	0.007855	0.35	3	583	4	435	5	348	7	241	10	139

Table 4:17 Hydrated Salt based PCM solutions technical details and freezing rates (Time in minutes) in FlatICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	FlatICE Container volume	TubeICE energy (kWh/TubeICE)	ΔT b/n PCM & hot water	Freezing time at 4 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 5 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 6 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 8 oC temp diff	ΔT b/n PCM & hot water	Freezing time at 10 oC temp diff
S117	117	160	232	64.4	0.004	0.26	3	630	4	500	5	400	7	260	10	160
S89	89	151	234	65.0	0.004	0.26	3	595	4	504	5	403	7	245	10	161
S83	83	141	226	62.8	0.004	0.25	3	555	4	487	5	390	7	229	10	156
S72	72	127	212	58.9	0.004	0.24	3	500	4	457	5	366	7	206	10	146
S70	70	110	185	51.4	0.004	0.21	3	433	4	399	5	319	7	179	10	128
S58	58	145	218	60.6	0.004	0.24	3	571	4	470	5	376	7	236	10	150
S50	50	100	160	44.4	0.004	0.18	3	394	4	345	5	276	7	163	10	110
S46	46	210	333	92.5	0.004	0.37	3	827	4	718	5	574	7	341	10	230
S44	44	100	158	43.9	0.004	0.18	3	394	4	341	5	272	7	163	10	109
S34	34	115	242	67.2	0.004	0.27	3	453	4	522	5	417	7	187	10	167
S32	32	200	292	81.1	0.004	0.32	3	788	4	629	5	503	7	325	10	201
S30	30	190	248	68.9	0.004	0.28	3	748	4	534	5	428	7	309	10	171
S27	27	183	280	77.8	0.004	0.31	3	721	4	603	5	483	7	297	10	193
S25	25	180	275	76.4	0.004	0.31	3	709	4	593	5	474	7	293	10	190
S23	23	175	268	74.4	0.004	0.30	3	689	4	578	5	462	7	284	10	185
S21	22	170	260	72.2	0.004	0.29	3	669	4	560	5	448	7	276	10	179
S19	19	160	243	67.5	0.004	0.27	3	630	4	524	5	419	7	260	10	168
S17	17	160	244	67.8	0.004	0.27	3	877	4	479	5	383	7	362	10	153
S15	15	160	242	67.2	0.004	0.27	3	630	4	522	5	417	7	260	10	167
S13	13	160	242	67.2	0.004	0.27	3	630	4	522	5	417	7	260	10	167

Table 4:18 and table 4:19 below show the technical details and melting rates of Organic (A) Range and Hydrated Salt PCMs respectively in FlatICE container based on the values obtained from the melting performance curve of figure 2:12.

The melting rates are influenced by the PCM characteristics such as latent heat capacity, phase change temperature and others.

Table 4:18 Organic (A) Range technical details and melting rates (Time in minutes) in FlatICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	FlatICE Container volume	FlatICE energy (kWh/TubeICE)	ΔT b/n PCM & hot water	Melting time at 3 oC temp diff	ΔT b/n PCM & hot water	Melting time at 5 oC temp diff	ΔT b/n PCM & hot water	Melting time at 6 oC temp diff	ΔT b/n PCM & hot water	Melting time at 7 oC temp diff	ΔT b/n PCM & hot water	Melting time at 10 oC temp diff
A40	40	230	186	51.7	0.004	0.21	3	320	4	250	5	200	7	130	10	90
A36	36	217	171	47.5	0.004	0.19	3	320	4	250	5	200	7	130	10	90
A32	32	130	110	30.6	0.004	0.12	3	320	4	250	5	200	7	130	10	90
A29	29	226	183	50.8	0.004	0.20	3	320	4	250	5	200	7	130	10	90
A26	26	150	119	33.1	0.004	0.13	3	320	4	250	5	200	7	130	10	90
A25H	25	226	183	50.8	0.004	0.20	3	320	4	250	5	200	7	130	10	90
A24	24	145	115	31.9	0.004	0.13	3	320	4	250	5	200	7	130	10	90
A23	23	145	114	31.7	0.004	0.13	3	320	4	250	5	200	7	130	10	90
A22H	22	216	177	49.2	0.004	0.20	3	320	4	250	5	200	7	130	10	90
A17	17	150	118	32.8	0.004	0.13	3	320	4	250	5	200	7	130	10	90
A16	16	213	162	45.0	0.004	0.18	3	320	4	250	5	200	7	130	10	90

Table 4:19 Hydrated salt based PCM solutions technical details and melting rates (Time in minutes) in FlatICE container

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m3)	PCM energy capacity per volume (kWh/m3	FlatICE Container volume	FlatICE energy (kWh/FlatICE)	ΔT b/n PCM & hot water	Melting time at 3 oC temp diff	ΔT b/n PCM & hot water	Melting time at 4 oC temp diff	ΔT b/n PCM & hot water	Melting time at 5 oC temp diff	ΔT b/n PCM & hot water	Melting time at 7 oC temp diff	ΔT b/n PCM & hot water	Melting time at 10 oC temp diff
S117	117	160	232	64.4	0.004	0.26	3	315	4	250	5	200	7	130	10	80
S89	89	151	234	65.0	0.004	0.26	3	297	4	252	5	202	7	123	10	81
S83	83	141	226	62.8	0.004	0.25	3	278	4	244	5	195	7	115	10	78
S72	72	127	212	58.9	0.004	0.24	3	250	4	228	5	183	7	103	10	73
S70	70	110	185	51.4	0.004	0.21	3	217	4	199	5	159	7	89	10	64
S58	58	145	218	60.6	0.004	0.24	3	285	4	235	5	188	7	118	10	75
S50	50	100	160	44.4	0.004	0.18	3	197	4	172	5	138	7	81	10	55
S46	46	210	333	92.5	0.004	0.37	3	413	4	359	5	287	7	171	10	115
S44	44	100	158	43.9	0.004	0.18	3	197	4	170	5	136	7	81	10	54
S34	34	115	242	67.2	0.004	0.27	3	226	4	261	5	209	7	93	10	83
S32	32	200	292	81.1	0.004	0.32	3	394	4	315	5	252	7	163	10	101
S30	30	190	248	68.9	0.004	0.28	3	374	4	267	5	214	7	154	10	86
S27	27	183	280	77.8	0.004	0.31	3	360	4	302	5	241	7	149	10	97
S25	25	180	275	76.4	0.004	0.31	3	354	4	296	5	237	7	146	10	95
S23	23	175	268	74.4	0.004	0.30	3	345	4	289	5	231	7	142	10	92
S21	22	170	260	72.2	0.004	0.29	3	335	4	280	5	224	7	138	10	90
S19	19	160	243	67.5	0.004	0.27	3	315	4	262	5	209	7	130	10	84
S17	17	160	244	67.8	0.004	0.27	3	438	4	240	5	192	7	181	10	77
S15	15	160	242	67.2	0.004	0.27	3	315	4	261	5	209	7	130	10	83

4.6.3 PCM selection for a particular application

Selecting PCM for an application should consider the following:

- The phase change temperature of the PCM should match the heating or cooling operating temperature.
- In a volumetric basis the latent heat should be high to minimise the physical size of the heat store
- Its thermal conductivity should be high to effectively assist the charging and discharging of the energy storage.

Table 4:20 and table 2:21 below describe the properties of PlusICE Organic (A) range and Hydrated Salt (S) range PCMs respectively. The colour coded PCMs in table 2:21 are those selected for the research project.

Table 4:20 PlusICE Organic (A) Range properties

PCM type	Phase change temp (°C)	Density (kg/m^3)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m^3)	Specific heat capacity (kJ/Kg °C)	Thermal conductivity (W/m K)	Maximum temp (°C)
A40	40	810	230	186	2.43	0.18	300
A36	36	790	217	171	2.37	0.18	300
A32	32	845	130	110	2.2	0.21	280
A29	29	810	226	183	2.15	0.18	300
A26	26	790	150	119	2.22	0.21	280
A25H	25	810	226	183	2.15	0.18	400
A24	24	790	145	115	2.22	0.18	280
A23	23	785	145	114	2.22	0.18	280
A22H	22	820	216	177	2.85	0.18	400
A17	17	785	150	118	2.22	0.18	250
A16	16	760	213	162	2.37	0.18	250

The selected PCMs for the project are S44, S34, S25, S21 S15, from PCM products Limited**.** The S44 and 34 PCMs were selected for the storage tanks to assist extract as much solar energy from the active solar system with the minimum solar radiation absorbed by the solar collectors. The S25, S21 and S15 PCMs selected for the heating pipes were based on the zones heating requirements.

Table 4:21 PlusICE PCM Hydrated Salt (S) Range Properties

PCM type	Phase change temp (°C)	Density (kg/m³)	Latent heat capacity (kJ/kg)	Volumetric heat capacity (MJ/m³)	Specific heat capacity (kJ/kg)	Thermal conductivity (W/mK)	Maximum operating temp (°C)
S117	117	1,450	160	232	2.61	0.700	140
S89	89	1,550	151	234	2.48	0.670	120
S83	83	1,600	141	226	2.31	0.620	120
S72	72	1,666	127	212	2.13	0.580	120
S70	70	1,680	110	185	2.10	0.570	120
S58	58	1,505	145	218	2.55	0.690	120
S50	50	1,601	100	160	1.59	0.430	120
S46	46	1,587	210	333	2.41	0.450	56
S44	44	1,584	100	158	1.61	0.430	120
S34	34	2,100	115	242	2.10	0.520	70
S32	32	1,460	200	292	1.91	0.510	60
S30	30	1,304	190	248	1.90	0.480	60
S27	27	1,530	183	280	2.20	0.540	60
S25	25	1,530	180	275	2.20	0.540	60
S23	23	1,530	175	268	2.20	0.540	60
S21	22	1,530	170	260	2.20	0.540	60
S19	19	1,520	160	243	1.90	0.430	60
S17	17	1,525	160	244	1.90	0.430	60
S15	15	1,510	160	242	1.90	0.430	60
S13	13	1,515	160	242	1.90	0.430	60

4.6.4 Solar collector selection for the research project

The efficiency and energy delivery of solar collectors were discussed in sections 3.4.3.3.4.1 and 3.4.3.3.4.2 respectively.

Several collectors were considered to select the best solar collector for the project based on their energy delivery efficiencies and factors affecting their operational capacities such as F_R which is the collector's heat removal factor, τ which is the transmittance of the cover, α which is the shortwave absorptivity of the absorber, G which is the global incident solar radiation on the collector, UL which is the overall heat loss coefficient of the collector, and ∆T being the temperature difference between the working fluid entering the collectors and the ambient air temperature.

These factors relate to the collector energy delivery equation of

$\dot{Q}_{coll}$ = F_R (τα) G - $F_R U_L$ ΔT. After considering the above factors and the overall efficiencies of multiple collectors, six collectors were selected to assess their energy delivery performance through a year cycle.

The solar collector details were supplied by the manufacturers.
The global incident solar radiation data was supplied by the meteorological department in London for three consecutive years covering 2010 to 2012. The collector performance assessment and analysis were carried out based on three years solar radiation data.
The global incident radiation data used for the calculation were supplied by the meteorological department from the following location data:
KEW GARDENS
NGR = 5185E 1772N
Altitude = 6 metres
Latitude = 51:48N
Longitude = 00:29W

Table 4:22 below describes the selected six collectors details sent by the manufacturers

Table 4:22 Detail description of the six selected solar collectors

Manufacturer	Collector type	Collector details		
		FR(τα)	FR UL	Collector aperture area (m2)
ACR	Glazed	0.6	3.73	1.72
Agena SA Energies	Glazed	0.85	4.12	2.463
Citrin	Glazed	0.73	3.67	1.9
Hoval Herzog	Evacuated tube	0.53	1.13	3.17
Shangdong Linuo	Evacuated tube	0.56	0.83	1
Thermomax	Evacuated tube	0.58	1.05	2.14

4.6.5 Monthly expected energy delivery by each solar collector

Daily expected energy delivery by each of the selected solar collector was calculated using the manufacturer's information and solar radiation data supplied by the meteorological depart in London. The calculation was based on how much energy that can be delivered by 1 m^2 aperture area of each collector. January 2010 data was used in the assessment and analysis. Any other month data could have been used to determine the energy performance of the solar collectors. Daily mean temperature and global solar radiation was used in the calculation.

Table 4:23 and table 4:24 are example calculations to determine expected daily energy delivery of Agena SA Energies and Thermomax solar collectors in January 2010.

Table 4:23 Daily expected energy delivery of Agena SA Energies solar collector in January 2010

Date	Daily Mean Temp (oC) (0900-0900)	Daily Maximum Temp (oC) (0900-0900)	Daily Minimum Temp (oC) (0900-0900)	Daily Total Global Radiation (KJ/m2)	FR(τα)	FR UL	Ti	Gcoll MJ/m2	Collector Aperture area (m2)	Total collector energy delivery (kWh/day)
01/01/2010	0.1	3.4	-0.5	4025	0.85	4.12	25	3.31	1.0	0.92
02/01/2010	1.9	5.7	-3.5	3663	0.85	4.12	25	3.01	1.0	0.84
03/01/2010	-1.8	2.7	-1.4	2550	0.85	4.12	25	2.05	1.0	0.57
04/01/2010	-3.4	2.6	-6.5	4648	0.85	4.12	25	3.83	1.0	1.06
05/01/2010	0.6	1.7	-6.7	1668	0.85	4.12	25	1.32	1.0	0.37
06/01/2010	-2.1	1.4	-3.0	721	0.85	4.12	25	0.50	1.0	0.14
07/01/2010	-1.1	0.5	-8.1	3353	0.85	4.12	25	2.74	1.0	0.76
08/01/2010	-1.5	1.0	-5.9	2588	0.85	4.12	25	2.09	1.0	0.58
09/01/2010	0.8	1.9	-3.3	3531	0.85	4.12	25	2.90	1.0	0.81
10/01/2010	1.2	2.3	-2.0	1274	0.85	4.12	25	0.98	1.0	0.27
11/01/2010	1.0	1.5	0.1	748	0.85	4.12	25	0.54	1.0	0.15
12/01/2010	1.0	2.1	0.3	990	0.85	4.12	25	0.74	1.0	0.21
13/01/2010	1.0	2.6	-0.3	1703	0.85	4.12	25	1.35	1.0	0.37
14/01/2010	2.2	3.7	-1.1	1225	0.85	4.12	25	0.95	1.0	0.26
15/01/2010	5.3	6.5	0.4	1910	0.85	4.12	25	1.54	1.0	0.43
16/01/2010	5.5	7.7	0.6	564	0.85	4.12	25	0.40	1.0	0.11
17/01/2010	3.5	10.1	0.1	4592	0.85	4.12	25	3.81	1.0	1.06
18/01/2010	5.5	6.5	-0.9	1539	0.85	4.12	25	1.23	1.0	0.34
19/01/2010	5.7	7.7	3.5	1080	0.85	4.12	25	0.84	1.0	0.23
20/01/2010	3.1	3.7	3.2	481	0.85	4.12	25	0.32	1.0	0.09
21/01/2010	5.6	7.7	2.1	3279	0.85	4.12	25	2.70	1.0	0.75
22/01/2010	7.5	8.3	2.8	579	0.85	4.12	25	0.42	1.0	0.12
23/01/2010	4.7	6.7	6.6	833	0.85	4.12	25	0.62	1.0	0.17
24/01/2010	4.3	6.1	2.9	2375	0.85	4.12	25	1.93	1.0	0.54
25/01/2010	3.0	4.5	1.6	1149	0.85	4.12	25	0.88	1.0	0.25
26/01/2010	0.1	3.0	1.0	4467	0.85	4.12	25	3.69	1.0	1.02
27/01/2010	4.5	7.0	-3.4	2140	0.85	4.12	25	1.73	1.0	0.48
28/01/2010	5.5	7.5	0.0	3233	0.85	4.12	25	2.66	1.0	0.74
29/01/2010	1.6	6.0	4.2	3024	0.85	4.12	25	2.47	1.0	0.69
30/01/2010	-0.7	4.5	-1.7	6447	0.85	4.12	25	5.37	1.0	1.49
31/01/2010	0.9	3.9	-4.7	5070	0.85	4.12	25	4.21	1.0	1.17
Total										16.98

Table 4:24 Daily expected energy delivery of Thermomax solar collector in January 2010

Date	Daily Mean Temp (oC) (0900-0900)	Daily Maximum Temp (oC) (0900-0900)	Daily Minimum Temp (oC) (0900-0900)	Daily Total Global Radiation (KJ/m2)	FR(τα)	FR UL	Ti	Gcoll MJ/m2	Collector Aperture area (m2)	Total collector energy delivered (kWh/day)
01/01/2010	0.1	3.4	-0.5	4025	0.58	1.05	25	2.31	1.0	0.64
02/01/2010	1.9	5.7	-3.5	3663	0.58	1.05	25	2.10	1.0	0.58
03/01/2010	-1.8	2.7	-1.4	2550	0.58	1.05	25	1.45	1.0	0.40
04/01/2010	-3.4	2.6	-6.5	4648	0.58	1.05	25	2.67	1.0	0.74
05/01/2010	0.6	1.7	-6.7	1668	0.58	1.05	25	0.94	1.0	0.26
06/01/2010	-2.1	1.4	-3.0	721	0.58	1.05	25	0.39	1.0	0.11
07/01/2010	-1.1	0.5	-8.1	3353	0.58	1.05	25	1.92	1.0	0.53
08/01/2010	-1.5	1.0	-5.9	2588	0.58	1.05	25	1.47	1.0	0.41
09/01/2010	0.8	1.9	-3.3	3531	0.58	1.05	25	2.02	1.0	0.56
10/01/2010	1.2	2.3	-2.0	1274	0.58	1.05	25	0.71	1.0	0.20
11/01/2010	1.0	1.5	0.1	748	0.58	1.05	25	0.41	1.0	0.11
12/01/2010	1.0	2.1	0.3	990	0.58	1.05	25	0.55	1.0	0.15
13/01/2010	1.0	2.6	-0.3	1703	0.58	1.05	25	0.96	1.0	0.27
14/01/2010	2.2	3.7	-1.1	1225	0.58	1.05	25	0.69	1.0	0.19
15/01/2010	5.3	6.5	0.4	1910	0.58	1.05	25	1.09	1.0	0.30
16/01/2010	5.5	7.7	0.6	564	0.58	1.05	25	0.31	1.0	0.09
17/01/2010	3.5	10.1	0.1	4592	0.58	1.05	25	2.64	1.0	0.73
18/01/2010	5.5	6.5	-0.9	1539	0.58	1.05	25	0.87	1.0	0.24
19/01/2010	5.7	7.7	3.5	1080	0.58	1.05	25	0.61	1.0	0.17
20/01/2010	3.1	3.7	3.2	481	0.58	1.05	25	0.26	1.0	0.07
21/01/2010	5.6	7.7	2.1	3279	0.58	1.05	25	1.88	1.0	0.52
22/01/2010	7.5	8.3	2.8	579	0.58	1.05	25	0.32	1.0	0.09
23/01/2010	4.7	6.7	6.6	833	0.58	1.05	25	0.46	1.0	0.13
24/01/2010	4.3	6.1	2.9	2375	0.58	1.05	25	1.36	1.0	0.38
25/01/2010	3.0	4.5	1.6	1149	0.58	1.05	25	0.64	1.0	0.18
26/01/2010	0.1	3.0	1.0	4467	0.58	1.05	25	2.56	1.0	0.71
27/01/2010	4.5	7.0	-3.4	2140	0.58	1.05	25	1.22	1.0	0.34
28/01/2010	5.5	7.5	0.0	3233	0.58	1.05	25	1.85	1.0	0.52
29/01/2010	1.6	6.0	4.2	3024	0.58	1.05	25	1.73	1.0	0.48
30/01/2010	-0.7	4.5	-1.7	6447	0.58	1.05	25	3.71	1.0	1.03
31/01/2010	0.9	3.9	-4.7	5070	0.58	1.05	25	2.92	1.0	0.81
Total										**11.95**

Table 4:23 and table 4:24 calculation method was used to calculate the expected monthly energy delivery of all the selected six solar collectors from January to December using 2010, 2011 and 2012 supplied solar radiation data from the meteorological department.

Table 4:25 to table 4:30 below shows the annual expected energy delivery of each of the six selected solar collectors and comparison is made to establish the differences between the years energy delivery from 2010 to 2012.

The results demonstrated the differences in energy delivery between the years to be minimal which suggests that the results of any of the years or the average value can be relied upon. The assessment of the solar collector energy delivery performance was based on 25 ^{o}C water inlet temperature.

Table 4:25 Monthly expected energy delivery of Agena SA Energies solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	16.98	14.54	20.37	14.38	-19.96	-40.12
February	24.07	22.13	23.10	8.06	4.03	-4.38
March	61.88	59.26	60.57	4.24	2.12	-2.21
April	104.48	108.61	106.54	-3.95	-1.98	1.90
May	117.59	132.51	126.07	-12.69	-7.21	4.86
June	136.32	123.18	110.07	9.64	19.25	10.64
July	124.06	121.11	120.98	2.38	2.49	0.11
August	98.26	98.26	114.63	0.00	-16.66	-16.66
September	74.34	74.34	85.64	0.00	-15.20	-15.20
October	48.11	48.11	43.19	0.00	10.24	10.24
November	19.14	22.07	24.69	-15.32	-28.99	-11.85
December	9.98	16.07	15.51	-61.04	-55.40	3.50
Annual Total (kWh	**835.22**	**840.19**	**851.36**	**-0.60**	**-1.93**	**-1.33**

Table 4:26 Monthly expected energy delivery of ACR solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	11.84	10.13	14.26	14.41	-20.48	-40.76
February	16.88	15.53	16.20	7.99	4.00	-4.34
March	43.61	41.76	42.68	4.24	2.12	-2.22
April	73.73	76.67	75.20	-3.98	-1.99	1.92
May	83.01	93.57	89.01	-12.73	-7.24	4.87
June	96.29	86.98	77.72	9.66	19.28	10.65
July	87.64	85.53	85.44	2.41	2.51	0.10
August	69.38	69.38	80.96	0.00	-16.68	-16.68
September	52.47	52.47	60.45	0.00	-15.20	-15.20
October	33.92	33.92	30.42	0.00	10.32	10.32
November	13.40	15.50	17.32	-15.67	-29.31	-11.79
December	6.89	11.23	10.82	-63.07	-57.17	3.61
Annual Total (kWh)	**589.04**	**592.67**	**600.49**	**-0.62**	**-1.94**	**-1.32**

Table 4:27 Monthly expected energy delivery of Citrin solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	14.58	12.48	17.50	14.39	-20.03	-40.20
February	20.68	19.01	19.85	8.05	4.02	-4.38
March	53.19	50.93	52.06	4.24	2.12	-2.21
April	89.82	93.37	91.59	-3.96	-1.98	1.90
May	101.09	113.93	108.38	-12.70	-7.22	4.86
June	117.21	105.90	94.63	9.65	19.26	10.64
July	106.67	104.12	104.01	2.39	2.49	0.11
August	84.48	84.48	98.55	0.00	-16.66	-16.66
September	63.91	63.91	73.63	0.00	-15.20	-15.20
October	41.35	41.35	37.12	0.00	10.25	10.25
November	16.44	18.96	21.21	-15.37	-29.03	-11.84
December	8.56	13.80	13.31	-61.30	-55.62	3.52
Annual Total (kWh	**717.96**	**722.25**	**731.85**	**-0.60**	**-1.93**	**-1.33**

Table 4:28 Monthly expected energy delivery of Hoval Herzog solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	10.89	9.33	12.95	14.33	-18.98	-38.88
February	15.26	14.02	13.78	8.18	9.72	1.68
March	38.85	37.21	36.34	4.22	6.47	2.35
April	65.40	67.94	66.56	-3.88	-1.77	2.03
May	73.57	82.87	78.84	-12.63	-7.17	4.85
June	85.20	77.01	68.84	9.61	19.20	10.61
July	77.52	75.71	75.63	2.34	2.44	0.11
August	61.44	61.44	71.65	0.00	-16.60	-16.60
September	46.52	46.52	53.60	0.00	-15.21	-15.21
October	30.18	30.18	27.14	0.00	10.08	10.08
November	12.18	13.96	15.63	-14.65	-28.36	-11.96
December	6.52	10.26	9.92	-57.28	-52.10	3.30
Annual Total (kWh)	**523.54**	**526.45**	**530.87**	**-0.56**	**-1.40**	**-0.84**

Table 4:29 Monthly expected energy delivery of Shangdong Linuo solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	11.57	9.92	13.74	14.32	-18.76	-38.60
February	16.19	14.86	15.87	8.20	1.96	-6.80
March	41.11	39.38	42.03	4.22	-2.24	-6.74
April	69.15	71.82	68.97	-3.87	0.26	3.97
May	77.78	87.59	83.34	-12.62	-7.16	4.85
June	90.04	81.40	72.77	9.60	19.19	10.61
July	81.93	80.02	79.93	2.33	2.43	0.11
August	64.95	64.95	75.72	0.00	-16.59	-16.59
September	49.18	49.18	56.66	0.00	-15.21	-15.21
October	31.93	31.93	28.72	0.00	10.04	10.04
November	12.92	14.79	16.57	-14.49	-28.21	-11.99
December	6.97	10.90	10.54	-56.43	-51.35	3.25
Annual Total (kWh)	**553.72**	**556.74**	**564.88**	**-0.55**	**-2.02**	**-1.46**

Table 4:30 Monthly expected energy delivery of Thermomax solar collector

Month	Monthly collector energy delivery (KWh) 2010	Monthly collector energy delivery (KWh) 2011	Monthly collector energy delivery (KWh) 2012	% Generated energy delivery difference 2010 & 2011	% Generated energy delivery difference 2010 & 2012	% Generated energy delivery difference 2011 & 2012
January	11.95	10.24	14.20	14.32	-18.87	-38.74
February	16.73	15.36	17.6	8.19	-5.17	-14.56
March	42.55	40.75	43.7	4.22	-2.70	-7.23
April	71.60	74.37	72.8	-3.88	-1.68	2.11
May	80.53	90.70	86.30	-12.63	-7.16	4.85
June	93.25	84.29	75.35	9.60	19.19	10.61
July	84.84	82.87	82.77	2.33	2.44	0.11
August	67.25	67.25	78.42	0.00	-16.60	-16.60
September	50.92	50.92	58.67	0.00	-15.21	-15.21
October	33.05	33.05	29.72	0.00	10.06	10.06
November	13.35	15.30	17.13	-14.57	-28.28	-11.97
December	7.18	11.26	10.89	-56.86	-51.73	3.27
Annual Total (kWh)	**573.21**	**576.37**	**587.56**	**-0.55**	**-2.50**	**-1.94**

The annual energy delivery results of the six selected collectors analysis demonstrated that Agena SA Energies solar collector performs best compare with the others.

4.7 Solar collector further performance comparison

The selected solar collectors include three flat plate collectors and three evacuated tubes. The flat plate collectors were ACR glazed collector, Agena SA Energies glazed collector and Citrin glazed collector. The evacuated tube collectors were Hoval evacuated tube, Shangdong evacuated tube and Thermomax evacuated tube.

The performances of the six collectors were further assessed based on 25, 30, 35, 50 and 70 ^{o}C hot water entry temperatures. For the purpose of this report only 25 and 50 ^{o}C hot water entry temperatures are demonstrated below. The 25 and 50 ^{o}C entry temperatures were selected at random just to demonstrate the effect that the entry water temperature has on the solar active system efficiency.

Figure 4:28 and figure 4:29 below demonstrate annual energy delivery performance of the collectors. The assessment established that the performance efficiency reduces as the inlet hot water temperature increases and this is substantiated in figure 4:29

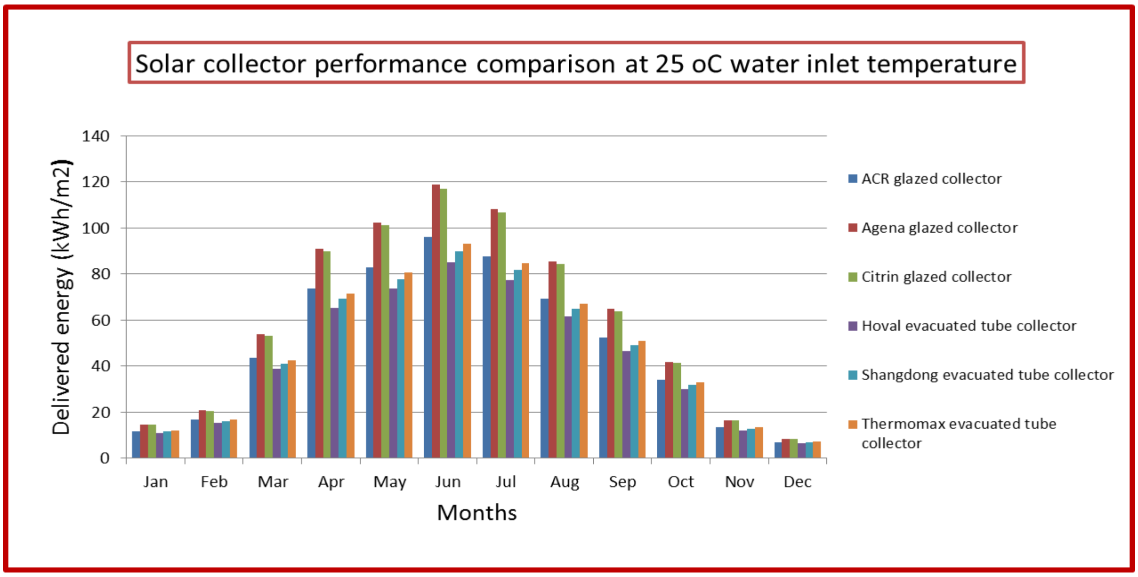

Figure 4:28 Solar collector performance comparison at hot water inlet temperature of 25 ^{o}C

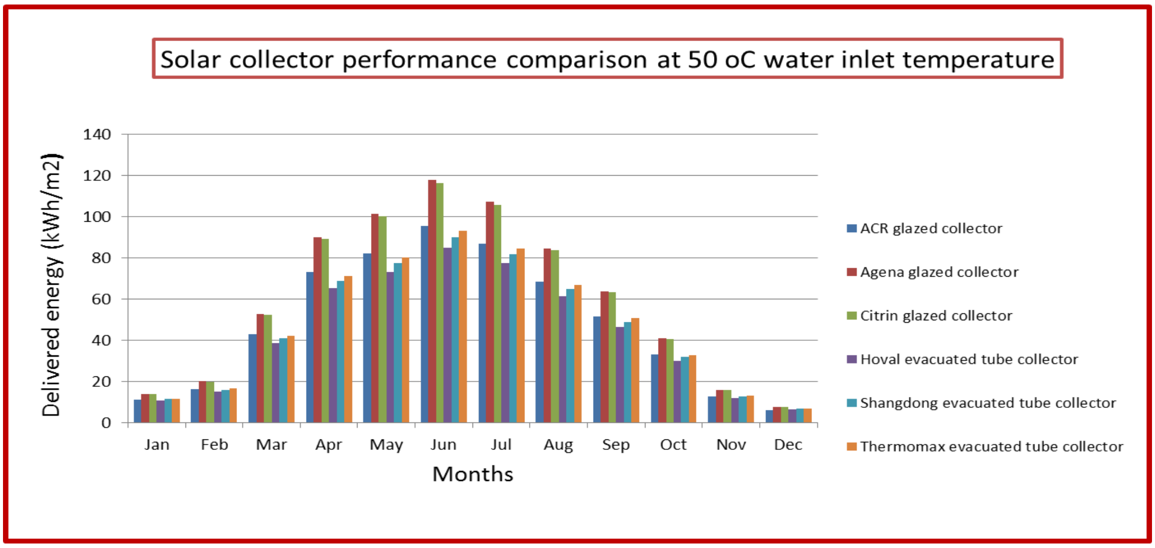

Figure 4:29 Solar collector performance comparison at hot water inlet temperature of 50 ^{o}C

The energy performance assessment and analysis of the solar collectors confirmed Agena SA Energies collector to be the best among the others. Now that Agena SA Energies solar collector has been selected it will be beneficial to carry out more detail performance assessment of the collector in terms of yearly solar incident radiation and hot water inlet temperature to the solar collector system.

The first assessment was to establish the difference in performance in relation to solar incident radiation data supplied by the meteorological department for the three years. It was established that the differences in performance of using 2010, 2011 and 2012 global solar radiation data supplied are minimal.

The second performance assessment was to establish the effect of water inlet temperature to the yearly energy delivery.

Table 4:31 illustrates the collector delivery energy at inlet water temperatures of 20 to 70 ^{o}C. The difference in the energy delivery is quite significant. For example, the difference between energy delivered at 25 and 30 ^{o}C is 2.1 kWh / m^2. The difference between the yearly energy delivered between 20 and 70 ^{o}C inlet temperatures is 21.7 kWh / m^2 which is quite significant.

This emphasise the need to keep the inlet water temperature to the solar collector system low. The result is graphically demonstrated in figure 4:30 below.

The assessment was based on 2012 solar incident radiation data obtained from the Metrological office London.

Table 4:31 Agena SA Energies solar collector system performance at 20 to 70 °C inlet water temperatures

Month	20 oC inlet water temp	25 oC inlet water temp	30 oC inlet water temp	35 oC inlet water temp	50 oC inlet water temp	60 oC inlet water temp	70 oC inlet water temp
January	20.55	20.37	20.19	20.02	19.48	19.13	18.78
February	23.26	23.10	22.94	22.78	22.30	21.98	21.66
March	60.75	60.57	60.39	60.22	59.68	59.33	58.97
April	106.71	106.54	106.37	106.20	105.68	105.34	105.00
May	126.24	126.07	125.89	125.71	125.18	124.83	124.47
June	110.25	110.07	109.90	109.73	109.22	108.87	108.53
July	121.16	120.98	120.80	120.62	120.09	119.74	119.38
August	114.80	114.63	114.45	114.27	113.74	113.38	113.03
September	85.81	85.64	85.47	85.30	84.78	84.44	84.10
October	43.36	43.19	43.01	42.83	42.30	41.94	41.59
November	24.86	24.69	24.51	24.34	23.83	23.48	23.14
December	15.69	15.51	15.33	15.16	14.62	14.27	13.91
Total (kWh /m2)	**853.45**	**851.36**	**849.27**	**847.18**	**840.92**	**836.74**	**832.56**

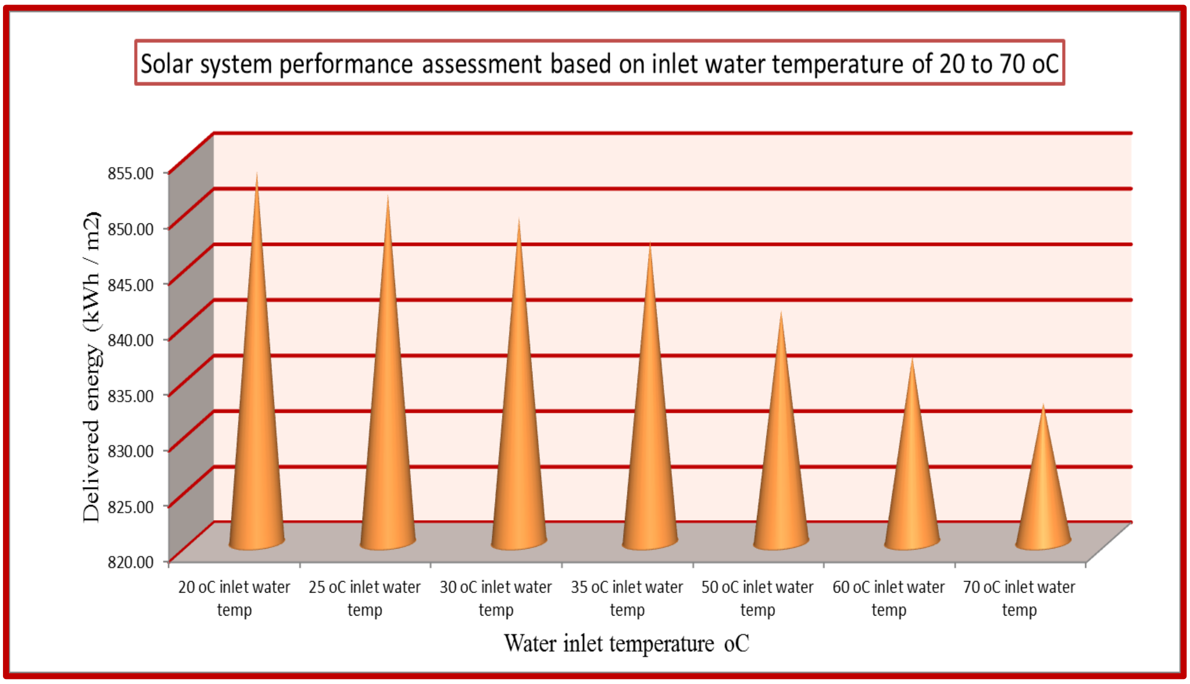

Figure 4:30 Agena SA Energies collector performance at 20 to 70 °C inlet temperatures

Summary of chapter 4

The heating system was designed to achieve nearly zero carbon emission energy generating system. The integrated environmental solutions (IES) software tool was used to design the glasshouse to determine zones hourly, daily and monthly heating demand, heating energy requirements, temperature and relative humidity profiles.

This was important to know the periods where the thermal energy generated and stored in the PCM storage tanks by the active solar thermal hot water system will be used.

The simulation results of the designed glasshouse established that the heating energy demand for the zones and solar gains are almost the same therefore enough solar energy could be useful if the solar gains in the glasshouse could be stored without venting it to the atmosphere.

The temperature and relative humidity profiles of the design simulation results were validated against the actual measured values of the glasshouse using a calibrated temperature and relative humidity measuring tool to ascertain that the designed glasshouse is a representative of the existing glasshouse in terms of shape, size and form.

The validation was satisfactory and this was the bases from which all the other calculations were based. This was to ensure that all calculations will be true representation of the building.

The hot water supply temperature of the heating system has been kept low to increase the solar hot water system efficiency and again minimise distribution losses through the pipework.

The system has been designed such that the little thermal energy generated by the solar collector system will still be useful thus why 44 and 34 ^{o}C phase change temperature PCMs have been selected.

This is to say that heat can still be transferred to the PCM storage tank (34 ^{o}C PCM tank) when the return hot water temperature from the solar collector system for example is 40 ^{o}C or lower unless the supply temperature falls below 35 ^{o}C.

The waste heat from the CHP plant close to the glasshouse under research will be used as a supplementary source of thermal energy in winter.

Basically the zones space set point temperatures will be maintained through freezing and melting of the PCM heating pipes. The water flow through the PCM heating pipes will be abled and disabled by motorised valves to ensure that thermal energy is appropriately distributed.

Grundfos MAGNA3 variable speed pump will regulate the flow efficiently to ensure system efficiency and it will measure the thermal energy generated by the solar collectors to determine the solar collector’s efficiency to assist future projects implementation.

The glasshouse simulation results demonstrate that the amount of solar gains in the zones is almost equal to the heating demand for the design year that is why efforts have been made to absorb this energy using PCM heating pipes.

The phase change material (PCM) was selected to absorb maximum heat from the solar collector system and the heat trapped in the glasshouse. The simulation results demonstrated that the heat energy stored in the PCM storage tanks will be needed in winter as the solar gains and absorbed heat energy stored by the PCM heating pipes is enough to meet the glasshouse heating demand from March to October.

The melting and freezing times of the selected PCMs were calculated to establish the length of time that it will take the PCM filled in the PCM heating pipe to melt and freeze. This will determine the heat that can be transferred or absorbed by the PCM heating pipes within specific period.

The amount of energy that could be delivered by the solar collector systems throughout the year from January to December was calculated to size the solar collector system that will produce enough energy to meet the glasshouse heating demand especially from March to October.

The designed heating system has demonstrated that managing the heat trapped inside the glasshouse effectively and supplying low inlet water temperature to the solar collector system (active system) will produce enough heat energy to meet the glasshouse heating demand.

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5 Glasshouse heat transfer analysis

5.1 Introduction

This section of the research is to analysis the processes and principles of the heat transfer to ascertain that sufficient heat can be produced to condition the glasshouse by the designed heating system to promote effective growth of the plants.

The IES software was used to calculate heating requirements of each of the twenty one zones. Table 5:1 below shows the maximum heating demand of each zone for the design outside temperature of -3.4 ^{o}C. Twenty per cent (20 %) heating load demand has been added to the calculated load to account for distribution losses through pipes, thermal energy storage tanks and other components in the heating system.

It is also to account for unforeseen future heating demand and calculation errors that might have occurred during the calculation process.

Table 5:1 includes heating load, low temperature hot water (LTHW) flow and return temperatures, mass and volume flow rates requirement for each zone. The flow and return temperatures of 32 ^{o}C and twenty 25 ^{o}C were selected to reduce distribution heat losses and also to improve the solar heating system performance.

The solar heating system efficiency is increased at low water entry temperature. The determination of the LTHW mass flow rate was important to select correct pipe sizes, flow velocities and provision of sufficient thermal energy to each zone to ensure system efficiency. Water properties are selected from the mean value of 32 and 25 ^{o}C which was calculated to be 28.5 ^{o}C and approximated to be 29 ^{o}C. Table 5:1 shows mass and volume flow rate requirements to each zone.

Table 5:1 Zones heating demand, mass flow rate and volume flow rate of LTHW

Zones	Heating load (KW)	Heating load plus 20% (KW)	LTHW flow temperature (oC)	LTHW return temperature (oC)	Specific heat capacity of water (kJ/kgK) at 29 oC	LTHW mass flow rate (Kg/s)	Water density at 29 oC (kg/m3)	LTHW volume flow rate (m3/s)
1	30	36	32	25	4.18	1.2303486	995.9	0.001235414
2	9.3	11.16	32	25	4.18	0.38140807	995.9	0.000382978
3	11.5	13.8	32	25	4.18	0.47163363	995.9	0.000473575
4	10	12	32	25	4.18	0.4101162	995.9	0.000411805
5	1.5	1.8	32	25	4.18	0.06151743	995.9	6.17707E-05
6	18	21.6	32	25	4.18	0.73820916	995.9	0.000741248
7	27	32.4	32	25	4.18	1.10731374	995.9	0.001111872
8	70	84	32	25	4.18	2.8708134	995.9	0.002882632
9	87	104.4	32	25	4.18	3.56801094	995.9	0.0035827
10	42	50.4	32	25	4.18	1.72248804	995.9	0.001729579
11	60	72	32	25	4.18	2.4606972	995.9	0.002470828
12	45	54	32	25	4.18	1.8455229	995.9	0.001853121
13	26	31.2	32	25	4.18	1.06630212	995.9	0.001070692
14	40	48	32	25	4.18	1.6404648	995.9	0.001647218
15	28	33.6	32	25	4.18	1.14832536	995.9	0.001153053
16	27	32.4	32	25	4.18	1.10731374	995.9	0.001111872
17	46	55.2	32	25	4.18	1.88653452	995.9	0.001894301
18	45	54	32	25	4.18	1.8455229	995.9	0.001853121
19	69	82.8	32	25	4.18	2.82980178	995.9	0.002841452
20	58	69.6	32	25	4.18	2.37867396	995.9	0.002388467
21	39	46.8	32	25	4.18	1.59945318	995.9	0.001606038
Total	**789.3**	**947.16**				**32.3704716**		**0.032503737**

5.1.1 Pipe sizing

The zones space heating emitting device or component in the glasshouse heating system is the heating pipes therefore appropriate sizing is important to achieve heating demand in the zones.

Table 5.2 below is a pipe selection guide to assist designers select appropriate pipe sizes to achieve effective flow velocity that will maintain system flow distribution efficiency.

The guide is produced by Building Services Research and Information Association (BSRIA). Typical water velocities for pipework shown in

Table 5:2 and table of internal diameters of steel and iron pipes from CIBSE Guide C Table 4.2 were used to select appropriate pipe sizes which will not only ensure that appropriate velocity is achieved but will also ensure that appropriate heat transfer is achieved in each zone.

Table 5:2 Typical water velocities for pipework (BSRIA[15])

Situation/diameter	Velocity (m·s–1)
Small bore	<1.0
Diam. 15 to 50 mm	0.75–1.15
Diam. > 50 mm	1.25–3.0
Heating/cooling coils	0.5–1.5

5.1.2 Determining the appropriate pipe sizes

To determine the minimum pipe size appropriate for water flow rate, the table of internal diameters of steel and iron pipes from CIBSE Guide C Table 4.2 and the following formulas were used.

The table of internal diameters of steel and iron pipes shown in table 5:4 below was used as a guide to select the appropriate pipe internal diameters of the heating pipes.

$$d_m = \sqrt{(4q / \pi C)} \qquad (5.1.1)$$

Where d_m is the minimum diameter required, q is volumetric flow rate, C is mean fluid velocity. Calculation of Reynolds number

$$Re = Cd_i/V \qquad (5.1.2)$$

Where C and V is the fluid mean velocity and viscosity respectively

If the flow is laminar ($Re < 2000$), f is obtained from

$$f = 64 / Re \qquad (5.1.3)$$

Where f is the friction factor

If the flow is turbulent ($Re > 3000$), f is obtained from

$$1 / \sqrt{f} = -1.8 \log [6.9/Re + (\lambda /d / 3.71)^{1.11}] \qquad (5.1.4)$$

λ is the fluid thermal conductivity

The pressure drop, or the pressure drop per unit length, is obtained from

$$\Delta p = f Pv/d_i \qquad (5.1.5)$$

Pv is the velocity pressure, d_i is the pipe internal diameter and f is the friction factor

Table 5:3 below illustrates the calculation of the annulus heating pipe internal diameters using excel spread sheet to achieve the required fluid velocity. All other heat transfer calculations are based on the selected internal pipe diameters in table 5:3.

The design of the annular pipe is bespoke to meet the required annular pipe area (Area between the inner and the outer pipe) to store the PCM solution as demonstrated in figure 5:4.

Table 5:3 Calculation of required inner diameter, velocity and pressure drop per unit of pipe length for hot water distribution to zones including Reynolds numbers

Zones	zones m.f.r	Density (kg/m3)	Cmax (m/s)	dmin (m)	Closer pipe size from table (m)	Actual vel (m/s)	Vel pressure Pv (Pa)	Viscosity V (m2/s)	Reynolds No (Re)	Roughness (mk)	(λ / d) /(3.71) raised to power 1.11	[6.9/Re+ ((k/d)/(3.71))1.11]	$1/\sqrt{f}$	f	Pressure drop ΔP (Pa)
1	0.861	995	2	0.023	0.042	0.624532	194.0448	8.005E-07	32767.43	0.046	0.00012074	0.00022102	6.58	0.023	106.7081
2	0.2669	995	2	0.013	0.022	0.705618	247.7033	8.005E-07	19392.36	0.046	0.000247495	0.00037898	6.158	0.026	296.8654
3	0.3301	995	2	0.015	0.022	0.872538	378.7579	8.005E-07	23979.8	0.046	0.000247495	0.00031091	6.313	0.025	431.9479
4	0.287	995	2	0.014	0.022	0.758729	286.3954	8.005E-07	20852	0.046	0.000247495	0.00035407	6.212	0.026	337.3888
5	0.0431	995	2	0.005	0.022	0.113809	6.443896	8.005E-07	3127.801	0.046	0.000247495	0.00222919	4.773	0.044	12.85529
6	0.5166	995	2	0.018	0.022	1.365711	927.921	8.005E-07	37533.61	0.046	0.000247495	0.000207	6.631	0.023	959.1748
7	0.7749	995	2	0.022	0.042	0.562079	157.1763	8.005E-07	29490.69	0.046	0.00012074	0.00024442	6.501	0.024	88.53772
8	2.0091	995	2	0.036	0.042	1.457241	1056.466	8.005E-07	76457.35	0.046	0.00012074	0.00010069	7.195	0.019	485.9482
9	2.497	995	2	0.04	0.042	1.811142	1631.917	8.005E-07	95025.56	0.046	0.00012074	8.3056E-05	7.345	0.019	720.1933
10	1.2055	995	2	0.028	0.042	0.874344	380.3278	8.005E-07	45874.41	0.046	0.00012074	0.00016085	6.828	0.021	194.2085
11	1.7221	995	2	0.033	0.042	1.249063	776.1793	8.005E-07	65534.87	0.046	0.00012074	0.00011573	7.086	0.02	368.0748
12	1.2916	995	2	0.029	0.042	0.936798	436.6008	8.005E-07	49151.15	0.046	0.00012074	0.00015083	6.879	0.021	219.6937
13	0.7462	995	2	0.022	0.042	0.541261	145.7492	8.005E-07	28398.44	0.046	0.00012074	0.00025341	6.473	0.024	82.81937
14	0.3731	995	2	0.015	0.022	0.986347	484.0082	8.005E-07	27107.6	0.046	0.000247495	0.00027771	6.402	0.024	536.8595
15	0.8036	995	2	0.023	0.042	0.582896	169.0346	8.005E-07	30582.94	0.046	0.00012074	0.00023606	6.529	0.023	94.42595
16	0.7749	995	2	0.022	0.042	0.562079	157.1763	8.005E-07	29490.69	0.046	0.00012074	0.00024442	6.501	0.024	88.53772
17	1.3203	995	2	0.029	0.042	0.957615	456.2209	8.005E-07	50243.4	0.046	0.00012074	0.00014778	6.895	0.021	228.5035
18	1.2916	995	2	0.029	0.042	0.936798	436.6008	8.005E-07	49151.15	0.046	0.00012074	0.00015083	6.879	0.021	219.6937
19	1.9804	995	2	0.036	0.042	1.436423	1026.497	8.005E-07	75365.1	0.046	0.00012074	0.000102	7.185	0.019	473.4902
20	1.6647	995	2	0.033	0.042	1.207428	725.2964	8.005E-07	63350.37	0.046	0.00012074	0.00011936	7.062	0.02	346.3017
21	1.1193	995	2	0.027	0.042	0.811891	327.9357	8.005E-07	42597.66	0.046	0.00012074	0.00017242	6.774	0.022	170.1505

Notes:

m.f.r: mass flow rate

Cmax: maximum velocity

d_{min}: minimum diameter

mk: pipe material roughness

Table 5:4 Internal diameters of steel and iron pipes

Non alloy steel (BS EN10255)				Ductile iron (BS EN 545)			Seamless and welded steel series 1(BS EN 10220)		
Nominal pipe size	Specified outside diameter (mm)	Inside diameter (mm)		Nominal outside diameter (mm)	Inside diameter (mm)		Inside diameter (mm)	Wall thickness (mm)	Inside diameter (mm)
		Medium	Heavy		Class 40	Type K9			
6	10.2	6.2	5.0				10.2	1.4	7.4
8	13.5	9.0	7.8				13.5	1.6	10.3
10	17.2	12.5	11.3				17.2	1.8	13.6
15	21.3	16.2	15.0				21.3	2.0	17.3
20	26.9	21.7	20.5				29.1	2.3	26.9
25	33.7	27.4	25.8				33.7	2.3	22.3
32	42.4	36.1	34.5				42.4	2.3	37.8
40	48.3	42.0	40.4	56	46.4	44.0	48.3	2.6	43.1
50	60.3	53.1	51.3	66	65.4	54.0	60.3	2.6	55.1
60				77	67.4	65.0	76.1	2.6	70.9
65	76.1	68.8	67.0	82	72.4	70.0			
80	88.9	80.8	78.8	98	88.4	86.08	88.9	2.6	83.7
100	114.3	105.1	103.3	118	108.4	106.0	114.3	2.9	108.5
125	139.7	129.7	128.9	144	134.4	132.0	139.7	3.2	133.3
150	165.1	155.2	154.4	170	160.0	158.0	168.3	3.2	161.9

Source: CIBSE Guide C Table 4.2

5.2 Heat transfer assessment and analysis in the glasshouse

5.2.1 Physical consideration of free convection

Fluid motion in free convection is due to buoyancy forces within the fluid whilst forced convection is externally imposed. In free or natural convection, the body force in practice is usually gravitational and could be centrifugal or Coriolis force.

Mass density gradient may arise in a fluid in several ways but temperature gradient is the most common. The density of gases and liquids depends on temperature which generally decreases as the temperature rises due to fluid expansion.

The presence of a fluid density gradient in a gravitational field is not a guarantee that free convection currents exist.

Free convection effects depend on the expansion coefficient β. The manner in which β is obtained depends on the fluid. For an ideal gas ρ = p/RT and

$\beta = 1/\rho\,(\partial\rho / \partial T)_p = 1/\rho\,p/RT^2 = 1/T$ (5.2.1)

Where T is the absolute temperature. For liquids and non-ideal gases, β must be obtained from appropriate property tables.

In free or natural convection Grashof number is defined as

$G_{rLn} = g\beta\,(Ts - T\infty)L_n^3 / \alpha v$ (5.2.2)

Grashof number plays the same role in free convection as that of Reynolds number plays in forced convection. In free or natural convection the heat transfer correlations is of the form: $N_{uL} = f(Gr_{Ln}, Pr)$ (5.2.3)

Using Newton's law of cooling for the local convection coefficient h, the local Nusselt number may be expressed as:

$$Nu_x = \frac{hx}{\lambda} = -\left(\frac{Gr_x}{4}\right)^{1/4} \frac{dT*}{d\eta}\bigg|_{\eta = 0} = \left(\frac{Gr_x}{4}\right)^{1/4} g(Pr) \qquad (5.2.4)$$

The average convection coefficient for a surface of length L_n is given by

$$\overline{Nu_{Ln}} = \frac{\bar{h}Ln}{\lambda} = \frac{4}{3}\left(\frac{Gr_{Ln}}{4}\right)^{1/4} g(Pr) \qquad (5.2.5)$$

A correlation that may be applied over the entire range of RaL_n has been recommended by Churchill and Chu [1] and is of the form

$$\overline{Nu_{Ln}} = \left\{0.825 + \frac{0.387\,Ra_{Ln}^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}}\right\} \qquad (5.2.6)$$

For most engineering calculations equation 5.2.6 is suitable but in laminar flow, slightly better accuracy may be obtained using the correlation below (5.2.7) [1]

$$\overline{Nu_D} = \left\{0.68 + \frac{0.670\,Ra_D^{1/4}}{[1+(0.492/Pr)^{9/16}]^{4/9}}\right\} \quad Ra_{Ln} \lesssim 10^9 \qquad (5.2.7)$$

Notes: For circular geometry the length L_n will be replaced by the diameter D

5.2.2 Empirical correlations: External free convection flows

Grashof number Gr and the Rayleigh number Ra are two dimensionless parameters introduced in free convection flows. They both appear in laminar and turbulent flow conditions and in geometries other than a flat plate in empirical correlations.
The correlations are suitable for many engineering calculations and are often of the form:

$$\overline{Nu}_{Ln} = \frac{\bar{h}Ln}{\lambda} = CRa_{Ln}^{n} \tag{5.2.8}$$

Where the Raleigh number is

$$Ra_{Ln} = Gr_{Ln}\,\mathrm{Pr} = \frac{g\beta(T_s - T_\infty)Ln^3}{va} \tag{5.2.9}$$

based on the characteristic length L_n of the geometry. Typically, n = 1/4 and 1/3 for laminar and turbulent flows respectively (Equation 5.2.8).
All properties are evaluated at the film temperature, $T_f = (Ts + T_\infty) / 2$ (5.2.10)

The Grashof number is a measure of the ratio of the buoyancy forces to the viscous forces acting on the fluid in free or natural convection.

5.2.3 Determining heat transfer coefficient in force convection

Local and average convection coefficients may be related by equations of the form:

$$\mathrm{Nu}_{\mathrm{Ln}} = f(\mathrm{x}^*, \mathrm{Re}_{\mathrm{x}}, \mathrm{Pr}) \tag{5.2.11}$$

$$\overline{\mathrm{Nu}}_{\mathrm{x}} = f(\mathrm{Re}_{\mathrm{x}}, \mathrm{Pr}) \tag{5.2.12}$$

Equation 5.2.11 and 5.2.12 refers to local and average convection coefficients respectively. A particular location on a surface is indicated by the subscript x and the overbar indicates an average from $x^* = 0$ where the boundary layer starts to develop to the interest location.

Data may then be represented by an algebraic expression of the form below of all the fluids.

$$\overline{Nu_x} = C\, Re_x^m\, Pr^n) \tag{5.2.13}$$

The values of *C*, *m*, and *n* are often independent of the fluid nature.
The variation of fluid properties can influence the heat transfer rate with temperature across the boundary layer.

The influence in the heat transfer rate can be solved by evaluating all the fluid properties at a mean boundary layer temperature called the film temperature represented by equation 5.2.14

$$T_f \equiv \frac{T_s + T_\infty}{2} \tag{5.2.14}$$

The average heat transfer coefficient is of the form in lamina flow

$$\overline{Nu_x} \equiv \frac{\bar{h}_x x}{\lambda} = 0.664\, Re_x^{1/2}\, Pr^{1/3} \quad Pr \gtrsim 0.6 \tag{5.2.15}$$

The subscript *x* may be replaced by L_n if the flow is laminar over the entire surface. The surface temperature is assumed to be uniform for flat plates

5.2.4 Methodology for a convection calculation

5.2.4.1 Cross flow over cylindrical shapes

Figure 5:1 demonstrates a cross flow over a cylinder where the free stream fluid is brought to rest at the forward stagnation point with an accompanying pressure rise.

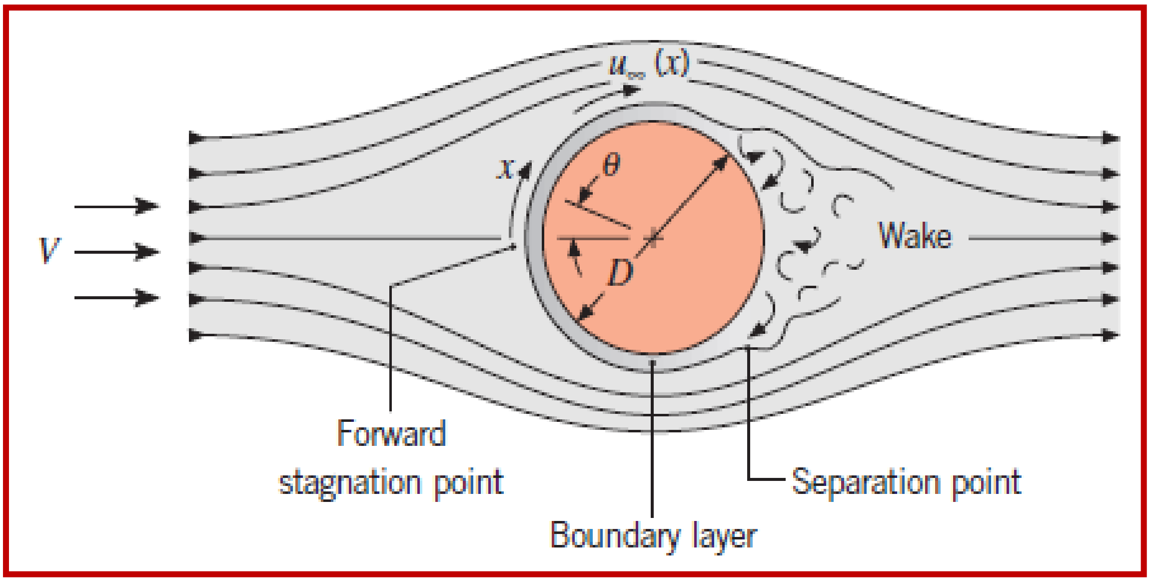


Figure 5:1 A cross flow over a cylinder

The boundary layer in figure 5:2 will develop under the influence of a pressure gradient ($dp / dx < 0$) as x increases and the pressure decreases from the stagnation point.

Further boundary layer will be developed towards the rear of the cylinder as minimum pressure is reached with an adverse pressure gradient of ($dp / dx > 0$).

The upstream velocity V and the free stream velocity u in figure 5:1 will differ with u_∞ from the stagnation point depending on the distance x.

From Euler's equation of an inviscid flow [2], the velocity $u_\infty (x)$ must exhibit opposite behaviour to that of p(x).

The fluid accelerates from $u_\infty = 0$ at the stagnation point as it has a favourable pressure gradient ($du_\infty / dx > 0$ when $dp / dx < 0$) until it reaches a maximum velocity when $dp / dx = 0$ and decelerates at the adverse pressure gradient ($du_\infty / dx < 0$ when $dp / dx > 0$). Figure 5:2 below describes the principles.

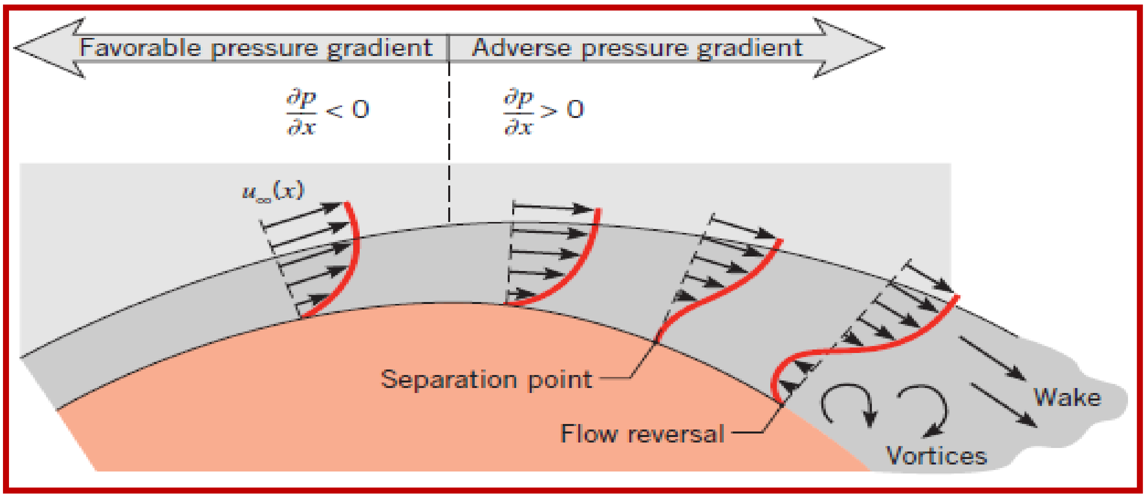


Figure 5:2 Velocity profile associated with separation on a circular cylinder in cross flow

The velocity gradient of the fluid decelerates as du / dy_y = 0 at the fluid surface and eventually becomes zero (Figure 5:2).

The downstream movement of the fluid becomes impossible as the fluid near the surface lacks sufficient momentum to overcome the pressure gradient at the separation point.

A wake is formed in the downstream region as the boundary layer detaches from the surface. The flow in this region is highly irregular and characterised by vortex formation.

The separation point is the location for which du / dy_y = 0. Coutanceau and Defaye [3] provided an excellent review of flow conditions in the wake of a circular cylinder.

The transition occurrence of the boundary layer depends on the Reynolds number and influences the separation point position. The characteristic length of a circular cylinder is the diameter and the Reynolds number is defined as:

$$Re_D \equiv \rho VD / \mu = VD / v \qquad (5.2.16)$$

5.2.5 Force convection heat transfer

Correlations may be obtained for the local Nusselt number and at the forward stagnation point for Pr ≥ 0.6. The boundary layer analysis [4] yields an expression of the following form for which is most accurate at low Reynolds number:

$$Nu_D(\theta = 0) = 1.15\, Re_D^{1/2}\, Pr^{1/3} \tag{5.2.17}$$

From the engineering standpoint calculations, the overall average condition is the point of interest. An empirical correlation has been modified to account for fluids of various Prandtl numbers due to Hilpert [5]

$$\overline{Nu_D} \equiv \frac{\bar{h}D}{\lambda} = C\, Re_D^m\, Pr^{1/3} \tag{5.2.18}$$

Equation 5.2.18 is widely used for Pr ≥ 0.7, where the constants C and m are listed in Table 5:5 below. Equation 5.2.18 may also be used for flow over cylinders of noncircular cross section, with the characteristic length D and the constants obtained.

In working with equations 5.2.16 and 5.2.17 all properties are evaluated at the film temperature. Other correlations have been suggested for the circular cylinder in cross flow [6, 7, 9].

The correlation due to Zukauskas [7] is of the form

$$Nu_D = C\, Re_D^m\, Pr^n \left(\frac{Pr}{Pr_s}\right)^{1/4}$$

$$\begin{bmatrix} 0.7 = \lesssim Pr \lesssim 500 \\ 1 \lesssim Re_D \lesssim 10^6 \end{bmatrix} \tag{5.2.19}$$

where all properties are evaluated at T_∞ except Pr_s, which is evaluated at T_s. Values of C and m are listed in table 5.5 below. If Pr ≤10, n = 0.37; if Pr ≥10, n = 0.36.

Table 5:5 Constants C and m of a circular cylinder in crossflow [5, 8] (Equation 5.2.18)

Reynolds number (Re_D	C	m
0.4-4	0.989	0.330
4-40	0.911	0.385
40-4000	0.683	0.466
4000-40,000	0.193	0.618
40,000-400,000	0.027	0.805

Table 5:6 Constants of circular cylinder in crossflow [9] (Equation 5.2.19)

Reynolds number (Re_D	C	m
1-40	0.75	0.4
40-1000	0.51	0.5
10^3-2 x 10^5	0.26	0.6
2 x 10^5-2 x 10^6	0.076	0.7

5.3 Glasshouse heat transfer analysis

5.3.1 Introduction

The glasshouse heating is by natural convection which is caused by buoyancy forces due to density differences caused by temperature variations in the fluid. As the fluid is heated the density change in the boundary layer as discussed above will cause the fluid to rise and be replaced by cooler fluid that will also be heated and rise. This continuous phenomenon is called free or natural convection.

The heat transfer per unit surface through convection was first described by Newton and the relation is known as the Newton's Law of Cooling.

Heat loss from plain pipe is by convection to the room air and by radiation exchange with the walls. Hence heat loss from a heating pipe $q = q_{conv} + q_{rad}$

Where q_{conv} is the convective heat transfer and q_{rad} is the radiant heat transfer

$$q = h(\pi DL)(T_s - T_\infty) + \mathcal{E}(\pi DL)\sigma(T_s^4 - T_{sur}^4) \tag{5.3.1}$$

The rate of heat transfer by free convection from the plain heat pipe to the room is given by Newton's law of cooling:

$$q = \bar{h} A_s (T_s - T_\infty) \tag{5.3.2}$$

where $\bar{h}$ may be obtained from knowledge of the Rayleigh number given by

$$Ra_{Ln} = Gr_{Ln}\,\mathrm{Pr} = \frac{g\beta(T_s - T_\infty)Ln^3}{va} \tag{5.3.3}$$

5.3.2 Sizing of heat emitters (Heating pipes)

Each zone has a heat requirement and should be met to maintain satisfactory environment to ensure effective growth of the plants in each zone. To effectively size the heat emitters which will be the heating pipes, the maximum heating demand of each zone, the mass flow rate of hot water to meet the maximum heating demand, heating pipe internal diameter that will meet good practice water flow velocity to avoid stagnation should be calculated.

It is also important to calculate the pressure drop in the pipe to select appropriate pump size to circulate the hot water. To select the best possible heating pipe for the heating system the minimum pipe size, the actual mean flow velocity and pressure drop are necessary to ensure effective system operation.

Table 5.3 demonstrates internal heating pipe diameter, pressure drops and actual pipe velocity result. The calculated pressure drop will assist to select the appropriate pump motor size to power the pump to distribute sufficient LTHW to heat each zone. Once the sizes of the heating pipes have been determined then heat transfer or losses from the heating pipe could be calculated.

5.3.3 Heat transfer or loss calculation of a plain and PCM filled heating pipes

Heat transfer must occur wherever temperature difference exists in a medium or between media. The heat transfer per unit surface through convection was first described by Newton and the relation is known as the Newton's Law of Cooling and the equation for convection heat transfer can be expressed as:

$$q_{conv} = h_c\, A\, dT \tag{5.3.4}$$

In free convection, the flow velocity is smaller compared to forced convection and the corresponding convection transfer rates are also smaller.

To minimise electrical energy consumption in the glasshouse force convection was not considered and again it will be difficult to determine the flow velocity as fan installed in the space will only agitate the air flow in the space.

Heat transfer or loss of heat from a heating pipe passing through a glasshouse is by convection and radiation and will be better explained using an example calculation. The calculation is carried out using an excel spread sheet. A heating pipe of outside diameter of 114.3 mm is constantly supplied with low temperature hot water of 32 ^oC and passes through a glasshouse.

Figure 5:3 illustrates the plain heating pipe installed in the greenhouse to transfer or emit heat to the space to maintain the required set point temperature in the zone. The zone set point temperature is 20 ^oC.

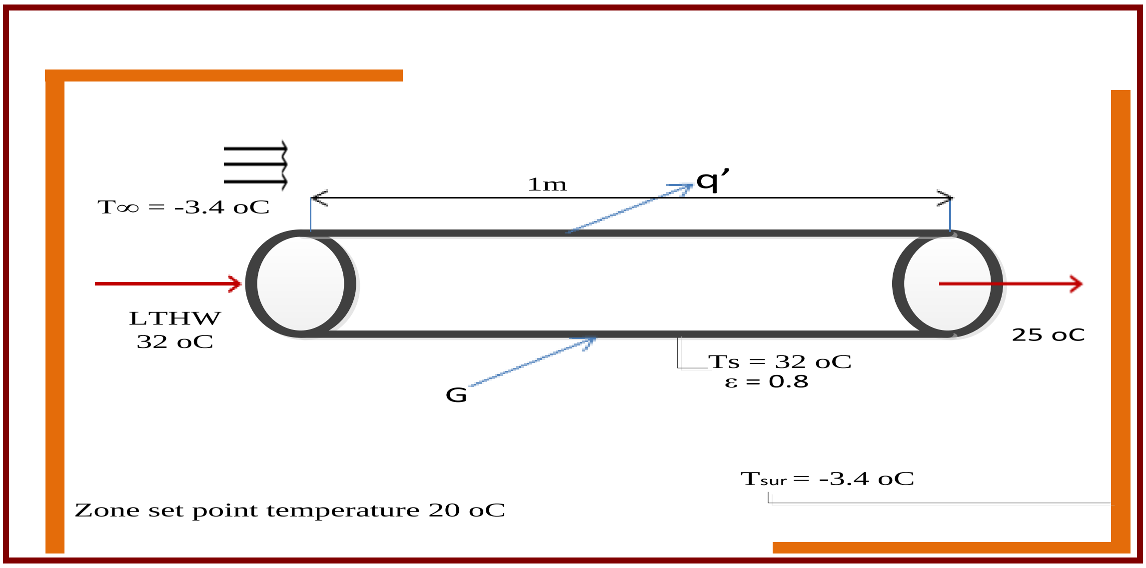


Figure 5:3 shows a plain heating pipe passing through a glasshouse zone 1

The surface temperature of the heating pipe (Figure 5:3 above) is assumed to be 32 ^oC and the surrounding space air temperature of the glasshouse is – 3.4 ^oC (Minimum outside design temperature), calculate the heat transfer coefficient, heat transfer rate to the space and the length of heating pipe required to meet zone 1 maximum heating demand of 36 kW.

Heat transfer assumptions:

1. Heating pipe surface temperature Ts is assumed to be constant throughout the process.
2. Constant properties.
3. Steady-state and heat transfer through pipe circumference.

Known data:

Heating pipe diameter is 114.3 mm, flow of LTHW temperature is 32 ^oC, fluid surrounding heating pipe temperature is -3.4 ^oC, and glasshouse zone heating demand is 36 kW. The heat transfer from the plain pipe to the space is by natural or free convection and radiation which is demonstrated below. The calculation is broken down into 4 steps

Step 1 calculation of Rayleigh number

The Rayleigh number is given by

$R_{aD} = G_{rD}\, Pr = [g\beta(T_s - T_\infty)D^3] / \nu\mu$

The calculation of the Rayleigh number is carried out using excel spread sheet as shown below. In circular pipe D represents length. The calculation of the Rayleigh number is demonstrated in table 5.7 below.

Table 5:7 Rayleigh number calculation

g	β	Ts	T∞	D	V	µ	vµ	Ra_D
m/s2	K-1	oC	oC	m	m2/s	kg/(s·m)		
9.8	0.0699301	32	-3.4	0.1143	0.00001589	0.0000225	3.58E-10	101327091

From the above calculation Rayleigh number is101327091.

Notes: Symbols and their meanings in table 5.7

Symbol	Meaning
β	The expansion coefficient
G	Acceleration due to gravity
T_s	Pipe surface temperature
T_∞	Fluid temperature
D	Pipe diameter
V	kinematic viscosity
µ	Fluid viscosity
Ra_D	Rayleigh number

All the properties of the fluid were taken from the film temperature (T_f) between the pipe surface and room air temperature which in this case assumed to be the design outside temperature of -3.4 ^{o}C. This is the outside design temperature to determine the maximum heating demand in the zone. That is in the absence of any internal heat gains the space temperature is equal to the outside design temperature.

The space maximum heating load was calculated using the IES software tool and it was based on the design outside temperature of -3.4 ^{o}C and space set point temperature of 9 ^{o}C in zone 1.

Step 2 calculation of mean Nusselt number

The average or mean Nusselt number could be obtained or calculated using the formula below.

$$\overline{Nu_{Ln}} = \left\{0.825 + \frac{0.387\, Ra_{Ln}^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}}\right\} \tag{1}$$

The Nusselt number was calculated using an excel spread sheet and the result illustrated in table 5:8 below. For the purpose of this calculation the mean Nusselt number formula above is designated as equation 1.

Table 5:8 Nusselt number using equation 1 result

RaD	0.387Ra1/6	Pr	0.492/Pr	[1+(0.492/Pr)9/16]8/27	{0.825 + [(0.387Ra1/6) / (1+	Mean Nusselt number
101327091.3	8.35605384	0.707	0.695898	1.193275205	7.827620858	61.3

$$\overline{Nu_{Ln}} = \left\{0.68 + \frac{0.670\, Ra_{Ln}^{1/4}}{[1+(0.492/Pr)^{9/16}]^{4/9}}\right\} \quad Ra_{Ln} \lesssim 10^9 \tag{2}$$

For the purpose of this calculation the mean Nusselt number formula above is designated as equation 2

Table 5:9 Nusselt number equation 2 result

RaD	0.670Ra1/4	Pr	0.492/Pr	[1+(0.492/Pr)9/16]4/9	{0.68 + [(0.670Ra1/4) /	Mean Nusselt number
101327091.3	67.22119	0.707	0.695898	1.303154563	52.26343606	52.3

Two equations are used to determine the Nusselt number to compare results for accuracy.

Step 3 Calculation of heat transfer coefficient and heat transfer rate

The mean convection heat transfer coefficient could be obtained or calculated when the Nusselt number is known from the equation below:

$$\overline{Nu_D} \equiv \frac{\bar{h}D}{\lambda} = C\, Re_D^m\, Pr^{1/3} \tag{3}$$

Where λ is the thermal conductivity, D diameter of heating pipe and $\overline{h_D}$ being the mean convection heat transfer coefficient. For the purpose of this calculation the equation to determine the heat transfer coefficient is designated as equation 3.

The convection heat transfer rate q_{conv} could be calculated using the equation given below

$$q_{conv} = \overline{h}A_s(T_s - T_\infty) \quad (4)$$

For the purpose of this calculation the equation to determine the heat transfer rate is designated as equation 4.

Table 5.10 below demonstrates the heat transfer coefficient and heat transfer rate result using equation 1 to determine the Nusselt number.

Table 5:10 Heat transfer coefficient and heat transfer rate

RaD	0.387Ra1/6	Pr	0.492/Pr	[1+(0.492/Pr)9/16]8/27	{0.825+[(0.387Ra1/6) / (1+0.492/Pr)9/16]8/27}	Mean Nusselt number	λ (w/m.k)	Heat transfer coefficient (W/m2K)	Heat transfer rate (W)
101327091	8.35605384	0.707	0.695898	1.193275205	7.827620858	61.2716483	0.0263	14.1	179.2

Table 5.11 below demonstrates the heat transfer coefficient and heat transfer rate result using equation 2 to determine the Nusselt number.

Table 5:11 Heat transfer coefficient and heat transfer rate

RaD	0.670Ra1/4	Pr	0.492/Pr	[1+(0.492/Pr)9/16]4/9	{0.68+[(0.670Ra1/4) / (1+0.492/Pr)9/16]4/9}	Mean Nusselt number	λ (w/m.k)	Heat transfer coefficient (W/m2K)	Heat transfer rate (W)
101327091	67.22119	0.707	0.695898	1.303154563	52.26343606	52.26343606	0.0263	12.0	152.9

Equation (1) is suitable for most engineering calculations but slightly better accuracy may be obtained for laminar flow by using equation 2 [8] but the Rayleigh number shows that the flow is not laminar flow.

Step 4 calculation of radiant heat transfer

Radiant heat transfer is calculated using the formula below

$$q_{rad} = \varepsilon\sigma A(T_s^4 - (T_{sur}^4) \quad (5)$$

For the purpose of this calculation the equation to determine the radiant heat transfer is designated as equation 5.

Table 5:12 Radiant heat transfer

Pipe emmisitivity	Stefan Boltzman constant	Pipe dia (m)	Pipe area (m2)	pipe surface temp (K)	Surrounding temp (K)	Radiant heat transfer per unit of pipe length (W)
0.8	5.67E-08	0.1143	0.359131	305	269.6	54.9

There is slight difference between the heat transfer rate using equation 1 and 2. The difference is about 15%. The radiant heat transfer to the surroundings was also calculated (Table 5:12) to know the overall heat transfer from the heating pipe to the environment or zone space.

The calculation of the overall heat transfer from the heating pipe to the space is demonstrated in table 5:13 below including heating pipe length required to meet maximum heating demand using both equation 1 and 2.

5.3.4 Determining heating pipe length to meet zone 1 heating demand

Now that zone 1 heat transfer coefficient, heat transfer rate and radiant heat transfer have been calculated for a unit of pipe length, the pipe length required to emit enough heat to meet the zone maximum heating demand using plain heating pipe is calculated using excel spread sheet.

The plain heating pipe length required is obtained by dividing the zone maximum heating demand by heat transfer rate per unit pipe length and this is demonstrated in table 5:13 below when zone 1 for example maximum heating load of 36 kW is divided by heat transfer rate of 221.08 W (Equation 1) to obtain 163 m of pipe length.

Using equation 2 of heat transfer rate of 196.01 W gave pipe length of 184 m. The same principle was used in calculating all the zones heating pipe lengths.

Table 5:13 Calculation of plain heating pipe length required to meet the zone 1 maximum heating demand

Zone 1	Pipe size (mm)	Pipe surface temp (oC)	Space temp (oC)	Heat transfer coefficient (W/m2) equation 1	Heat transfer coefficient (W/m2) equation 2	Conv Heat transfer rate (W) equation 1	Conv Heat transfer rate (W) equation 2	Radiant heat transfer (W)	Total heat transfer (W) equation 1	Total heat transfer (W) equation 2	Zone max heating load (kW)	Pipe length required (m) equation 1	Pipe length required (m) equation 2
Plain pipe	114.3	32	-3.4	14.16	12.07	169.83	144.76	51.25	221.08	196.01	36.00	163	184

Table 5.14 demonstrates the heating pipe length required to transfer sufficient heat to meet zone 1 heating demand of 36 kW using PCM filled heating pipe with phase change temperature of 15 ^{o}C.

The zone heating set point is 9 ^{o}C and vent opening temperature is 15 ^{o}C. Thus the vent should open to allow some heat out when the space temperature reaches 15 ^{o}C. As a result only 15 ^{o}C phase change temperature PCM heating pipe will be installed in zone 1.

Investigating in detail into zone 1 requirements established that the space temperature could rise to 19 ^{o}C with no effects to the condition of the plants in the zone.

This means that vents could be opened at 19 ^{o}C to allow the 15 ^{o}C PCM heating pipe to absorb enough thermal energy before vents open. The advantage in the new heating system is that when the space temperature exceeds 15 ^{o}C the PCM heating pipe will start to absorb heat in the space retarding the rise of the temperature to 19 ^{o}C or higher.

The first parameter calculated was the Rayleigh number followed by the Nusselt number and subsequently, the heat transfer coefficient and heat transfer rate as demonstrated below.

Table 5:14 Calculation of 15 ^{o}C phase change temperature PCM heating pipe length required to meet the maximum heating demand (36 kW) in zone 1

g	β	Ts	T∞	D	V	α	vα	RaD	
m/s2	K-1	oC	oC	m	m2/s	m2/s			
9.8	0.17241379	15	-3.4	0.1143	0.00001589	0.0000225	3.58E-10	129851861.8	
Equation 1									
RaD	0.387Ra1/6	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]8/27	{0.825 + [(0.387Ra1/6) / (1+ 0.492/Pr)9/16]8/27 }	Mean Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
129851862	8.70873415	0.707	0.695898	1.193275205	8.123177412	65.98601126	0.0263	15.2	100.3
Equation (2)									
RaD	0.670Ra1/4	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]4/9	{0.68 + [(0.670Ra1/4) / (1+ 0.492/Pr)9/16]4/9 }	Mean Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
129851862	71.5215385	0.707	0.695898	1.303154563	55.56338878	55.56338878	0.0263	12.8	84.5
Radiant heat transfer									
pipe emissivity	Stefan Boltzman constant	Pipe dia (m)	Pipe area (m2)	pipe surface temp (K)	Heat pipe Surrounding air temp (K)	Radiant heat loss per unit of pipe (W)			
0.8	5.67E-08	0.1143	0.359131	288	269.6	26.01083103			
Total heat transferred using equation 1 (W)							**126.3**		
Total heat transferred using equation 2 (W)							**110.5**		
Maximum zone heating load (kW)							**36**		
Pipe length required for using equation 1 results (m)							**285**		
Pipe length required for using equation 2 results (m)							**326**		

The 15 ^{o}C PCM heating pipe length required is obtained by dividing the zone maximum heating demand by heat transfer rate per unit pipe length and this is demonstrated in table 5.14 above when zone 1 maximum heating load of 36 kW is divided by heat transfer rate of 126.3 W (Equation 1) to obtain 285 m of pipe length.

Using equation 2 with heat transfer rate of 110.5 W gave pipe length of 326 m. The same principle was used in calculating all the zones heating pipe lengths.

5.3.5 Determining heat pipe length required for zone 8 using PCM filled heat pipes

Currently, the glasshouse under research is heated by plain heat pipes supplied with low temperature hot water (LTHW) of 85 ^{o}C and returned at 75 ^{o}C. This plain heating pipe will be replaced with phase change temperature (PCM) filled heating pipes which has the capacity to store excess energy gained through solar gains in the glasshouse zones which would have been otherwise vented to the atmosphere wasted.

The calculation method used to compute heating pipe length required to produce sufficient heat to meet the maximum heat demand of zone 1 and zone 8 will be applied to all the other zones.

Table 5:15 below demonstrates the heating pipe length required to transfer sufficient heat to meet zone 8 heating demand of 84 kW using PCM filled heating pipe with phase change temperature of 25 ^{o}C.

The zone heating set point temperature is 20 ^{o}C and vent opening temperature of 27 ^{o}C (Thus the vent should be opened to allow some heat out when the space temperature reaches 27 ^{o}C but it could go up to 29 ^{o}C).

To calculate the length of heating pipe required, the total heat transfer rate per unit of pipe length which includes convection and radiation heat transfer should be calculated.

Figure 5:4 illustrates a unit length of PCM heating pipe designed to provide sufficient heat to maintain space set point temperature of 20 ^{o}C in zone 8. LTHW of 32 ^{o}C is supplied through the inner pipe during the charging process to melt the PCM in the annular pipe shown in figure 5:4.

Alternatively, the installed PCM heating pipes could be charged when the space temperature of the zone exceeds the PCMs phase change temperatures of 25 ^{o}C. The PCM heating pipes installed in zone 8 are 22 and 25 ^{o}C phase change temperature PCMs.

The calculation in table 5.15 and table 5.16 show 25 ^{o}C and 22 ^{o}C PCM length of pipe required in meeting the zone heating demand of 84 kW respectively. Figure 5:4 demonstrates 25 ^{o}C phase change temperature PCM heating pipe. The heat transfer from the PCM heating pipe to the space is by free or natural convection and radiation. The environment temperature is -3.4 ^{o}C (Heating system external design temperature).

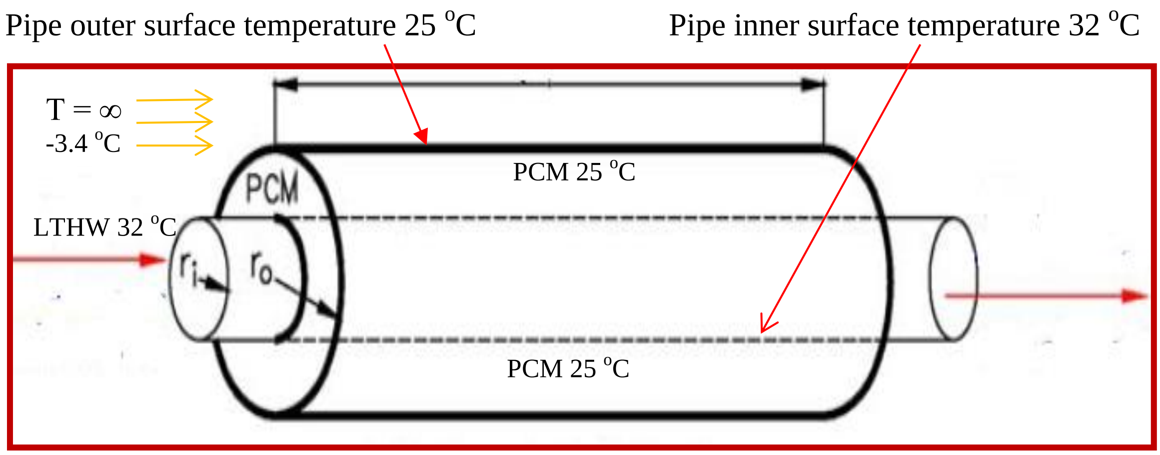


Figure 5:4 PCM filled annular heating pipe with LTHW flow temperature of 32 °C

Table 5:15 Calculation of 25 °C phase change temperature PCM heating pipe length required to meet the maximum zone 8 heating demand of 84kW

g	β	Ts	T∞	D	V	α	να	RaD	
m/s2	K-1	oC	oC	m	m2/s	m2/s			
9.8	0.0925926	**25**	-3.4	0.1143	0.00001589	0.0000225	3.58E-10	107634856.5	
Equation 1									
RaD	0.387Ra1/6	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]8/27	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
107634856	8.4405834	0.707	0.695898	1.193275205	7.898459129	62.38565661	0.0263	14.4	146.4
Equation (2)									
RaD	0.670Ra1/4	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]4/9	{0.68 + [(0.670Ra1/4) / (1+ 0.492/Pr)9/16]4/9 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
107634856	68.243774	0.707	0.695898	1.303154563	53.04813503	53.04813503	0.0263	12.2	124.5
Radiant heat transfer									
pipe emissivity	Stefan Boltzman constant	Pipe dia (m)	Pipe area (m2)	pipe surface temp (K)	Surrounding temp (K)	Radiant heat loss per unit of pipe (W)			
0.8	5.67E-08	0.1143	0.359131	298	269.6	42.40595714			
Total heat transferred using equation 1 (W)							**188.8**		
Total heat transferred using equation 2 (W)							**166.9**		
Maximum zone heating load (kW)							**84**		
Pipe length required for using equation 1 results (m)							**445**		
Pipe length required for using equation 2 results (m)							**503**		

The 25 °C PCM heating pipe length required is obtained by dividing the zone maximum heating demand by heat transfer rate per unit pipe length and this is demonstrated in table 5.15 above when zone 8 maximum heating load of 84 kW is divided by heat transfer rate of 188.8 W (Equation 1) to obtain 445 m of pipe length. Using equation 2 with heat transfer rate of 166.9 W gave pipe length of 503 m. The same principle was used in calculating all the zones heating pipe lengths.

Table 5:16 Calculation of 22 ^{o}C phase change temperature PCM heating pipe length required to meet the maximum zone 8 heating demand of 84kW

g	β	Ts	T∞	D	V	α	vα	RaD	
m/s2	K-1	oC	oC	m	m2/s	m2/s			
9.8	0.1075269	**22**	-3.4	0.1143	0.00001589	0.0000225	3.58E-10	111791586.5	
Equation 1									
RaD	0.387Ra1/6	Pr	0.492/Pr	[1+ (0.492/Pr)9/ 16]8/27	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
111791587	8.4940569	0.707	0.695898	1.193275205	7.943271544	63.09556282	0.0263	14.5	132.4
Equation (2)									
RaD	0.670Ra1/4	Pr	0.492/Pr	[1+ (0.492/Pr)9/ 16]4/9	{0.68 + [(0.670Ra1/4) / (1+ 0.492/Pr)9/16]4/9 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)	Heat transfer rate (W)
111791587	68.893315	0.707	0.695898	1.303154563	53.54657238	53.54657238	0.0263	12.3	112.4
Radiant heat transfer									
pipe emissivity	Stefan Boltzman constant	Pipe dia (m)	Pipe area (m2)	pipe surface temp (K)	Surrounding temp (K)	Radiant heat loss per unit of pipe (W)			
0.8	5.67E-08	0.1143	0.359131	295	269.6	37.31039724			
Total heat transferred using equation 1 (W)							**169.7**		
Total heat transferred using equation 2 (W)							**149.7**		
Maximum zone heating load (kW)							**84**		
Pipe length required for using equation 1 results (m)							**495**		
Pipe length required for using equation 2 results (m)							**561**		

It could be seen from table 5:16 that the maximum heating pipe length required to produce sufficient heat to meet zone 8 heating demand is 561m using equation 2. This length of pipe is required when 22 ^{o}C phase change temperature PCM alone is assumed to be used and 503 metre length will be required when 25 ^{o}C phase change temperature PCM heating pipe is used.

In zone 8, 25 and 22 ^{o}C phase change temperature PCM heating pipes will be used to condition the zone. The reason is that the rate of heat transfer from the PCM heating pipe is slow and minimal compared to plain heating pipes as demonstrated using zone 1as an example.

The 25 ^{o}C PCM heating pipe will start to give out heat to the space when the space temperature falls below 25 ^{o}C. If the space temperature continues to fall below 22 ^{o}C then the 22 ^{o}C PCM heating pipe will join in to transfer more heat to maintain space temperature at 21 ^{o}C.

The 561 metre length heating pipe was selected to meet worse case heating demand. The longer length PCM heating pipes will be installed in all zones to ensure that adequate heat is provided to maintain plants in good condition even in worse weather conditions. The calculation method used in calculating zone 8 heat transfer rate and PCM heating pipe length required was used in calculating all the other zones heat transfer rate and heating pipe length required to maintain good environment for the plants.

Table 5:17 shows length (In metres) of PCM heating pipe required to emit enough heat to meet each zone heating demand using plain pipe, 25, 22 and 15 ^{o}C phase change temperature PCM heating pipes. Zone 1, 5, 6 and 7 use 15 ^{o}C PCM heating pipes

Table 5:17 Shows length (In metres) of PCM heating pipe required

Zone	Plain heating pipe		25 ^{o}C PCM heating pipe		22 ^{o}C PCM heating pipe		15 ^{o}C PCM heating pipe	
	Equation 1	Equation 2	Equation 1	Equation 2	Equation 1	Equation 2	Equation 1	Equation 2
1	163	184					284	325
2	49	56	58	66	65	73		
3	63	71	74	84	82	94		
4	54	61	63	72	70	80		
5	9	10					16	18
6	98	110					171	195
7	146	165					256	293
8	380	428	445	503	495	561		
9	470	530	551	623	613	694		
10	228	257	267	302	297	337		
11	326	367	381	431	424	481		
12	244	276	286	323	318	361		
13	140	158	164	186	182	207		
14	70	80	82	93	92	267		
15	152	171	178	201	198	224		
16	146	165	172	194	191	216		
17	248	280	291	330	324	367		
18	244	275	285	323	318	360		
19	375	423	439	497	488	554		
20	314	355	368	417	410	465		
21	212	238	248	280	275	312		

5.4 PCM melting rate calculation

5.4.1 Introduction

The first law of thermodynamics states that the total energy of a system is conserved and the amount can only be changed when energy crosses the boundaries. For a closed system (Fixed mass region), the two ways that energy can cross the boundary is by heat transfer and work done on or by the system.

The first law statement for a closed system is given by:

$$\Delta E_{st}^{tot} = Q - W \qquad (5.4.1)$$

Where the system stored energy change is ΔE_{st}^{tot}, net heat transferred to the system is Q, and net work done by the system is W. System (a) in Figure 5:5 below is illustrated by equation 5.4.1

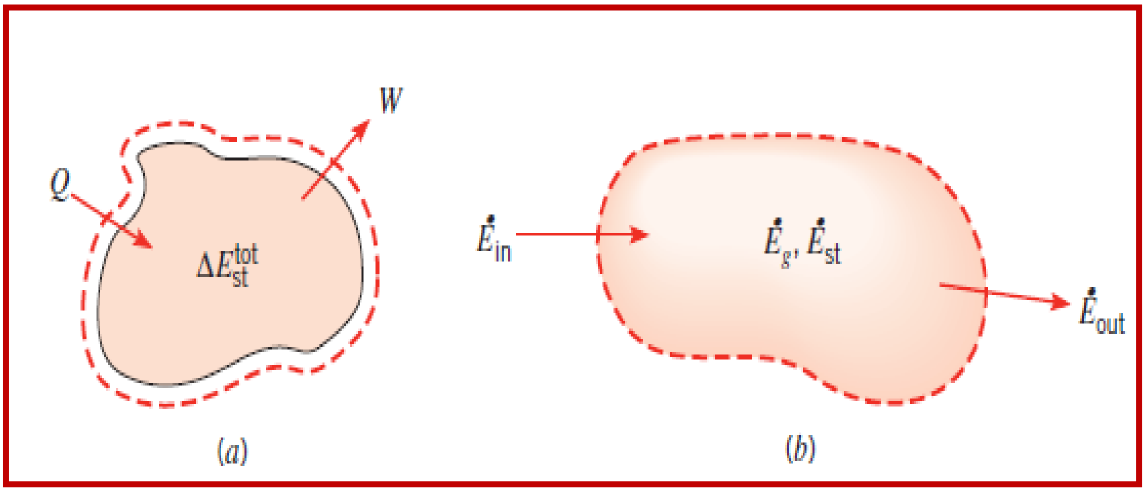


Figure 5:5 Conservation of energy: (*a*) for a closed system over a time interval and (*b*) for a control volume at an instant.
Source: The fundamental of heat and mass transfer

A space of region bounded by a control surface through which mass may pass is a control volume or open system and the principle of thermodynamics can be applied. System (b) in figure 5.5 illustrates energy in and energy out. Energy advection is the energy carried by a mass entering and leaving a control volume.

This is the third way in which energy can cross the boundaries of a control volume. The first law of thermodynamics can be stated for both the control volume and a closed system.

5.4.2 PCM melting rates

The current heating method of the glasshouse is by a plain pipe heating system where low temperature hot water (LTHW) is supplied and returned at 85 ^{o}C and 75 ^{o}C respectively. The proposed heating system (Research heating system) will supply low temperature hot water at 32 ^{o}C and return at 25 ^{o}C to increase the solar thermal heating system efficiency. Heat energy will be stored in an annulus heating pipes filled with PCM of cylindrical shape of 114.3 mm outer diameter and inner pipe diameter of 42 mm.

The annulus pipe filled with phase change material (PCM) will be melted by either the hot water flowing through the inner pipe of the annulus heating pipe at a temperature of 32 ^{o}C or by warm air surrounding the PCM heating pipe if the temperature of the surrounding air exceeds the phase change temperature of the PCM.

Thus, instead of venting the excess heat trapped in the space through the vents as done currently, will be absorbed by the PCM heating pipes rather than venting it to the atmosphere to waste.

The heat will be stored in the PCM and used when needed. This is the main focus of the research to maximise the use of the solar thermal energy trapped in the space to heat the glasshouse rather than using fossil fuel to heat the glasshouse.

The principle will be best explained by using examples. For this example, the melting point temperature or the phase change temperature of the PCM is 25 ^{o}C and has been assumed to be the surface temperature of the outer pipe of the annulus heating pipe.

We will consider a mass M of phase change material with phase change temperature or fusion temperature of 25 ^oC ($T_{pcm} = 25\ ^oC$) and is enclosed in an annulus heating pipe of outer diameter 114.3mm and wall thickness mL to be 2.6 mm and thermal conductivity λ to be 66 W/m.K (Steel pipe).

The low temperature hot water (LTHW) flowing through the inner pipe of the annular pipe has a supply temperature of 32 ^oC and return temperature of 25 ^oC.

The average surface temperature of the inner pipe is calculated to be 28.5 ^oC approximated to be 29 ^oC. Figure 5:4 above demonstrates the pipe configuration. If the inner pipe surface temperature $T_{hw} > T_{pcm}$ to melt the phase change material, how long will it take to melt the entire mass of the PCM in a metre (m) pipe length. The outer and inner diameter of the annular heat pipe is 114.3 and 42 mm respectively.

To determine the melting time t_m, the first law thermodynamics should be applied over the time interval $\Delta t = t_m$. Hence, applying equation 5.4.2 to a control volume, it follows

that $E_{in} = \Delta E_{st} = U_{lat}$ (5.4.2)

Where E_{in} is thermal energy in, ΔE_{st} is the change in thermal energy and U_{lat} latent heat of fusion. Where the increase in energy stored within the control volume is due exclusively to the change in latent energy associated with conversion from solid to liquid state. Heat is transferred to the PCM by means of conduction through the inner pipe wall.

Since the temperature difference across the inner pipe wall is assumed to remain at $T_{hw} > T_{pcm}$ throughout the melting process, the rate of heat transfer is assumed to be constant. T_{hw} is the mean of the supply and return water temperatures calculated to be approximately 29 ^oC. The heat from the hot water will be transferred to the PCM solution in the PCM heating pipe to melt it before the PCM will deliver the heat to the space.

We will assume that the heat transfer from the hot water to the PCM is Q1 and also assume that the heat transfer from the PCM to the space is Q2 therefore the effective heat transfer (Q_{eff}) to melt the PCM is Q1 – Q2.

$$Q1 = (T_{hw} - T_{pcm}) / R_{t1} \tag{5.4.3}$$

Where T_{hw} is the mean hot water temperature, T_{pcm} is the PCM temperature and R_{t1} is the total resistance to heat transfer to the PCM and includes water film resistance at the internal pipe surface and pipe resistance.

R_{t1} = The water film resistance plus pipe resistance

$$= 1 / h_{ci}\, A + \mathrm{In}(r_o / r_i) / 2\, \pi\, \lambda \tag{5.4.4}$$

Where h_{ci} is the heat transfer coefficient of the inner pipe, λ thermal conductivity of the steel pipe, A heat transfer surface area of the pipe. The pipe resistance = $\mathrm{In}(r_o / r_i) / 2\, \pi\, \lambda$ where r_o and r_i are the outer and inner radius of the annulus inner pipe. Thus $r_o = r_i$ plus pipe thickness. $1 / h_{ci}\, A$ = water film resistance across the pipe surface area.

$$Q2 = (T_s - T_{sur}) / R_{t2} \tag{5.4.5}$$

Where T_s is the outer pipe surface temperature which has been assumed to be the PCM phase change temperature. The PCM temperature at the inner surface of the outer pipe will remain almost constant during the phase change period from solid to liquid.

If T_s is the outer pipe outer surface temperature, the difference in temperature between the PCM and the outer pipe outer surface temperature will be very minimal as the pipe thickness is just 2.6 mm. This will not affect the calculation results significantly. R_{t2} is the total resistance to heat transfer from the PCM to the space and includes outer pipe surface film resistance and pipe resistance.

R_{t2} = The outer pipe surface film resistance plus pipe resistance

$$= 1 / h_{cs}\, A + \mathrm{In}(r_o / r_i) / 2\, \pi\, \lambda \tag{5.4.6}$$

The pipe resistance = In(r_o/r_i) / 2 π λ where r_o and r_i are the outer and inner radius of each pipe (The inner and outer pipes of the annular pipe) Thus $r_o = r_i$ plus pipe thickness.
For the inner pipe of the annulus pipe r_i is 21 mm and r_o is 23.6 and r_i for the outer pipe is 54.55 mm and r_o is 57.15 mm (Refer to figure 5:6 below).

The amount of energy inflow

$$E_{in} = Q1 - Q2 = Qeff \tag{5.4.7}$$

The PCM stored energy is $E_{st} = Mh_{sf}$ (5.4.8)

Where M is the PCM mass (kg) and h_{sf} latent heat of fusion (kJ/kg)

The rate of time to melt the PCM

$$t_m = Mh_{sf} / Qeff \tag{5.4.9}$$

The amount of energy required to effect such a phase change per unit mass of solid is termed the latent heat of fusion h_{sf}.

The effective heat transfer (Q_{eff}) to melt the PCM will be Q1 – Q2 (Refer to figure 5.6 below) where Q1 is the heat transfer from the hot water to the PCM and Q2 is the heat transfer or lost from the PCM to the space.

5.4.2.1 Calculation of the heat transfer coefficient

For turbulent flow within circular and non-circular tubes and ducts, the following correlation is used $Nu_D = 0.023\, Re_D^{4/5}\, Pr^n$ where n = 0.4 for heating ($\theta_s > \theta_m$) and 0.3 for cooling ($\theta_s < \theta_m$) and θ_m is the mean fluid temperature, evaluated as the average of the inlet and outlet fluid temperatures of the tube: $\theta_m = (\theta t_i + \theta t_o) / 2$
The flow is laminar when Reynolds number *Re* < 2000 and turbulent when *Re* > 3000

The Reynolds numbers for the hot water flow to each zone were taken from calculated Reynolds numbers in table 5:3 above.

The Reynolds (Re) number for zone 8 from table 5:3 above for example was 76457 which suggest that the flow is turbulent as the Reynolds number is greater than 3000. The prandtl number of hot water at 29 ^{o}C is 5.52.

$$\bar{N}u_D = 0.023\, Re_D^{4/5}\, Pr^n \tag{5.4.10}$$

where n = 0.4 for heating, Pr is 5.52

$\bar{N}u_D = 0.023 \times 76457^{4/5} \times 5.52^{0.4} = 367.5$

The mean convection heat transfer coefficient could be obtained or calculated when the Nusselt number is known from the equation:

$$\overline{Nu}_D \equiv \frac{\bar{h}D}{\lambda} = C\, Re_D^m\, Pr^{1/3} \tag{5.4.11}$$

Where λ is the fluid thermal conductivity, D diameter of heating pipe and $\bar{h}_D$ being the mean convection heat transfer coefficient. For the purpose of this calculation the equation to determine the heat transfer coefficient is designated as equation 5.4.11

From CIBSE Guide C Table 2.2 Properties of water at saturation, the prandtl number for 29 ^{o}C is 5.52. The average Nusselt number is calculated using equation 5.4.10 above. Therefore the average heat transfer coefficient of the inner pipe h_{ci} = Average Nusselt number multiply by fluid thermal conductivity and divided by the pipe diameter = (367.5*0.61941) / 0.042 = 5420 W/m^{2}K.

The calculated thermal heat transfer coefficient (h_{cs}) at the outside surface of the PCM heating pipe from table 5:18 below is 7 W/m^{2}K. The properties of the air used for the calculation were obtained from the mean air values of 25 and 20 ^{o}C [(25+20) / 2].
The heat transfer from the outer pipe surface to the space is mainly by natural convection and the Rayleigh number is given by

$$R_{aD} = G_{rD}\, Pr = [g\beta(T_s - T_\infty)D^3] / \nu\alpha \tag{5.4.12}$$

And the mean Nusselt number is calculated from equation below

$$\overline{Nu_D} = \left\{0.825 + \frac{0.387\, Ra_D^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}}\right\} \tag{5.4.13}$$

The heat transfer coefficient is calculated as per equation 5.4.11 as the Nusselt number is calculated from equation 5.4.13 above. All the air properties were taken from the mean temperature of 25 and 20 ^{o}C. The Rayleigh number was calculated as per equation 5.4.12 above. The calculation is shown in table 5:18

Table 5:18 Calculation of heat transfer coefficient of 25 ^{o}C PCM and space temperature of 20 ^{o}C

g	β	Ts	T∞	D	V	α	vα	RaD
m/s2	K-1	oC	oC	m	m2/s	m2/s		
9.8	0.0444444	25	20	0.1143	0.00001589	0.0000225	3.58E-10	9095903.366
Equation 1								
RaD	0.387Ra1/6	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]8/27	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)
9095903.4	5.5914047	0.707	0.695898	1.193275205	5.510762881	30.36850753	0.0263	7.0

Heat transfer analysis assumptions:

1. Inner pipe surface temperature T_{hw} is assumed to be constant throughout the process.
2. Constant properties.
3. Steady-state and conduction through inner pipe circumference.
4. Conduction areas are the inner and outer pipes surface areas

Known data:

PCM (25 ^{o}C melting point) latent heat of fusion180 kJ / kg, density 1530 kg / m^3, inner pipe mean surface temperature 29 ^{o}C, PCM phase change temperature 25 ^{o}C, pipe thermal conductivity of 66 W / m.K and pipe thickness 2.6 mm.

Mass of PCM is the outer pipe cross-sectional area minus inner pipe cross-sectional area multiply by unit length of pipe and PCM density.

$M = \pi/4(D_o^2 - D_i^2) * \rho = 3.142/4(0.109.1^2 - 0.042^2) *1*1530 = 12.2\text{kg}$

$t_m = Mh_{sf} / Q_{eff}$

The heat transfer of Q1, Q2, Q_{eff} and R_{t1}, R_{t2} and melting time were calculated using excel spread sheet. All the heat transfer parameters such as heat transfer coefficient have been calculated above.

Table 5:19 demonstrates the calculation and figure 5:6 below shows the PCM heating pipe configuration details. The calculated melting time per metre of the 25 ^{o}C PCM heating pipe is 15.4 minutes as demonstrated in table 5:19.

Table 5:19 Rate of time to melt a metre length of 25 ^{o}C melting point PCM heating pipe

inner pipe inner radius (m)	inner pipe outer radius (mm)	Heat transfer coefficient (W/m2K)	Thermal conductivity of pipe (W/m,K)	Rt1	Hot water Temp (oC)	PCM temp (oC)	Q1 (W)	Outer pipe inner radius (mm)	Outer pipe outer radius (mm)	Heat transfer coefficient (W/m2K)	Thermal conductivity (W/m,K)	Rt2	Space temp	Q2 (W)	Qeff (W)	PCM mass	Latent heat	PCM stored energy (kJ)	Melting Time (min)
21	23.6	5420	66	0.00168	29	25	2382	54.55	57.15	7	66	0.416858	20	11.994	2370	12.2	180	2196.0	15.4

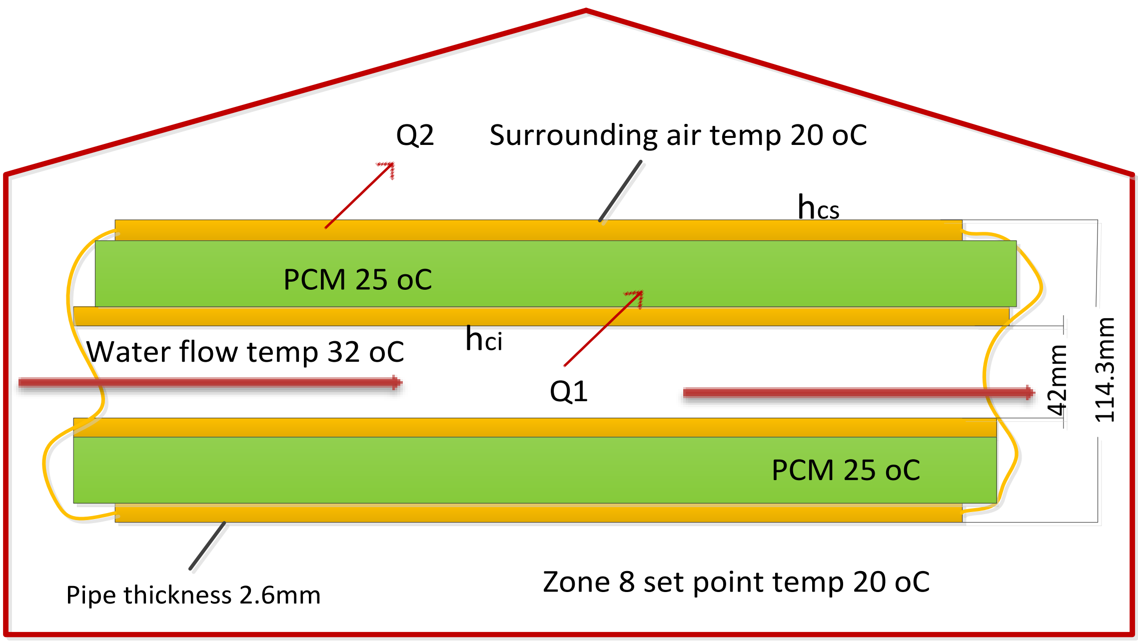


Figure 5:6 PCM heating pipe

5.4.2.2 Calculation of melting time for 22 ^{o}C phase change material (PCM)

Most of the zones use 25 and 22 ^{o}C PCM heating pipes, therefore the rate of melting of the 22 ^{o}C phase change temperature heating pipe is important to determine the length of time that the 22 ^{o}C PCM will melt.

The surface heat transfer coefficient of the outer heating pipe will change but all the other parameters will remain the same as the 25 ^{o}C PCM heating pipe.

Table 5:20 shows that the heat transfer coefficient of the outer pipe surface area of the 22 ^{o}C PCM heating pipe is 5.5 W/m^{2}K and the latent heat of fusion is170 kJ/kg

Table 5:20 Heat transfer coefficient calculation of the outer pipe surface of the 22 ^{o}C PCM heating pipe

g	β	Ts	T∞	D	V	α	vα	RaD
m/s2	K-1	oC	oC	m	m2/s	m2/s		
9.8	0.047619	22	20	0.1143	0.00001589	0.0000225	3.58E-10	3898244.3
Equation 1								
RaD	0.387Ra1/6	Pr	0.492/Pr	[1+ (0.492/Pr)9/16]8/27	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	Nusselt number	λ (w/m.k)	Heat gtransfer coefficient (w/m.K)
3898244.3	4.8550235	0.707	0.695898	1.193275205	4.893653644	23.94784599	0.0263	5.5

The melting rate of 22 ^{o}C phase change temperature PCM base on the same calculation method as that of 25 ^{o}C PCM is 8.3 minutes as shown in table 5:21 below.

Table 5:21 Rate of time to melt a metre length of 22 ^{o}C melting point PCM heating pipe.

inner pipe inner radius (m)	inner pipe outer radius (mm)	Heat transfer coefficient (W/m2K)	Thermal conductivity of pipe (W/m,K)	Rt1	Hot water Temp (oC)	PCM temp (oC)	Q1 (W)	Outer pipe inner radius (mm)	Outer pipe outer radius (mm)	Heat transfer coefficient (W/m2K)	Thermal conductivity (W/m,K)	Rt2	Space temp	Q2 (W)	Qeff (W)	PCM mass	Latent heat	PCM stored energy (kJ)	Melting Time (min)
21	23.6	5420	66	0.00168	29	22	4168	54.55	57.15	5.5	66	0.530516	20	3.7699	4164	12.2	170	2074.0	8.3

In zone 8 two PCM heating pipes with phase change temperatures of 22 ^{o}C and 25 ^{o}C will be installed respectively. The length of the PCM filled heating pipe that is required to emit enough heat to meet the zone maximum heating demand of 84 kW using 25 and 22 ^{o}C phase change temperatures are 503m and 561m respectively.

Thus 503m length of heating pipe will be required if 25 ^{o}C PCM heating pipe only is used to heat the space. Alternatively 561m length of the heating pipe will be required if 22 ^{o}C PCM only is used.

To ensure that sufficient heat is produced, 561m length of PCM pipe is selected.

The configuration of the pipework in zone 8 and zone 1 is demonstrated in figure 5:7 and figure 5:8 respectively below.

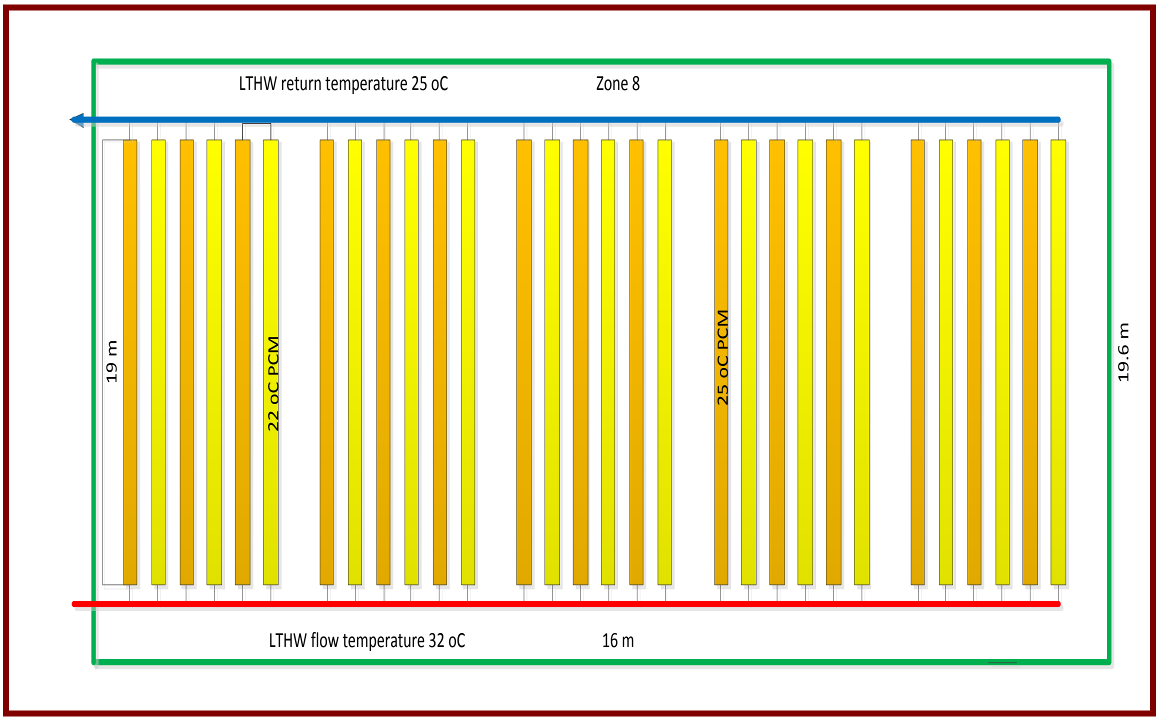


Figure 5:7 Zone 8 PCM heating pipe configuration

The length of each heating pipe is 19 m and the number of such pipes needed is 30 as shown in figure 5.7 above.

The pipe configuration is based on the zone dimensions. The number of 19 m PCM heating pipes needed was calculated by dividing 561 by 19 m which gives the result to be 29.5 (Approximately 30 pieces). The same principle was used in determining PCM heating pipe length and number of pipes needed in zone 1 below and other zones.

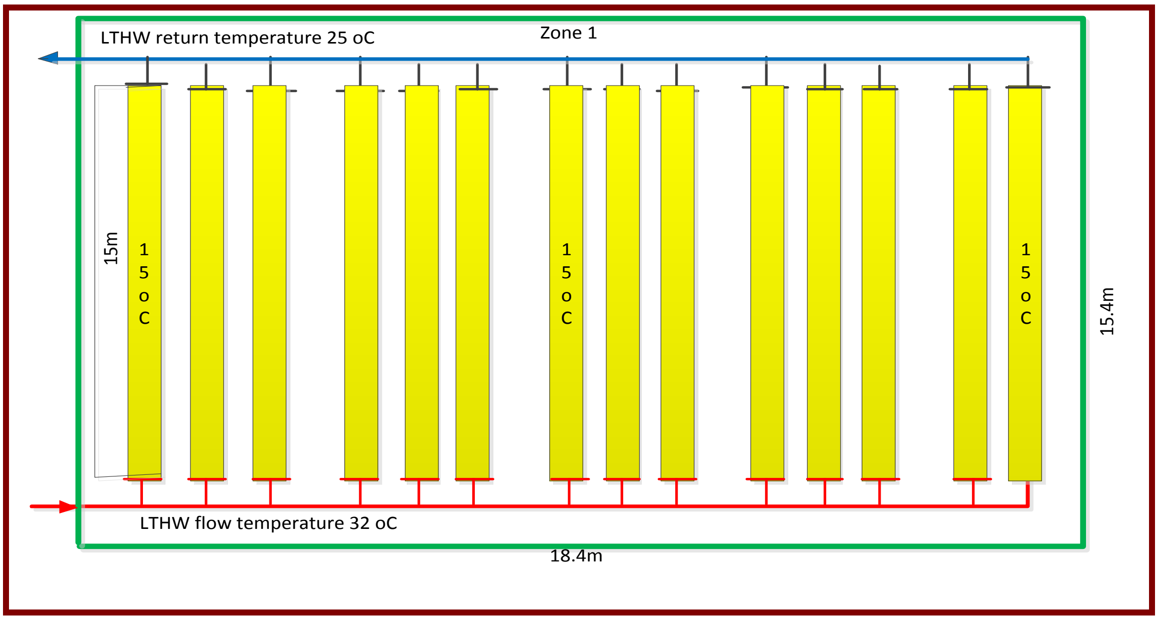


Figure 5:8 Zone 1 PCM heating pipe configuration of 15 metre pipe length

It has been established from table 5:19 and table 5:21 that with a temperature difference of 4 and 7 ^{o}C between the PCM solution and hot water temperature, the melting rate or time per metre length of 25 ^{o}C and 22 ^{o}C PCM heating pipes are 15.4 and 8.3 minutes respectively.

Table 5:22Table 5:22 below shows PCM heating pipe lengths and number of such lengths required for each zone.

Table 5:22 Zones PCM heating pipe lengths (In metres) and number required

Zone	PCm heating Pipe lengths	No of PCM heating pipes required for each zone
1	14	15
2	6.5	3
3	7	13
4	7	12
5	7	6.3
6	17	12
7	18	16
8	15	19
9	15	46
10	15	22
11	11	32
12	15	24
13	15	14
14	15	18
15	11	21
16	11	20
17	20	18
18	20	18
19	20	28
20	20	23
21	18	18

The time required to melt each length of the PCM heating pipe has been calculated based on the time it takes to melt a metre length of 25, 22 and 15 ^{o}C PCM heating pipes and this is shown in table 5:23 below.

Table 5:23 Melting rate of PCM heating pipes per meter length and design length of PCM heating pipes in each zone

Zones	PCM heating pipe design length (m)	Time to melt 25 oC PCM heating Pipe metre length (min)	Time to melt 22 oC PCM heating Pipe metre length (min)	Time to melt 15 oC PCM heating Pipe metre length (min)	Time to melt 25 oC PCM heating Pipe design length (min)	Time to melt 22 oC PCM heating Pipe design length (min)	Time to melt 15 oC PCM heating Pipe design length (min)
1	14			3.9			54.6
2	6.5	15.4	8.3		100.1	53.95	
3	7	15.4	8.3		107.8	58.1	
4	7	15.4	8.3		107.8	58.1	
5	7			6.6			46.2
6	17			6.6			112.2
7	18			3.9			70.2
8	19	15.4	8.3		292.6	157.7	
9	15	15.4	8.3		231	124.5	
10	15	15.4	8.3		231	124.5	
11	11	15.4	8.3		169.4	91.3	
12	15	15.4	8.3		231	124.5	
13	15	15.4	8.3		231	124.5	
14	15	15.4	8.3		231	124.5	
15	11	15.4	8.3		169.4	91.3	
16	11	15.4	8.3		169.4	91.3	
17	20	15.4	8.3		308	166	
18	20	15.4	8.3		308	166	
19	20	15.4	8.3		308	166	
20	20	15.4	8.3		308	166	
21	18	15.4	8.3		277.2	149.4	

The heating system has been designed such that there is always enough thermal energy stored in the PCM heating pipes and the PCM storage thanks so the length of time to melt the whole PCM in the heating pipe is not critical.

The longest time to melt 20 m length of 25 ^{o}C PCM heating pipe from table 5:23 calculation is 308 minutes. The point is that the system will not wait till the whole heat energy in the PCM heating pipe is completely exhausted before charging it.

5.4.3 Validation of PCM melting time or rate

From table 5:19 and table 5:21, the heat transfer rate from the hot water to the 25 and 22 ^{o}C PCM heating pipe with temperature difference of 4 and 7 ^{o}C between the hot water and the PCM heating pipes is 2.37 and 4.164 kW respectively.

The mass flow rate of hot water to produce 2.37 kW heat energy at a temperature difference of 4 ^{o}C between the average hot water temperature of 29 ^{o}C and the 25 ^{o}C PCM heating pipe is 0.141746 kg/s. Similarly the mass flow rate of hot water to produce 4.164 kW heat energy at a temperature difference of 7 ^{o}C between the average hot water temperature of 29 ^{o}C and the 22 ^{o}C PCM heating pipe is 0.14231 kg/s.

From Table 5:24 below shows the calculation of the mass flow rate and melting times.

Table 5:24 Mass flow rate and melting time of PCM heating pipes

Hot water mass flow rate to meet heat transfer rate (kg/s)	Spec heat capacity (kJ/kgK)	Hot water entry temp (oC)	PCM phase change temp (oC)	Water heat transfer rate (kW)	PCm melting time per metre length of pipe (m)	PCm heating pipe length (m)	Melting time (mins)
0.14175	4.18	29	25	2.37	15.4	19	292.6
0.14231	4.18	29	22	4.164	8.3	19	157.7

The energy stored in the TubeICE container (PCM tested melting rate in TubeICE container) from calculation (Table 5:25 below) is 540.817 kJ. The rate of heat transfer to melt the tested PCM with stored energy of 504.817 is 0.036054 kW based on 250 minutes melting time.

The stored energy in a metre length of the PCM heating pipe is calculated to be 2193.3 kJ and requires heat transfer rate of 0.14622 kW to melt it in 250 minutes at temperature difference of 4 ^{o}C between the PCM and the hot water.

The mass flow rate of hot water to produce 0.14622 kW at 4 ^{o}C temperature difference could be obtained from the formula McpΔT where M is the mass flow rate, cp is specific heat capacity of water and ΔT is the temperature difference between the hot water and the

PCM filled in the heating pipe which is 4 ^{o}C. The mass flow rate (M) = 0.14622 / 4 x 4.18 = 0.008745 kg/s.

This is the mass flow rate to melt the PCM solution in a metre length of the PCM heating pipe in 250 minutes. The equivalent mass flow rate to melt the PCM solution in a metre length of the PCM heating pipe at 15.4 minutes as calculated in table 5:24 will require 0.141964 kg / s (250 / 15.4 x 0.008745 kg / s).

From table 5:24 the required mass flow rate to achieve 15.4 minutes in melting the PCM solution in the PCM heating pipe with temperature difference of 4 ^{o}C between the hot water and the PCM solution in the PCM heating pipe as explained above is 0.141964 kg / s.

The difference between the mass flow rate based on the tested PCM melting rate results of 0.143357 kg / s and the calculated value obtained in table 5:24 is 0.97 % which shows that the calculated melting time 15.4 minutes to melt a metre length of the solution in the PCM heating pipe is accurate and can be relied on.

Table 5:25 below shows PCM melting rate validation. The table contains the properties of the selected PCM being used to store energy for the research heating system. These include phase change temperature, latent heat of fusion and density. Figure 2:9 in chapter 2 shows test result of the melting rate profile of hydrated salt in TubeICE container. The test melting rates (time) were obtained from figure 2:9 and populated into table 5:25 to assist in the calculation of the melting rate of the PCM solution in the heating pipes. The size of the TubeICE container is 50 mm diameter by 1000 mm in length as illustrated in figure 2:6. The volume of the TubeICE container was calculated to be 0.0019638 m^3 and the PCM density is 1530 kg / m^3.

The calculated PCM mass content in the tubeICE container is 3 kg approximately. The stored energy was calculated to be 504.817 kJ (Latent heat of fusion x the mass of the PCM solution) and the time taken to melt 25 ^{o}C PCM solution from the test result is 140 minutes when the temperature difference between the TubeICE container and the hot water is 7 ^oC.

The PCM solution volume in the heating pipe is calculated to be 0.00796 m^3. The calculated mass of the PCM solution in the heating pipe is 12.185 kg with the PCM density of 1530 kg / m^3. This suggest a stored energy of 2193.3 kJ (Latent heat of fusion x the mass of the PCM solution)

Table 5:25 shows the melting time or rates using the test results as the bases of calculation or measure and compare with the theoretical calculation. The difference is very close which suggest that the calculated rate of melting is accurate. For example, the calculated time to melt a metre length of PCM solution in the PCM heating pipe is 15.4 minutes compare with test results value of 15.25 minutes. The difference is just 0.97%.

Table 5:25 PCM melting rate validation

PCM type	Phase change temp (oC)	Latent heat capacity (kJ/kg)	TubeICE Container volume (m3)	PCM density (kg/m3)	Mass of PCM in TubeICE (kg)	PCM energy stored per tubeICE (KJ)	PCM heating pipe volume (kg/m3)	PCM heating pipe mass (kg)	PCM energy stored in pipe (kJ)	ΔT b/n PCM & hot water (oC)	TubeICE melting time at various temp diff (min)	Rate of heat transfer to tubeICE PCM (kW)	Rate of heat transfer to melt PCM in annular pipe (kW)	Water mass flow rate to melt PCM in heating pipe (kg/s)	Time to melt a metre of PCM heating pipe (min)	Calculated mass flow rate to melt PCM in heating pipe in 15.4 min (kg/s)	Mass flow rate to achieve 15.4 mins with tested PCM heat transfer	PCM heating pipe melting time based on tested results (min)
S25	25	180	0.0019638	1530	3.00454	540.817	0.00796	12.185	2193.3	3	320	0.028168	0.11423	0.00911	20.3	0.14329676	0.1446995	20.15
S21	22	170	0.0019638	1530	3.00454	510.771	0.00796	12.185	2071.4	3	302	0.028188	0.11432	0.009116	14.9	0.18513067	0.1947368	14.14
S15	15	160	0.0019638	1510	2.96526	474.442	0.00796	12.026	1924.1	3	281	0.02814	0.11412	0.009101	8.2	0.31094073	0.3139845	8.14
S25	25	180	0.0019638	1530	3.00454	540.817	0.00796	12.185	2193.3	4	250	0.036054	0.14622	0.008745	15.4	0.14196748	0.1433572	15.25
S21	22	170	0.0019638	1530	3.00454	510.771	0.00796	12.185	2071.4	4	236	0.036071	0.14629	0.008749	11.6	0.17767838	0.1868979	11.05
S15	15	160	0.0019638	1510	2.96526	474.442	0.00796	12.026	1924.1	4	220	0.035943	0.14577	0.008718	6.4	0.29786708	0.3007828	6.38
S25	25	180	0.0019638	1530	3.00454	540.817	0.00796	12.185	2193.3	5	200	0.045068	0.18277	0.008745	12.7	0.13756489	0.1389115	12.59
S21	22	170	0.0019638	1530	3.00454	510.771	0.00796	12.185	2071.4	5	189	0.045042	0.18267	0.00874	9.3	0.17749036	0.1867001	8.85
S15	15	160	0.0019638	1510	2.96526	474.442	0.00796	12.026	1924.1	5	175	0.045185	0.18325	0.008768	5.1	0.29956918	0.3025016	5.07
S25	25	180	0.0019638	1530	3.00454	540.817	0.00796	12.185	2193.3	7	140	0.064383	0.26111	0.008924	8.9	0.14037234	0.1417464	8.81
S21	22	170	0.0019638	1530	3.00454	510.771	0.00796	12.185	2071.4	7	132	0.064491	0.26155	0.008939	6.5	0.18152423	0.1909433	6.18
S15	15	160	0.0019638	1510	2.96526	474.442	0.00796	12.026	1924.1	7	123	0.064288	0.26072	0.00891	3.6	0.30444022	0.3074203	3.57
S25	25	180	0.0019638	1530	3.00454	540.817	0.00796	12.185	2193.3	10	90	0.100151	0.40617	0.009717	5.7	0.15284988	0.1397913	6.3
S21	22	170	0.0019638	1530	3.00454	510.771	0.00796	12.185	2071.4	10	85	0.100151	0.40617	0.009717	4.2	0.19732751	0.1909433	4.3
S15	15	160	0.0019638	1510	2.96526	474.442	0.00796	12.026	1924.1	10	79	0.100093	0.40593	0.009711	2.3	0.33180131	0.3034541	2.5

5.4.4 Maintaining supply and return temperatures of the LTHW and reducing PCM melting time

The heat transfer to the PCM to change the phase from solid to liquid is by means of conduction as demonstrated above. The heat from the supply hot water is conducted through the inner pipe wall to the PCM solution. Since the temperature difference across the inner pipe wall to the PCM is assumed to remain at $T_{hw} > T_{pcm}$ throughout the melting process, the pipe wall conduction rate is constant.

Table 5:24 shows the calculated mass flow rate required to achieve hot water supply and return temperatures of 32 and 25 ^{o}C respectively with an average heat transfer temperature of 29 ^{o}C to the PCM in the annular pipe.

For example, to maintain flow and return temperatures of 32 and 25 ^{o}C of the hot water passing through 19 m 25 ^{o}C PCM heating pipe to achieved 292.6 minutes melting rate will require a flow rate of 0.141964 kg/s as demonstrated in table 5:24.

The melting time can be changed by regulating the hot water flow rate. For example if the flow rate of the hot water is reduced from 0.141964 kg/s to 0.09567kg/s, the melting time will be 434 minutes. Alternatively, the melting rate could be reduced if the flow rate is increased.

5.4.5 Calculation of mean space temperature above PCM phase change temperature of a zone

The IES design software tool was used in assessing and analysing each zone space conditioning load, energy demand, solar gains, relative humidity and temperature profiles throughout the design period from January to December.

Figure 5:9 below shows the heating load, solar gains and space temperature profile on 26 March in zone 1. In zone 1, 15 ^{o}C PCM filled heating pipe will be installed with the phase change temperature of 15 ^{o}C. The 15 ^{o}C phase change temperature PCM will start to melt when the space temperature exceeds 15 ^{o}C.

The calculation of the mean temperature as can be seen from Figure 5:9 started from 1030 hours when the space temperature was 15.27 ^{o}C through to 1630 hours when the temperature was 15.5 ^{o}C. The mean temperature from 1030 to 1630 hours was calculated to be 18.4 ^{o}C and this occurred for six hours.

Time	Heating load (kW)	Solar gain (kW)	Temp oC
00:30	3.699	0	9.42
01:30	2.9238	0	9.49
02:30	2.3111	0	9.6
03:30	1.981	0	9.67
04:30	1.9737	0	9.67
05:30	2.2696	0.5193	9.64
06:30	1.7742	1.9559	9.75
07:30	0.2298	4.8264	10.18
08:30	0	10.8956	11.94
09:30	0	13.9869	13.65
10:30	0	19.9387	15.27
11:30	0	30.7765	18.78
12:30	0	30.1391	20.68
13:30	0	23.5897	20.39
14:30	0	17.3172	19.24
15:30	0	9.9836	17.32
16:30	0	5.1133	15.5
17:30	0	2.4701	13.7
18:30	0	0.7804	12.3
19:30	0	0	11.25
20:30	0	0	10.68
21:30	0	0	10.37
22:30	0	0	10.15
23:30	0	0	10.11

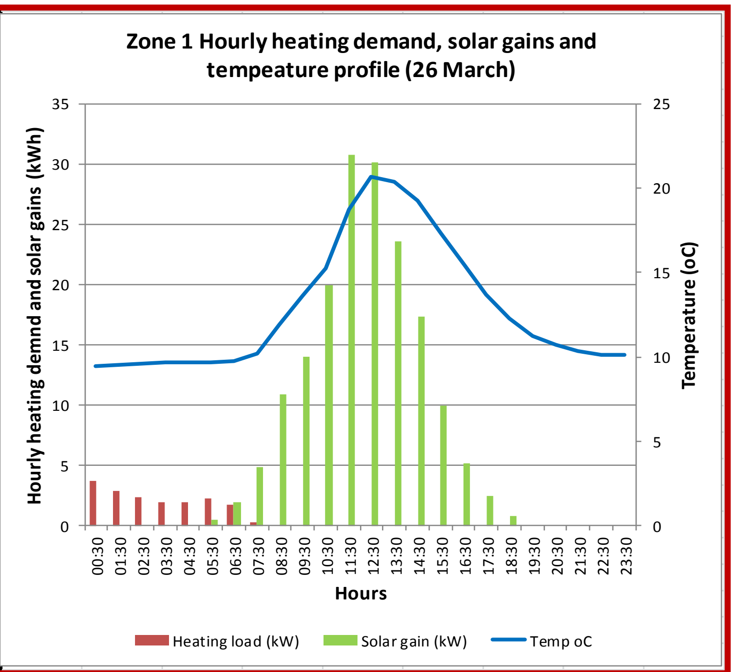


Figure 5:9 Zone 1 mean space temperature above PCM phase change temperature and period occurred on 26 March

Another example of calculating mean space temperature above PCM phase change temperature is demonstrated using zone 8 temperature profile in 16 April design day. This is demonstrated in Figure 5:10. In this zone, two PCM filled heating pipes are installed with the phase change temperatures of 22 and 25 ^{o}C.

The 22 ^{o}C and 25 ^{o}C phase change temperature PCM will start to melt when the space temperature exceeds 22 ^{o}C and 25 ^{o}C.

The calculation of the mean temperature as can be seen from figure 5:10 started from 1130 hours when the temperature was 27.7 ^{o}C through to 1730 hours when the temperature was 28.67 ^{o}C. The mean temperature from 1130 to 1730 hours was calculated to be 35 ^{o}C and this occurred for six hours as demonstrated in Figure 5:10 below.

Time	Heating load (kW)	Solar gain (kW)	Temp oC
00:30	20.4302	0	17.69
01:30	22.014	0	17.59
02:30	23.5278	0	17.5
03:30	24.8497	0	17.43
04:30	25.0756	0	17.41
05:30	25.1382	0	17.4
06:30	21.8208	2.7472	17.66
07:30	14.4063	7.1533	18.25
08:30	8.5297	11.4151	19.07
09:30	3.3877	15.4852	19.96
10:30	0	23.764	22.28
11:30	0	38.3175	27.7
12:30	0	51.3912	33.68
13:30	0	56.7239	37.28
14:30	0	60.0539	40.12
15:30	0	57.1948	41.1
16:30	0	43.02	36.5
17:30	0	19.8463	28.67
18:30	0	5.7733	22.09
19:30	4.874	1.0471	19.19
20:30	9.9246	0	18.59
21:30	12.5557	0	18.33
22:30	14.7529	0	18.17
23:30	16.0508	0	18.06

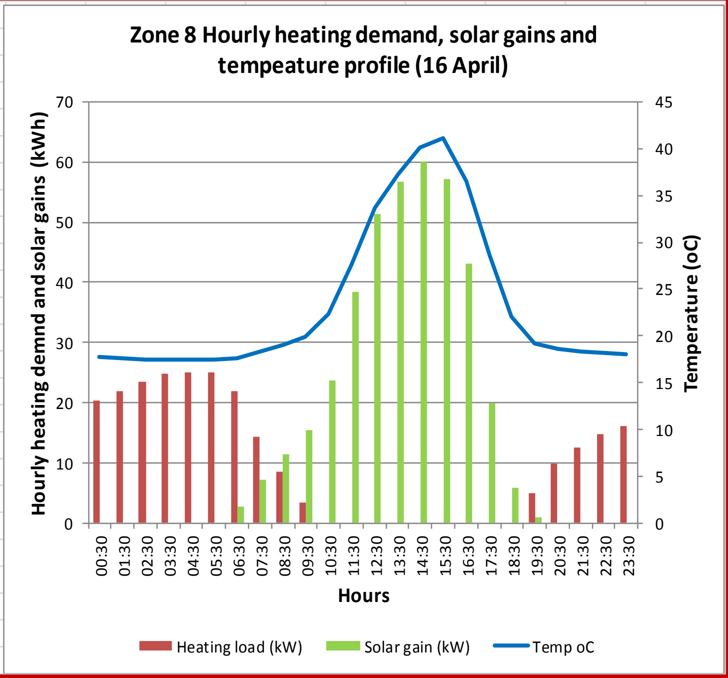


Figure 5:10 Zone 8 mean temperature above PCM phase change temperature and period occurred on 16 April

The daily mean temperature when the space temperature exceeds the PCM phase change temperature as demonstrated in Figure 5:9 and Figure 5:10 were calculated from March to October across the twenty-one zones.

From March to October are the months when the solar gains in the zones are sufficient to raise the space temperature above the installed PCMs phase change temperatures. The determination of the mean temperature when the space temperature exceeds the installed PCM phase change temperature in the zones is important to establish periods where the PCM filled heating pipes could be charged.

Table 5:26 below shows the mean temperatures that occurred within a period where the space temperature exceeded the phase change temperature of the PCM.
The table shows the mean temperature calculation of zone 1 based on the IES software calculation space temperature profile of the zone from March to October.
Zone 1 space temperature set point is 9 ^{o}C and the heating pipe installed in the zone contains 15 ^{o}C phase change temperature PCM which means that space temperature exceeding 15 ^{o}C will cause the PCM to melt.

This assessment will assist to calculate the thermal energy that can be transferred or absorbed by the PCM heating pipe in daily bases when the space temperature exceeds the PCM phase change temperature.

This free energy absorbed by the PCM heating pipe will contribute to a greater extent reduce zone heating energy demand required to be produced by the active solar hot water system. The excessive energy trapped in the space due to solar gains would have been vented to the atmosphere wasted without the installation of PCM heating pipes to store the energy.

This energy absorbed by the PCM heating pipes is free, inexhaustible and sustainable. The absorbed energy would have been produced by burning fossil fuel emitting tonnes of Carbon dioxide (CO_2) into the atmosphere causing environmental hazards but this has become useful energy using PCM storage techniques.

Table 5:26 shows mean space temperatures above PCM phase change temperature of 15 ^{o}C and the period that the mean temperature occurred in Zone 1. The method used in calculating the daily mean space temperature above PCM phase change temperature was used in calculating the mean temperature values of all the other zones.

Table 5:26 Daily space mean temperature above PCM phase change temperature and hours occurred

Mar	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp							17	18	18	18	21	22	21	15		16							16	16	19	18	19	27	29	23	22
Hours occurred							6	10	4	6	6	6	6	1	0	2	0	0	0	0	0	0	4	2	5	6	6	10	12	10	8
Apr	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	23	27	27	15	22	24	21	17	26	23	21	24	20	16	18	25	16	22	18	15	15	25	20	21	17	20	19	20	23	19	
Hours occurred	8	10	10	1	8	10	6	4	10	6	8	6	10	4	8	10	4	10	4	1	2	6	8	8	6	10	4	6	8	10	0
May	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	20	30	24	15	20	23	21	17	27	28	34	34	34	37	27	20	23	20	26	31	26	27	29	25	26	20	18		16		
Hours occurred	6	14	8	1	10	12	10	8	12	13	14	16	16	16	22	16	10	6	12	12	12	16	14	14	14	12	8	0	6	0	0
Jun	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	16	31	27	35	27	28	32	15	16	33	31	18	24	35	22	20	26	22	28	34	21	20	19	21	18	29	30	36	23	26	
Hours occurred	8	14	12	16	12	14	14	6	4	16	19	16	16	16	18	10	14	14	18	16	18	12	14	12	12	16	16	16	18	16	0
Jul	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	27	23	23	22	21	20	23	23	22	21	18	18	23	27	16	27	18	25	26	20	22	26	24	28	25	21	20	20	22	21	29
Hours occurred	14	14	12	12	14	12	12	14	12	14	12	14	14	16	6	14	10	14	16	16	14	14	10	16	14	14	12	14	16	10	14
Aug	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	29	16	26	24	22	24	28	24	21	21	31	25	26	20	28	28	30	24	21	28	18	24	23	17	22	19	26	20	25	24	22
Hours occurred	12	8	16	18	16	15	14	15	12	12	16	20	18	16	16	16	16	18	18	16	10	12	12	10	12	10	14	14	14	12	10
Sep	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	18	20		25	29	19	24	17	22	18	20	21	20	24	20	17	17	19	27	27	27	27	26	20	19	20	22	22	22		
Hours occurred	8	10	0	10	10	10	10	4	8	8	8	8	8	10	12	10	8	8	10	10	12	10	10	10	10	8	8	8	8	0	0
Oct	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	20	22	22	23	22	22	18	19		16	16	18	22			19	18	23	15	20		16		17	16	20	16			16	
Hours occurred	8	8	8	8	8	8	4	6	0	4	2	4	6	0	0	6	6	4	4	8	0	2	0	4	4	6	2	0	0	2	0

The amount of heat that can be transferred from the space to the 15 ^{o}C phase change temperature PCM throughout March to October can be calculated using the mean space temperature data from table 5:26 above.

5.4.6 Daily heat transfer and energy absorbed by PCM heating pipes calculation example using zone 8 details

Two examples of heat transferred and energy absorbed calculation using zone 8 details are demonstrated in table 5:27 and table 5:28. In the first example (Table 5:27), the daily mean space temperature above 22 ^{o}C PCM phase change temperature is 26 ^{o}C and the mean space temperature occurred for 4 hours.

Using equation 1to determine the mean Nusselt number below.

$$\overline{Nu}_{Ln} = \left\{0.825 + \frac{0.387\, Ra_{Ln}^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}}\right\} \qquad (1)$$

The heat transfer is 5.19kW using equation 1and the space mean temperature above the PCM melting point occurred for 4 hours therefore the heat energy that can be absorbed by the PCM heating pipe is 20 kWh (5.19 kW x 4). The calculation is demonstrated in Table 5:27 below.

Using equation 2 to determine the mean Nusselt number

$$\overline{Nu}_D = \left\{0.68 + \frac{0.670\, Ra_D^{1/4}}{[1+(0.492/Pr)^{9/16}]^{4/9}}\right\} \; Ra_{Ln} \lesssim 10^9 \qquad (2)$$

The heat transfer is 4.91kW using equation 2 and the space mean temperature above the PCM melting point occurred for 4 hours therefore the heat energy that can be absorbed by the PCM heating pipe is approximately 20 kWh (4.91 x 4). The calculation is demonstrated in table 5:27 below.

Table 5:27 Heat transfer and absorbed energy by 22 ^{o}C PCM heating pipe on 26 March design day

g	β	Ts	T∞	D	V	α	vα	RaD					
m/s2	K-1	oC	oC	m	m2/s	m2/s							
9.8	0.04166667	22	26	0.1143	0.00001589	0.0000225	3.575E-10	6821927.524					
Equation 1													
RaD	0.387Ra1/6	Pr	0.492	[1+ (0.492	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	$\overline{Nu}_D$	λ (w/m.k)	Heat transfer coefficient	Heat transfer rate (kW)	Hours heat transfer occurred (hrs)	Length of pipe (m)	Total heat transfered to PCM (kW)	Energy stored (kWh)
6821928	5.32963859	0.71	0.7	1.193275	5.29139515	27.998863	0.0263	6.442432961	0.009254699	4	561	5.1918863	20.768
Equation 2													
RaD	0.670Ra1/4	Pr	0.492	[1+ (0.492/Pr)9/16]4/9	{0.68 + [(0.670Ra1/4) / (1+ 0.492/Pr)9/16]4/9 }	$\overline{Nu}_D$	λ (w/m.k)	Heat transfer coefficient	Heat transfer rate (W)	Hours heat transfer occurred (hrs)	Length of pipe (m)	Total heat transfered to PCM (kW)	Energy stored (kWh)
6821928	34.2414051	0.71	0.7	1.303155	26.95578194	26.955782	0.0263	6.202424017	0.008909921	4	561	4.9984657	19.994

The second example is demonstrated in table 5:28. In zone 8, 22 and 25 ^{o}C phase change temperature PCM heating pipes will be installed. On 23 July design day, the mean space temperature above 22 and 25 ^{o}C PCM phase change temperature is 35 ^{o}C and this space mean temperature occurred for 6 hours.

Using equation 1 for the second example, the heat transfer is 15.8 kW and the space mean temperature above the PCM melting point occurred for 6 hours therefore the heat energy that can be absorbed by the PCM heating pipe is approximately 95 kWh. The calculation is demonstrated in table 5.28 below.

Using equation 2 for the second example, the heat transfer is 14.8 kW and the space mean temperature above the PCM melting point occurred for 6 hours therefore the heat energy that can be absorbed by the PCM heating pipe is approximately 89 kWh. The calculation is demonstrated in table 5:28.

Table 5:28 Heat transferred and energy absorbed by the 25 ^{o}C PCM heating pipe on 23 July design day

g	β	Ts	T∞	D	V	α	να	RaD					
m/s2	K-1	oC	oC	m	m2/s	m2/s							
9.8	0.03333333	25	35	0.1143	0.00001589	0.0000225	3.575E-10	13643855.05					
Equation 1													
RaD	0.387Ra1/6	Pr	0.492	[1+ (0.492	{0.825 + [(0.387Ra1/6) /(1+ 0.492/Pr)9/16]8/27 }	$\overline{Nu_D}$	λ (w/m.k)	Heat transfer coefficient	Heat transfer rate (kW)	Hours heat transfer occurred (hrs)	Length of pipe (m)	Total heat transfered to PCM (kW)	Energy stored (kWh)
13643855	5.98231843	0.71	0.7	1.193275	5.838360207	34.08645	0.0263	7.843163891	0.028167202	6	561	15.8018	94.811
Equation 2													
RaD	0.670Ra1/4	Pr	0.492	[1+ (0.492/Pr)9/16]4/9	{0.68 + [(0.670Ra1/4) / (1+ 0.492/Pr)9/16]4/9 }	$\overline{Nu_D}$	λ (w/m.k)	Heat transfer coefficient	Heat transfer rate (W)	Hours heat transfer occurred (hrs)	Length of pipe (m)	Total heat transfered to PCM (kW)	Energy stored (kWh)
13643855	40.7201226	0.71	0.7	1.303155	31.92734684	31.927347	0.0263	7.346362396	0.026383035	6	561	14.800883	88.805

The above method was used to calculate the heat transfer and energy that can be absorbed by the PCM heating pipes for the 21 zones.

5.4.7 Daily heat transfer to PCM heating pipes in all the zones due to internal heat gains

The energy that can be absorbed by the PCM heating pipes should be known to determine the energy that has to be stored by the PCM storage thanks to size it appropriately.
Table 5:29 shows calculation of heat transfer and heat energy that can be absorbed daily by the PCM heating pipes of zone 8 using table 5:28 heat transfer calculation method.

The table shows daily heat transfer and heat energy that can be absorbed (using equation 1and 2) by the PCM heating pipes in zone 8 for all the months starting from March to October where sufficient solar gain is possible.

Table 5:29 and table 5:30 show calculated daily heat transfer and heat energy that can be absorbed by the PCM heating pipes in March and April respectively. May to October calculated daily heat transfer and heat energy that can be absorbed by the PCM heating pipes of zone 8 is demonstrated in appendix D.

Table 5:29 shows calculated daily heat transfer and heat energy that can be absorbed by the PCM heating pipes in March

	Days of the month																														
Mar	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp							23	24	24	25	27	29	27										22		26	26	26	36	40	32	31
Hours occurred							2	2	2	4	6	6	6										1		4	4	4	8	8	8	8
PCM 22 oC																															
Heat transferred (kW) Eq 1							1	3	3	3	8	11	8										0		5	5	5	24	34	16	13
Heat transferred (kW) Eq 2							1	3	3	3	5	11	5										0		5	5	5	24	32	16	13
Heat energy stored (kWh) Eq 1							3	5	5	13	42	64	42										0		21	21	21	196	267	130	114
Heat energy stored (kWh) Eq 2							3	5	5	13	40	58	40										0		21	21	21	183	243	122	106
PCM 25 oC																															
Heat transferred (kW) Eq 1										0	3	5	3												0	0	0	19	26	11	8
Heat transferred (kW) Eq 2										0	3	5	3												0	0	0	16	24	11	8
Heat energy stored (kWh) Eq 1										0	13	29	13												3	3	3	143	209	82	66
Heat energy stored (kWh) Eq 2										0	13	29	13												3	3	3	132	193	77	64

Table 5:30 Show calculated daily heat transfer and heat energy that can be absorbed by the PCM heating pipes in April

	Days of the month																														
Apr	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	32	33	33		30	32	28	23	34	30	29	31	28		24	32		31	24			34	27	29	24	29	26	28	30	26	
Hours occurred	6	8	10		4	8	4	2	8	6	8	6	8		2	8		8	4			6	6	6	2	6	4	4	8	8	
PCM 22 oC																															
Heat transferred (kW) Eq 1	16	19	19		13	16	8	0	21	13	11	13	8		3	16		13	3			21	8	11	3	11	5	8	13	5	
Heat transferred (kW) Eq 2	16	16	16		11	16	8	0	19	11	11	13	8		3	16		13	3			19	5	11	3	11	5	8	11	5	
Heat energy stored (kWh) Eq 1	98	146	183		50	130	34	3	164	74	85	85	69		5	130		114	8			122	42	64	5	64	21	34	98	42	
Heat energy stored (kWh) Eq 2	90	135	169		48	122	32	3	151	69	79	79	66		5	122		106	8			111	40	58	5	58	21	32	93	40	
PCM 25 oC																															
Heat transferred (kW) Eq 1	11	13	13		8	11	3		13	8	5	8	3			11		8				13	3	5		5	0	3	8	0	
Heat transferred (kW) Eq 2	11	11	11		5	11	3		13	5	5	8	3			11		8				13	3	5		5	0	3	5	0	
Heat energy stored (kWh) Eq 1	61	95	119		26	82	13		111	40	40	50	29			82		66				82	13	29		29	3	13	53	8	
Heat energy stored (kWh) Eq 2	58	90	114		26	77	13		103	37	40	48	26			77		64				79	13	29		29	3	13	50	8	

5.4.8 Analysing heat energy absorbed and stored by the passive system using the PCM heating pipes

Table 5:31 shows calculated heat energy that could be absorbed and stored by the PCM heating pipes installed in each zone from March to October. These are the periods that the solar gains in the glasshouse are sufficient to raise the space temperature above the installed PCM solution in the heating pipes phase change temperature to charge them. The calculation is based on space mean temperature above the PCM installed phase change temperature in the zones.

Table 5:31 Passive system PCM heating pipes calculated monthly energy absorbed in each zone

Zone	March absorbed energy (kWh)	April absorbed energy (kWh)	May absorbed energy (kWh)	June absorbed energy (kWh)	July absorbed energy (kWh)	August absorbed energy (kWh)	september absorbed energy (kWh)	October absorbed energy (kWh)	Total (MWh)
1	429	571	2320	2644	1705	2155	917	145	10.886
2	156	642	2618	1568	921	829	1100	61	7.895
3	589	507	1044	558	200	269	412	145	3.724
4	1180	910	1815	956	323	467	717	275	6.643
5	5	10	68	101	62	86	27	6	0.365
6	277	401	1865	2060	1444	1754	661	84	8.546
7	804	1393	4226	5459	3251	4329	1830	516	21.808
8	755	1457	5805	7169	3273	4783	1637	314	25.193
9	359	764	4344	5811	2149	3465	1032	97	18.021
10	76	86	1041	1544	529	932	157	8	4.373
11	349	610	3177	4166	4921	7216	1992	318	22.749
12	523	904	3660	4728	2308	3219	1147	285	16.774
13	164	111	957	1564	524	786	178	22	4.306
14	33	36	578	824	97	186	41	0	1.795
15	76	114	936	1317	517	815	191	39	4.005
16	267	484	1752	2345	1312	1696	682	395	8.933
17	93	486	1843	1190	2991	2201	339	262	9.405
18	115	138	1344	1906	645	1079	244	36	5.507
19	76	65	968	1906	645	1079	244	36	5.019
20	266	468	2739	3737	1573	2455	654	166	12.058
21	66	160	1458	1996	779	1292	249	34	6.034
Total	6658	10317	44558	53549	30169	41093	14451	3244	**204.039**

It could be seen from table 5:31 above that the total absorbed energy from the twenty-one zones is 204 MWh which is quite a significant energy that could be saved.

5.5 Analysing and sizing of the active system and PCM storage tanks

The sizing of the active system will be determined based on the solar gains into the glasshouse and heat energy that the passive system could absorb and store especially from periods where solar gain in the greenhouse is sufficient to charge the PCM filled heating pipes installed.

Figure 5:11 below shows that the difference between the glasshouse heating energy demand and the solar gains is minimal and just about 5.5%. This suggests that with careful design and effective solar energy management, most of the glasshouse heating requirement could be met using solar gains. The IES design software tool was used to determine the heating demand and solar gains in all the twenty-one zones. The design of the glasshouse using the IES design software tool demonstrated the period where the solar gain is not enough to meet the glasshouse heating requirement.

These periods are from October to March (Figure 5:11) where the stored energy in the PCM storage tanks will be useful. From April to the end of September, the solar gains and energy stored in the PCM heating pipes will be the primary heating energy source.

The stored energy in the PCM storage tanks will be used when needed to charge the PCM heating pipes in the zones if the solar gains and the stored energy in the PCM heating pipes are not sufficient to meet the heating requirements.

The waste heat from the CHP (Engine jacket cooling medium) and the active solar system will be used to charge the PCM storage tanks that contain PCM with phase change temperatures of 44 and 34 ^{o}C. The heat loss from the storage tanks will be met by the 20% excess load added to the actual building heating load.

In winter the CHP recovered waste heat will be supplementary heat source to the active solar system. Because the heat energy obtained from the CHP system is a recovered waste heat, the system could be classified as zero carbon emission heating system with the exception of the small electrical energy that will be used by the hot water circulation pump.

Months	Space conditioning sensible energy demand (MWh)	Solar gains (MWh)	% difference between space heating demand and solar gains
Jan 01-31	411.7	46.2	88.8
Feb 01-28	362.8	63.9	82.4
Mar 01-31	302.4	183.1	39.5
Apr 01-30	179.0	263.3	-47.1
May 01-31	100.3	376.9	-275.8
Jun 01-30	57.3	389.5	-579.9
Jul 01-31	58.4	303.9	-420.9
Aug 01-31	54.3	312.0	-475.1
Sep 01-30	107.1	204.4	-90.8
Oct 01-31	212.9	135.0	36.6
Nov 01-30	248.7	48.7	80.4
Dec 01-31	402.1	32.2	92.0
Summed total	**2497.0**	**2359.2**	5.5

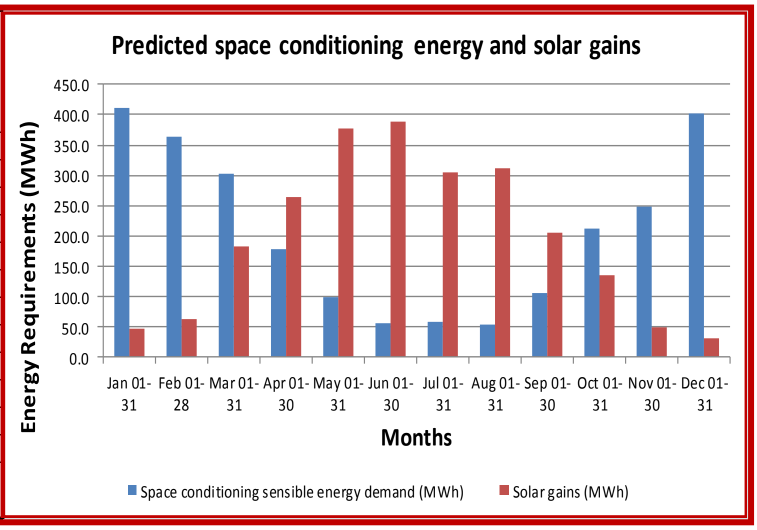


Figure 5:11 Predicted space heating energy demand and solar gains

Figure 5:12 to figure 5:16 demonstrate monthly zones heating energy demand and the energy that could be absorbed by the PCM heating pipes from March to October. The heat energy that could be absorbed by the PCM heating pipes in each zone is shown in Table 5:31 above but the graphs from figure 5:12 to figure 5:14 compare the zone heating energy demand against the heat energy absorbed by the PCM heating pipes.

The graphs did not cover all zones as they display similar pattern of absorbed energy and behaviours. Seventeen out of the twenty-one zones energy demand against the heat energy absorbed by the PCM heating pipes are demonstrated in the report.

Figure 5:12, 5.13 and 5.14 below show zone 1, 5 and 6 monthly heating energy demand against PCM heating pipes energy absorbed from March to October of the design year. The other fourteen zones graphs of monthly heating energy demand against PCM heating pipes energy absorbed is demonstrated in appendix D.

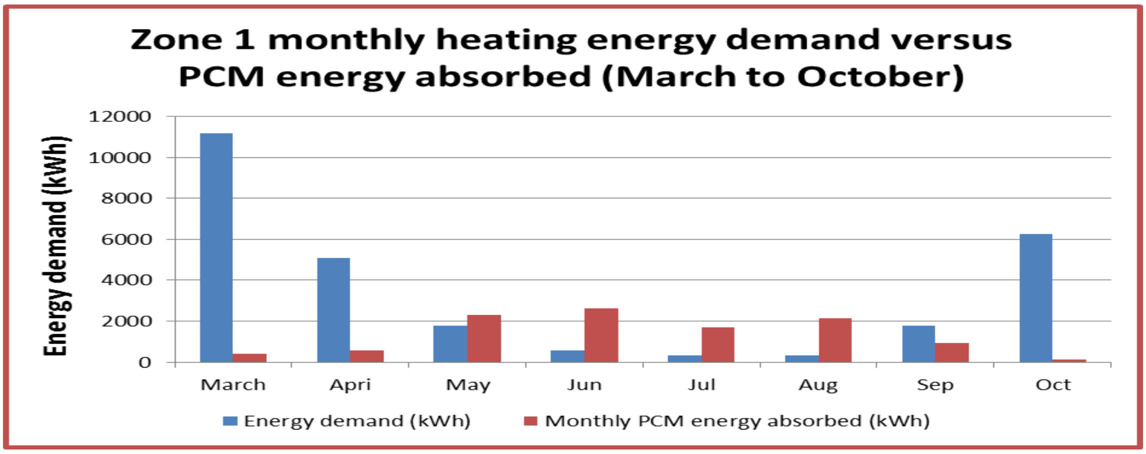


Figure 5:12 Zone 1 monthly heating energy demand versus PCM heating pipe absorbed energy

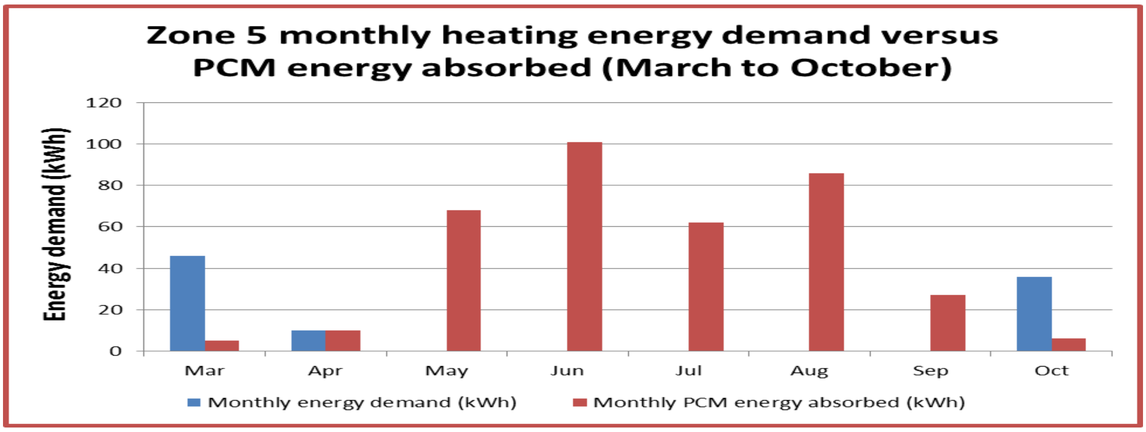


Figure 5:13 Zone 5 monthly heating energy demand versus PCM heating pipe absorbed energy

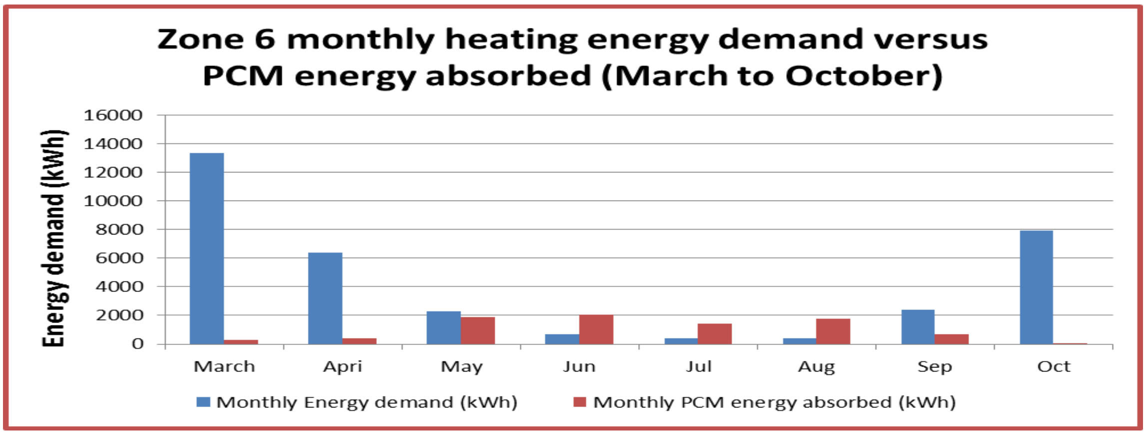


Figure 5:14 Zone 6 monthly heating energy demand versus PCM heating pipe absorbed energy

5.6 Heating system critical analysis

The above graphs (Figure 5:12 to 5.14) demonstrate the monthly heating energy demand and PCM absorbed energy considered only the periods where the space temperature was above the PCM phase change temperature. These are periods where the solar energy gain inside the glasshouse will be vented to the atmosphere wasted if the passive system is not installed to absorb this free and sustainable energy.

Figure 5:15 below shows the 21 zones monthly heating energy demand, solar gains and energy absorbed by the PCM heating pipes. These three entities are very important in the heating system analysis and sizing.

It could be seen from figure 5:15 that the solar gains from October to March is all useful as the glasshouse heating demand exceeds the solar gains and this is the period that additional source of energy is required from the active solar system and the CHP waste heat.

Within these months of the year only few zones temperatures rise above the installed PCM phase change temperature to charge it. As a result energy absorbed by the PCM heating pipes is minimal. The useful solar gains from October to March will be considered in sizing the active solar thermal hot water system.

	Monthly Space conditioning sensible energy demand (MWh)	Monthly Solar gains (MWh)	Monthly PCM heating pipe eneergy absorbed (MWh)
Jan 01-31	411.7	46.2	0
Feb 01-28	362.8	63.9	0
Mar 01-31	302.4	183.1	6.638
Apr 01-30	179.0	263.3	10.295
May 01-31	100.3	376.9	44.205
Jun 01-30	57.3	389.5	53.046
Jul 01-31	58.4	303.9	27.165
Aug 01-31	54.3	312.0	36.688
Sep 01-30	107.1	204.4	13.235
Oct 01-31	212.9	135.0	3.05
Nov 01-30	248.7	48.7	0
Dec 01-31	402.1	32.2	0

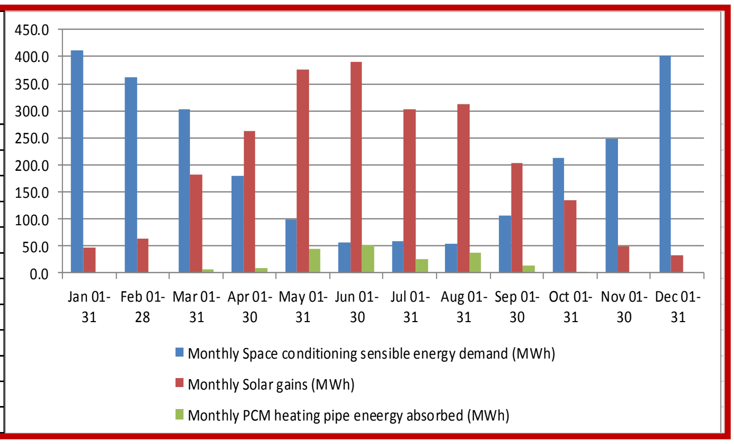


Figure 5:15 Showing monthly heating energy demand, solar gains and PCM heating pipe absorbed energy

The active solar system design will be based on heating energy requirement of the glasshouse from March to October where solar energy is significant. The CHP recovery waste heat recovery as mentioned above will supplement the heat energy that will be produced by the active solar system.

The sizing will be based on the zones heating energy demand in March, April and May. The assessment and analysis will be based on the heating requirements of these months to decide which solar collector size will be the most cost effective.

The system design will be linked to the close circuit low temperature hot water (LTHW) system in Kew Gardens so that in the event that the CHP breaks down in winter and the active solar thermal hot water system could not meet the heating demand then the close circuit hot water system can supplement to meet the glasshouse heating demand.

Table 5:31 shows how much energy that can be absorbed by the passive system for a year. This absorbed energy compare to the solar gains in the glasshouse in some instances are small. This is because the energy transfer to the PCM heating pipes is only possible when the space temperature rises above the PCM phase change temperature.

For example, in zone 1 the heating set point temperature is 9 ^{o}C and it could be seen from Figure 5:9 that at 0630 hours the solar gains in the glasshouse is sufficient to maintain the space temperature at 9.75 ^{o}C which means that no heat is required from any source.

The solar gain inside the zone is greater than the heating load and therefore no heat is required from neither passive nor the active solar system. The space temperature continues to rise as the solar gain increases but the PCM heating pipe could not absorb the heat energy trapped inside the zone until the temperature exceeds the phase change temperature of the PCM in the heating pipe which in this case is 15 ^{o}C.

The PCM will start to be charged or absorb heat energy from the space at 1030 hours till 1630 when the space temperature begins to fall below 15 ^{o}C.

The solar gain in the zone was sufficient to maintain the required space temperature until 0030 hours when the solar gain was insufficient to maintain the space temperature and heating was required to maintain the space set point temperature.

Comparatively, it could be seen from figure 5:15 that the heat absorbed by the PCM heating pipes is less in relation to the solar heat gains in the glasshouses.
Even though the heat absorbed by the PCM heating pipes compared to the solar gains is less, yet it is quite significant and useful when the solar gain is insufficient or nil for example in the nights to maintain the space set point temperature.

A similar story is told in figure 5:16 below. From 0730 to 1930 hours the solar gain in zone 18 May 2 design day is sufficient that no other heat source is required but the PCM heating pipes could only be charged when the space temperature reaches 22.3 ^{o}C at 0830 hours. The charging of the PCM heating pipes will continue till 1830 hours.

The installed PCM heating pipes in this zone have 22 and 25 ^{o}C phase change temperatures. The 22 ^{o}C phase change temperature PCM will start charging when the space temperature reaches 22.3 ^{o}C and the 25 ^{o}C PCM phase change temperature will start charging when the space temperature reaches 26.65 at 1030 hours till 1830 hours.

The space set point temperature of this zone is 20 ^{o}C so the solar gain is sufficient to maintain the set point temperature until 1930 hours. Heating will be required after 1930 hours to maintain the set point temperature.

Time	Heating load (kW)	Solar gain (kW)	Temp oC
00:30	25.1933	0	16.91
01:30	26.956	0	16.81
02:30	27.4775	0	16.78
03:30	27.7485	0	16.75
04:30	28.5703	0	16.7
05:30	27.9566	0.0394	16.77
06:30	22.0513	2.5924	17.24
07:30	8.4802	12.6355	18.45
08:30	0	25.2466	22.3
09:30	0	27.7016	24.78
10:30	0	30.4622	26.65
11:30	0	44.3114	31.5
12:30	0	52.304	36.48
13:30	0	42.6912	36.58
14:30	0	37.1217	34.54
15:30	0	39.5591	35.13
16:30	0	39.8292	34.84
17:30	0	31.6968	32.3
18:30	0	13.842	26.08
19:30	1.4386	2.5986	20.68
20:30	8.5381	0.0294	18.9
21:30	12.1856	0	18.22
22:30	14.1823	0	17.85
23:30	16.8317	0	17.53

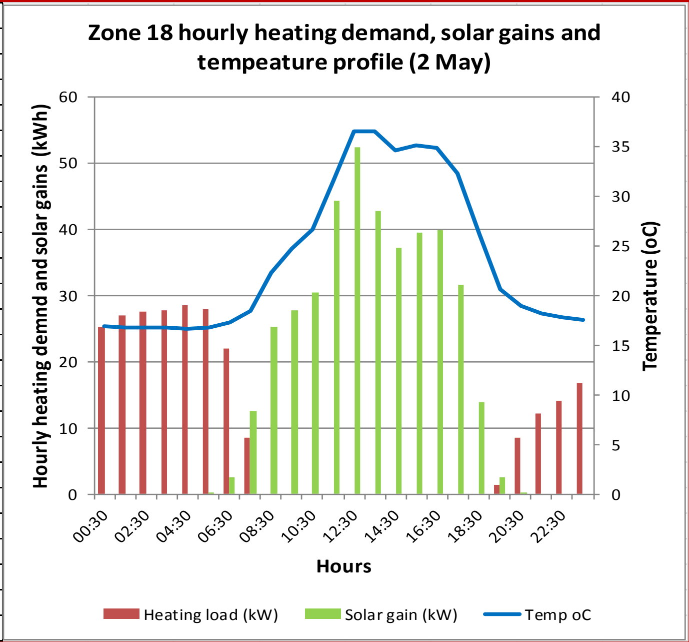


Figure 5:16 Heating demand, solar gains and temperature profile of zone 18 design day 02 May

The space set point temperature in zone 9 is 21 ^{o}C. In figure 5:17 below the period that the solar gain is sufficient to maintain the space set point temperature is from 1030 when the space temperature start rising from 20.15 ^{o}C.

The PCM heating pipes start charging from 1130 hours when the space temperature is 24.11 ^{o}C. The phase change temperature of the PCM heating pipes installed in this zone is 22 and 25 ^{o}C.

Time	Heating load (kW)	Solar gains (kW)	Temp oC
00:30	27.5134	0	17.48
01:30	29.5563	0	17.37
02:30	31.6565	0	17.26
03:30	33.0571	0	17.19
04:30	35.2248	0	17.08
05:30	37.9577	0	16.95
06:30	39.6862	0	16.88
07:30	37.3808	2.0463	17.02
08:30	30.4323	5.5728	17.41
09:30	17.3262	12.8489	18.21
10:30	3.2075	22.1074	20.15
11:30	0	27.8575	24.11
12:30	0	30.997	26.13
13:30	0	29.694	28.05
14:30	0	24.8875	28.94
15:30	0	17.5177	27.33
16:30	0	9.2117	23.13
17:30	5.8895	3.9602	20.15
18:30	13.4416	0.9214	18.81
19:30	18.4148	0	18.09
20:30	20.5547	0	17.92
21:30	23.5646	0	17.73
22:30	26.0932	0	17.58
23:30	28.0845	0	17.47

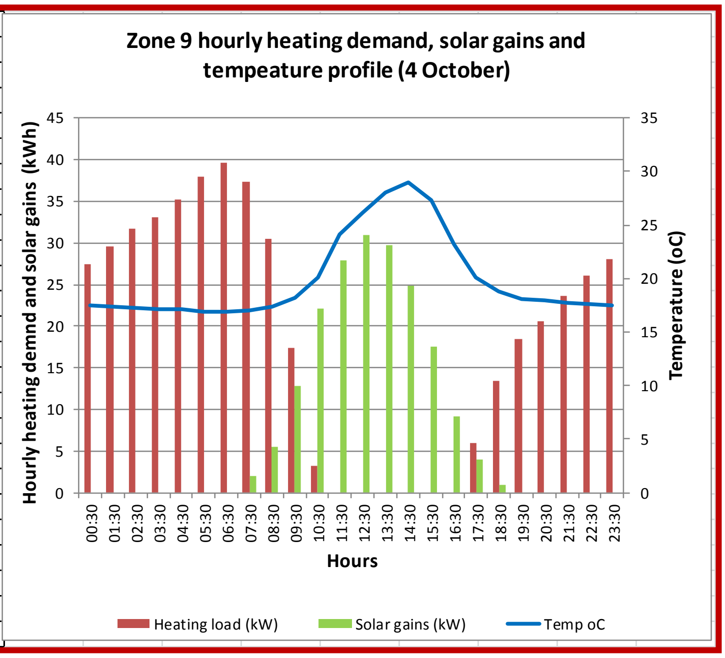


Figure 5:17 Heating demand, solar gains and temperature profile of zone 9 design day 04 October

The solar gain is sufficient to maintain the space temperature to the required standard between 1030 to 1730 hours but heat could only be absorbed from 1130 to 1630 by the PCM heating pipes which again emphasise the difference between the solar energy trapped in the glasshouse and the energy that could be absorbed by the PCM heating pipes.

5.7 Sizing of the active solar thermal hot water system

The calculated monthly space conditioning sensible energy demand of the glasshouse using the IES Software tool is shown in table 5.32 below

Table 5:32 Monthly glasshouse sensible space conditioning energy demand

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Sensible energy demand (MWh)	411.7	362.3	302.4	179	100.3	57.3	58.4	54.3	107.1	212.9	248.7	402

The sizing option of the solar collectors will be based on January, February, March, April and May heating energy demand of the glasshouse. The monthly heating load assessment and analysis will determine the appropriate collector size that will be technically and economically viable for the project.

Table 5:33 demonstrates the amount of solar collector area required based on water inlet temperature, monthly glasshouse sensible space conditioning energy demand and monthly solar delivered energy. It could be seen from the table that the higher the inlet water temperature the greater the number of solar collector area needed as the efficiency of the active solar energy system reduces at higher inlet temperatures. Cost analysis was carried out to determine the cost effectiveness of the solar collector area selected.

Table 5:33 Solar collector sizing based on water inlet temperature and monthly space conditioning energy demand

Inlet water temp oC	Jan collector delivered energy (kWh)	Feb collector delivered energy (kWh)	Mar collector delivered energy (kWh)	Apr collector delivered energy (kWh)	May collector delivered energy (kWh)	Collector area to meet Jan heating demand of 411660 kWh	Collector area to meet Feb heating demand of 362829 kWh	Collector area to meet Mar heating demand of 302442 kWh	Collector area to meet Apr heating demand of 179040 kWh	Collector area to meet May heating demand of 100296 kWh
20	17.83	20.20	52.87	92.96	110.00	23088	17962	5720	1926	912
25	17.64	20.03	52.69	92.79	109.81	23337	18111	5740	1930	913
30	17.46	19.87	52.50	92.61	109.63	23577	18263	5760	1933	915
35	17.28	19.70	52.32	92.43	109.45	23829	18417	5781	1937	916
50	16.72	19.20	51.77	91.89	108.89	24616	18896	5842	1948	921
60	16.35	18.87	51.40	91.54	108.52	25171	19229	5884	1956	924
70	15.99	18.54	51.03	91.18	108.16	25751	19575	5927	1964	927

The cost analysis concentrated more on 20, 25 and 30 ^{o}C inlet water temperatures as the active solar hot water heating system has been designed to have inlet water temperature of 25 ^{o}C. The 20 and 30 ^{o}C inlet water temperature is included in the assessment to determine the effects on the system should the inlet water temperature for unseen circumstances reduces to 20 or increases to 30 ^{o}C.

From table 5:33 the solar collector area for 20, 25 and 30 ^{o}C inlet water temperature in January is 23088, 23337and 23577 metre squared respectively which is not technically and economically viable as the site will not have this size of space to install the collectors and again the thermal energy that will be delivered in summer will be over the glasshouse heating demand and will be wasted.

The technical and economic benefits for January will not be far different from February so the design will not be based on these two months.

5.7.1 Analysis of selected solar collector

AZUR 8+ AC 2.8H solar collector is produced by Agena Energies SA. It is an efficient solar collector used by several contractors and figure 5:18 shows an example of the collector installation and table 5:18 demonstrates the collector module construction details. With this collector module the monthly delivered energy is demonstrated in Table 5:34 and the size of the collector required to meet monthly glasshouse heating demand is also demonstrated.

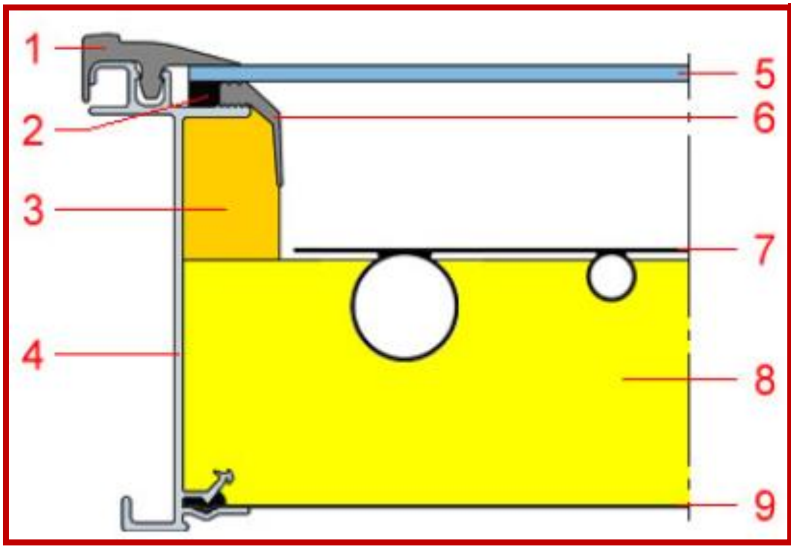
Table 5:34 solar collector sizing did not consider the heat energy absorbed by the passive system due to solar gains in the spaces but table 5:35 considered the thermal energy absorbed by the passive system due to solar gains in the spaces. The sizing of the solar collectors is based on the entry water temperature to the solar collector system, monthly collector delivered energy and monthly heating energy requirement of the glasshouse.

Table 5:36 shows the number of solar collector modules requirement to meet the monthly heating demand. For example, to meet the heating demand in March, the collector area required is 1,977 m^2 at 20 ^oC water entry temperature.

The selected collector aperture area per module is 2.463 m^2, therefore the number of modules required is 1977 divided by 2.463 m^2 which gives the result to be 803 modules.

Table 5:36 demonstrates the number of modules required for each month.

Figure 5:18 An example installation of solar collector Model: AZUR 8+ AC 2.8H



1 Glass fixing
2 Sealing
3 Lateral thermal insulation
4 Frame
5 Glazing
6 Glass support rail
7 Absorber
8 Thermal insulation
9 Rear panel

Figure 5:19 Solar collector module construction details

Table 5:34 below shows selected collector monthly delivered energy and collector size requirement to meet heating demand without considering heat energy absorbed by the passive system

Table 5:34 Selected collector monthly delivered energy and collector size requirement

	Jan collector delivered energy (kWh)	Feb collector delivered energy (kWh)	Mar collector delivered energy (kWh)	Apr collector delivered energy (kWh)	May collector delivered energy (kWh)	Collector area to meet Jan heating demand (m2)	Collector area to meet Feb heating demand (m2)	Collector area to meet Mar heating demand (m2)	Collector area to meet Apr heating demand (m2)	Collector area to meet May heating demand (m2)
Monthly heating demand (MWh)						411.7	362.8	302.4	179.0	100.3
Water inlet temp oC 20	50.61	57.30	149.62	262.83	310.94	8134	6332	2021	681	323
25	50.18	56.90	149.19	262.41	310.50	8204	6376	2027	682	323
30	49.74	56.51	148.75	261.99	310.07	8276	6421	2033	683	323

Table 5:35 below shows selected module monthly delivered energy and collector size requirement to meet heating demand considering the heat energy absorbed by the passive system.

Table 5:35 Selected module monthly delivered energy and collector size requirement

	Jan collector delivered energy (kWh)	Feb collector delivered energy (kWh)	Mar collector delivered energy (kWh)	Apr collector delivered energy (kWh)	May collector delivered energy (kWh)	Collector area to meet Jan heating demand (m2)	Collector area to meet Feb heating demand (m2)	Collector area to meet Mar heating demand (m2)	Collector area to meet Apr heating demand (m2)	Collector area to meet May heating demand (m2)
Monthly heating demand (MWh)						411.7	362.8	302.4	179.0	100.3
PCM heating pipes						0	0	6.6	10.3	44.2
Water inlet temp oC 20	50.61	57.30	149.62	262.83	310.94	8134	6332	1977	642	180
25	50.18	56.90	149.19	262.41	310.50	8204	6376	1983	643	181
30	49.74	56.51	148.75	261.99	310.07	8276	6421	1989	644	181

Table 5:36 Number of solar collector modules to meet monthly heating demand

	Jan collector delivered energy (kWh)	Feb collector delivered energy (kWh)	Mar collector delivered energy (kWh)	Apr collector delivered energy (kWh)	May collector delivered energy (kWh)	No of collector modules to meet Jan heating demand	No of collector modules to meet Feb heating demand	No of collector modules to meet Mar heating demand	No of collector modules to meet Apr heating demand	No of collector modules to meet May heating demand
Monthly heating demand (MWh)						411.7	362.8	302.4	179.0	100.3
PCM heating pipes monthly absorbed energy (MWh)						0	0	6.6	10.3	44.2
Water inlet temp oC 20	50.61	57.30	149.62	262.83	310.94	3302	2571	803	261	73
25	50.18	56.90	149.19	262.41	310.50	3331	2589	**803**	**261**	**73**
30	49.74	56.51	148.75	261.99	310.07	3360	2607	807	261	73

5.7.2 Solar collector system water flow rates analysis

The maximum, nominal and minimum flow rates of the selected solar collector module (AZUR 8+AC 2.8H) are 200 litres per hour (0.05556 l/sec), 40 litres per hour (0.1111 l/sec) and 20 litres per hour (0.00556 l/sec) respectively. Table 5:37 and table 5:38 shows the flow rates requirement for the number of modules selected to meet January, February, March, April and May design heating demands.

Table 5:37 shows maximum flow requirements and table 5:38 shows the nominal flow requirement. The minimum flow requirement to meet March designed solar collectors of 803 is 6 litres per second approximately 6 kg per second. The heating system design flow rate to meet peak heating demand is 32 litres per second which is well above the minimum flow rate so under no circumstance that the system flow rate will fall below the minimum solar active system flow requirement.

The glasshouse heating system as mentioned above will not be designed to meet January and February heating requirements as the size of solar collectors will not be technically and economically viable. Again there is free waste thermal energy to use from the CHP plant.

The design will concentrate on March, April and May heating demands. The heating system design hot water flow rate to meet the maximum heating demand in January of 914 kW is approximately 32 litres per second with flow and return temperatures of 32 and 25 ^{o}C respectively.

The designed hot water flow rate fits between the maximum and the nominal flow rate requirements of the selected solar collector modules to meet March, April and May heating requirements.

Table 5:37 Maximum flow rates to meet selected modules flow requirements for each month

Water inlet temp (°C)	Module nominal flow rate (l/sec)	Maximum flow rate to meet Jan design collectors (l/sec)	Maximum flow rate to meet Feb design collectors (l/sec)	Maximum flow rate to meet Mar design collectors (l/sec)	Maximum flow rate to meet Apr design collectors (l/sec)	Maximum flow rate to meet May design collectors (l/sec)
20	0.05556	183	143	45	14	4
25	0.05556	183	143	45	14	4
30	0.05556	183	143	45	14	4

Table 5:38 Nominal flow rates to meet selected modules flow requirements for each month

Water inlet temp (°C)	Module nominal flow rate (l/sec)	Nominal flow rate to meet Jan design collectors (l/sec)	Nominal flow rate to meet Feb design collectors (l/sec)	Nominal flow rate to meet Mar design collectors (l/sec)	Nominal flow rate to meet Apr design collectors (l/sec)	Nominal flow rate to meet May design collectors (l/sec)
20	0.01111	37	29	9	3	1
25	0.01111	37	29	9	3	1
30	0.01111	37	29	9	3	1

5.7.2.1 Actual water flow rate through the solar collector system

The actual hot water flow rate through the solar system will depend on the collect peak power, the designed inlet and outlet hot water temperature to achieve. The system is designed to have inlet water temperature of 25 °C to the solar collector system and returned at 50 °C to charge the PCM storage tanks.

The hot water will enter the 44 °C phase change temperature PCM storage tank at 50 °C and leave at 45 °C to enter the 34 °C phase change temperature PCM storage tank and leave at 32 °C to serve the PCM filled heating pipes in the zones. The set point temperatures will be achieved by varying the flow rate.

The collector peak power is illustrated in figure 5:20 below. The power generation of the solar collector system depends on the temperature difference between the mean fluid temperature of the solar collector system and the ambient. It could be seen from the graph that less power is generated as the temperature difference between the fluid and the ambient increases.

Figure 5:21 illustrates relative efficiency of the collector module based on solar collector gross, aperture and absorber areas. The peak power of the solar module is 2092W (Wpeak). The flow rate requirement will depend on the water inlet temperature to the solar collector system which table 5:39 demonstrates the flow rate requirements of 20, 25 and 30 ^{o}C inlet temperatures.

The flow rate increases as the temperature difference between the inlet and the system outlet designed temperature (50 ^{o}C) decreases. Table 5:39 assumes that the maximum collector power is achieved and calculates the flow rate required.
The maximum power (2092 W) per collect is multiplied by the number of collector modules required to achieve the design month heating demand.

For example, the number of collector modules to meet March heating demand is 803 so the maximum power generated by the number of collector is approximately 1680 kW with mass flow rate of 13, 16 and 20 kg/s when the inlet water temperature is 20, 25 and 30 ^{o}C respectively. Table 5:39 illustrates the flow rates calculation to meet March, April and May maximum power generated based on the number of collector requirements for each of the months.

Table 5:39 Flow rates requirement to meet maximum power generated by the system with designed outlet temperature of 50 ^{o}C

Flow rate requirement at 20 oC inlet temperature						
Design months being considered	No of modules required	Maximum Power generatred (kW)	Spec heat capacity of water (KJ/kgK)	Inlet water temp (oC)	Outlet water temp (oC)	Flow rate required (Kg/s)
March	803	1679.876	4.18	20	50	13
April	261	546.012	4.18	20	50	4
May	73	152.716	4.18	20	50	1
Flow rate requirement at 25 oC inlet temperature						
Design months being considered	No of modules	Power generatred (kW)	Spec heat capacity of water (KJ/kgK)	Inlet water temp (oC)	Outlet water temp (oC)	Flow rate required (Kg/s)
March	803	1679.876	4.18	25	50	16
April	261	546.012	4.18	25	50	5
May	73	152.716	4.18	25	50	1
Flow rate requirement at 30 oC inlet temperature						
Design months being considered	No of modules	Power generatred (kW)	Spec heat capacity of water (KJ/kgK)	Inlet water temp (oC)	Outlet water temp (oC)	Flow rate required (Kg/s)
March	803	1679.876	4.18	30	50	20
April	261	546.012	4.18	30	50	7
May	73	152.716	4.18	30	50	2

The flow rate of the solar system has been assessed against the maximum collector flow requirement and maximum power generated. The assessment falls within the design flow rate of 32 litres per second to achieve the glasshouse heating demand.

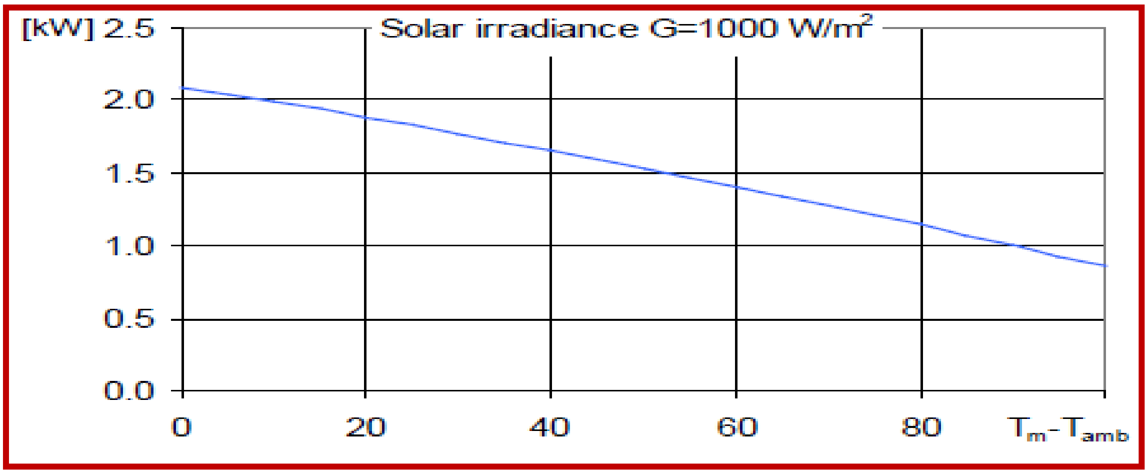


Figure 5:20 Peak power per collector unit (Wpeak)

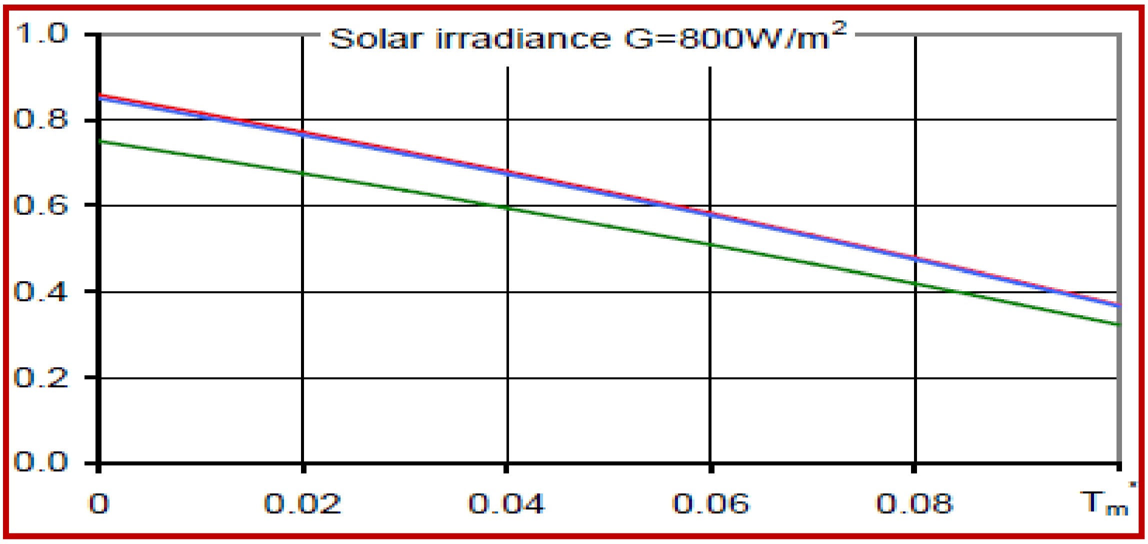


Figure 5:21 Solar collector relative efficiency

The selected solar collector is described as follows:

Model: AZUR 8+ AC 2.8H

Type: Flat plate

Manufacturer: Agena Energies SA

Collector Test date: April 2014

Table 5:40 and 5:41 describe the selected solar collected technical details and specifications respectively.

Table 5:40 Solar collector technical details

Peak power Wpeak (W)	Thermal capacity kJ/K	Relative efficiency η			System heat loss a^1(Wk^{-}m^{-2})			Temperature heat loss a^2 (Wk^{-}m^{-2})		
		Gross	Aperture	Absorber	Gross	Aperture	Absorber	Gross	Aperture	Absorber
2092	6.8	0.750	0.849	0.857	3.64	4.12	4.16	0.008	0.0088	0.0089

Table 5:41 Solar collector specifications

Total length (m)	Total Width (m)	Gross area (m^2)	Aperture area (m^2)	Absorber area (m^2)	Maximum flow rate (l/h)	Nominal flow rate (l/h)	Minimum flow rate (l/h)	Maximum operating pressure (bar)	Stagnant temperature (°C)
1.189	2.347	2.791	**2.263**	2.44	200	40	20	6	195 °C

Summary of chapter 5

The IES software tool for environmental design was used to calculate the heating energy demand of the twenty one zones of the glasshouse as demonstrated in table 5:1. Twenty per cent (20%) heating load was added to the calculated load to account for distribution losses through pipes, PCM thermal energy storage tanks and other components in the heating system.

It is also to account for unforeseen future heating demand and calculation errors that might have occurred during the calculation process.

Building Services Research and Information Association (BSRIA) pipe sizing guide was used to size all the heating pipes to ensure that appropriate velocity and heat transfer is achieved.

The glasshouse heat transfer generally is by natural or free convection. Fluid motion in free convection is due to buoyancy forces within the fluid whilst forced convection is externally imposed. The combined presence of a fluid density gradient and a body force that is proportional to density is called buoyancy and body force in practice is usually gravitational and could be centrifugal or Coriolis force.

Mass density gradient may arise in a fluid in several ways but the most common situation is due to temperature gradient. The density of gases and liquids depends on temperature which generally decreasing (Due to fluid expansion) with increasing temperature. Because the heat transfer in the glasshouse is natural or free convection and not force convection more attention was drawn to the study of free convection.

Heat loss from a heating pipe is by convection to the room air and by radiation exchange with the walls. Hence heat loss from a heating pipe $q = q_{conv} + q_{rad.}$ Where q_{conv} is the convective heat transfer and q_{rad} is the radiant heat transfer. This was the principle from which the heat transfer from the PCM heating pipes to the space was calculated.

Once the heat transfer rate of the PCM heating pipes have been calculated for a unit of pipe length, the pipe length required to emit enough heat energy to meet each zone maximum heating demand was calculated.

As the zones will be heated by PCM heating pipes and it was therefore necessary to calculate the solidification and melting rates of the PCM filled in the heating pipes. This will establish the amount of heat energy that can be absorbed or delivered to the space within a specific time.

The change in energy stored within the PCM heating pipe is due exclusively to the change in latent energy associated with conversion from the solid to liquid state. Heat is transferred to the PCM in the heating pipe by means of conduction through either the inner pipe wall using low temperature hot water of temperature 32 ^{o}C or through the outer surface when the space temperature rises above the PCM phase change temperature.

The thermal energy that could be absorbed by the PCM heating pipes were also calculated using Newton Law of cooling to determine the total heat energy that each zone heating pipes could achieve. This was achieved by establishing the mean daily space temperature above the PCM filled in the heating pipes phase change temperature and the duration that the average temperature occurred.

The sizing of the active system was determined based on the solar gains into the glasshouse and the heat energy that the passive system could absorb and store especially from periods where solar gain in the greenhouse is sufficient to charge the PCM filled heating pipes installed.

The difference between the glasshouse heating energy demand and the solar gains is minimal and just about 5.5 % so it has been established that with careful design and effective solar energy management, most of the glasshouse heating requirement could be met using solar gains. The design of the glasshouse using the IES design software tool demonstrates the periods where the solar gains are not enough to meet the heating requirement.

These periods are from October to March and these are the periods where the stored energy in the PCM storage tanks will be useful. From April to the end of September the solar gains and energy stored in the PCM heating pipes will be the primary energy heating source.

The stored energy in the PCM storage tanks will be used when needed to charge the PCM heating pipes in the zones if the solar gains and the stored energy in the PCM heating pipes are not sufficient to meet the heating requirements.

The waste heat from the CHP (engine jacket cooling medium) and active solar system will be used to charge the PCM storage tanks that contain PCM with phase change temperatures of 44 and 34 ^{o}C.

The heat loss from the storage tanks will be managed by the 20% additional load to the glasshouse heating load. In winter the CHP recovered waste heat will be supplementary heat energy source as the active solar system output will be low within this period. Because the heat energy obtained from the CHP system is a recovered waste heat the system could be classified as zero carbon emission heating system.

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6 Project Appraisal

6.1 Introduction

Project appraisal is carried out to access proposed project's potential success, viability and risks. The appropriateness of a project can be checked by several methods by considering funds available and the economic climate. A good project should be able to pay for itself and maximise profit of shareholders.

Project appraisal check through assumptions made, the preparation method and data used carefully. It is an in-depth review of cost estimates, proposed capital and the work plan,

It is a process to ascertain whether the project is technically sound, financially justified and environmentally acceptable. The main project appraisal techniques are undiscounted and discounted.

Undiscounted techniques includes payback period, profit and loss account.

Discounted techniques takes into account the time value of the money invested which includes net present value (NPV), internal rate of return (IRR), benefit cost ratio (BCR), sensitivity analysis (Taken into consideration uncertainties).

Simple payback is the least used in project appraisal.

6.1.1 Purpose of financial appraisal

The purpose of project financial appraisal is that several projects or investment opportunities may have to be implemented by organisations and capital investment of the project is more than funds available. In such a situation a choice must be made between the projects.

Financial appraisal is the method to assist making choices. The costs and benefits of the projects are combined to produce a measure of financial return. The four objectives that financial appraisal wants to achieve are explained below.

6.1.2 Identify best investments

Financial appraisal will assist identify the investment with best use of the capital available. Energy efficiency projects such as the research project is likely to be considered alongside all other projects requiring capital investment that is why it is important to carry out financial appraisal of the research project.

6.1.3 Optimise benefits from each investment

All the relevant factors will be examined by the appraisal to help identify the projects with the maximum benefit. Detail project appraisal analysis was carried under the research project to identify all the benefits possible.

6.1.4 Risk minimisation

Project success requires investigation to highlight any financial risk and the measures that could be taken to minimise the risk.

6.1.5 Performance analysis

Detail project appraisal is a benchmark for assessing the project performance once implemented.

6.2 Projects appraisal methods

6.2.1 Simple payback

Simple payback period is the evaluation method which simply divides the capital cost by the annual project savings.

6.2.1.1 Advantages of simple payback

It is easy to understand and simple to calculate and normally expressed in years. The project lifetime, inflation and interest rates, and others factors that have effects on the investment are not considered.

6.2.1.2 Disadvantage

It does not take into consideration that future savings will be worthless compared to today's savings. It does not consider savings made after the payback period. Simple payback considers when the time cash flow becomes positive as a risk measure.

6.2.2 Undiscounted financial calculation

This method actually compares initial investment capital with undiscounted cash flow over the project lifetime but this will not be considered under this research study, as it is not detailed enough to highlight projects risk and project life cycle benefits.

They normally expressed in four parameters:

- Gross return on capital: That is total project benefit over its lifetime divided by the capital cost.
- Net return on capital: That is total benefits of the project over its lifetime minus the capital cost divided by the capital cost.
- Gross annual average rate of return: That is the gross return on capital divided by the project life in years.
- Net annual average rate of return: That is the gross return on capital less the capital divided by the capital and project life in years.

6.2.3 Discounted financial calculation

The discounted financial project appraisal compares initial capital investment with discounted cash flow over the lifetime of the project. This method is used widely for financial project appraisal calculation for project investment. This method was used to assess the financial prospects of the research heating system project. Examples are demonstrated in table 6.10.

6.2.3.1 Effects of inflation on investment

Inflation affects investment therefore consideration should always be given to assess the effect it will have on the investment. For example, if the inflation on maintenance is 2.5%, the cost of maintenance for 'n' years will be annual maintenance cost x $(1+ 0.025)^n$, where ‘n’ in the formula is the years raised as power.

For the research project for example, the calculated maintenance savings on replacing boiler heating system with solar collector heating system is £2577 per annum and the savings in maintenance cost in year 3 equals £2577 x $(1+0.025)^3$ = £2816 (Table 6.10). The same principle applies to fuel savings.

For another example, if the energy savings of the proposed glasshouse heating system is £104,640 per annum and fuel inflation is 20 %, the savings in energy cost in year 3 will be £104,640 x $(1+0.20)^3$ = £180,818.

The net savings is the addition of the energy plus maintenance savings. For the above example the net savings in year three is £2816 + £180818 = £183634. If the maintenance cost of the proposed project was an increase, then net savings would have been £180818-2816 = £178,002.

Present Value (PV): The present value is the discounted savings in 'n' years. The present value is discounted by 5 per cent (5%) if the interest rate is 5%. The discounted value takes into consideration the interest paid on the project loan.

For the above example, $PV = (S+M) / (1+ 0.05)^n$ where S and M are project fuel and maintenance annual savings respectively. The interest rate has been assumed an average value of 5 % over the life cycle of the project.

Net present value (NPV): The net present value is the total project cost minus the present value for year on year. Where the figure changes to be positive is the payback period and that is where profits on investment start to be realised.

The method usually used in project appraisal is the discounted cash flow method typically Net Present Value (NPV) or Internal Rate of Return (IRR) normally expressed as a percentage.

In discounting to present day values, both the interest rate and inflation rate need to be considered.

6.2.3.2 Prioritising projects using net present value and capital cost

Where several energy efficiency projects are being considered, a suitable basis for comparison is the Index of Profitability (IOP).

IOP = Discounted net savings sum / Capital cost = PV / Capital cost. This is demonstrated in all the project appraisal calculations of the research study. For example, in table 6.8 below, the IOP is 3 and this ratio is an indication of the project's profitability and the greater the number the more profitable will be the project. It is used in prioritising investment projects.

6.2.3.3 Internal rate of return

The discounted rate at which the net present value (NPV) reduces to zero is the internal rate of return (IRR). IRR represents the rate of return on investment and it is a measure that organisations use when investing. Better investments produce higher internal rate of return.

6.3 Project qualified incentives and tax reliefs

6.3.1 Renewable heat incentive (RHI)

The Non-Domestic Renewable Heat Incentive (RHI) is a government environmental programme that provides financial incentives to encourage generating heat from renewable sources such as solar thermal.

The programme provides a subsidy payable for 20 years in producing heat through renewable source for the non-domestic sector. Renewable heat generators including producers of biomethane for injection based systems are eligible in Great Britain. Ofgem is responsible for implementing and administering the scheme on behalf of the Department of Energy and Climate Change (DECC).

6.3.1.1 The renewable heat incentive scheme purpose

The scheme purpose is to increase heat generated from renewable sources. It is an incentive that the government introduced to encourage heating through renewables. The government wants to bring change in the heating sector dominated by fossil fuel technologies.

The scheme is open to the industrial, commercial, public sector and not-for-profit organisations.

6.3.1.2 Payments to RHI scheme

The payment under the RHI scheme is for 20 years of the project lifetime and they are made on quarterly basis.

The installation will be assessed and a tariff level will be accredited based on the type of technology installed such as solar thermal systems, biomass, heat pump and others. The actual system heat output dictates the payments and starts from the date of accreditation. The current RHI rate is 10.16p / kWh of energy generated by solar collector system.

Table 6.1 demonstrates expected cash value for various solar thermal system installations.

Table 6:1 Renewable heat incentive (RHI) cash value for installing various solar collector sizes and energy delivered per year

No of modules	Yearly energy delivered at 20 oC inlet water temp (kWh)	Yearly energy delivered at 25 oC inlet water temp (kWh)	Yearly energy delivered at 30 oC inlet water temp (kWh)	RHI at 20 oC inlet water temp (£)	RHI at 25 oC inlet water temp (£)	RHI at 30 oC inlet water temp (£)
1	2102.04	2096.90	2091.75	214	213	213
73	153449	153073	152698	15590	15552	15514
261	548633	547290	545947	55741	55605	55468
803	1687939	1683808	1679677	171495	171075	170655

6.3.2 Enhance capital allowance (ECA)

Under the enhanced capital allowance (ECA), twenty per cent (20%) of the capital cost of the item purchased can be claimed.

The enhance capital allowance support businesses to invest in energy-saving plant or machinery that might have been too expensive for them if not supported.

Businesses can claim 100% cost of the plant or machinery against taxable profits in a single tax year for the first year allowance. This suggests that the new plant or machinery cost can be written off against the business's taxable profits in the financial year that the plant or the machinery was bought.

Businesses that pay income or corporation tax are eligible to claim 100% capital allowance of the first year that the product was purchased provided the product is listed in the energy technology list (ETL) at the time of purchase.

The Carbon Trust manages ETL on behalf of the Department of Energy and Climate Change (DECC). DECC carryout annual reviews of technologies and products that qualify to be included in the energy technology list.

The current products included in the list are:

- Air to air energy recovery
- Automatic monitoring and targeting (AMT) equipment
- Boiler equipment
- Combined heat and power (CHP)
- Compressed air equipment
- Heat pumps
- Heating, ventilation and air conditioning (HVAC) equipment
- High speed hand air dryers
- Lighting
- Motors and drives
- Pipework insulation
- Refrigeration equipment
- Solar thermal systems
- Uninterruptible power supplies
- Warm air and radiant heaters

6.3.2.1 Solar thermal systems eligibility criteria

The solar thermal system was added to the energy technology product list in 2002 and revised it in 2014.

6.3.2.1.1 Solar Thermal Systems and Collectors

Solar thermal systems and products are designed to capture solar radiation and convert it to useful energy to heat water.

Under the ECA Scheme two categories of solar products qualify:

- Individual solar collectors that could be assembled by an installer to build solar thermal system.
- Solar thermal system that is complete, ready to install or fixed configuration

Eligibility criteria

The product must be in the criteria below to be eligible:

- Solar thermal systems and components or Solar collectors and general requirements that comply with the requirements of BS EN 12975-1:2006
- Solar thermal system that complies with the requirements of BS EN 12976-1:2006 and sold as complete, ready to install or fixed configuration

A solar thermal system may include the following components:

- One or more solar collectors.
- One or more appropriately sized storage vessels (where required).
- The pipework and valves forming the connection loop between the solar collector(s) and storage vessel(s), including any non-return valves, control valves, pressure relief valves, air bleed valves etc., as required for the effective operation of the product.
- Circulation pumps (Where required).
- Any controls or sensors (And their associated power supplies) needed to:
  - Stop circulation when the yield is low
  - Ensure compliance with Health & Safety Executive (HSE) requirements

6.3.3 Climate change levy and rates

6.3.3.1 Introduction

A tax on energy delivered to non-domestic users in the United Kingdom (UK) is termed as Climate Change Levy (CCL).

The CCL is an energy efficiency incentive provided by the UK government to reduce carbon emissions in the UK. There have been ongoing calls however, to replace it with a proper carbon tax.

It was introduced under the Finance Act 2000 on 1 April 2001 with the forecast to reduce 2.5 million tonnes of CO_2 emissions annually by 2010. The levy applies to most UK energy users with the exception of those in the domestic and transport sectors.

Nuclear electricity generation causes no direct carbon emissions but still taxed. Renewables and cogeneration schemes which used to be exempted from the tax will no longer be and it is anticipated to raise £450m per year according to July 2015 budget.

This will apply to renewable generated electricity supplies to organisations that are eligible to pay climate change levy but do not currently pay because the electricity supply to them are generated from renewable source. It is currently in transitional phase till 31 March 2018.

6.3.3.2 Main rates and eligible business

Climate change levy are paid at the main rate on:

- Electricity
- Gas
- Solid fuels - like coal, lignite, coke and petroleum coke

The main rates of CCL are paid if your business is in one of the following sectors:

- Industrial
- Commercial
- Agricultural
- Public services

The main rates of CCL are not paid if the business is in one of the following sectors:

- Business that uses small amounts of energy
- Domestic energy user
- Charity engaged in non-commercial activities

6.3.3.3 Fuels that are exempted

Electricity, gas and solid fuels are normally exempted from CCL main rates if:

- The fuel will not be used in the UK
- They are supplied to or from CHP quality assurance (CHPQA)
- The electricity is generated from renewable sources
- They are used to produce 2 MW of electricity or greater capacity in a generating station
- They are used in certain forms of transport

6.4 Proposed project cost and savings analysis

6.4.1 Introduction

Solar thermal system can last for a very long time after installation and has less few moving parts. The technology has been around for some time and operates to a very high standard. The service requirements on solar thermal systems are minimal compared to central heating systems using boilers to generate thermal energy. In solar thermal system test on the solar circuit, electrical circuits, collector fixings, pipework insulation and examination of the control parameters is required.

The solar thermal system would last in excess of 25 years and could still function over 30 years. The economic life cycle costing will consider twenty and twenty five years of operation. The renewable heat incentive is paid by the UK government for twenty (20) years of a solar collector system. The twenty five years is the recommended period that the manufacturer proposes that the collector's efficiency will still be effective to generate enough thermal energy.

The current heating system uses Purewell atmospheric boilers to generate low temperature hot water (LTHW) to heat the glasshouse.

The manufacturer Hamworthy gives economic life cycle cost analysis of 3 X Purewell 95 kW atmospheric boilers in Figure 6:1 below. The operational cost of the plant is £19,326 over the life time of the plant which is twenty-five years. This suggests yearly operational cost of £773 for the three boilers. This operational cost includes maintenance, parts replacement and others excluding gas energy consumption.

The heating load of the glasshouse from the IES software calculation is 914 kW which means that 10 X Purewell 95 kW atmospheric boilers will be needed to meet the glasshouse heating load. Therefore the operational cost for the ten boilers will proportionately be £2577 per annum. This money will be saved with the solar thermal heating system as there will be no boiler to service.

The solar collector maintenance is almost negligible as the dirt collected on the solar collector will be washed away by rainfall and no maintenance will be required.

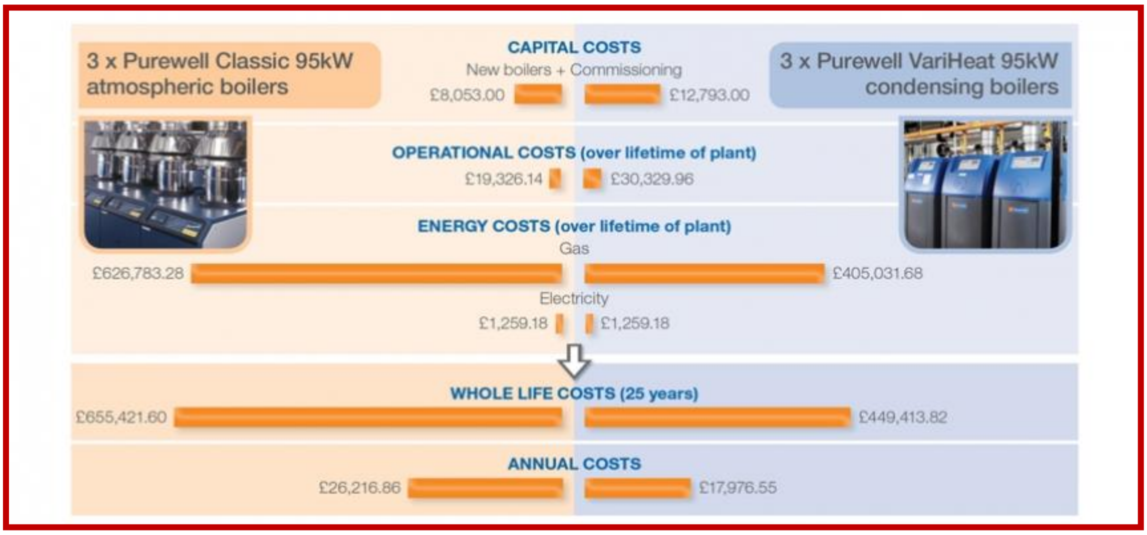


Figure 6:1 3 X Purewell 95 kW atmospheric boilers economic life cycle cost

The current heating system serving the glasshouse energy consumption was calculated to be 2421 MWh as demonstrated in table 4.7 in section 4.3.1.1.

6.4.2 Project material and installation cost

The project design is based on two scenarios or strategies. The thermal energy generated by solar collector system is weather dependant therefore the design has to take into account situations or periods where the solar radiation falling on the solar collectors will be insufficient to generate enough thermal energy required to heat the glasshouse especially in summer periods where the CHP plant is shut off.

The first design strategy is to design a system that the PCM storage tanks have the capacity to store two times the daily maximum heating energy demand of the glasshouse (N+1). Thus the system will have sufficient heat energy to heat the glasshouse for two days without any thermal energy input into the heating system.

The second strategy is to store enough thermal energy in the PCM storage thanks to meet three days maximum heating demand without thermal energy input from any source.

This means that the heating system will be capable to heat the glasshouse for three days if insufficient or no thermal energy is generated by the solar collector system.

The calculated daily energy demand includes 20% excess requirement to deal with unexpected increase in heating demand and losses through pipes and tanks.

Table 6:2 below illustrates monthly, daily and up to five days heating energy requirements from January to December of the design year. The sizing of the PCM thermal storage tank is based on the month and the day where maximum heating energy demand is required which is January in this case.

The daily heating energy demand is assessed without taken into consideration the heat energy that will be absorbed by the PCM heating pipes. It is anticipated that as the solar radiation becomes insufficient, the absorbed solar energy by the PCM heating pipes will also be minimal.

The heating energy that can be stored by 100 m^3 PCM storage tank is 4.5 to 5 MWh (Source: PCM products Limited) therefore 40 MWh of heating demand requires eight (8) 100 m^3 PCM storage tanks. Figure 6:2 shows the storage tank that will be used for the research project and table 6:2 below shows monthly and daily maximum heating energy requirements up to five days.

To meet two and three days heating energy demand requires 27 and 40 MWh of thermal energy storage thanks respectively. This suggests that 6 and 8 100 m^3 PCM storage tanks will be needed to store two and three days heating energy demand respectively.

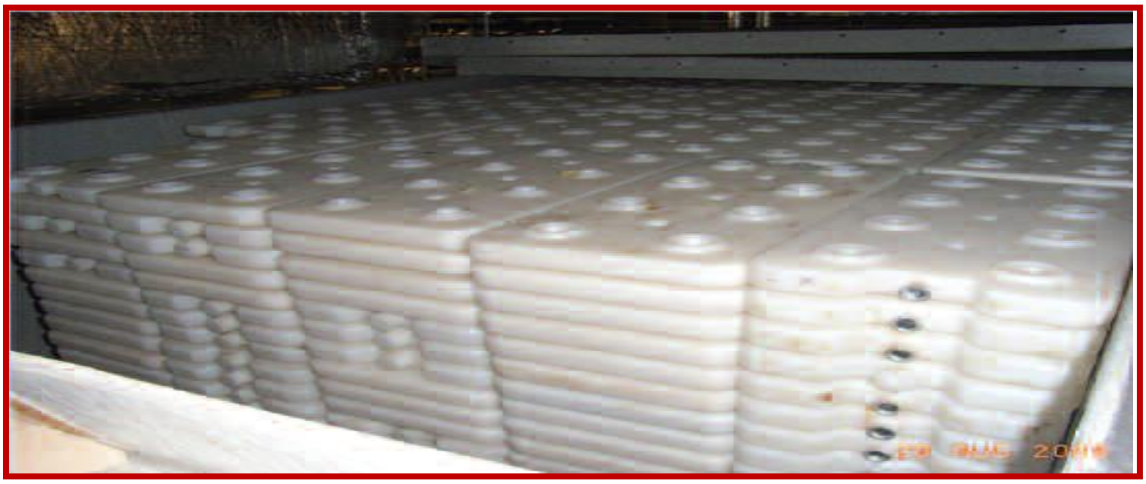


Figure 6:2 100 m^3 storage tank capable of storing 4.5-5 MWh of thermal energy

Source: PCM Products Ltd

Table 6:2 Heating energy demand without considering PCM heating pipes daily absorbed solar energy trapped in the space

Months	Monthly heating demand (MWh)	Daily heating demand (MWh)	Two days heating demand (MWh)	Three days heating demand (MWh)	Four days heating demand (MWh)	Five days daily heating demand (MWh)
Jan 01-31	411.7	13	**27**	**40**	53	66
Feb 01-28	362.8	12	23	35	47	59
Mar 01-31	302.4	10	20	29	39	49
Apr 01-30	179.0	6	12	17	23	29
May 01-31	100.3	3	6	10	13	16
Jun 01-30	57.3	2	4	6	7	9
Jul 01-31	58.4	2	4	6	8	9
Aug 01-31	54.3	2	4	5	7	9
Sep 01-30	107.1	3	7	10	14	17
Oct 01-31	212.9	7	14	21	27	34
Nov 01-30	248.7	8	16	24	32	40
Dec 01-31	402.1	13	26	39	52	65

6.4.2.1 Project estimation cost

The project cost estimate was based on the material requirements, installation and commissioning cost. Most of the materials cost were obtained from the manufacturers and those that could not be obtained from the manufacturers were obtained from the forty-sixth (46^{th}) edition of SPON'S Mechanical and Electrical Services Price Book 2015 which continues to cover the widest range and depth of engineering services.

This book assists the building services industry to estimate or calculate project cost especially when tendering for a project.

The standard rates may need adjustments by considering the time, location, local conditions, site constraints and any other factors likely to affect the costs of the project. The research project cost is based on London rates as the project under consideration is located in London, Kew Gardens.

Table 6:3 and table 6:4 show the research project cost estimation of the glasshouse heating system that could store two and three days thermal energy to meet maximum heating demand respectively.

Table 6:2 above demonstrates size of PCM thermal storage tanks that will be able to store enough thermal energy to meet the glasshouse heating demand for two and three days without any thermal energy input into the heating system.

This does not mean that daily heat energy generated by the solar thermal and PCM heating pipes will not be stored. The increase in thermal energy storage capacity is a precautionary measure to ensure that sufficient thermal energy is available to heat the glasshouse as demanded at all times.

The cost estimate for the heating system that could store two and three days thermal energy to meet the glasshouse maximum heating demand is calculated to be £3,944,783 and £4,739,783 respectively.

Table 6:3 and table 6:4 below demonstrate the cost calculation including source of prices. The installation cost of the project has been assumed to be 30% of the material cost (Source: Spon's Mechanical and Electrical Price Book 2015).Where the installation cost is zero means that the material requires no installation or the installation cost includes the material cost. For example, the insulation pipe and supporting brackets prices include installation cost.

Table 6:3 Project costing for two days glasshouse heating energy demand storage system

Item	Material cost/unit (£)	Number required	Installation cost (£)	Total cost including inst (£)	Cost source
Insulated Steel pipe	72.33	300	0	21699	Spon's Mechanical and Electrical services Price Book 2015
PCM heating pipe	103	6459	199583.1	864860.1	PCM Products Ltd (www.pcmproducts.net)
PCM solution cost/kg	3	15617	0	46851	PCM Products Ltd (www.pcmproducts.net)
PCM cost/100m3 tank	300,000	6		1800000	PCM Products Ltd (www.pcmproducts.net)
PCM storage tank cost	75000	6	135000	585000	PCM Products Ltd (www.pcmproducts.net)
Solar collector cost	200	803	48180	208780	SOLTOP Schuppisser AG (www.soltop.ch)
Heat exchanger cost	1916	1	574.8	2490.8	STOKVIS ENERGY SYSTEMS (www.stokvisboilers.com)
Shut off valves Valves	18.99	60	341.82	1481.22	Spon's Mechanical and Electrical services Price Book 2015
Flanges	77.52	30	697.68	3023.28	Spon's Mechanical and Electrical services Price Book 2015
Solar collector Pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Heat exchanger circulation pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Heating system hot water circultion pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Bypass valve	44.99	6	80.982	350.922	Screwfix (http://www.screwfix.com/)
Motorised valves	103	21	648.9	2811.9	Spon's Mechanical and Electrical services Price Book 2015
Control sensors	25	40	300	1300	Plumb Center (http://www.plumbcenter.co.uk/)
Pressure relief valve	43	5	64.5	279.5	Unvented Component Europe (http://www.unventedcomponentseurope.com/)
Air bleed valve	41	4	49.2	213.2	https://www.google.co.uk/search?q=air+bleed+off+valves+prices&biw
Supporting brackets	15.47	25836	0	399682.92	Spon's Mechanical and Electrical services Price Book 2015
Total project cost				**3944783**	

Table 6:4 Project costing for three days glasshouse heating energy demand storage system

Item	Material cost/unit (£)	Number required	Installation cost (£)	Total cost including inst (£)	Cost source
Insulated Steel pipe	72.33	300	0	21699	Spon's Mechanical and Electrical services Price Book 2015
PCM heating pipe	103	6459	199583.1	864860.1	PCM Products Ltd (www.pcmproducts.net)
PCM solution cost/kg	3	15617	0	46851	PCM Products Ltd (www.pcmproducts.net)
PCM cost/100m3 tank	300,000	8		2400000	PCM Products Ltd (www.pcmproducts.net)
PCM storage tank cost	75000	8	180000	780000	PCM Products Ltd (www.pcmproducts.net)
Solar collector cost	200	803	48180	208780	SOLTOP Schuppisser AG (www.soltop.ch)
Heat exchanger cost	1916	1	574.8	2490.8	STOKVIS ENERGY SYSTEMS (www.stokvisboilers.com)
Shut off valves Valves	18.99	60	341.82	1481.22	Spon's Mechanical and Electrical services Price Book 2015
Flanges	77.52	30	697.68	3023.28	Spon's Mechanical and Electrical services Price Book 2015
Solar collector Pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Heat exchanger circulation pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Heating system hot water circultion pump	1528	1	458.4	1986.4	Anglian Pump Services (http://www.anglianpumping.com/)
Bypass valve	44.99	6	80.982	350.922	Screwfix (http://www.screwfix.com/)
Motorised valves	103	21	648.9	2811.9	Spon's Mechanical and Electrical services Price Book 2015
Control sensors	25	40	300	1300	Plumb Center (http://www.plumbcenter.co.uk/)
Pressure relief valve	43	5	64.5	279.5	Unvented Component Europe (http://www.unventedcomponentseurope.com/)
Air bleed valve	41	4	49.2	213.2	https://www.google.co.uk/search?q=air+bleed+off+valves+prices&biw
Supporting brackets	15.47	25836	0	399682.92	Spon's Mechanical and Electrical services Price Book 2015
Total project cost				**4739783**	

6.4.2.2 The proposed heating system annul energy cost savings

Table 6:5 below shows how much the proposed heating system project could save in a year. The total energy consumption of the current system from calculation is 2,421,000 kWh and because the proposed heating system will generate this heat energy from zero carbon emission sources no climate change levy will be paid and that will be a significant savings. Again no payment will be made for the energy consumed as it comes from free sources.

The CO_2 saved for using zero carbon emission thermal energy could be sold and the current sales price is £16 per tonne of CO_2 (£16 / tonne CO2) under the current carbon reduction commitment (CRC) carbon credits purchase.

The overall energy cost savings for a year will be £104,640 as shown in table 6:5 below. This energy cost savings will increase year by year due to increase in fuel cost but for the purpose of this research project the savings have been assumed constant throughout the project life.

There are other charges such as distribution, commodity, transmission and transportation charges that will be saved for using free source of energy but these charges are excluded from the calculated energy cost savings in table 6:5.

Table 6:5 Energy cost savings per annum calculation

Glasshouse space conditioning energy demand per annum (kWh)	Climate change levy per kWh (£)	Gas energy tariff per kWh (£)	CO2 savings (£/tonne CO2)	Total energy cost savings (£)
2421000	0.00549	0.03	16	104640

6.5 Project cost and benefits savings

The project economic life cycle costing and benefits savings are calculated using the discounted method of project appraisal. The yearly thermal energy savings assessment were based on climate change levy tax exemption for using thermal energy with zero carbon dioxide (CO_2) emissions, using free solar thermal energy and CHP waste heat recovery energy and sale of CO_2 emissions saved.

6.5.1 Systems design thoughts or reasoning

In order to ensure that the research project is reliable, viable, cost effective and profitable to encourage investment, different system design scenarios, strategies and options were considered.

The design criteria was based on solar collector requirements, project cost, benefits and the month from which the active and passive systems are capable of providing sufficient heating energy to meet the glasshouse heating demand. A project that is technically, economically and environmentally sound.

As it has already been explained above, the site has CHP plant that runs in winter and shut down in summer. The CHP engine jacket cooling water is currently wasted and the research heating system design will maximise the use of this waste energy to minimise the requirements of solar collectors to meet the heating demand of the glasshouse especially in January and February where the heating demand reaches maximum peak.

As a result three different systems have been proposed based on the glasshouse heating requirement in March, April and in May. The solar collector size to meet January and February heating demand without any auxiliary heating energy source such as the CHP waste heat energy may not be technically and economically viable as large size of solar collectors will be needed to provide heating energy for the glasshouse.

To minimise the environmental impact of the heating system and at the same time maximise profit the following have been proposed:

System 1: Design to meet glasshouse heating requirements from March to the winter season when the heating energy demand will be significant to make use of the CHP waste heat energy.

With this system the additional heating energy requirement to meet January and February heating demand will not be significant to impact the environment even if boilers are to be used to support the active and passive systems. This is an important consideration as the proposed glasshouse heating system is aimed to achieve zero carbon emission. No fossil fuel heating energy will be used unless the unforeseeable happens for example, if the CHP becomes faulty or shut down for maintenance which is normally carried out in summer where thermal energy generated by the solar collectors and heat absorbed by the PCM heating pipes is sufficient to heat the glasshouse.

With this system eight hundred and three (803) solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to be supplemented by the CHP waste heat to meet the heating demand.

System 2: Design to meet glasshouse heating requirements from April to winter season when the heating demand is significantly increased. With this system two hundred and sixty one (261) solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the CHP waste heat.

System 3: Design to meet glasshouse heating requirements from May to winter season when the heating demand is significantly increased. With this system seventy three (73) solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the CHP waste heat.

6.5.2 Discounted project appraisal method explanation

The calculation method considers the project cost, enhance capital allowance (ECA) which is deducted from the project calculated capital cost as this money will be paid by the government and will not be part of the borrowed capital. The method assesses if the proposed project will increase or decrease maintenance cost and for this project the maintenance cost will be reduced by £2,577 as explained above due to boiler maintenance and repairs.

Renewable heat incentive (RHI) as explained earlier is a payment by the UK government for using solar collector system to generate thermal energy. The payment will continue for twenty years and the current rate of payment is 10.16 pence per kilowatt hour (p / kWh) of thermal energy generated using solar collector system.

The following have also been assumed in the calculation:

- Project loan interest rate 5% (Averaged throughout the project life)
- Maintenance inflation rate 3% (Averaged throughout the project life)
- Fuel inflation rate 20% (Averaged throughout the project life)

The project payback period and life cycle benefits for each proposed design project has been assessed, analysed and demonstrated using tables and graphs. The point where the net present value (NPV) turns to be zero is the payback period and the internal rate of return (IRR).

The payback period and IRR is indicated with red arrow in all the project appraisal graphs displaying savings, PV, NPV and IRR. Figure 6:4 is an example of such graphs. IRR is the point where investment profits starts to accrue. It is normally calculated as a percentage.

For each of the proposed design projects, two separate assessments and analysis have been carried out for economic life cycle of twenty and twenty five years. The twenty years economic life cycle assessment and analysis includes renewable heat incentive (RHI) as this incentive is paid for twenty years by the UK government.

The twenty five years economic life cycle assessment and analysis includes renewable heat incentive (RHI) payment up to 20 years and the rest of the five years do not include RHI benefits. The solar thermal project life cycle goes beyond 25 years but up to twenty five years economic life cycle savings and benefits have been assessed to establish that the project is economical and viable to implement.

The index of profitability (IOP) was also calculated to ascertain that the project will generate more profit and benefits. This will assist to prioritise project or projects that need to be implemented first.

To obtain the Net present Value (NPV), the discounted present value (PV) is subtracted from the capital cost at the end of every year as demonstrated in

Table 6:6 below and where the NPV equals zero is the payback period. This is illustrated in figure 6:4 of the proposed designed project to meet March heating demand using 803 solar collectors.

The net present value is zero at 9.5 years and that is the payback period of the project. This again is the point where the net present value line crosses the x-axis.

The energy savings in kilowatt hours (kWh) is the annual fuel cost savings under the annual savings cash value in

Table 6:6 divided by the fuel unit cost (p / kWh) and the CO_2 emission savings is the energy savings multiply by the CO_2 conversion factor of the fuel used.

These are demonstrated in

Table 6:6 and all the discounted project appraisal calculations. The bottom of the table displays the total energy cost savings at the end of the project life, total CO_2 savings, total climate change levy savings, renewable heat incentive savings, enhance capital allowance, project payback period and index of profitability.

6.5.3 Systems appraisal results

The three proposed systems as explained above are designed to meet March, April and May glasshouse heating demand and the financial appraisal results are demonstrated in detail in tables and graphical forms.

Under each system two economic life cycle assessments and analysis are carried out which includes 20 and 25 years economic life cycle costing analysis. The systems are explained below.

System 1: Design to meet glasshouse heating requirements from March to the winter season when the CHP system waste heat energy will supplement the heat energy generated by the active and passive systems. With this system, 803 solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the CHP waste heat.

Under this system, 20 years economic life cycle cost analysis is displayed in

Table 6:6 which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:3 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance. Figure 6:4 below displays the project payback period, cost savings, PV, NPV and IRR.

Table 6:6 March design project appraisal of 20 years economic life cycle cost analysis (system 1 of 803 solar collectors)

Capital cost of project (£)			4,737,990										
Capital cost of project less ECA (£)			3790392										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			171,495										
Years	Maintennace Cash value	Annual savings cash value	Net savings cash value	PV	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,790,392				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	171495	-3496781				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	171495	-3186133				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	171495	-2856008				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	171495	-2503616				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	171495	-2125767				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	171495	-1718819				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	171495	-1278605				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	171495	-800369				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	171495	-278669				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	171495	292708				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	171495	920868				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	171495	1613927				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	171495	2381165				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	171495	3233183				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	171495	4182101				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	171495	5241766				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	171495	6428006				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	171495	7758909				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	171495	9255146				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	171495	10940342				0.03	133721549	0.186	24872208
Total savings at the end of project life				**11300834**	**3429900.0**						**784889291**		**145989408**
Total energy cost savings at the end of project life						£	**10,940,342**						
Total CO2 emission cost savings (£16/tCO2)						£	**2335831**						
Total climate change (CCL) saved						£	**1,530,534**						
Renewable heat incentive						£	**3,429,900**						
Enhance capital allowance (ECA)						£	**947,598**						
Project payback period							**9.5yrs**						
Index of profitability (IOP)							**3**						

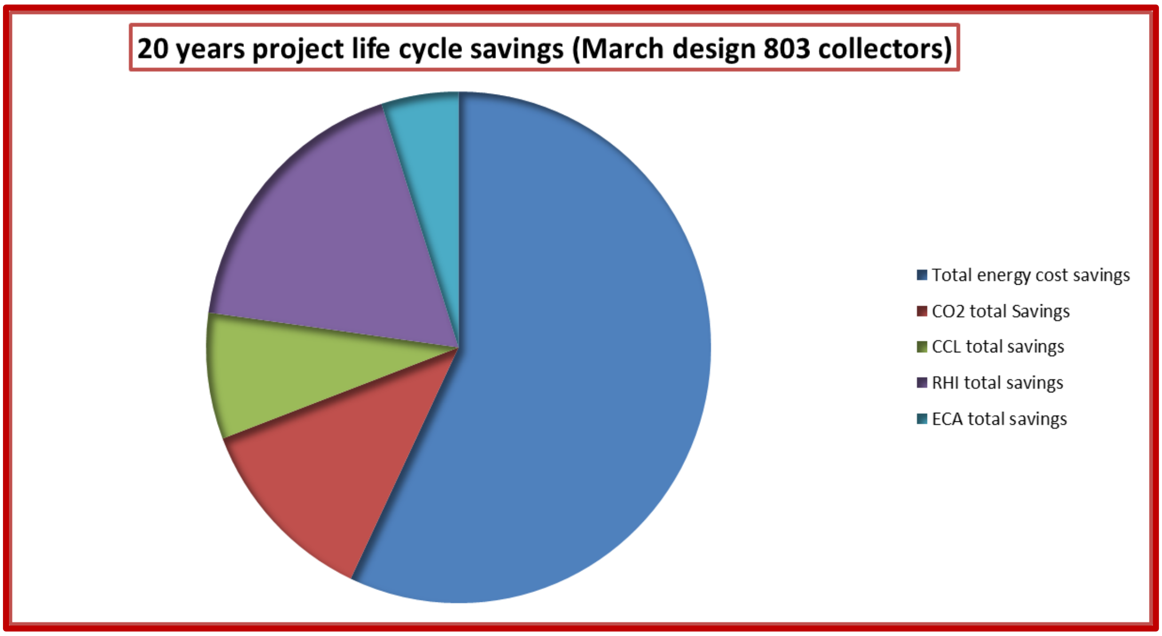


Figure 6:3 Twenty years project economic life cycle savings (March heating demand design of 803 collectors)

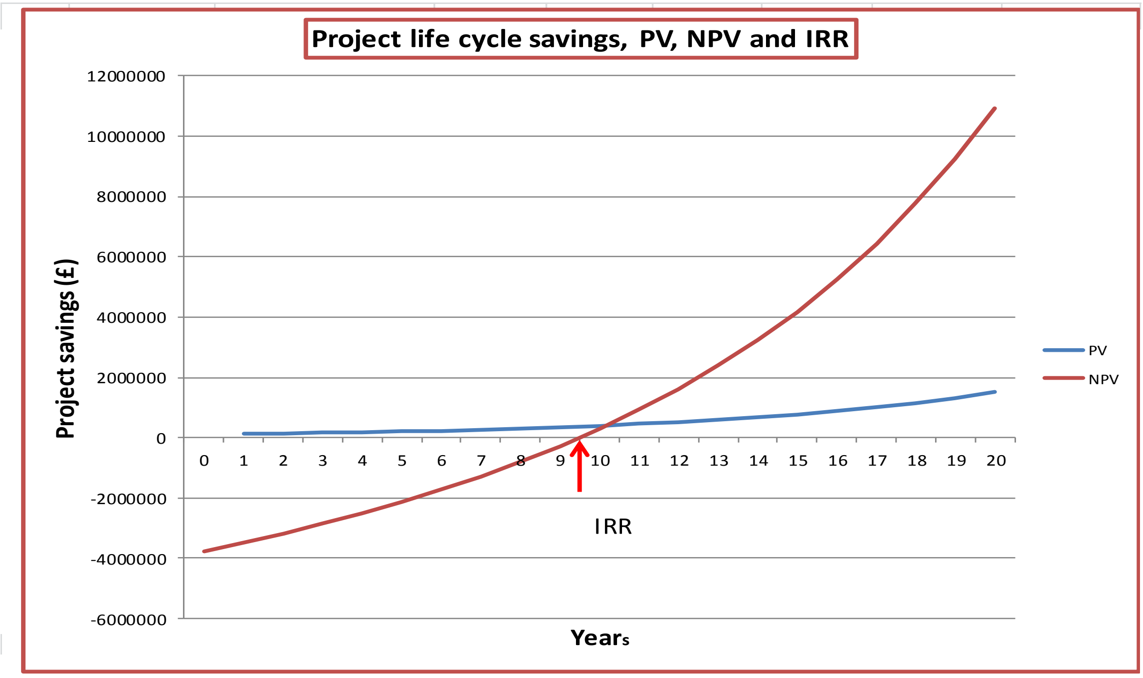


Figure 6:4 Twenty (20) years project life cycle savings, PV, NPV and IRR (March design of 803 collectors)

System 1: Twenty five years project economic life cycle cost analysis (March heating demand design of 803 collectors).

Twenty five years economic life cycle cost analysis is displayed in table 6:7 below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:5 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance.

Figure 6:6 below displays the project payback period, cost savings, PV, NPV and IRR.

Table 6:7 March design project appraisal of 25 years economic life cycle cost analysis (System 1 of 803 solar collectors)

Capital cost of project (£)			4,737,990										
Capital cost of project less ECA (£)			3790392										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			171,495										
Years	maintennace Cash value	Annual savings cash value	Net savings cash value	PV (£)	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,790,392				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	171495	-3496781				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	171495	-3186133				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	171495	-2856008				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	171495	-2503616				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	171495	-2125767				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	171495	-1718819				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	171495	-1278605				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	171495	-800369				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	171495	-278669				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	171495	292708				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	171495	920868				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	171495	1613927				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	171495	2381165				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	171495	3233183				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	171495	4182101				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	171495	5241766				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	171495	6428006				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	171495	7758909				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	171495	9255146				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	171495	10940342				0.03	133721549	0.186	24872208
21	2654	4813976	4816630	1728893		12669235				0.03	160465858	0.186	29846650
22	2734	5776771	5779505	1975723		14644958				0.03	192559030	0.186	35815980
23	2816	6932125	6934941	2257818		16902776				0.03	231070836	0.186	42979175
24	2900	8318550	8321451	2580215		19482991				0.03	277285003	0.186	51575011
25	2987	9982260	9985248	2948671		22431662				0.03	332742004	0.186	61890013
Total savings at the end of project life				**22792154**	**3429900**						**1979012022**		**368096236**
Total energy cost savings at the end of project life						**£**	**22,431,662**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**5889540**						
Total climate change (CCL) saved						**£**	**3,859,073**						
Renewable heat incentive						**£**	**3,429,900**						
Enhance capital allowance (ECA)						**£**	**947,598**						
Project payback period							**9.5 yrs**						
Index of profitability (IOP)							**6**						

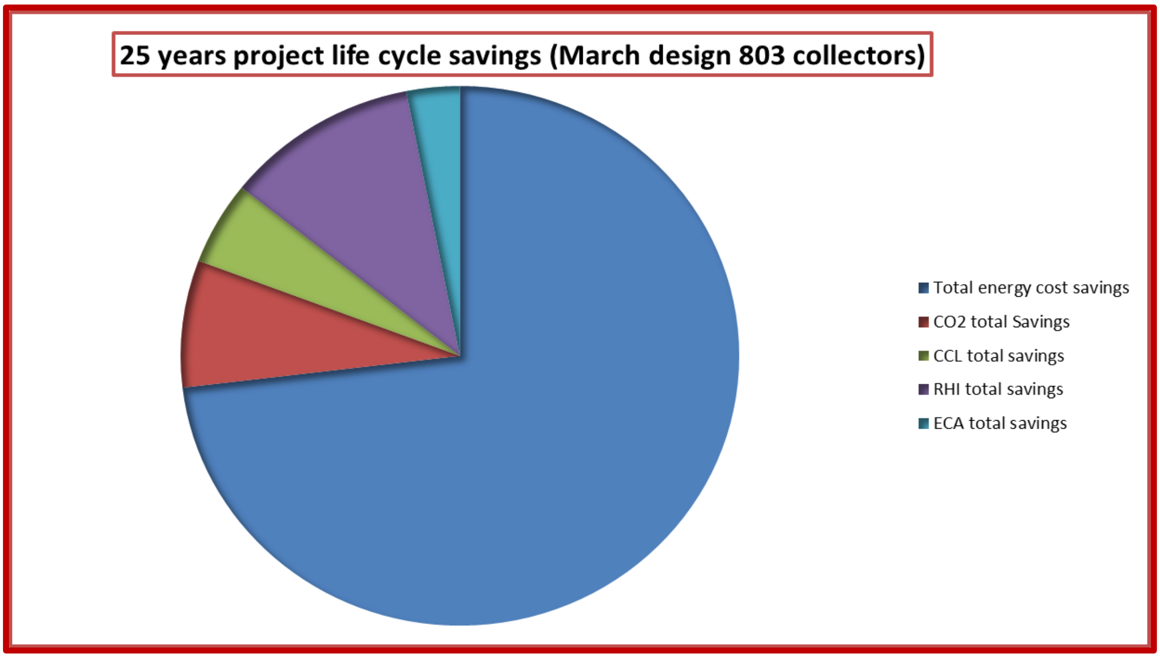


Figure 6:5 Twenty five years project economic life cycle savings (March heating demand design of 803 solar collectors)

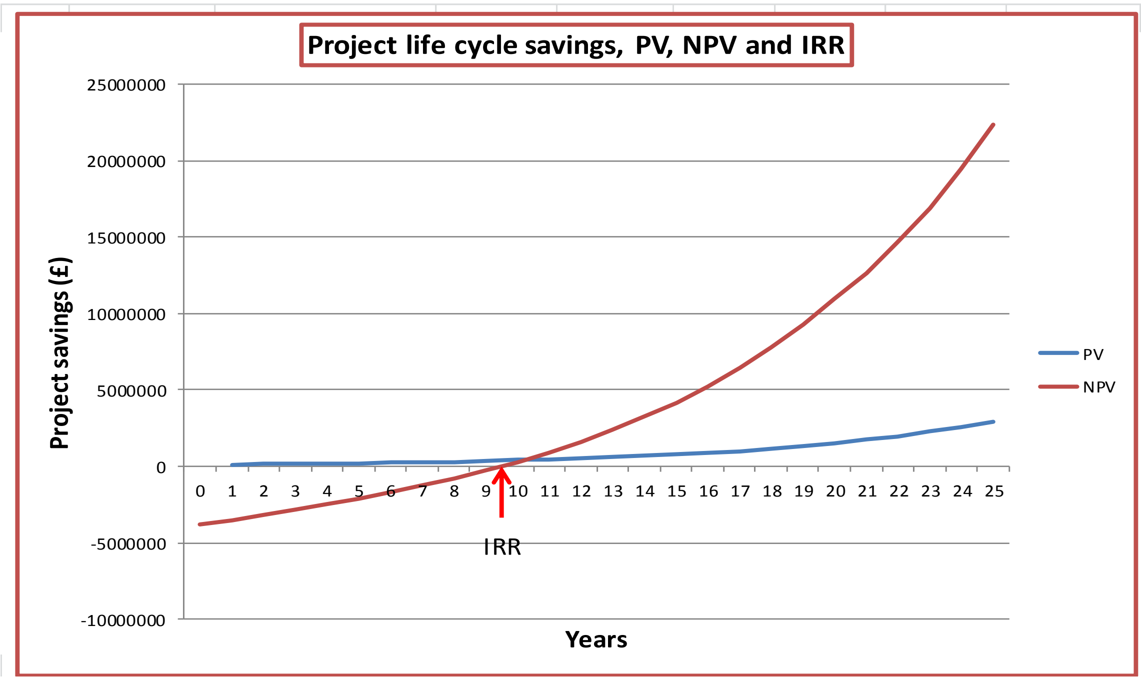


Figure 6:6 Twenty five (25) years project life cycle savings, PV, NPV and IRR (March design of 803 collectors)

System 2: Design to meet glasshouse heating requirements from April to winter season when the heating demand is significantly increased. With this system, 261 solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the CHP waste heat. Twenty years project economic life cycle cost analysis (April heating demand design of 261 collectors)

Twenty years economic life cycle cost analysis is displayed in Table 6:8 below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:7 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance. Figure 6:8 below displays the project payback period, savings, PV, NPV and IRR.

Table 6:8 April design project appraisal of 20 years economic life cycle cost analysis (System 2 of 261 solar collectors)

Capital cost of project (£)			4,716,975										
Capital cost of project less ECA (£)			3773580										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost saving			104,640										
Renewable heat incentive			55,741										
Years	Maintennace Cash value	Annual savings cash value	Net savings cash value	PV	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,773,580				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	55741	-3595723				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	55741	-3400829				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	55741	-3186458				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	55741	-2949820				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	55741	-2687725				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	55741	-2396531				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	55741	-2072071				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	55741	-1709589				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	55741	-1303643				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	55741	-848020				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	55741	-335614				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	55741	241691				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	55741	893175				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	55741	1629439				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	55741	2462603				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	55741	3406514				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	55741	4477000				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	55741	5692149				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	55741	7072632				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	55741	8642074				0.03	133721549	0.186	24872208
Total savings at the end of project life				**11300834**	**1114820**						**784889291**		**145989408**
Total energy cost savings at the end of project life						**£**	**8,642,074**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**2335831**						
Total climate change (CCL) saved						**£**	**1,530,534**						
Renewable heat incentive						**£**	**1,114,820**						
Enhance capital allowance (ECA)						**£**	**943,395**						
Project payback period							**11.5 yrs**						
Index of profitability (IOP)							**3**						

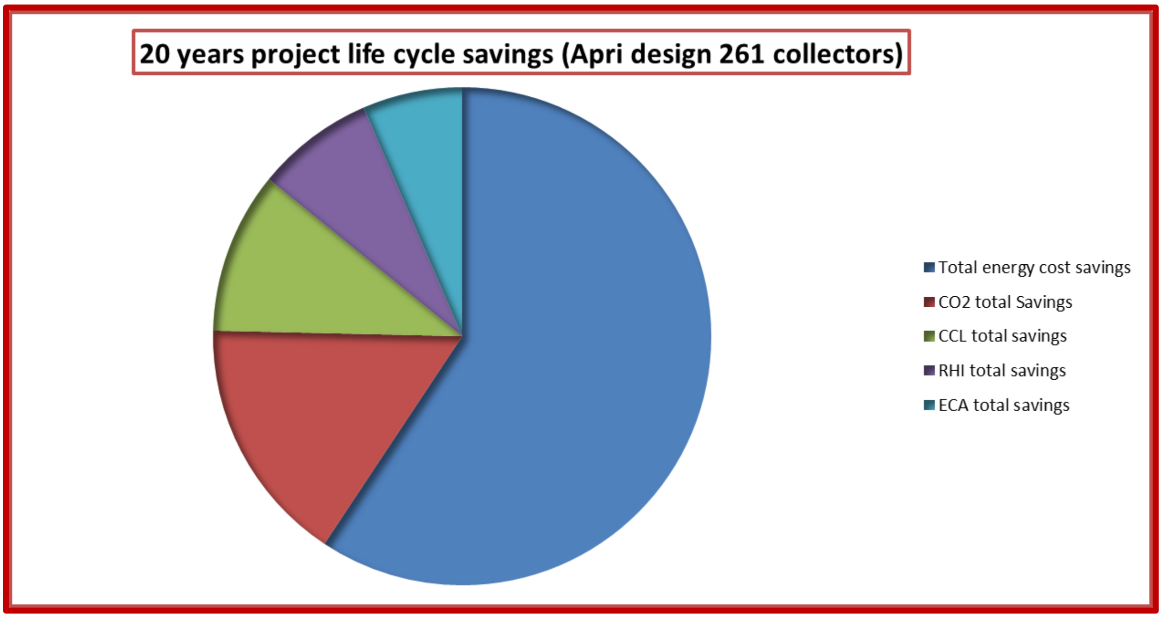


Figure 6:7 Twenty (20) years project life cycle savings (April heating demand design of 261 solar collectors)

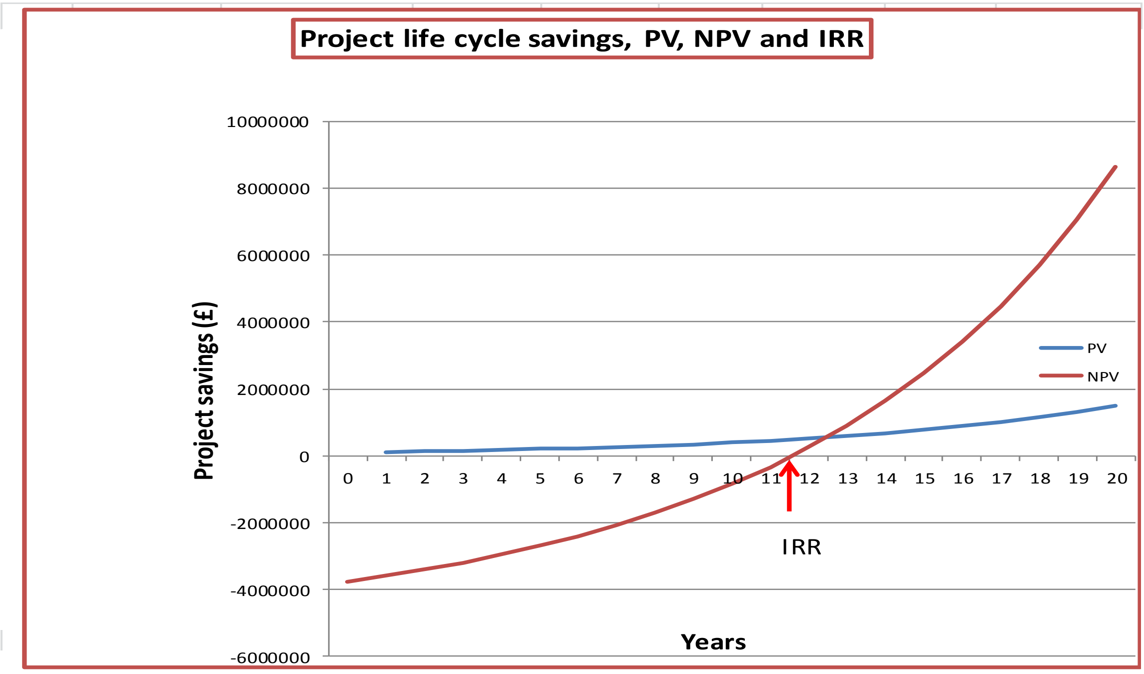


Figure 6:8 Twenty years project economic life cycle savings, PV, NPV and IRR (April design of 261 solar collectors)

System 2: Twenty five years project economic life cycle cost analysis (April heating demand design of 261 collectors)

Twenty five years life cycle cost analysis is displayed in Table 6:9 below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:9 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance. Figure 6:10 below displays the project payback period, savings, PV, NPV and IRR.

Table 6:9 April design project appraisal of 25 years economic life cycle analysis (System 2 of 261 solar collectors)

Capital cost of project (£)			4,716,975										
Capital cost of project less ECA (£)			3773580										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			55,741										
Years	maintennace Cash value	Annual savings cash value	44592.8	PV (£)	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	4,716,978	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,773,580				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	55741	-3595723				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	55741	-3400829				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	55741	-3186458				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	55741	-2949820				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	55741	-2687725				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	55741	-2396531				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	55741	-2072071				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	55741	-1709589				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	55741	-1303643				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	55741	-848020				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	55741	-335614				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	55741	241691				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	55741	893175				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	55741	1629439				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	55741	2462603				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	55741	3406514				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	55741	4477000				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	55741	5692149				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	55741	7072632				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	55741	8642074				0.03	133721549	0.186	24872208
21	2654	4813976	4816630	1728893		10370967				0.03	160465858	0.186	29846650
22	2734	5776771	5779505	1975723		12346690				0.03	192559030	0.186	35815980
23	2816	6932125	6934941	2257818		14604508				0.03	231070836	0.186	42979175
24	2900	8318550	8321451	2580215		17184723				0.03	277285003	0.186	51575011
25	2987	9982260	9985248	2948671		20133394				0.03	332742004	0.186	61890013
Total savings at the end of project life				**22792154**	**1114820**						**1979012022**		**368096236**
Total energy cost savings at the end of project life						**£**	**20,133,394**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**5889540**						
Total climate change (CCL) saved						**£**	**3,859,073**						
Renewable heat incentive						**£**	**1,114,820**						
Enhance capital allowance (ECA)						**£**	**943,395**						
Project payback period							**11.5 yrs**						
Index of profitability (IOP)							**6**						

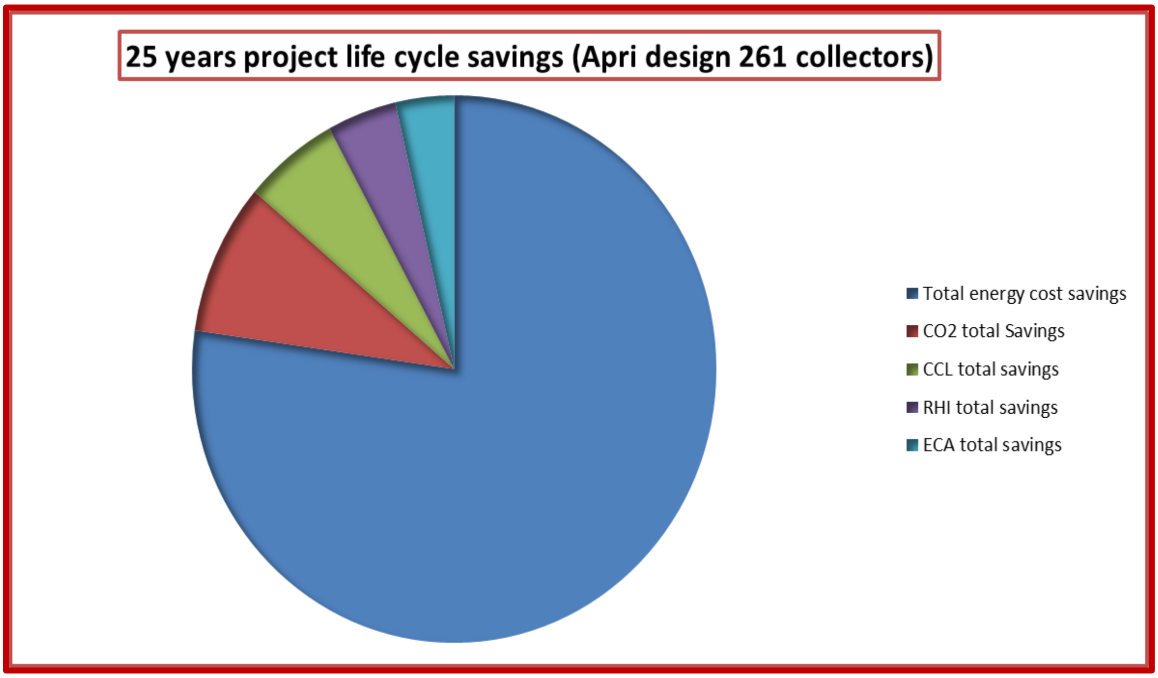


Figure 6:9 Twenty five years project economic life cycle savings (April heating demand design of 261 solar collectors)

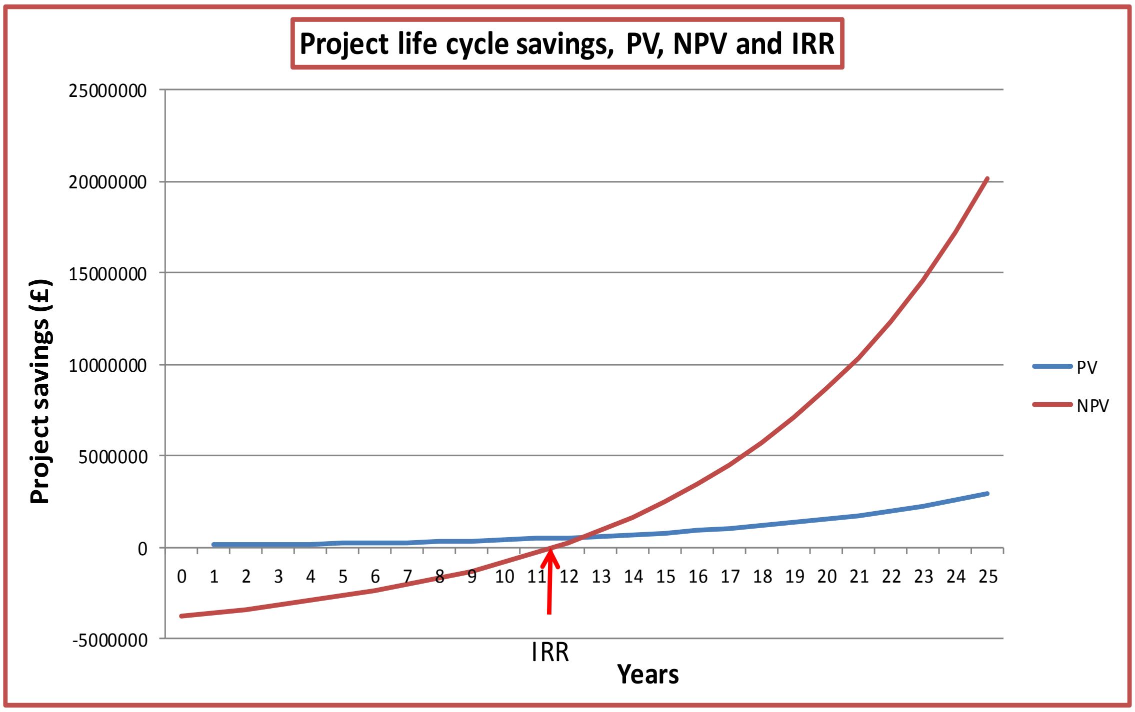


Figure 6:10 Twenty five years project economic life cycle savings, PV, NPV and IRR (April design of 261 solar collectors)

System 3: Design to meet glasshouse heating requirements from May to winter season when the heating demand is significantly increased. With this system, 73 solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the CHP waste heat.

Twenty years project economic life cycle cost analysis (May heating demand design of 73 collectors). Twenty years economic life cycle cost analysis is displayed in Table 6:10 below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:11 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance. Figure 6:12 below displays the project payback period, savings, PV, NPV and IRR.

Table 6:10 May design project appraisal of 20 years economic life cycle cost analysis (System 3 of 73 solar collectors)

Capital cost of project (£)			4,668,095										
Capital cost of project less ECA (£)			3734476										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost saving			104,640										
Renewable heat incentive			15,590										
Years	Maintennace Cash value	Annual savings cash value	Net savings cash value	PV	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,734,476				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	15590	-3596770				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	15590	-3442027				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	15590	-3267807				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	15590	-3071320				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	15590	-2849376				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	15590	-2598333				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	15590	-2314024				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	15590	-1991693				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	15590	-1625898				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	15590	-1210426				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	15590	-738171				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	15590	-201017				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	15590	410316				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	15590	1106429				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	15590	1899442				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	15590	2803202				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	15590	3833537				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	15590	5008535				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	15590	6348867				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	15590	7878158				0.03	133721549	0.186	24872208
Total savings at the end of project life				**11300834**	**311800**						**784889291**		**145989408**
Total energy cost savings at the end of project life						**£**	**7,878,158**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**2335831**						
Total climate change (CCL) saved						**£**	**1,530,534**						
Renewable heat incentive						**£**	**311,800**						
Enhance capital allowance (ECA)						**£**	**933,619**						
Project payback period							**12.5 yrs**						
Index of profitability (IOP)							**3**						

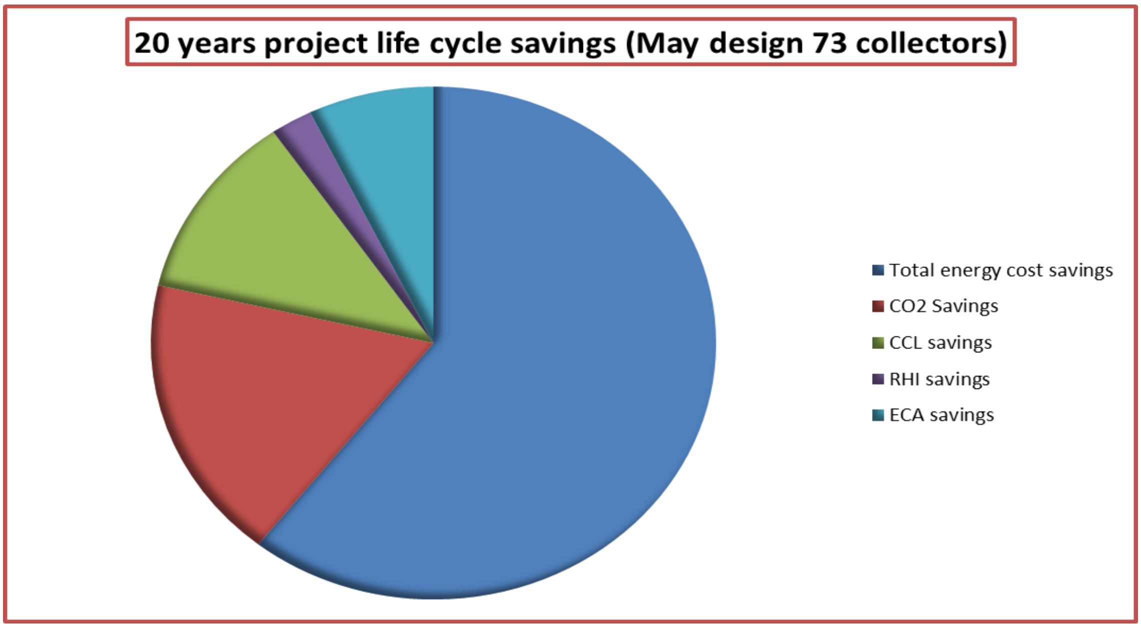


Figure 6:11 Twenty years project economic life cycle savings (May heating demand design of 73 solar collectors)

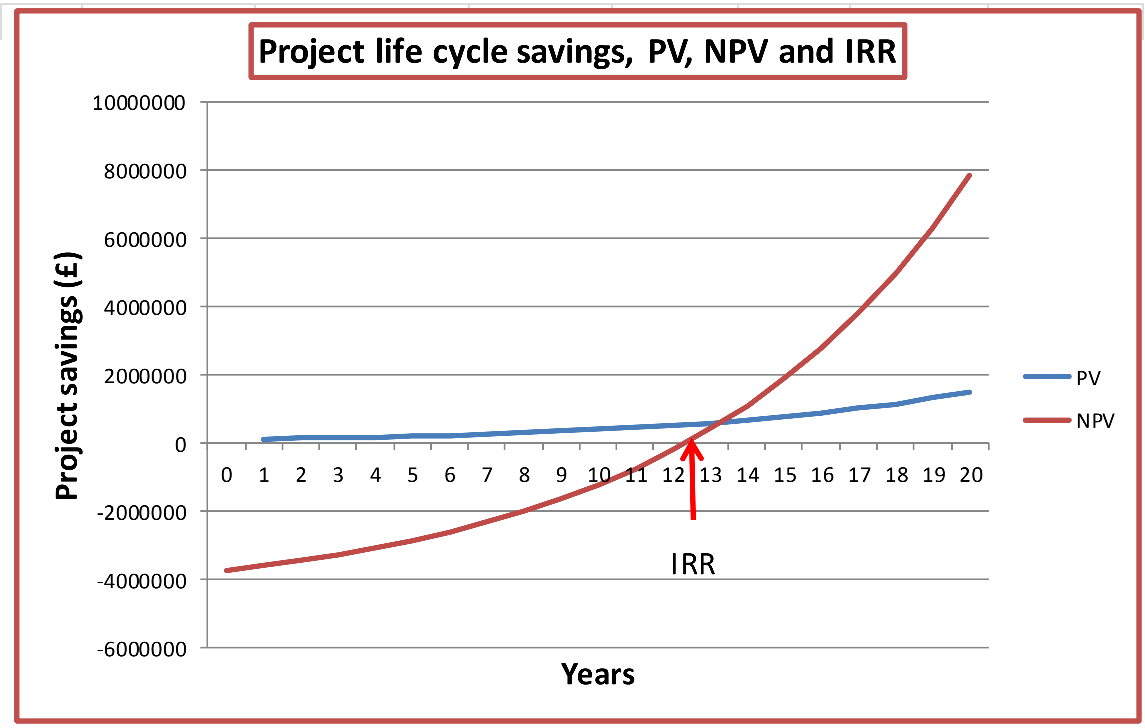


Figure 6:12 Twenty years project economic life cycle savings, PV, NPV and IRR (April design of 73 solar collectors)

System 3: Twenty five years project economic life cycle cost analysis (May heating demand design of 73 collectors)

Twenty five years economic life cycle cost analysis is displayed in Table 6:11below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:13 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance. Figure 6:14 below displays the project payback period, savings, PV, NPV and IRR.

Table 6:11 May design project appraisal of 25 years economic life cycle analysis (System 3 of 73 solar collectors)

Capital cost of project (£)			4,668,095										
Capital cost of project less ECA (£)			3734476										
Decerase in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			15,590										
Years	Maintennace Cash value	Annual savings cash value	Net savings cash value	PV (£)	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	Nat gas CO2 conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,734,476				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	15590	-3596770				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	15590	-3442027				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	15590	-3267807				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	15590	-3071320				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	15590	-2849376				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	15590	-2598333				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	15590	-2314024				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	15590	-1991693				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	15590	-1625898				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	15590	-1210426				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	15590	-738171				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	15590	-201017				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	15590	410316				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	15590	1106429				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	15590	1899442				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	15590	2803202				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	15590	3833537				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	15590	5008535				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	15590	6348867				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	15590	7878158				0.03	133721549	0.186	24872208
21	2654	4813976	4816630	1728893		9607051				0.03	160465858	0.186	29846650
22	2734	5776771	5779505	1975723		11582774				0.03	192559030	0.186	35815980
23	2816	6932125	6934941	2257818		13840592				0.03	231070836	0.186	42979175
24	2900	8318550	8321451	2580215		16420807				0.03	277285003	0.186	51575011
25	2987	9982260	9985248	2948671		19369478				0.03	332742004	0.186	61890013
Total savings at the end of project life				**22792154**	**311800**						**1979012022**		**368096236**
Total energy cost savings at the end of project life						**£**	**19,369,478**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**5889540**						
Total climate change (CCL) saved						**£**	**3,859,073**						
Renewable heat incentive						**£**	**311,800**						
Enhance capital allowance (ECA)						**£**	**933,619**						
Project payback period							**12.5 yrs**						
Index of profitability (IOP)							**6**						

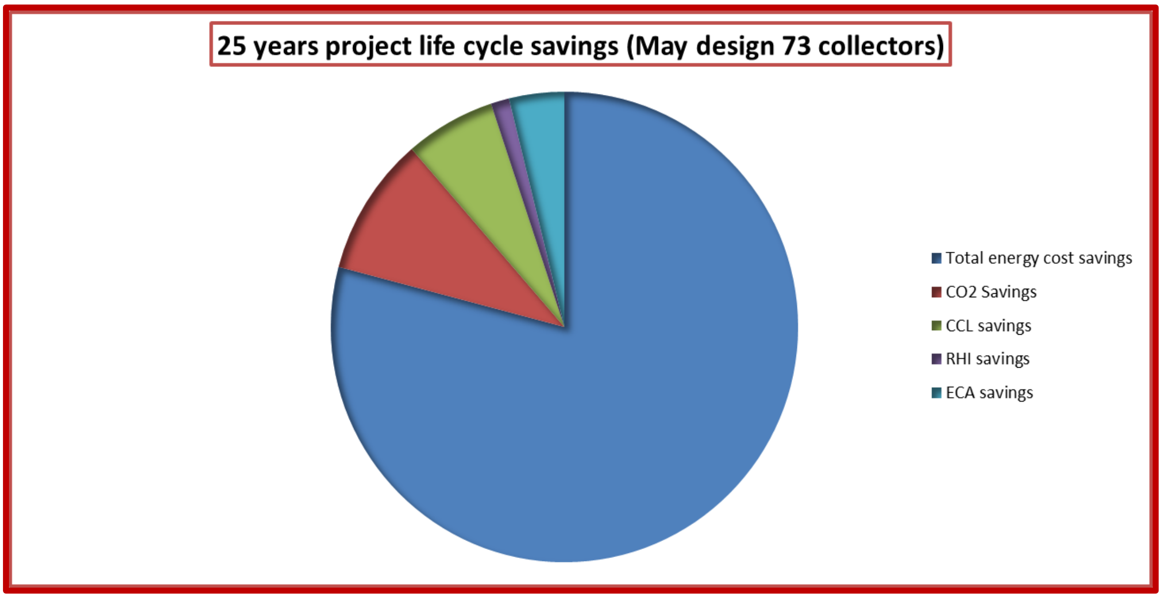


Figure 6:13 Twenty five years project economic life cycle savings (May heating demand design of 73 solar collectors)

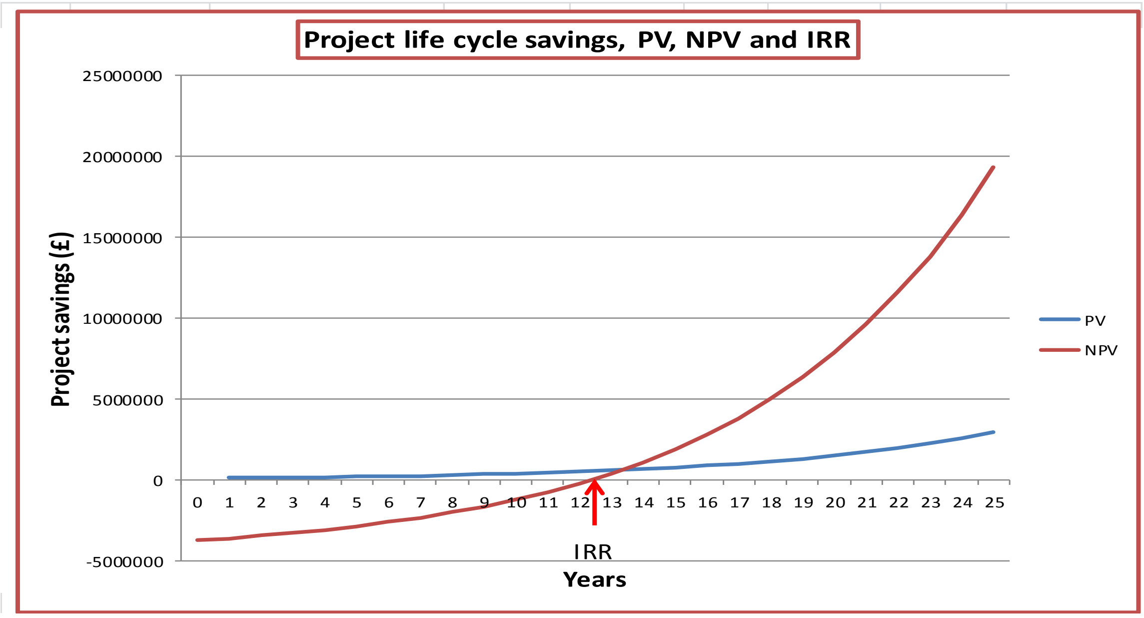


Figure 6:14 Twenty five years project economic life cycle savings, PV, NPV and IRR (May design of 73 collectors)

6.5.4 Systems designed to store two days glasshouse heating demand project economic life cycle cost analysis

The economic life cycle cost analysis under this design to store two days heating demand is similar to those under three days storage capacity. The only difference is that the capital cost of the systems will be decreased resulting to less payback period compared to the systems to store three days heating demand.

Two systems economic life cycle analysis is calculated to demonstrate the effect of reducing the storage capacity from three days to two days. The two days storage capacity will require six 100 m^3 PCM storage tanks whilst the 3 days storage capacity needs eight 100 m^3 PCM storage tanks.

System 1 under two days storage capacity project appraisal for twenty years economic life cycle costs analysis

System design to store two days heating energy capacity to meet glasshouse heating requirements from March to the winter season life cycle cost analysis is calculated below.

With this system, 803 solar collector (AZUR 8+ AC 2.8H) modules will be needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the supplementary CHP waste heat.

Under this system, twenty years economic life cycle cost analysis is displayed in table 6:12 which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:15 below shows all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance.

Figure 6:16 below displays the project payback period, savings, PV, NPV and IRR. Table 6:12 shows March design project appraisal for twenty years economic life cycle cost analysis (System 1 of 803 solar collectors). The capital cost is based on the PCM thermal energy storage tank storing two days of the glasshouse maximum heating demand.

Table 6:12 March design project appraisal to meet two days glasshouse heating demand, using 803 solar collectors 20 years economic life cycle cost analysis

Capital cost of project (£)			3,944,783										
Capital cost of project less ECA (£)			3155826.4										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			171,495										
Years	Maintennace Cash value	Annual savings cash value	Net savings cash value	PV	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversion factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,155,826				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	171495	-2862215				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	171495	-2551567				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	171495	-2221443				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	171495	-1869050				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	171495	-1491202				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	171495	-1084253				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	171495	-644040				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	171495	-165803				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	171495	355896				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	171495	927274				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	171495	1555433				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	171495	2248493				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	171495	3015730				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	171495	3867749				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	171495	4816666				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	171495	5876332				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	171495	7062572				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	171495	8393474				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	171495	9889711				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	171495	11574908				0.03	133721549	0.186	24872208
Total savings at the end of project life				**11300834**	**3429900**						**784889291**		**145989408**
Total energy cost savings at the end of project life						**£**	**11,574,908**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**2335831**						
Total climate change (CCL) saved						**£**	**1,530,534**						
Renewable heat incentive						**£**	**3,429,900**						
Enhance capital allowance (ECA)						**£**	**788,957**						
Project payback period							**8.5 yrs**						
Index of profitability (IOP)							**4**						

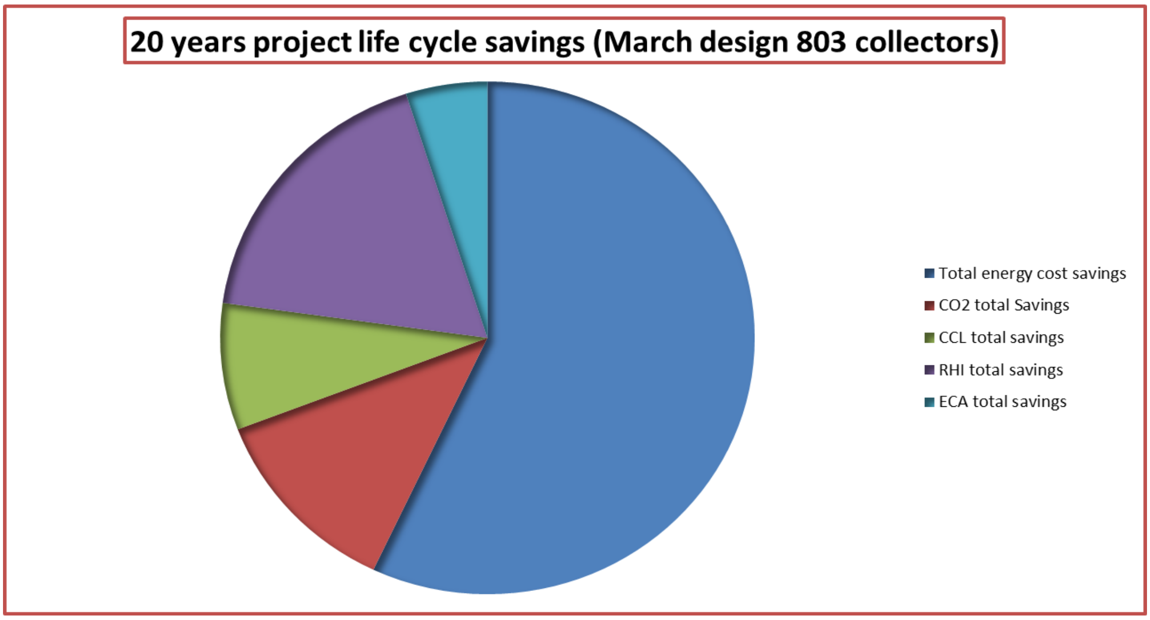


Figure 6:15 Twenty years project economic life cycle savings (March heating demand design of 803 solar collectors)

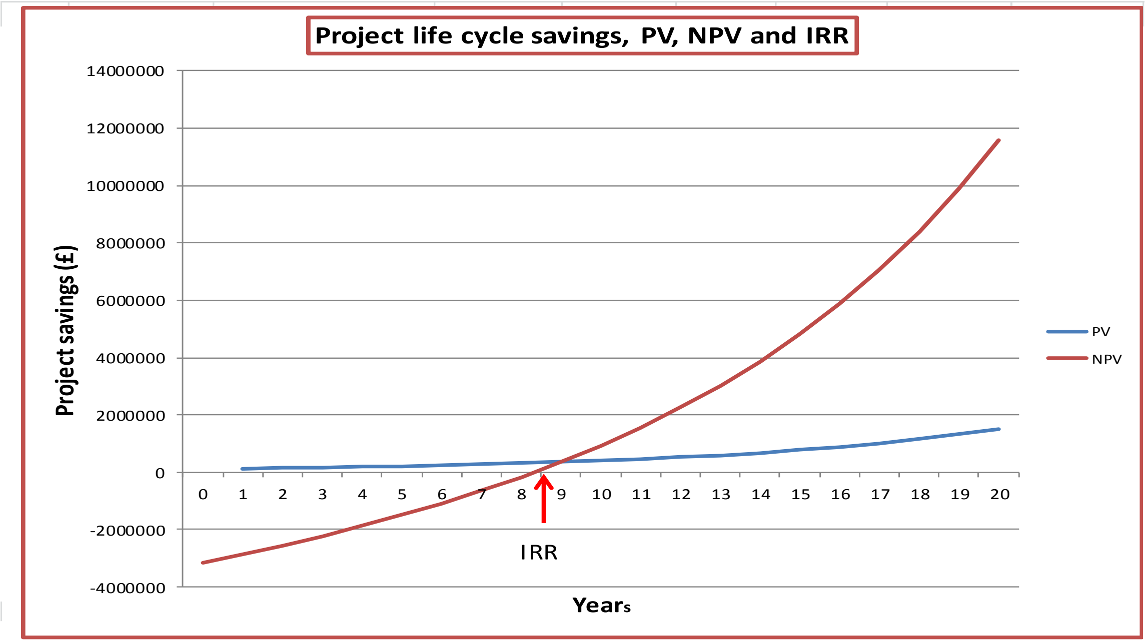


Figure 6:16 Twenty years project economic life cycle savings (March heating demand design of 803 solar collectors)

System 1 under two days storage capacity project appraisal for twenty five years economic life cycle costs analysis

System design to store two days heating energy capacity to meet glasshouse heating requirements from March to the winter season economic life cycle cost analysis is calculated below. With this system, 803 solar collector (AZUR 8+ AC 2.8H) modules is needed to generate enough thermal energy to meet the glasshouse heating demand before the winter season sets in to use the supplementary CHP waste heat.

Twenty five years economic life cycle cost analysis is displayed in table 6:13 below which includes the project capital cost, capital cost less enhance capital allowance, decrease in maintenance cost, renewable heat incentive and all the benefits, payback period and index of profitability at the bottom of the table.

Figure 6:17 below illustrates all the benefits which include the total energy cost savings, total climate change levy savings, renewable heat incentive and enhance capital allowance.

Figure 6:18 below displays the project payback period, savings, PV, NPV and IRR. The capital cost is based on the PCM thermal energy storage tank storing two days of the glasshouse maximum heating demand.

Table 6:13 March design project appraisal to meet two days glasshouse heating demand, using 803 solar collectors 20 years economic life cycle cost analysis

Capital cost of project (£)			3,944,783										
Capital cost of project less ECA (£)			3155826.4										
Decrease in maintenance cost(£)			2,577										
Calculated annual energy cost savings(£)			104,640										
Renewable heat incentive			171,495										
Years	maintennace Cash value	Annual savings cash value	Net savings cash value	PV (£)	RHI	NPV	Interest rate	Fuel inflation rate	Maintenance inflation rate	Cost/ kWh	Energy savings	conversi on factor	CO2 savings
	(£)	(£)	(£)	(£)	(£)	(£)	0.05	0.2	0.03	0.03	KWh		kg CO2
0	2,577	104,640				-3,155,826				0.03	3488000	0.186	648768
1	2654	125568	128222	122116	171495	-2862215				0.03	4185600	0.186	778522
2	2734	150682	153416	139152	171495	-2551567				0.03	5022720	0.186	934226
3	2816	180818	183634	158630	171495	-2221443				0.03	6027264	0.186	1121071
4	2900	216982	219882	180897	171495	-1869050				0.03	7232717	0.186	1345285
5	2987	260378	263365	206354	171495	-1491202				0.03	8679260	0.186	1614342
6	3077	312453	315530	235454	171495	-1084253				0.03	10415112	0.186	1937211
7	3169	374944	378113	268718	171495	-644040				0.03	12498135	0.186	2324653
8	3264	449933	453197	306742	171495	-165803				0.03	14997762	0.186	2789584
9	3362	539919	543282	350204	171495	355896				0.03	17997314	0.186	3347500
10	3463	647903	651367	399883	171495	927274				0.03	21596777	0.186	4017000
11	3567	777484	781051	456664	171495	1555433				0.03	25916132	0.186	4820401
12	3674	932981	936655	521565	171495	2248493				0.03	31099358	0.186	5784481
13	3784	1119577	1123361	595742	171495	3015730				0.03	37319230	0.186	6941377
14	3898	1343492	1347390	680524	171495	3867749				0.03	44783076	0.186	8329652
15	4015	1612191	1616206	777423	171495	4816666				0.03	53739691	0.186	9995583
16	4135	1934629	1938764	888170	171495	5876332				0.03	64487630	0.186	11994699
17	4259	2321555	2325814	1014745	171495	7062572				0.03	77385155	0.186	14393639
18	4387	2785866	2790253	1159408	171495	8393474				0.03	92862186	0.186	17272367
19	4519	3343039	3347557	1324742	171495	9889711				0.03	111434624	0.186	20726840
20	4654	4011646	4016301	1513702	171495	11574908				0.03	133721549	0.186	24872208
21	2654	4813976	4816630	1728893		13303801				0.03	160465858	0.186	29846650
22	2734	5776771	5779505	1975723		15279524				0.03	192559030	0.186	35815980
23	2816	6932125	6934941	2257818		17537341				0.03	231070836	0.186	42979175
24	2900	8318550	8321451	2580215		20117556				0.03	277285003	0.186	51575011
25	2987	9982260	9985248	2948671		23066227				0.03	332742004	0.186	61890013
Total savings at the end of project life				**22792154**	**3429900**						**784889291**		**145989408**
Total energy cost savings at the end of project life						**£**	**23,066,227**						
Total CO2 emission cost savings (£16/tCO2)						**£**	**2335831**						
Total climate change (CCL) saved						**£**	**1,530,534**						
Renewable heat incentive						**£**	**3,429,900**						
Enhance capital allowance (ECA)						**£**	**788,957**						
Project payback period							**8.5yrs**						
Index of profitability (IOP)							**7**						

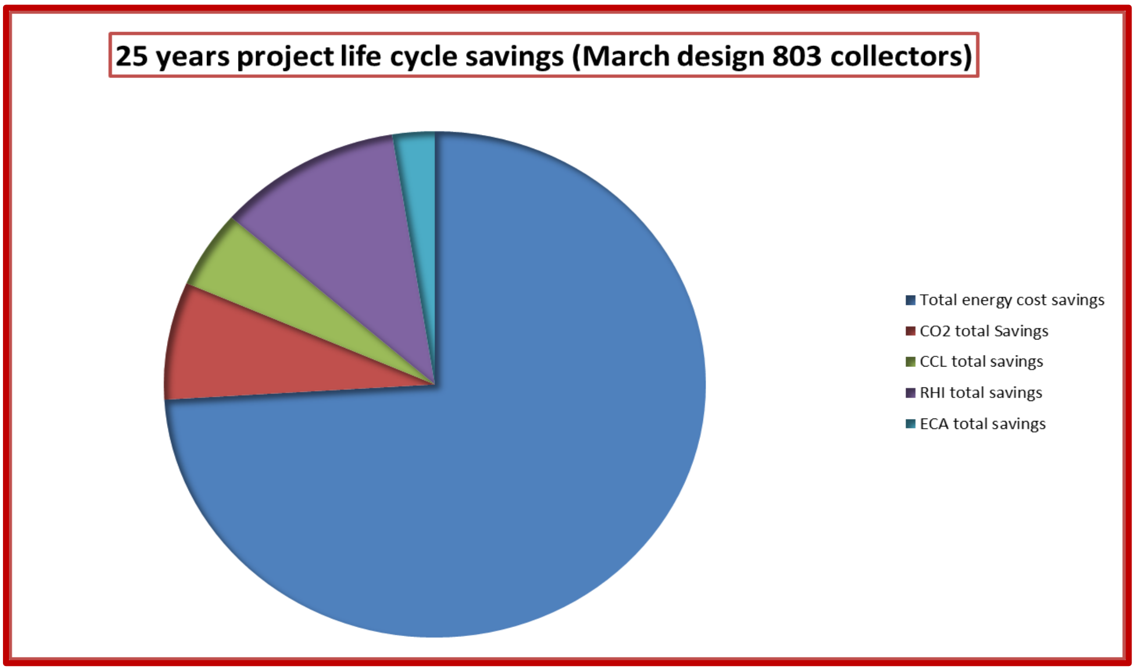


Figure 6:17 Twenty five years project economic life cycle savings (March heating demand design of 803 solar collectors)

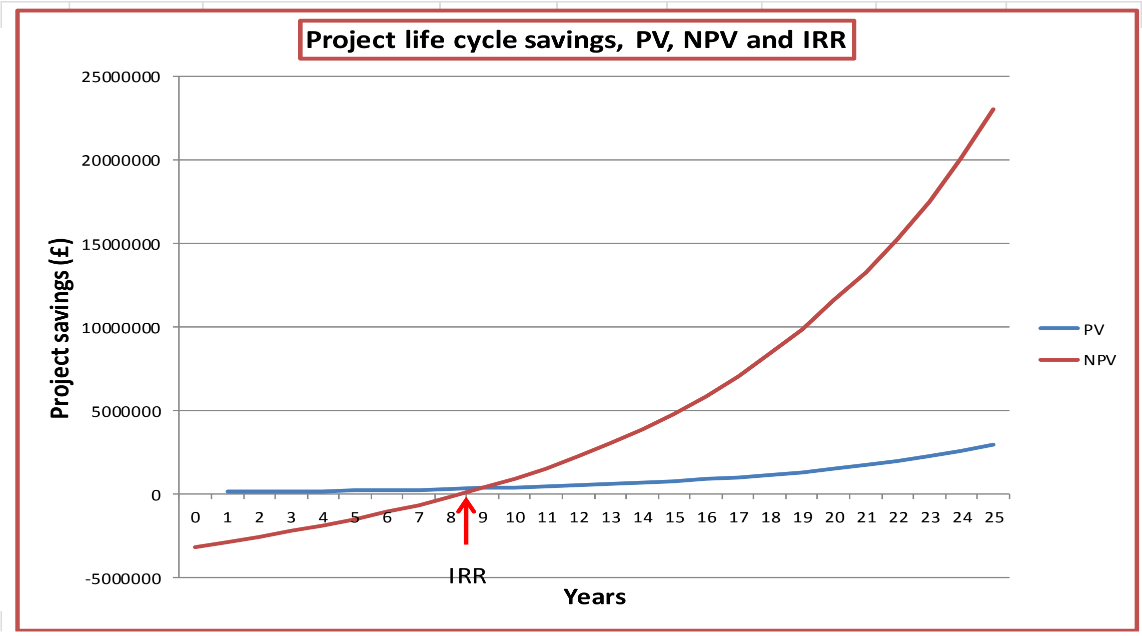


Figure 6:18 Twenty five years project economic life cycle savings, PV, NPV and IRR (March design of 803 solar collectors)

6.5.5 Summary of two of the research proposed projects potential benefits.

Table 6:14 below gives summary of two of the research proposed projects potential benefits. The two proposed design systems are capable of storing two and three days heating energy requirement of the glasshouse respectively.

The economic life cycle cost analysis for 20 and 25 years of the two design systems are demonstrated in table 6.14 below showing all the benefits, payback periods, project cost and index of profitability (IOP).

Table 6:14 Summary of two proposed research project potentials

	Two days storage capacity system 20 years (803 solar collectors)	Three days storage capacity system 20 years (803 solar collectors)	Two days storage capacity system 25 years (803 solar collectors)	Three days storage capacity system 25 years (803 solar collectors)
Capital cost of project (£)	3,944,783	4,737,990	3,944,783	4,737,990
Total energy cost savings at the end of project life (£)	11,574,908	10,940,342	23,066,227	22,431,662
Total CO2 emission cost savings [£16/tCO2] (£)	2335831	2335831	5889540	5889540
Total climate change (CCL) saved (£)	1,530,534	1,530,534	3,859,073	3,859,073
Renewable heat incentive (£)	3,429,900	3,429,900	3,429,900	3,429,900
Enhance capital allowance [ECA] (£)	788,957	947,598	788,957	947,598
Project payback period	8.5 yrs	9.5yrs	8.5yrs	9.5 yrs
Index of profitability (IOP)	4	3	7	6

The project appraisal has demonstrated that the heating system potential benefits are significant. It is technically sound, financially justified and environmentally acceptable.

6.5.6 Environmental impact assessment of the existing and research design heating systems

The proposed research heating system will have the following environmental benefits compared to the existing heating system:

- Increase opportunities in resource efficiency
- Reduce fast depleting of natural resources such as fossil fuel and improving operational efficiency with less resources
- Reduction in CO_2 emissions
- Reduce energy demand, conserve natural resources and improve health and social wellbeing
- Avoid or reduce health risk factors and improve human lives
- Effective energy management will save energy, cost and reduce CO_2 emissions leading to good environmental management.
- Assist Royal Botanical Gardens meeting their social responsibility to protect the environment from air pollution and noise and promote community wellbeing
- The heating system will has lower embodied environmental impact
- The heating system will use less resources in maintenance and in operations
- The survival of humans and thousands of other living species whose lives are threatened locally, regionally and globally by air and water pollution will be saved
- The heating system will reduce global warming depletion
- The earth integrity, its biodiversity, security of nations, and the heritage of future generations will be less threatened.
- The environmental changes caused by CO_2 emissions resulting to poverty aggravation in many regions in the world will be reduced.

Designers addressing the fundamental problems associated with heating systems using fossil fuels will reverse the catastrophic consequences of global warming, stabilise human population and will improve living standards.

Summary of project appraisal

The research project implementation requires project appraisal which is a process of checking the basic data, assumptions and methodology used in the preparation carefully. It is an in-depth study and review of the work plan, cost estimates, benefits and the proposed capital requirement.

It is a process of given thought to the financial details, economic and social benefits. The suitability of the project should be assessed as to whether the project is technically sound, financially justified and environmentally acceptable.

The techniques normally used for project appraisal are undiscounted, discounted and simple payback and because this project is a multi-million pound (£m) project, the discounted method of project appraisal is used.

The discounted method takes into consideration the time value of the money invested which includes net present value (NPV), benefit cost ratio (BCR), internal rate of return (IRR), sensitivity analysis (Taken into consideration uncertainties). The BCR was not shown in the analysis yet it was considered.

For the project to be acceptable, it should prove to be profitable to service any investment made therefore all the savings and qualified benefits needed to be assessed. The project will benefit from financial incentives such as renewable heat incentive, enhance capital allowance, climate change tax exemption and sale of the CO_2 emissions saved.

Solar thermal system would last in excess of 25 years and could still function over 30 years and qualifies with the above stated benefits through the project life with exemption of renewable heat incentive which will last for 20 years.

The renewable heat incentive is paid by the UK government for twenty (20) years of the solar collector system.

The current heating system yearly energy consumption was calculated to be 2421 MWh and maintenance cost of £2,577. The proposed heating system based on the current heating system energy performance will save £104,640 per annum which is quite significant amount of money.

The proposed project capital cost calculation was carried out to establish the capital investment cost. The total savings cost of each of the proposed systems designed was calculated including enhance capital allowance.

The actual capital cost used in the project appraisal calculation was the calculated cost less the enhance capital allowance as this money is written off against the business taxable profits in the financial year that the purchase was made.

Detailed project costing was carried out and all the materials cost for the calculation were obtained from the manufacturers or the suppliers therefore they are authentic and not assumptions.

The assumed interest rates, fuel and maintenance inflation rates are based on the current and expected future economic climate conditions. The proposed projects based on the detail life cycle cost analysis are economically sound and profitable.

The payback period of system 1 March design of 803 collectors under the two days heating energy storage capacity is 8.5 years whilst that of the three days storage capacity is 9.5 years. The reason is that the cost to implement the two days storage capacity system is less compared to three days storage capacity system.

The total overall savings of the proposed projects are significant and encouraging. The prospects of the proposed projects are overwhelming and this is an indication of the research success.

7 Conclusion and Further Study

7.1 Conclusion

Royal Botanic Gardens as an institution manages and run glasshouses in both Kew Gardens and Wakehurst. RBG requirement to improve the performance of the heating systems in the glasshouses has been the origin of the research.

The glasshouses in Kew Gardens and Wakehurst are heated using fossil fuels. They are costly to run and the environmental impact is significant.

The European Commission is also seeking cost effective ways to make the European economy less energy consuming to make it more climate-friendly. They want to cut greenhouse gas emissions through the use of clean technologies.

One of such clean technologies could be thermal storage using phase change material (PCM) thermal energy storage techniques where waste thermal or solar energy could be stored and use on demand.

The research sought to assist Royal Botanical Gardens to achieve low or zero carbon emission heating system aimed at achieving the following:

- Zero carbon dioxide emission heating system using phase change material (PCM) thermal energy storage techniques
- Space temperature controlled by melting and freezing of the PCM heating pipes
- Using space trapped solar energy to charge PCM filled heating pipes to maximise the use of solar energy

The aims have been achieved which suggest the success of the research.

The research first studied the environmental requirements of green plants, the current glasshouse heating system, fuel use, the system efficiency and energy consumption.

The IES environmental design software tool was used to design the glasshouse which was of the same size and shape as the existing glasshouse with the same construction properties.

The heating energy demand for each zone was calculated using the IES software tool and designed a heating system that will meet the heating requirement of each zone.
Each zone hourly, daily and monthly heating demand, energy requirements, temperatures and relative humidity profiles were also determined using the IES software tool.

This was to assist in knowing the periods when thermal energy needs to be generated to warm the glasshouse and periods where the solar gains are sufficient to warm the glasshouse and store the excess heat gains trapped inside the space to the PCM heating pipes rather than venting it through the roof to the atmosphere.

The active and passive systems were sized to generate enough heating energy to meet the glasshouse heating demand even in the worse cases situation and for the unexpected.

The solar collectors selected are efficient, have life expectancy over 25 years and qualify for enhance capital allowance (ECA) and renewable heat incentive (RHI).

The PCM storage tanks were sized to meet three days heating demand of the glasshouse should the active and passive systems generate zero heating energy for three days which is unlikely to happen.

The heating energy that could be stored by the PCM heating pipes were also calculated to ascertain how much trapped solar energy in the glasshouse could be stored daily, monthly and throughout the year.

The melting and freezing times of the selected PCMs were calculated to ensure that sufficient energy could be stored or delivered to the glasshouse when needed and store heating energy during periods when solar radiation is sufficient to charge the systems.

The life expectancy of the selected PCM is long if managed efficiently and has the following required properties:

- High latent heat of fusion per unit volume
- Relatively high thermal conductivity
- Small volume changes on melting.
- They are not corrosive, compatible with plastics and only slightly toxic.
- They are inexpensive and cheap
- Congruent melting (When the anhydrous salt is completely soluble in its water of hydration at the melting temperature)

The space set point temperatures will be maintained by melting and freezing of the PCM heating pipes which will ensure operational efficiency without heat losses as heat will only be delivered by the PCM heating pipes when the space temperature falls below the phase change temperature of the phase change material.

The actual capital cost used in the project appraisal calculation was the calculated capital cost less the enhance capital allowance as this money is paid by the UK government against business taxable profits in the financial year that the purchase is made. The project costing was detailed and materials cost were obtained from the manufacturers or the suppliers with no assumptions.

Project appraisal was carried out to assess the proposed project's economic potentials, its viability and risks.

The economic life cycle cost analysis has demonstrated the potentials of the research heating system. The project aim to achieve 100% zero carbon emission heating system is nearly achieved with the exception of the electrical energy that will be used by the hot water circulation pumps.

The research has been successful and could be applied to commercial glasshouses and similar buildings.

The originality of the research which is to achieve low or zero carbon dioxide emission heating system using phase change material thermal energy storage techniques, space temperature controlled by melting and freezing of the PCM heating pipes and use space trapped solar energy to charge PCM heating pipes have been successful.

The research aim has been achieved and will add knowledge to glasshouse heating systems. The research design principles could effectively be applied to domestic and non-domestic buildings heating systems.

7.2 Further studies

Two passive systems were considered during the initial planning of the project which includes PCM heating pipes and black painted pipes. The black painted pipes will absorb solar radiation to pre-heat or warm the heating circulating hot water prior to entering the solar collectors of the active solar system or divert it through the PCM thermal energy storage system depending on the hot water exiting temperature from the black painted pipes.

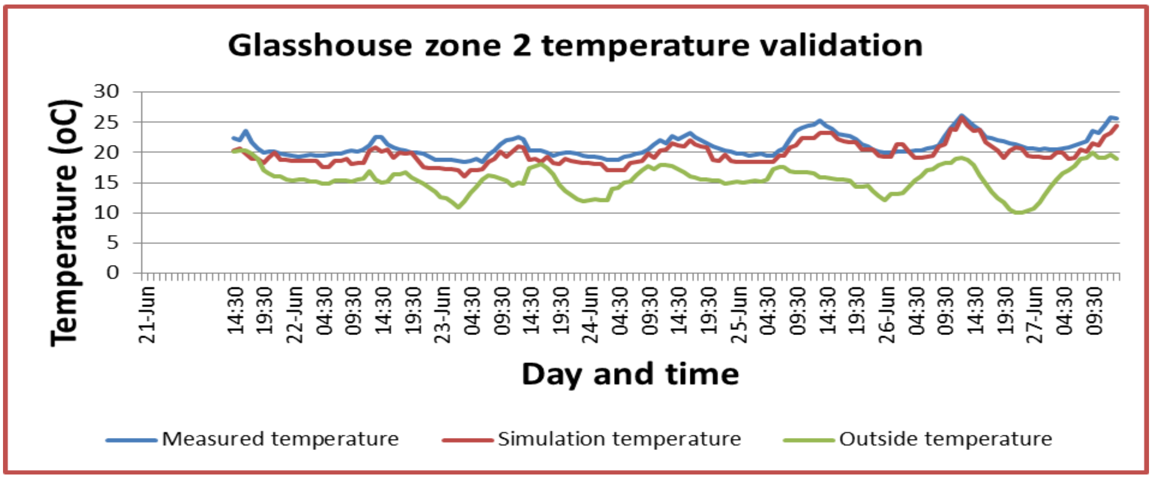
The black painted pipes could also transfer radiant heat to warm the bodies in the zones which include the plants, the plant soil and the ground. The heat energy that could be produced by the black painted heating pipes could not be analysed due to research time constrains and therefore suggest further study to analyse the heat energy that could be produced by the black painted heating pipes as part of the passive heating system.

The design principle of using PCM heating pipes to control space temperature by melting and freezing could effectively be applied to domestic and non-domestic buildings heating radiators.

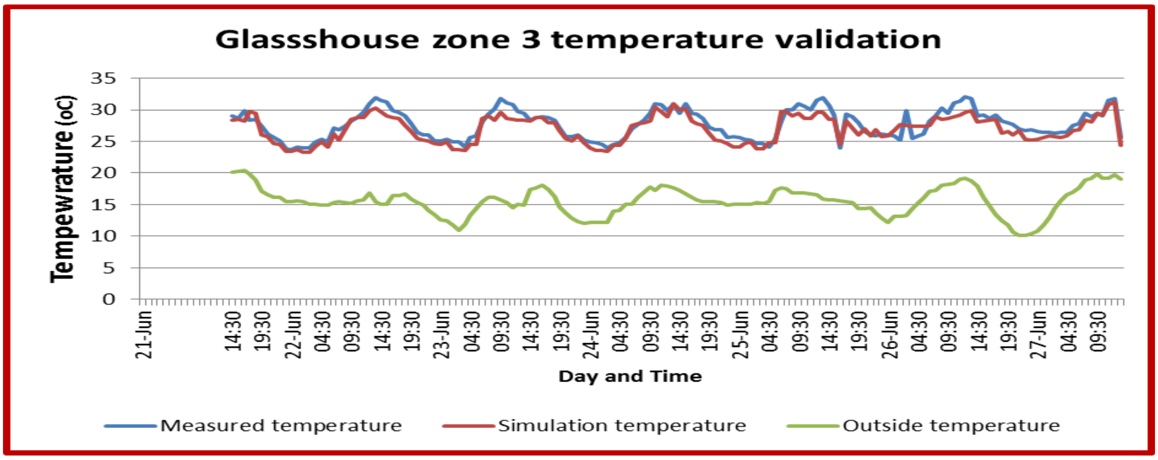
For example, radiators could be filled with PCM solution to maintain building space temperatures. It will be energy efficient and economical as it will only give heat out when the space temperature falls below the PCM phase change temperature to maintain the room temperature and will absorb heat energy from internal heat gains when the space temperature rises above the phase change temperature of the PCM filled in the radiator.

The system will be effective and useful for both heating and cooling modes as space temperature rising above an acceptable level in summer will be controlled by the PCM radiators absorbing the excessive heat generated in the space and give the heat out any time the space temperature falls below space temperature set point requirement.

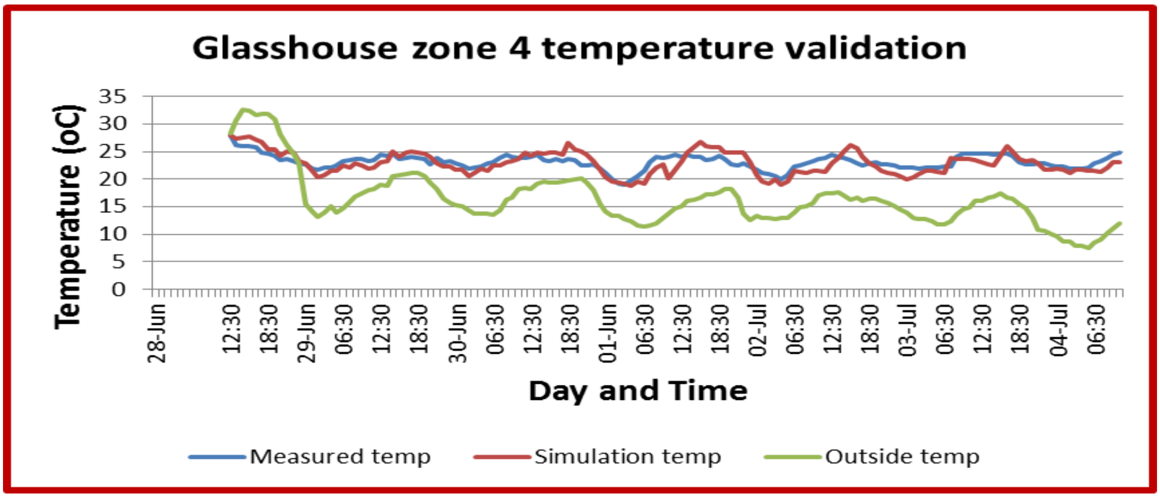
Appendix A Glasshouse zones temperature validation



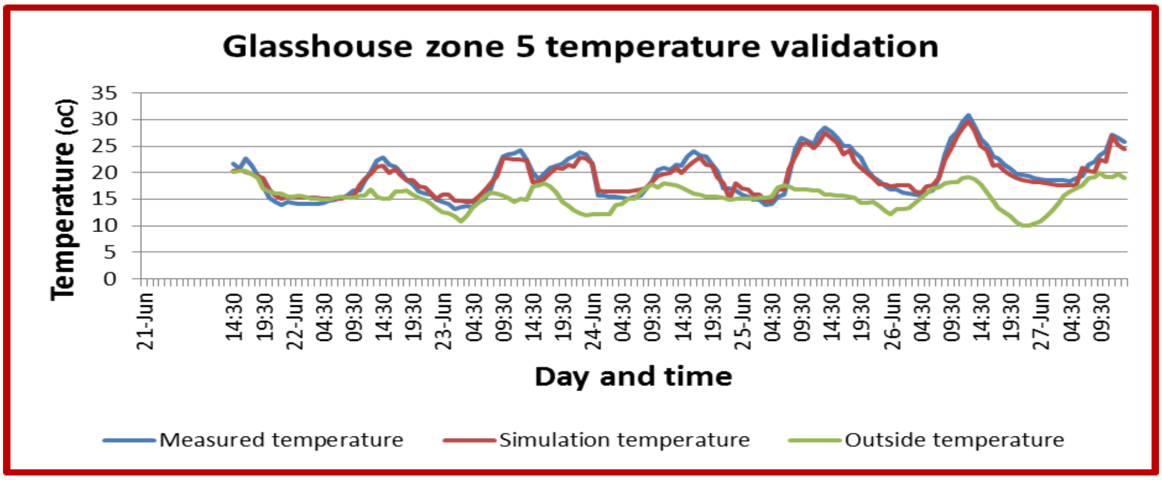
Appendix A1 Zone 2 temperature validation



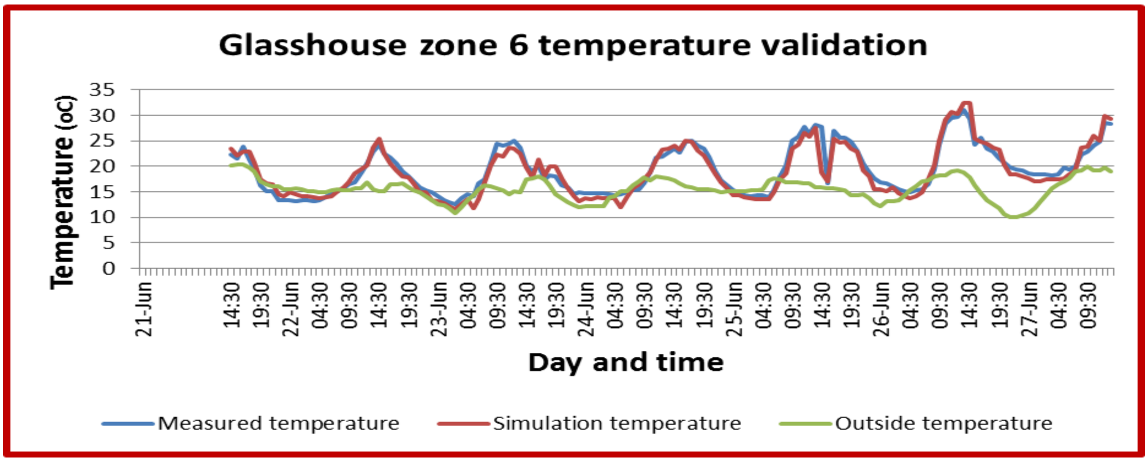
Appendix A2 Zone 3 temperature validation



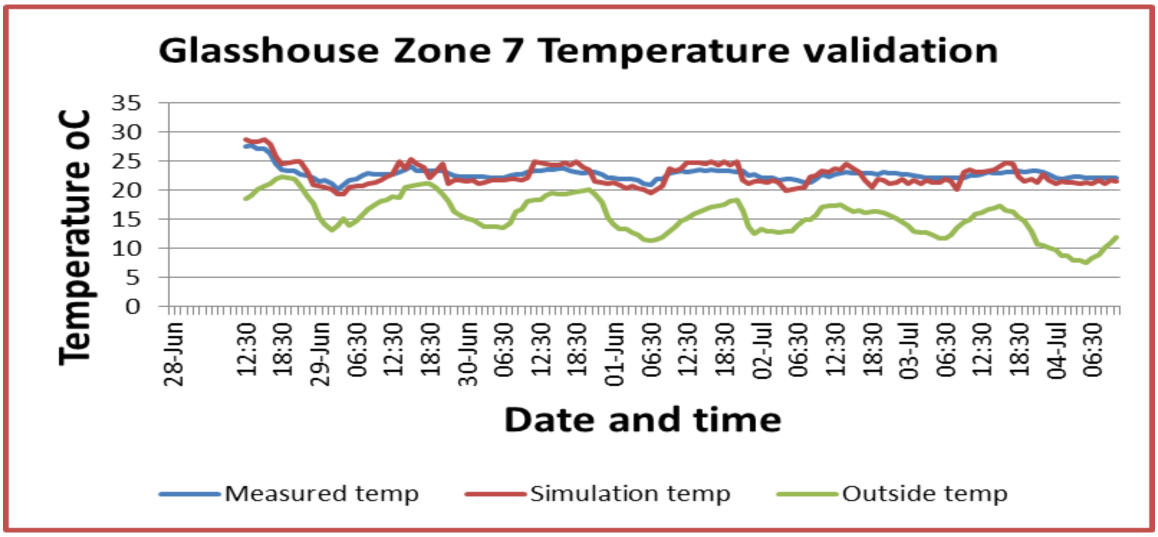
Appendix A3 Zone 4 temperature validation



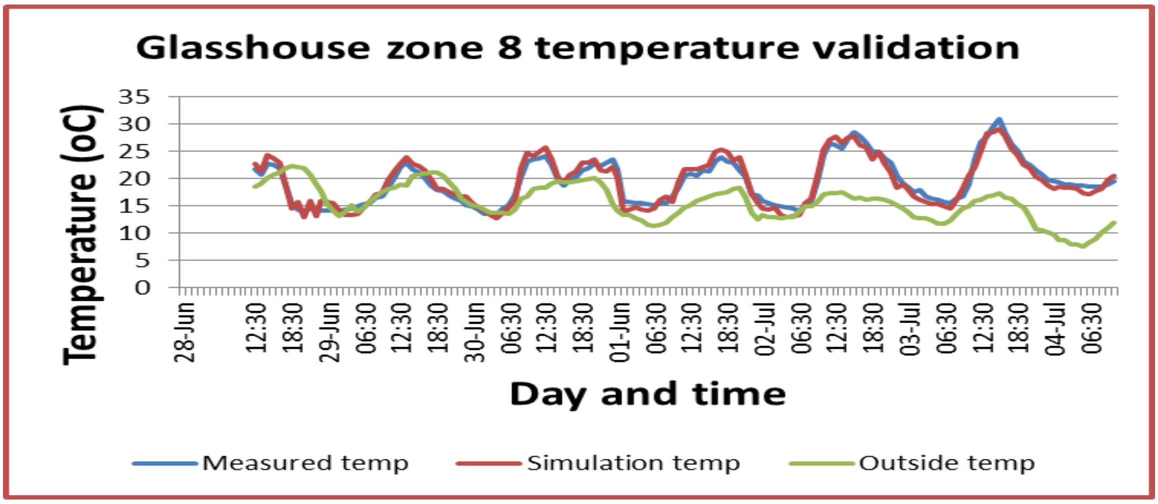
Appendix A4 Zone 5 temperature validation



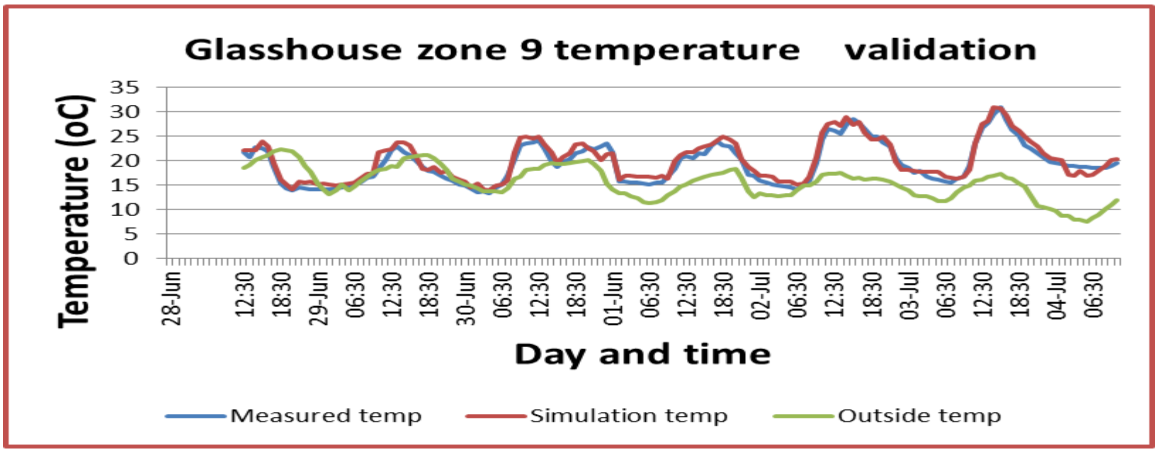
Appendix A5 Zone 6 temperature validation



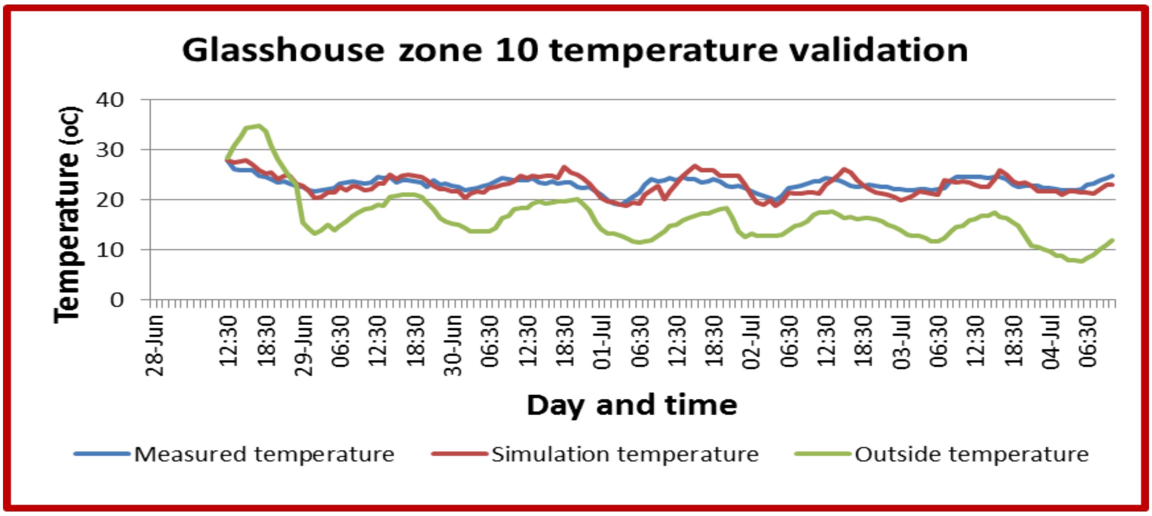
Appendix A6 Zone 7 temperature validation



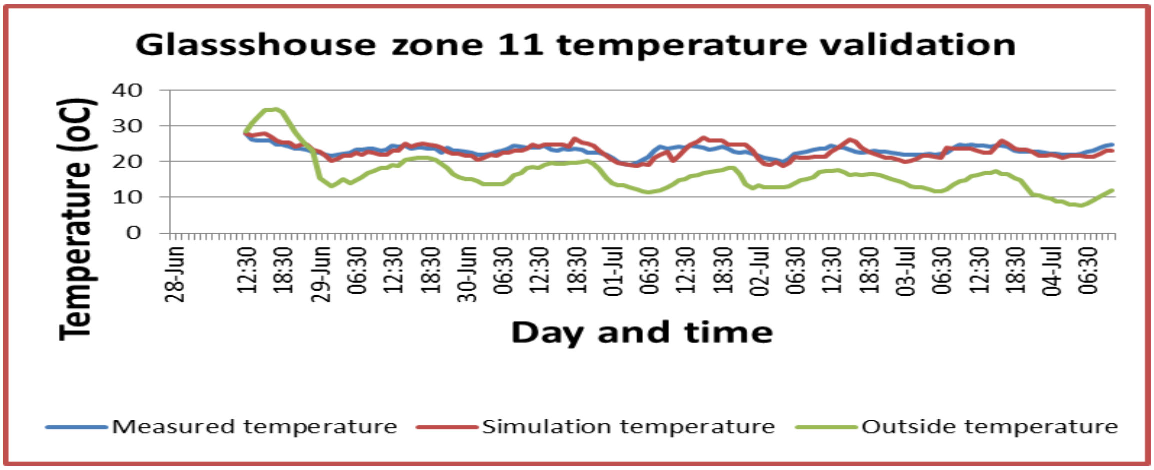
Appendix A7 Zone 8 temperature validation



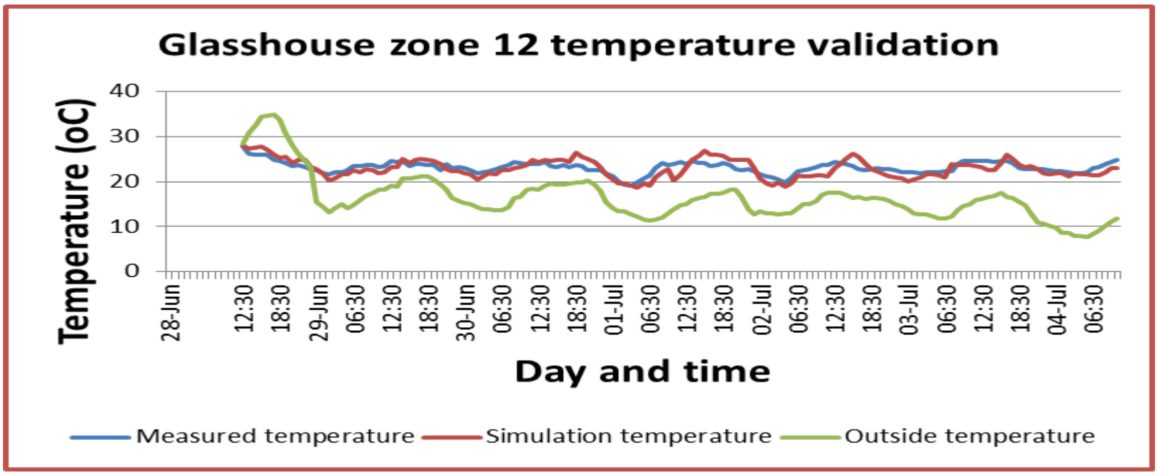
Appendix A8 Zone 9 temperature validation



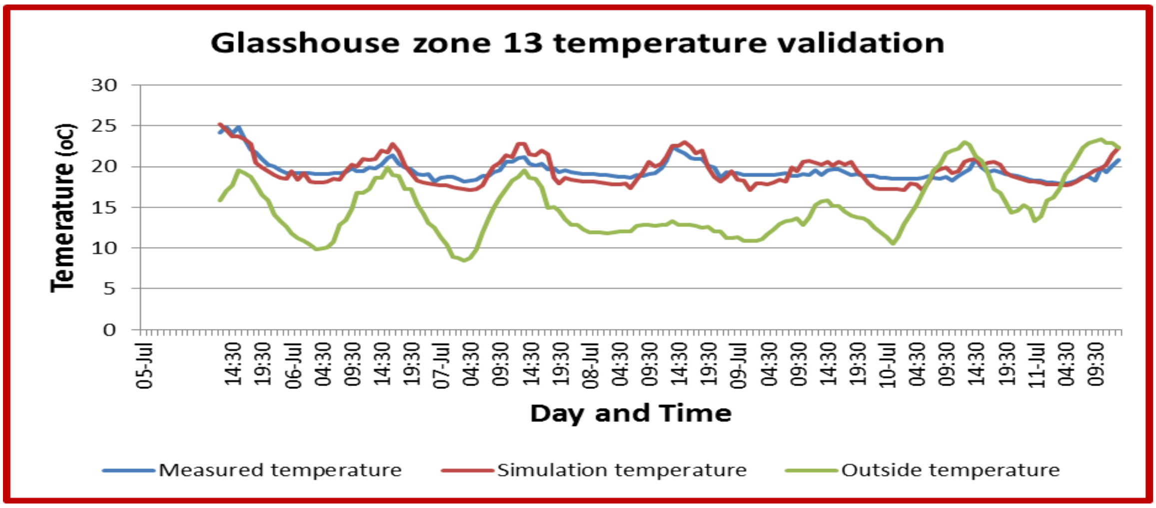
Appendix A9 Zone 10 temperature validation



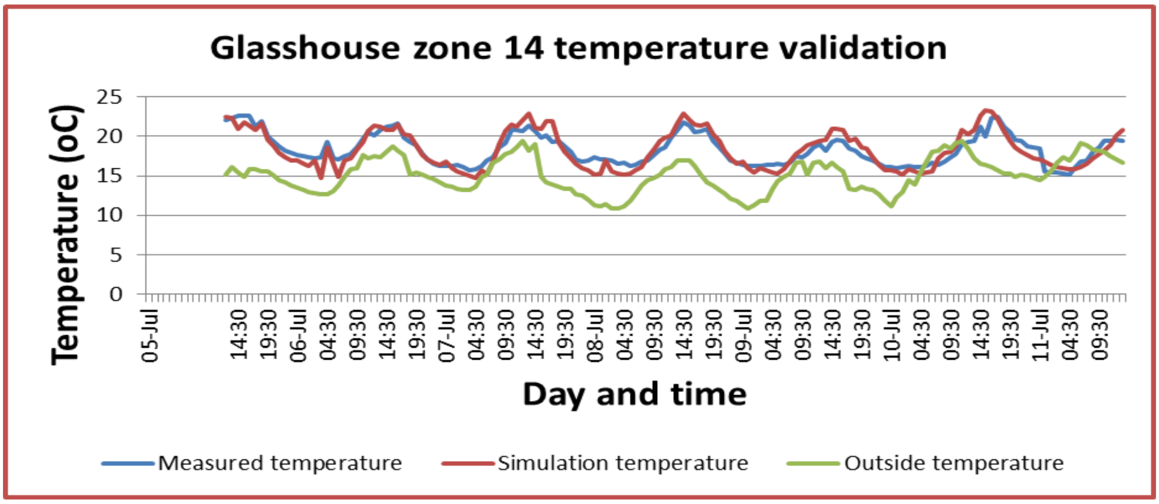
Appendix A10 Zone 11 temperature validation



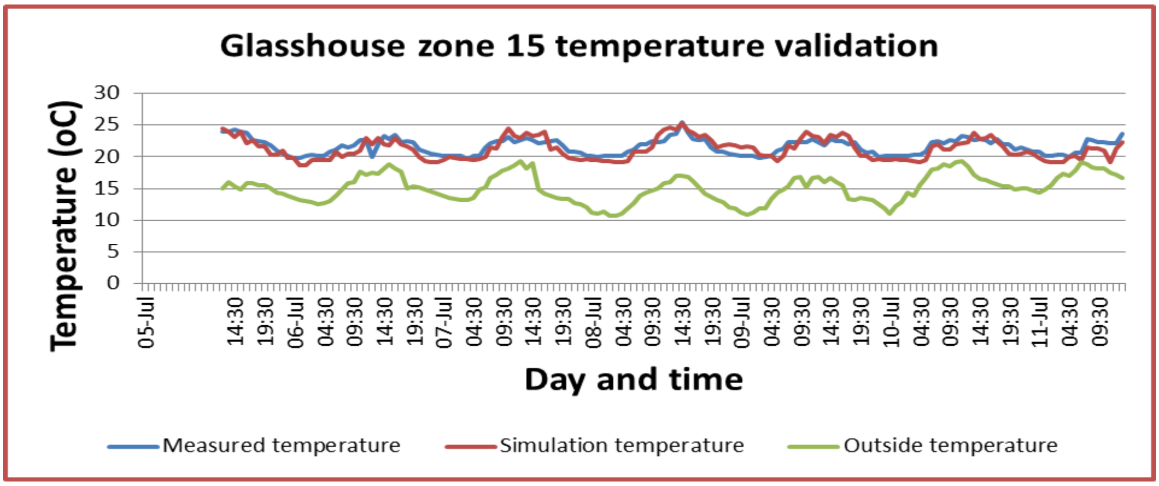
Appendix A11 Zone 12 temperature validation



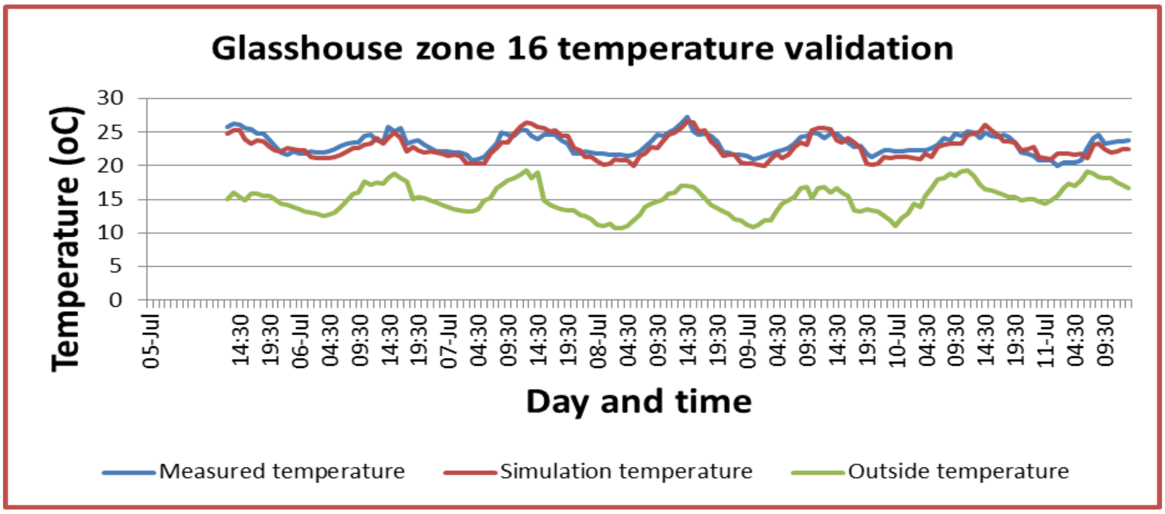
Appendix A12 Zone 13 temperature validation



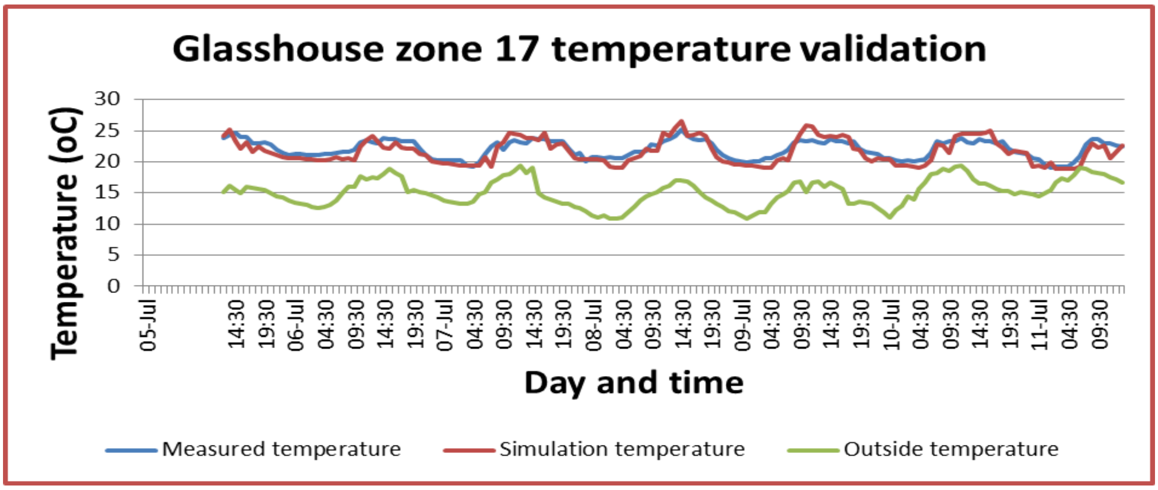
Appendix A13 Zone 14 temperature validation



Appendix A14 Zone 15 temperature validation

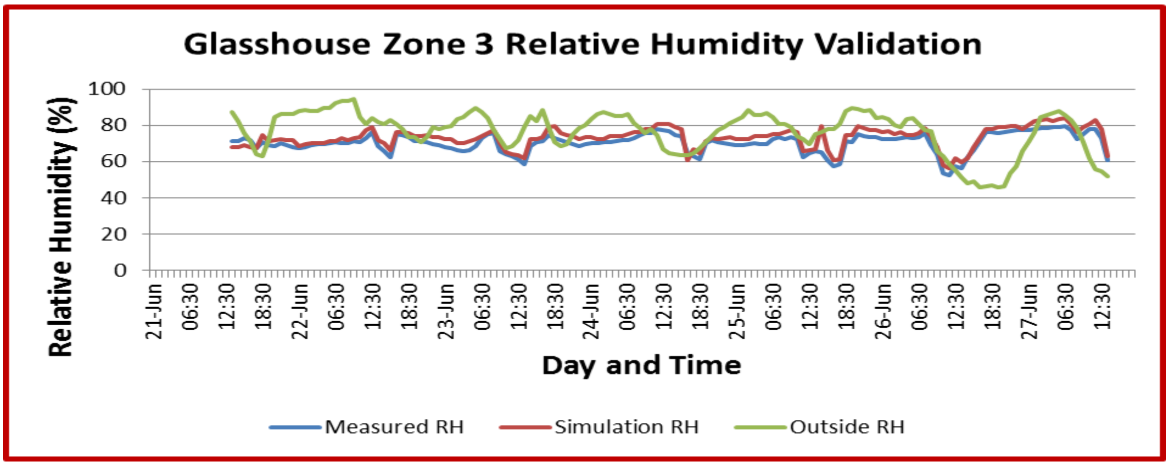


Appendix A15 Zone 16 temperature validation

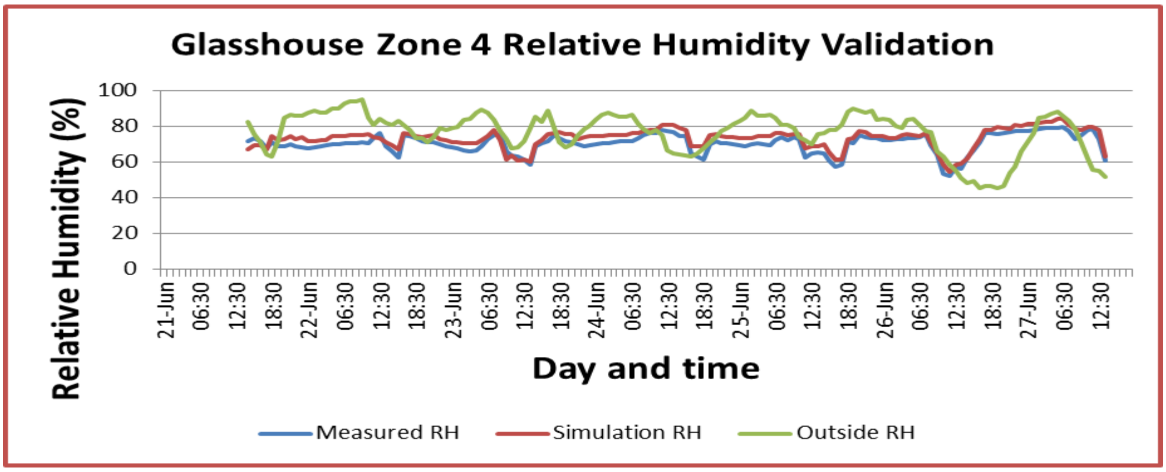


Appendix A16 Zone 17 temperature validation

Appendix B Glasshouse zones relative humidity validation

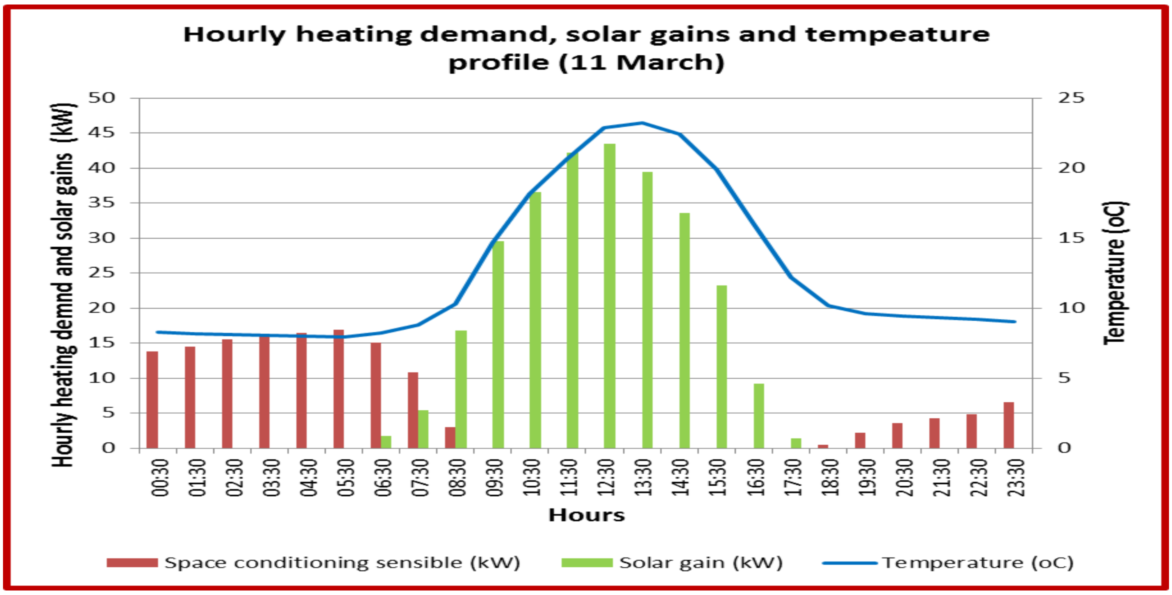


Appendix B1 Zone 3 Relative humidity validation

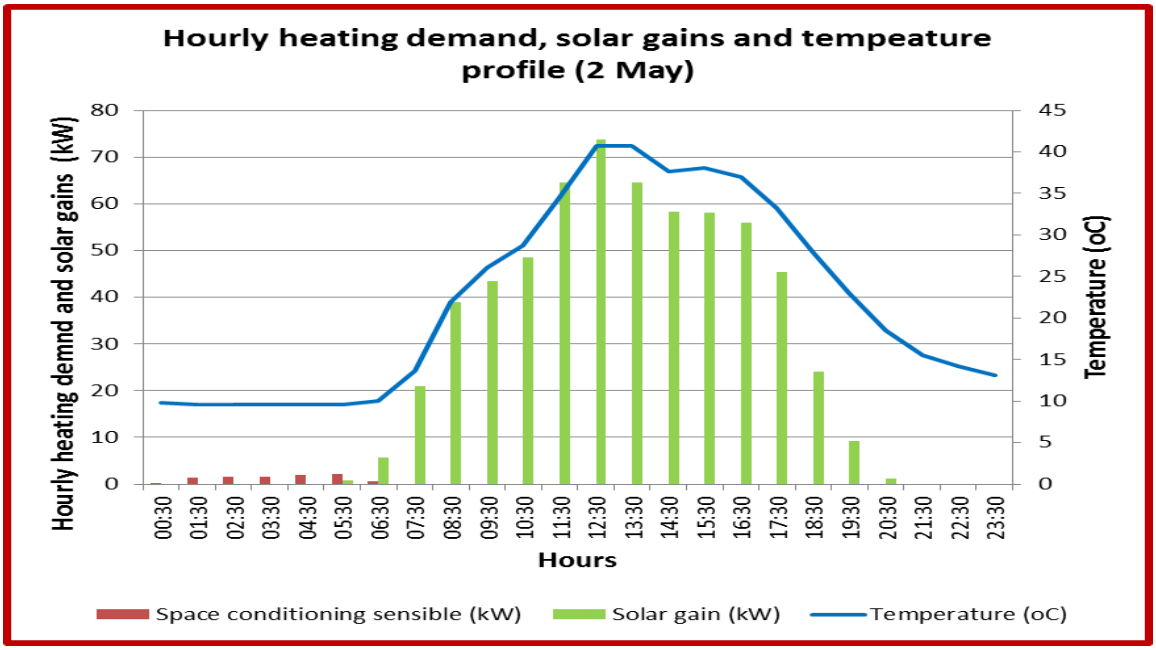


Appendix B2 Zone 4 Relative humidity validation

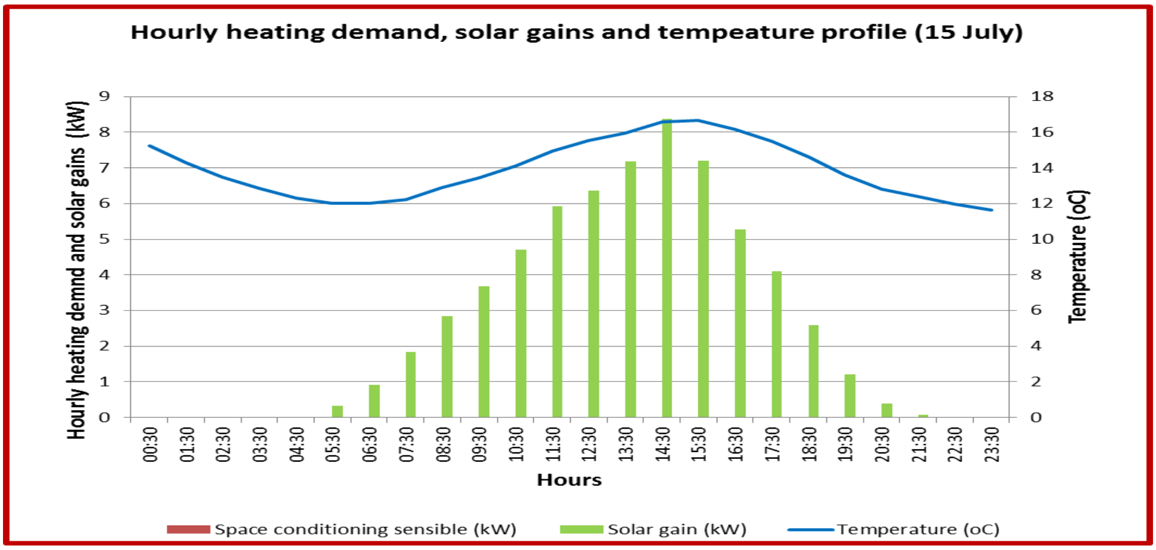
Appendix C Hourly heating demand, solar gains and temperature profile of zones



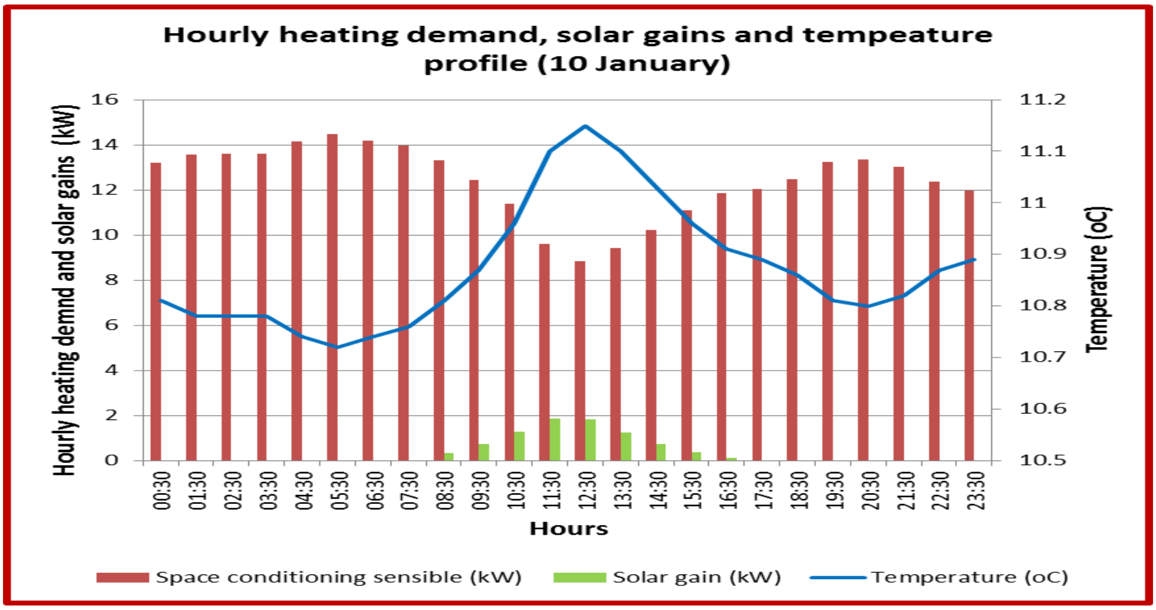
Appendix C1 Zone 1 hourly heating demand, solar gains and temperature profile with space temperature set point of 9 ^{o}C in 11 March



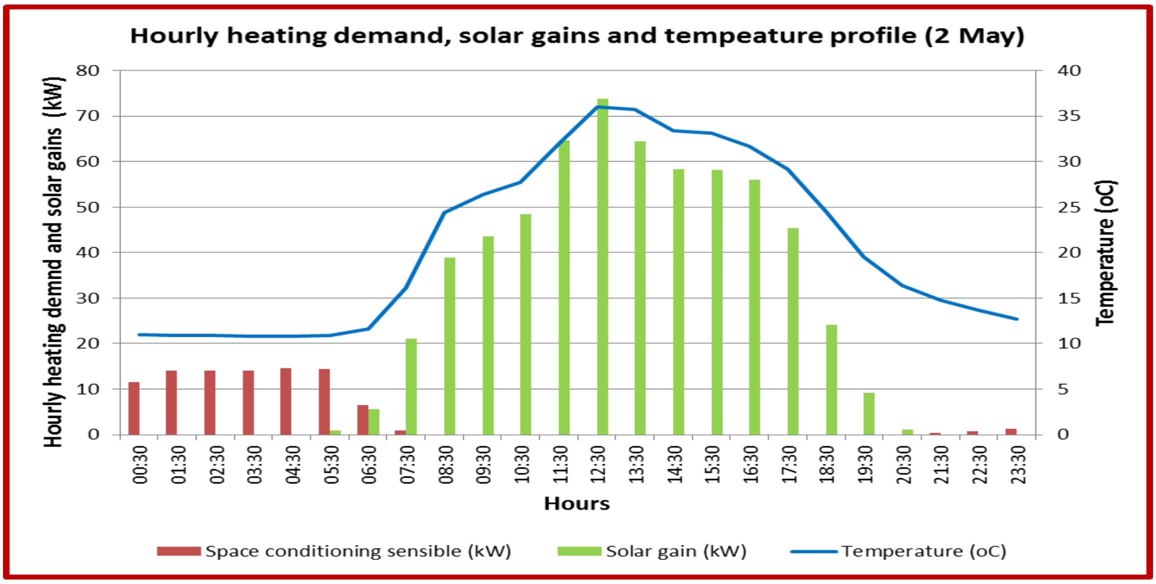
Appendix C2 Zone 1 hourly heating demand, solar gains and temperature profile with space temperature set point of 9 ^{o}C in 2 May



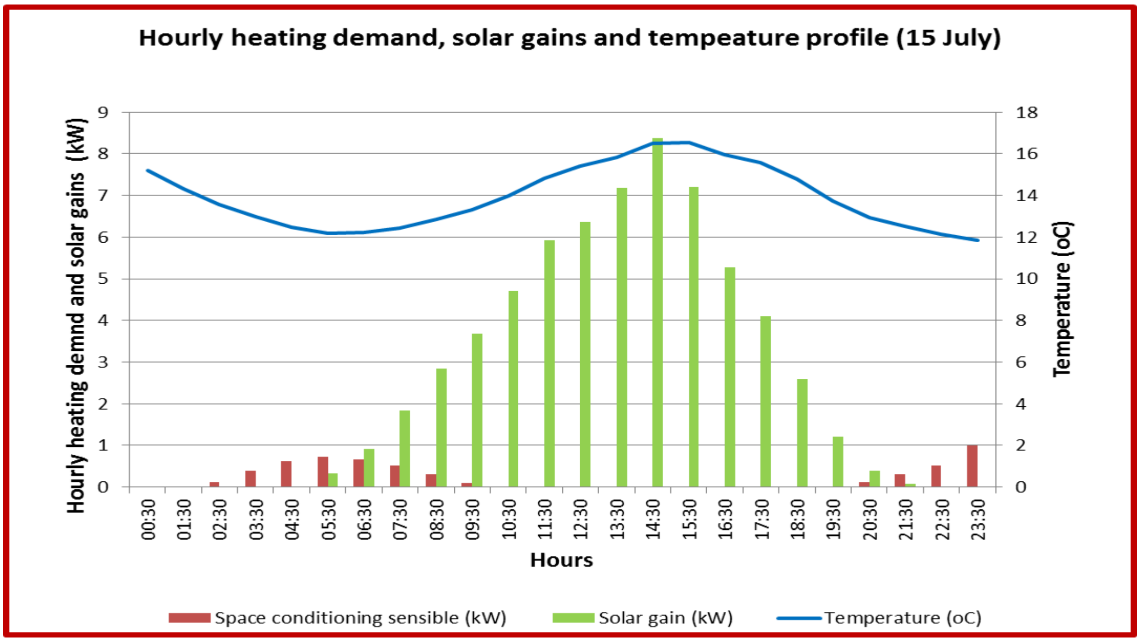
Appendix C3 Zone 1 hourly heating demand, solar gains and temperature profile with space temperature set point of 9 ^{o}C in 15 July



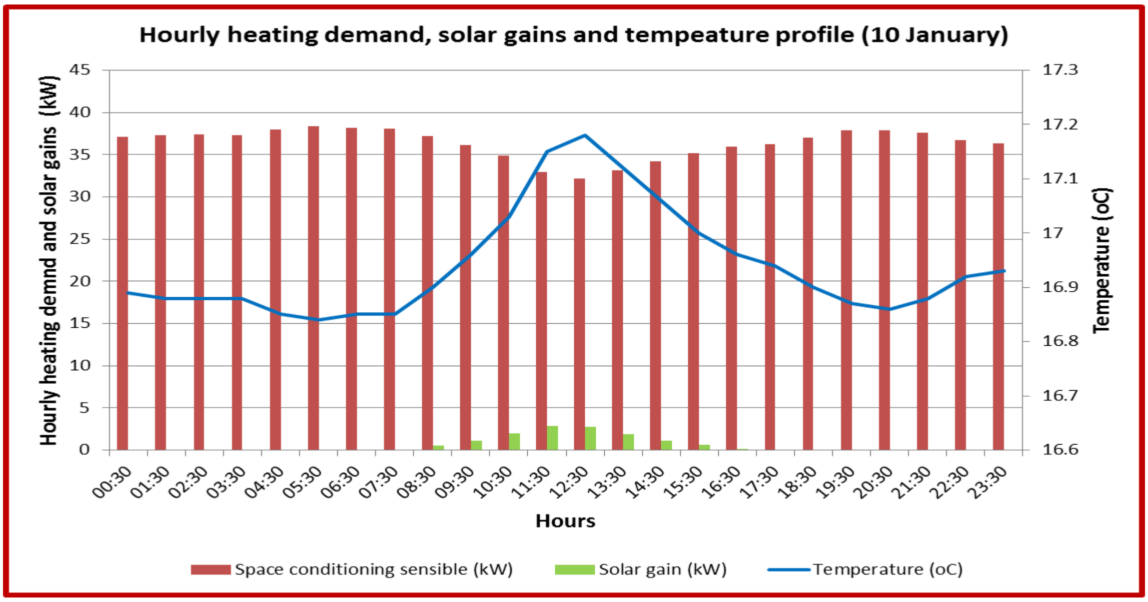
Appendix C4 Zone 6 hourly heating demand, solar gains and temperature profile with space temperature set point of 11 ^{o}C in 10 January



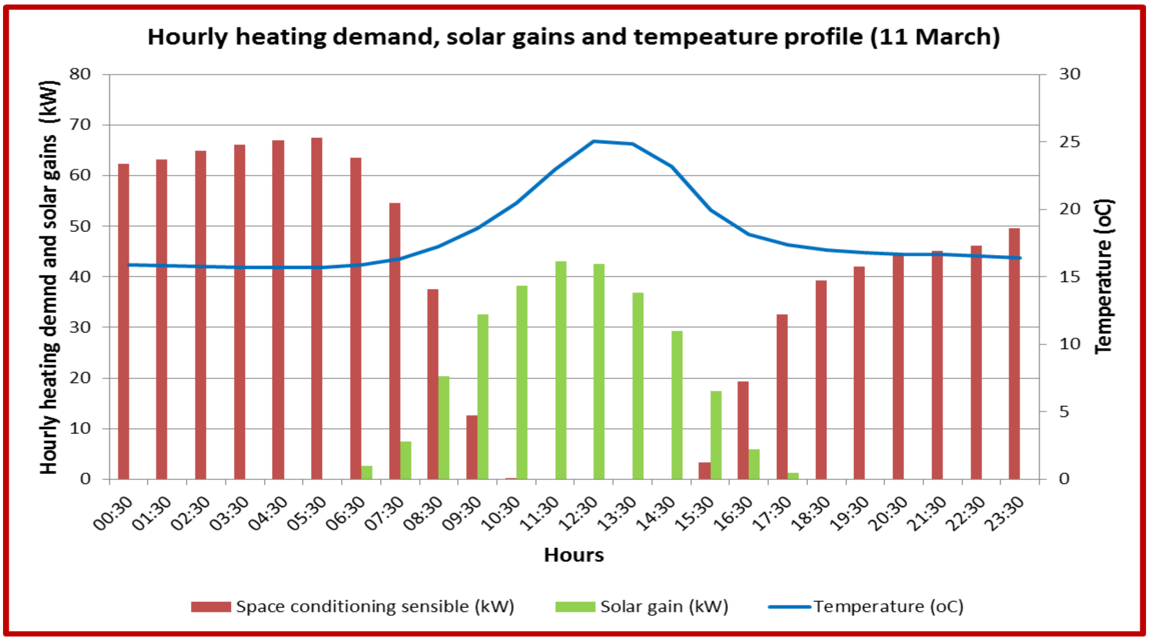
Appendix C5 Zone 6 hourly heating demand, solar gains and temperature profile with space temperature set point of 11 ^{o}C in 2 May



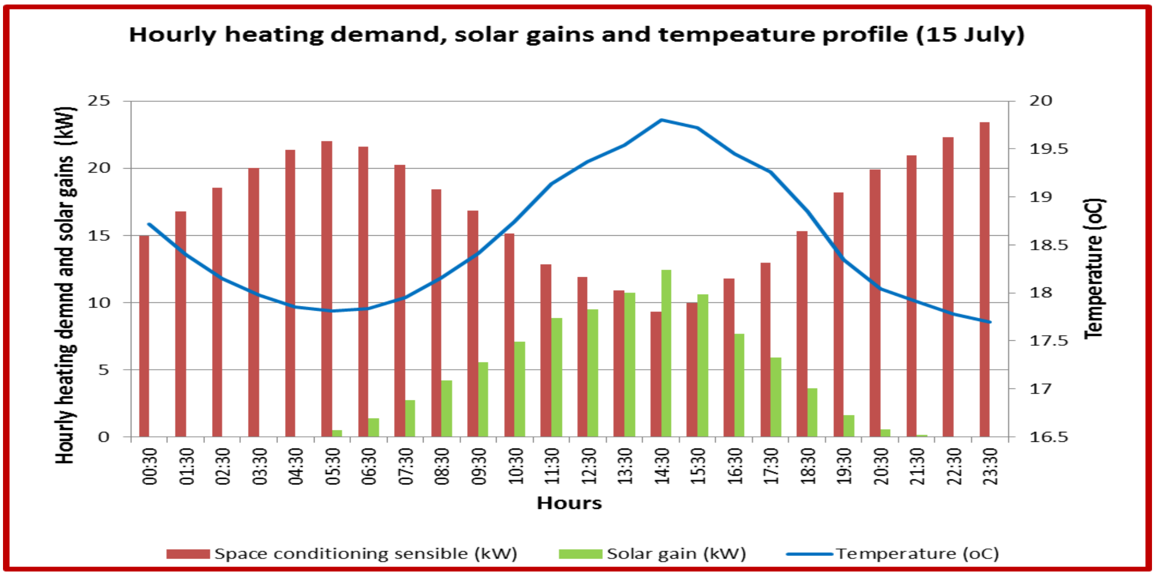
Appendix C6 Zone 6 hourly heating demand, solar gains and temperature profile with space temperature set point of 11 ^{o}C in 15 July



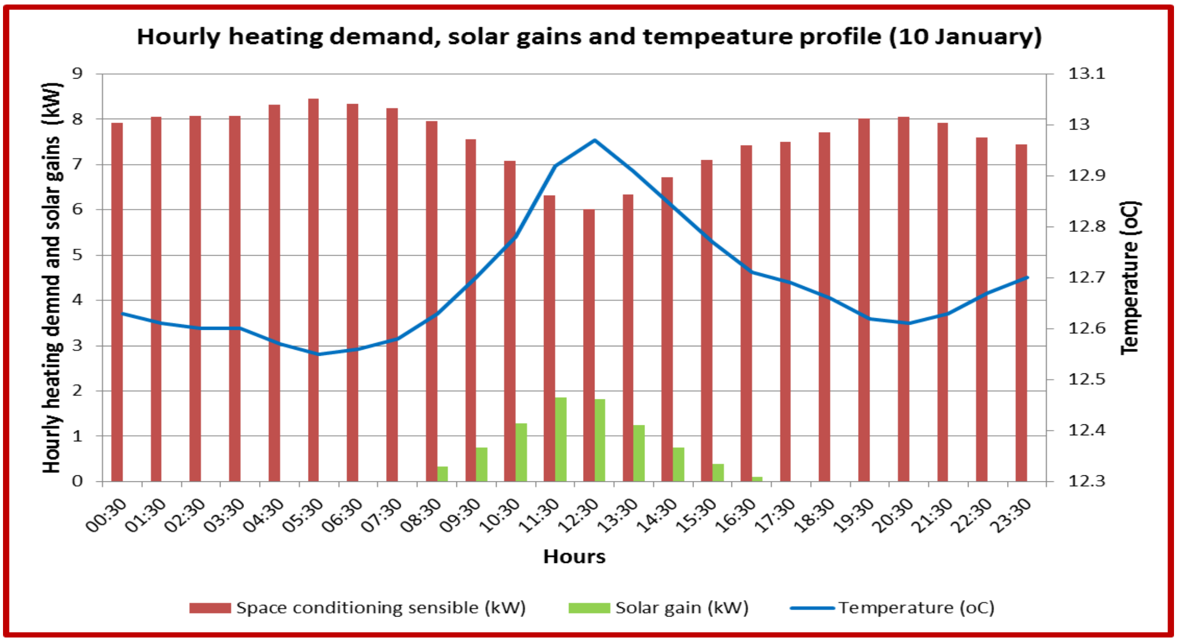
Appendix C7 Zone 9 hourly heating demand, solar gains and temperature profile with space temperature set point of 21 °C in 10 January



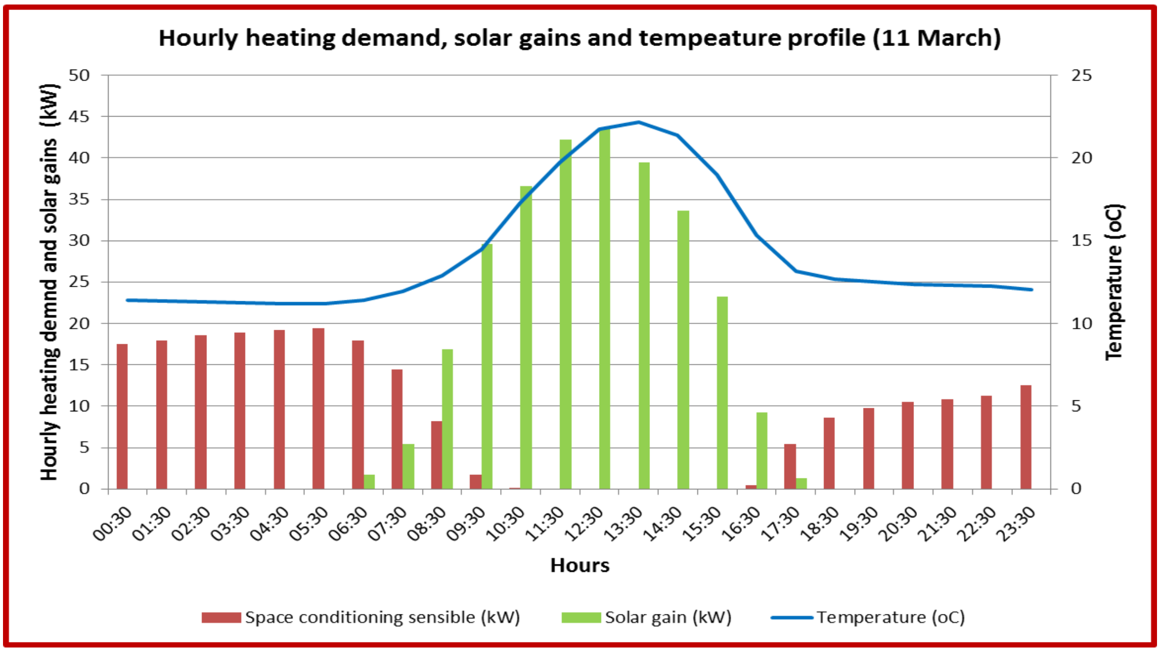
Appendix C8 Zone 9 hourly heating demand, solar gains and temperature profile with space temperature set point of 21 °C in 11 March



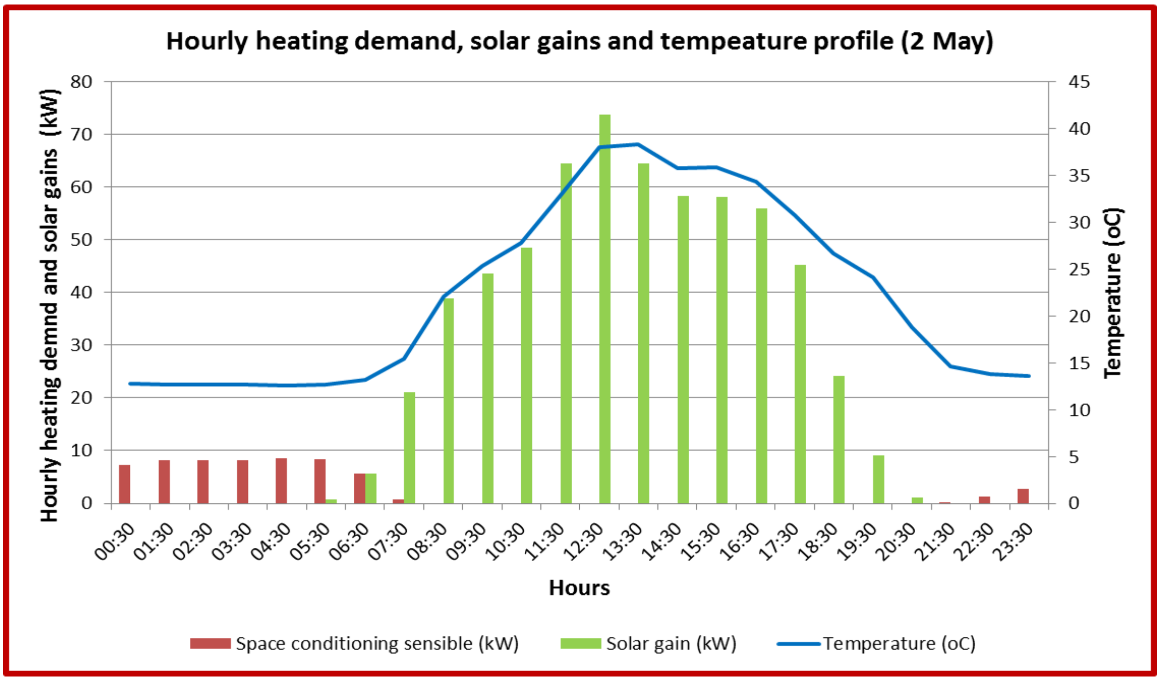
Appendix C9 Zone 9 hourly heating demand, solar gains and temperature profile with space temperature set point of 21 ^{o}C in 15 July



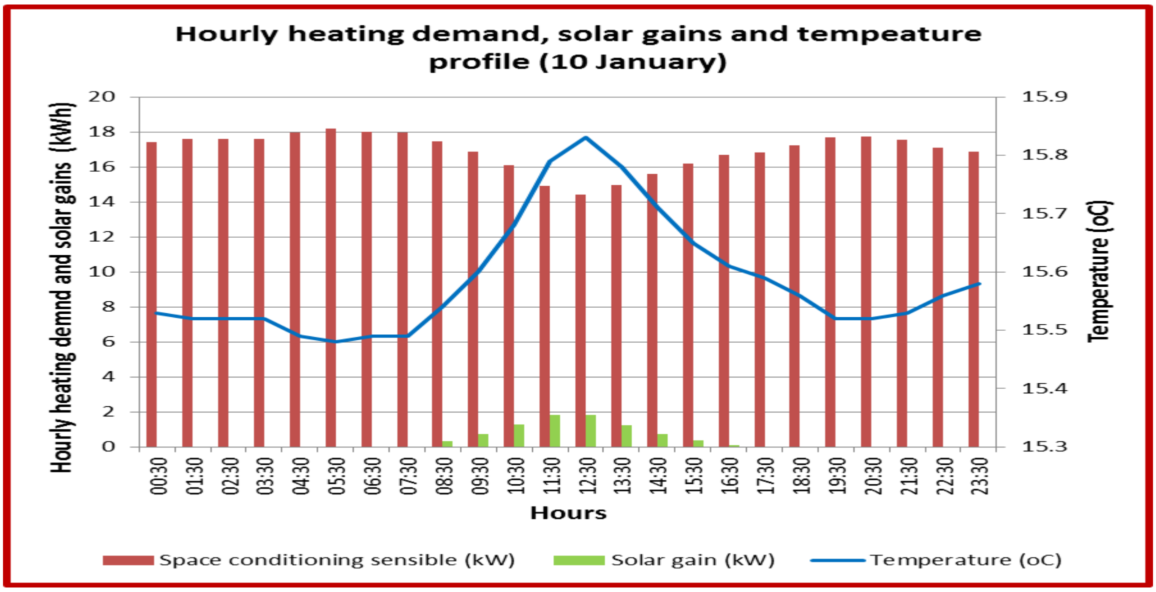
Appendix C10 Zone 15 hourly heating demand, solar gains and temperature profile with space temperature set point of 14 ^{o}C in 10 January



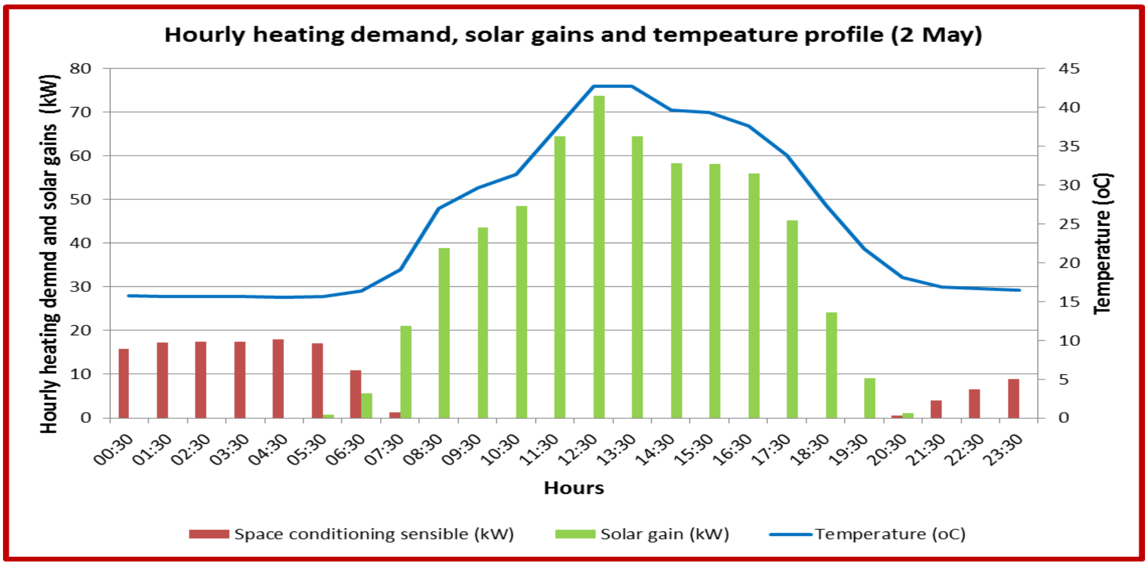
Appendix C11 Zone 15 hourly heating demand, solar gains and temperature profile with space temperature set point of 14 ^{o}C in 11March



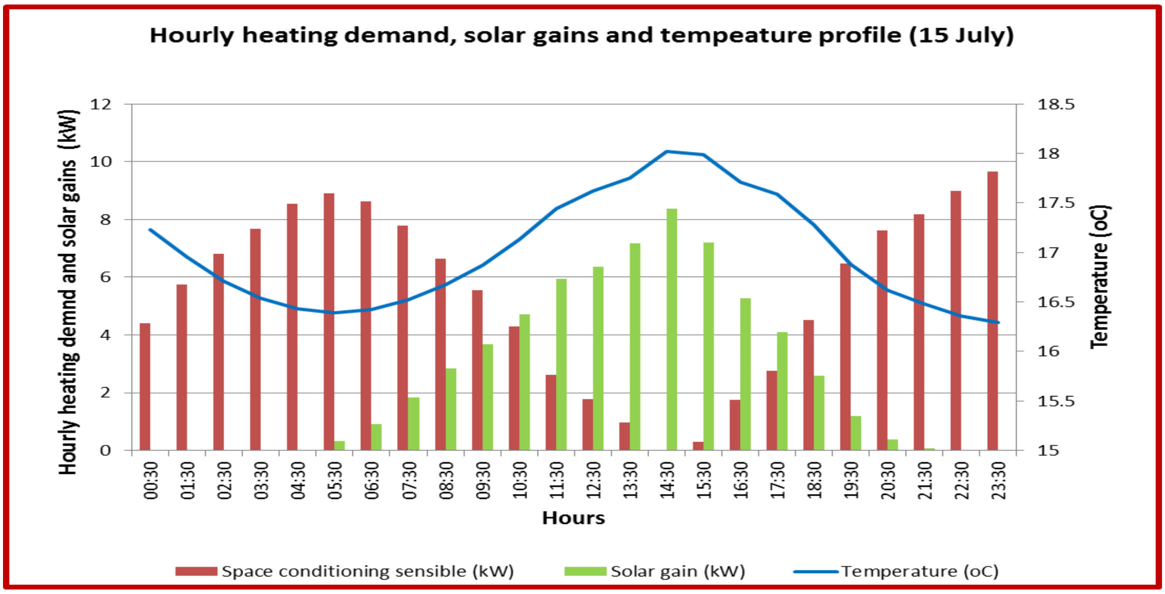
Appendix C12 Zone 15 hourly heating demand, solar gains and temperature profile with space temperature set point of 14 ^{o}C in 2 May



Appendix C13 Zone 17 hourly heating demand, solar gains and temperature profile with space temperature set point of 18 ^{o}C in 10 January



Appendix C14 Zone 17 hourly heating demand, solar gains and temperature profile with space temperature set point of 18 ^{o}C in 2 May



Appendix C15 Zone 17 hourly heating demand, solar gains and temperature profile with space temperature set point of 18 ^{o}C in 15 July

Appendix D Shows calculated daily heat transferred and energy absorbed by the PCM heating pipes from May to October

Appendix D1 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in May

	Days of the month																													
May	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	27	40	32		26	30	29	23	36	37	43	44	45	44	39	29	32	26	33	40	36	38	38	35	37	27	24		22		
Hours occurred	4	12	8		8	8	6	2	12	12	12	14	14	16	14	6	6	4	12	12	10	12	14	12	12	7	4		1		
PCM 22 oC																															
Heat transferred (kW) Eq 1	8	34	16		5	13	11	0	24	26	40	42	45	42	32	11	16	5	19	34	24	29	29	21	26	8	3		0		
Heat transferred (kW) Eq 2	5	32	16		5	11	11	0	24	24	37	40	40	40	29	11	16	5	16	32	24	26	26	21	24	5	3		0		
Heat energy stored (kWh) Eq 1	26	400	130		42	98	64	3	296	320	482	595	627	680	437	64	98	21	220	400	246	347	405	270	320	48	8		0		
Heat energy stored (kWh) Eq 2	26	365	122		40	93	58	3	273	296	439	540	569	617	400	58	90	21	204	365	228	318	373	249	296	45	8		0		
PCM 25 oC																															
Heat transferred (kW) Eq 1	3	26	11		0	8	5		19	19	32	34	37	34	24	5	11	0	13	24	19	21	21	16	19	3					
Heat transferred (kW) Eq 2	3	24	11		0	5	5		16	19	29	32	34	32	21	5	11	0	11	156	16	21	21	16	19	3					
Heat energy stored (kWh) Eq 1	8	312	82		8	53	29		214	238	389	487	516	556	336	29	61	3	143	312	177	262	304	191	238	16					
Heat energy stored (kWh) Eq 2	8	288	77		8	50	29		198	222	357	445	474	508	310	29	58	3	135	288	167	243	283	177	222	13					

Appendix D2 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in June

	Days of the month																													
Jun	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Mean temp		41	36	46	36	37	42			43	42	24	30	45	32	26	35	29	40	44	28	26	26	28	23	39	41	47	30	35
Hours occurred		12	10	14	8	12	12			14	14	6	12	14	10	6	12	10	14	14	12	6	8	8	4	14	14	14	12	14
PCM 22 oC																														
Heat transferred (kW) Eq 1		34	24	48	24	26	37			40	37	3	13	45	16	5	21	11	34	42	8	5	5	8	0	32	34	50	13	21
Heat transferred (kW) Eq 2		32	24	42	24	24	34			37	34	3	11	40	16	5	21	11	32	40	8	5	5	8	0	29	32	45	11	21
Heat energy stored (kWh) Eq 1		429	246	659	196	320	455			561	529	13	148	627	164	32	270	106	468	595	103	32	42	69	3	437	497	693	148	315
Heat energy stored (kWh) Eq 2		389	228	598	183	296	415			511	484	11	140	569	151	29	249	98	426	540	98	29	40	66	3	400	455	627	140	291
PCM 25 oC																														
Heat transferred (kW) Eq 1		29	19	40	19	19	29			32	29		8	37	11	0	16	5	26	34	3	0	0	3		24	29	42	8	16
Heat transferred (kW) Eq 2		26	16	37	16	19	29			29	29		5	34	11	0	16	5	24	32	3	0	0	3		21	26	37	5	16
Heat energy stored (kWh) Eq 1		339	177	548	143	238	363			455	423		79	516	101	5	191	50	365	487	42	5	8	29		336	394	580	79	222
Heat energy stored (kWh) Eq 2		312	167	500	132	222	333			418	392		77	474	95	5	177	48	336	445	40	5	8	26		310	363	529	77	206

Appendix D3 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in July

	Days of the month																														
Jul	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	34	30	30	29	28	25	30	30	28	29	24	22	31	36		37	24	33	34	27	28	35	35	39	34	27	26	26	30	27	38
Hours occurred	14	10	10	10	10	4	10	12	10	8	2	2	10	12		10	6	10	12	8	10	10	6	12	10	8	8	8	10	6	12
PCM 22 oC																															
Heat transferred (kW) Eq 1	21	13	13	11	8	3	13	13	8	11	3	0	13	24		26	3	19	21	8	8	21	21	32	21	8	5	5	13	8	29
Heat transferred (kW) Eq 2	19	11	11	11	8	3	11	11	8	11	3	0	11	24		24	3	16	19	5	8	21	21	29	19	5	5	5	11	5	26
Heat energy stored (kWh) Eq 1	286	124	124	106	87	13	124	148	87	85	5	0	124	296		267	13	183	243	56	87	225	135	373	204	56	42	42	124	42	347
Heat energy stored (kWh) Eq 2	265	116	116	98	82	13	116	140	82	79	5	0	116	273		246	13	169	228	53	82	209	124	341	188	53	40	40	116	40	318
PCM 25 oC																															
Heat transferred (kW) Eq 1	13	8	8	5	3	0	8	8	3	5			8	19		19		13	13	3	3	16	16	24	13	3	0	0	8	3	21
Heat transferred (kW) Eq 2	13	5	5	5	3	0	5	5	3	5			5	16		19		11	13	3	3	13	16	21	13	3	0	0	5	3	21
Heat energy stored (kWh) Eq 1	193	66	66	50	34	0	66	79	34	40			66	214		198		119	167	16	34	159	95	286	138	16	8	8	66	13	262
Heat energy stored (kWh) Eq 2	183	64	64	48	34	0	64	77	34	40			64	198		185		114	156	16	34	148	90	265	130	16	8	8	64	13	243

Appendix D4 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in August

	Days of the month																														
Aug	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	39		34	33	30	34	36	33	30	27	40	36	34	29	40	38	39	34	31	39	24	29	30	24	29	26	35	27	33	33	29
Hours occurred	12		12	12	12	10	12	10	6	8	12	12	12	6	10	12	12	12	10	12	4	10	10	2	8	4	10	8	10	10	8
PCM 22 oC																															
Heat transferred (kW) Eq 1	32		21	19	13	21	24	19	13	8	34	24	21	11	34	29	32	21	13	32	3	11	13	3	11	5	21	8	19	19	11
Heat transferred (kW) Eq 2	29		19	16	11	19	24	16	11	5	32	24	19	11	32	26	29	19	13	29	3	11	11	3	11	5	21	5	16	16	11
Heat energy stored (kWh) Eq 1	373		243	220	148	204	296	183	74	56	400	296	243	64	333	347	373	243	143	373	8	106	124	5	85	21	225	56	183	183	85
Heat energy stored (kWh) Eq 2	341		228	204	140	188	273	169	69	53	365	273	228	58	304	318	341	228	135	341	8	98	116	5	79	21	209	53	169	169	79
PCM 25 oC																															
Heat transferred (kW) Eq 1	24		13	13	8	13	19	13	8	3	26	19	13	5	26	21	24	13	8	24		5	8		5	0	16	3	13	13	5
Heat transferred (kW) Eq 2	21		13	11	5	13	16	11	5	3	24	16	13	5	24	21	21	13	8	21		5	5		5	0	16	3	11	11	5
Heat energy stored (kWh) Eq 1	286		167	143	79	138	212	119	40	16	312	212	167	29	259	262	286	167	85	286		50	66		40	3	159	16	119	119	40
Heat energy stored (kWh) Eq 2	265		156	135	77	130	198	114	37	16	288	198	156	29	241	243	265	156	79	265		48	64		40	3	148	16	114	114	40

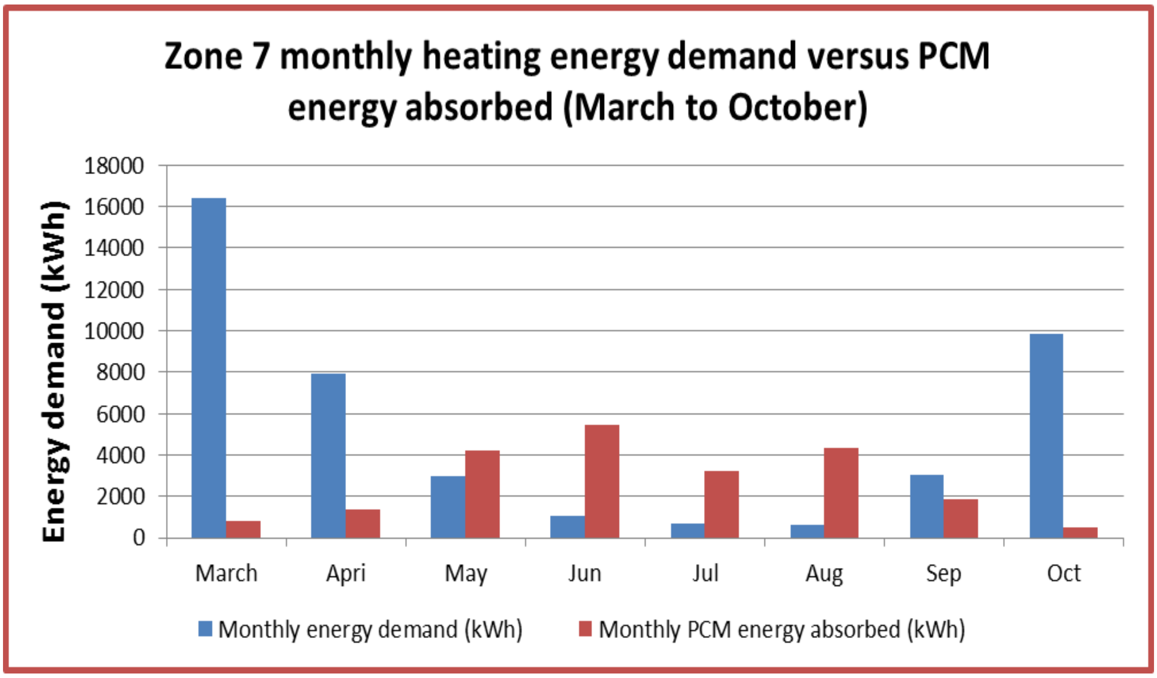
Appendix D5 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in September

	Days of the month																												
Sep	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Mean temp	24	27		32	38	26	31		29	23	27	29	25	30	27		22	24	33	33	34	35	34	26	24	25	26	28	30	
Hours occurred	8	6		10	8	4	8		8	6	4	6	6	8	6		2	4	10	10	10	8	8	6	4	6	8	6	6	
PCM 22 oC																														
Heat transferred (kW) Eq 1	3	8		16	29	5	13		11	0	8	11	3	13	8		0	3	19	19	21	21	21	5	3	3	5	8	13	
Heat transferred (kW) Eq 2	3	5		16	26	5	13		11	0	5	11	3	11	5		0	3	16	16	19	21	19	5	3	3	5	8	11	
Heat energy stored (kWh) Eq 1	19	42		164	233	21	114		85	5	26	64	21	98	42		0	8	183	183	204	180	164	32	8	21	42	53	74	
Heat energy stored (kWh) Eq 2	16	40		151	212	21	106		79	5	26	58	21	93	40		0	8	169	169	188	167	151	29	8	21	40	50	69	
PCM 25 oC																														
Heat transferred (kW) Eq 1		3		11	21	0	8		5		3	5	0	8	3				13	13	13	16	13	0		0	0	3	8	
Heat transferred (kW) Eq 2		3		11	21	0	8		5		3	5	0	5	3				11	11	13	16	13	0		0	0	3	5	
Heat energy stored (kWh) Eq 1		13		101	175	3	66		40		8	29	0	53	13				119	119	138	127	111	5		0	8	21	40	
Heat energy stored (kWh) Eq 2		13		95	161	3	64		40		8	29	0	50	13				114	114	130	119	103	5		0	8	21	37	

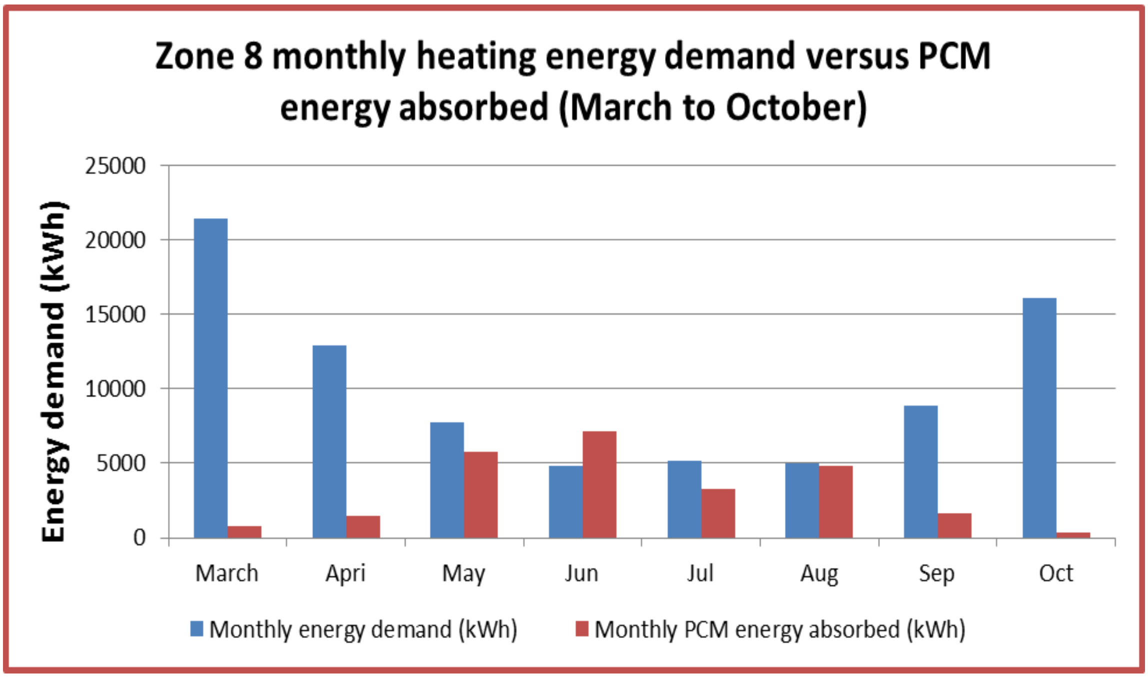
Appendix D6 shows calculated daily heat transferred and energy absorbed by the PCM heating pipes in October

	Days of the month																														
Oct	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Mean temp	25	28	28	29	29	29	23	25		23		25	27			25	24	26		26				22	22	23					
Hours occurred	6	6	8	6	6	6	2	6		1		2	6			2	2	4		6				2	1	4					
PCM 22 oC																															
Heat transferred (kW) Eq 1	3	8	8	11	11	11	0	3		0		3	8			3	3	5		5				0	0	0					
Heat transferred (kW) Eq 2	3	8	8	11	11	11	0	3		0		3	5			3	3	5		5				0	0	0					
Heat energy stored (kWh) Eq 1	21	53	69	61	61	61	3	21		0		8	42			8	5	21		32				0	0	3					
Heat energy stored (kWh) Eq 2	21	50	66	58	58	58	3	21		0		8	40			8	5	21		29				0	0	3					
PCM 25 oC																															
Heat transferred (kW) Eq 1	0	3	3	5	5	5		0				0	3			0		0		0											
Heat transferred (kW) Eq 2	0	3	3	5	5	5		0				0	3			0		0		0											
Heat energy stored (kWh) Eq 1	0	21	29	29	29	29		0				0	13			0		3		5											
Heat energy stored (kWh) Eq 2	0	21	26	29	29	29		0				0	13			0		3		5											

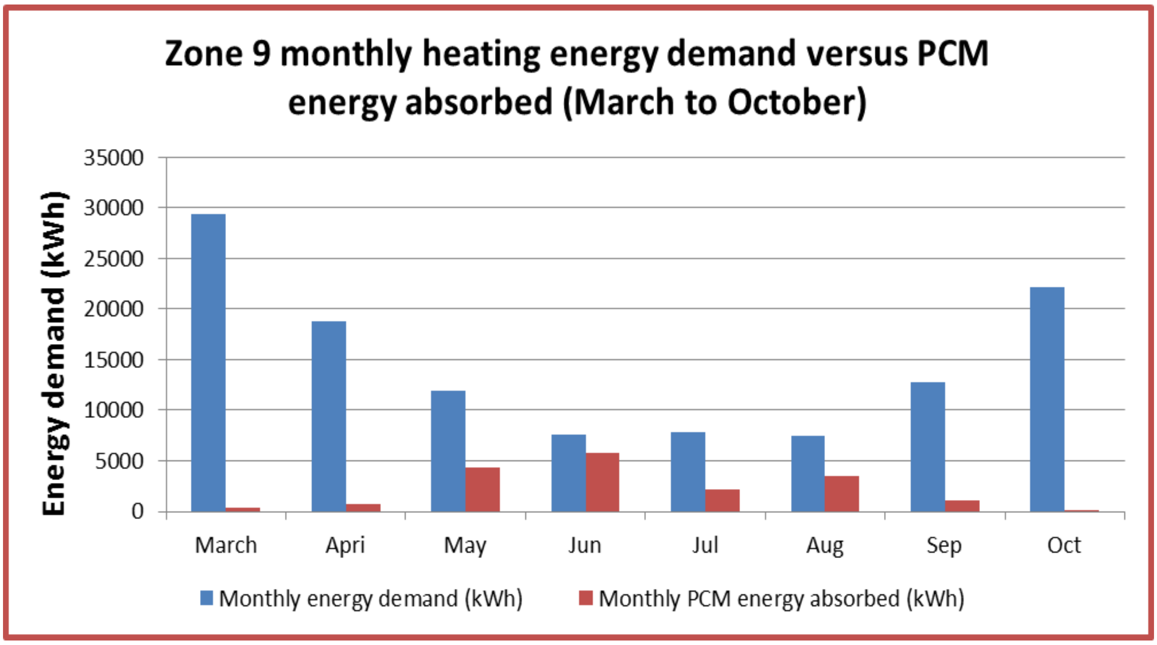
Appendix E Monthly heating energy demand against PCM heating pipes energy absorbed from March to October of the design year



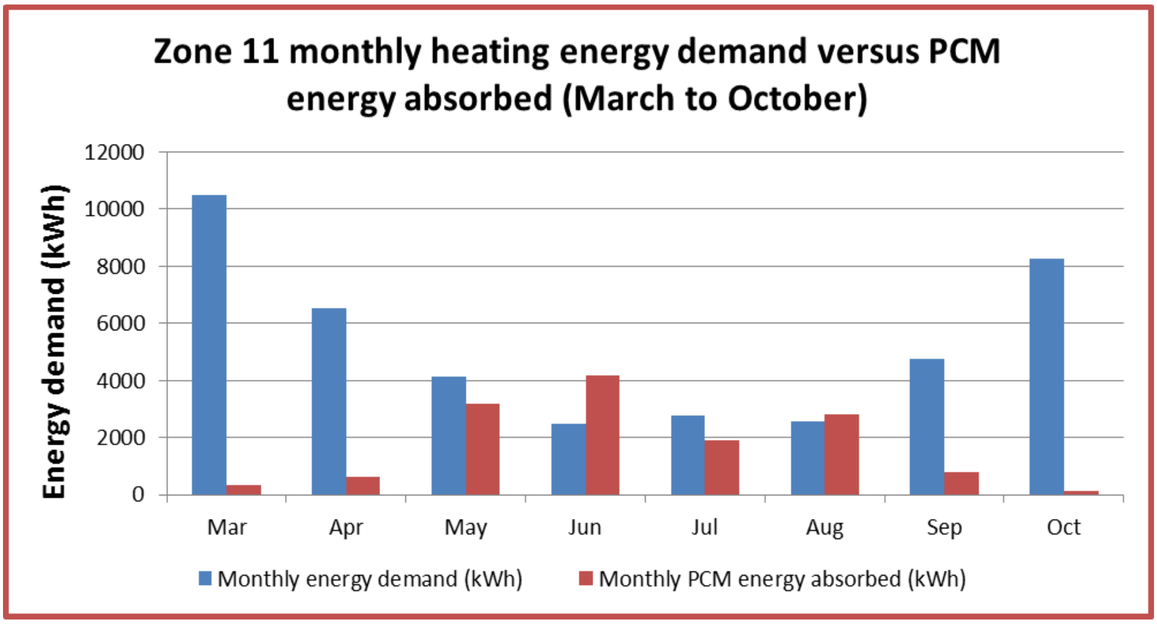
Appendix E1 Monthly heating energy demand versus PCM heating pipe absorbed energy



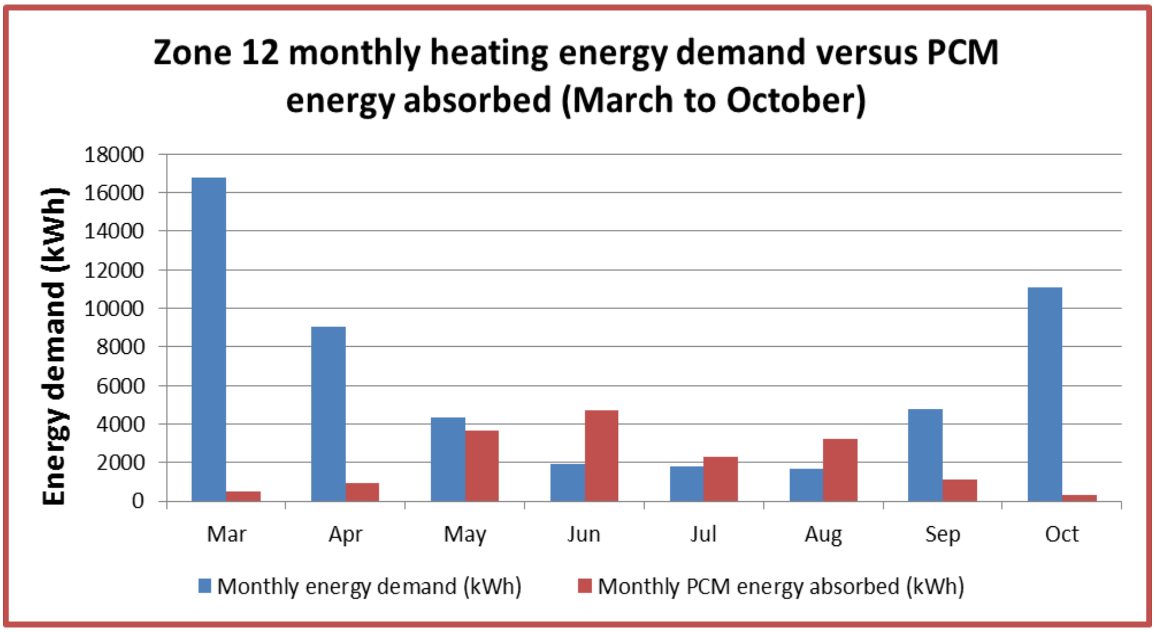
Appendix E2 Monthly heating energy demand versus PCM heating pipe absorbed energy



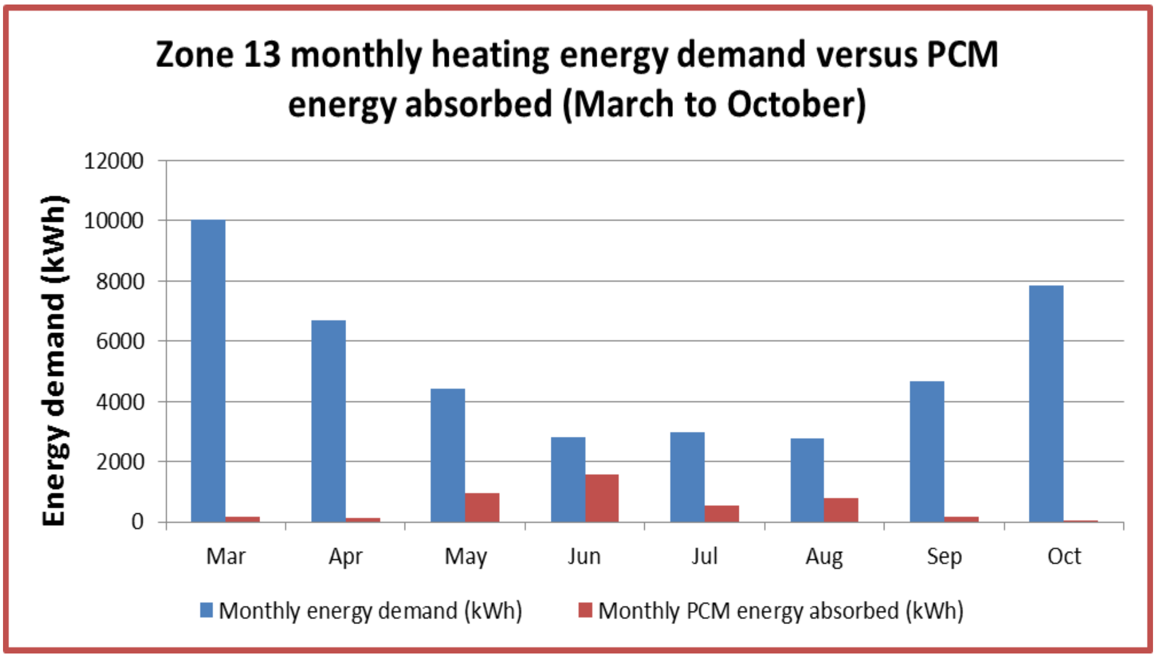
Appendix E3 Monthly heating energy demand versus PCM heating pipe absorbed energy



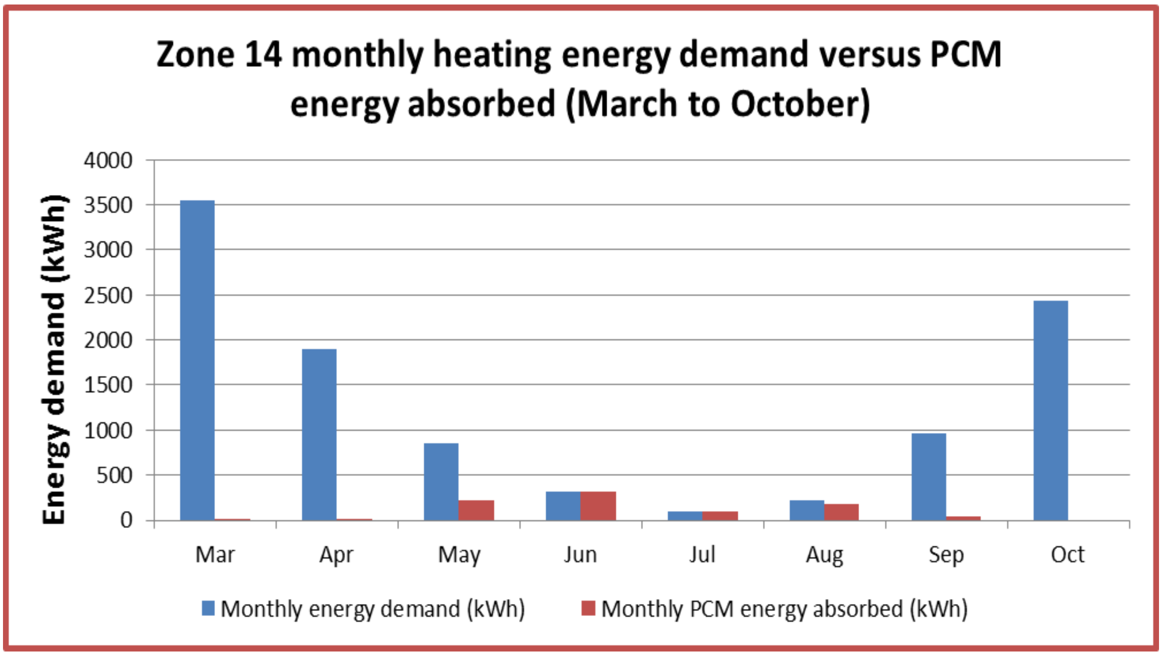
Appendix E4 Monthly heating energy demand versus PCM heating pipe absorbed energy



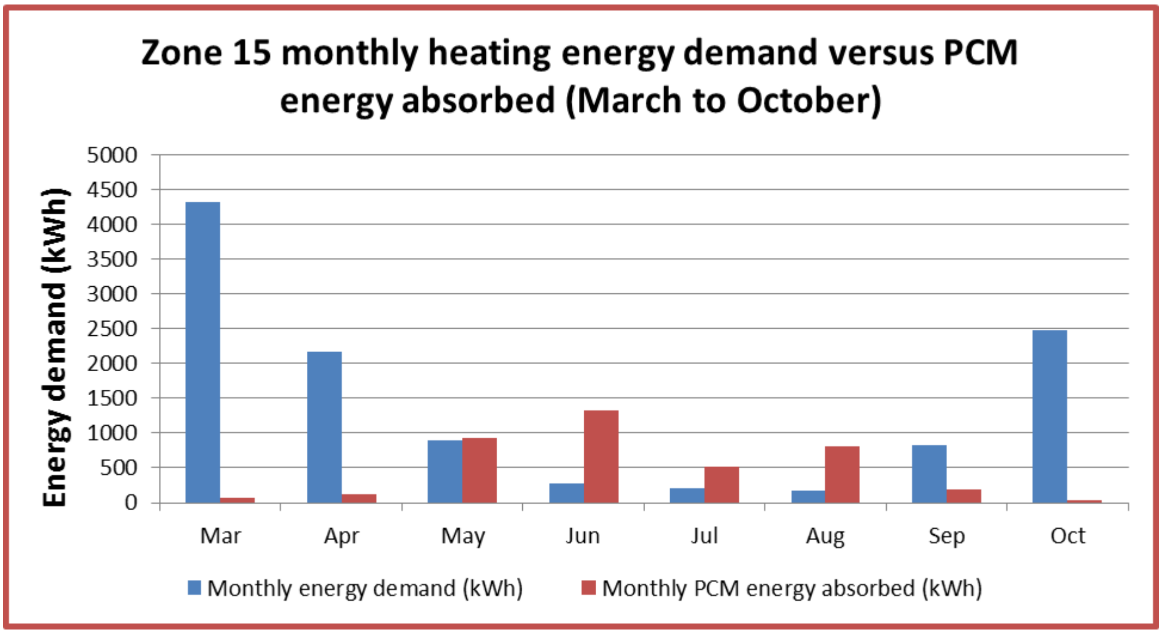
Appendix E5 Monthly heating energy demand versus PCM heating pipe absorbed energy



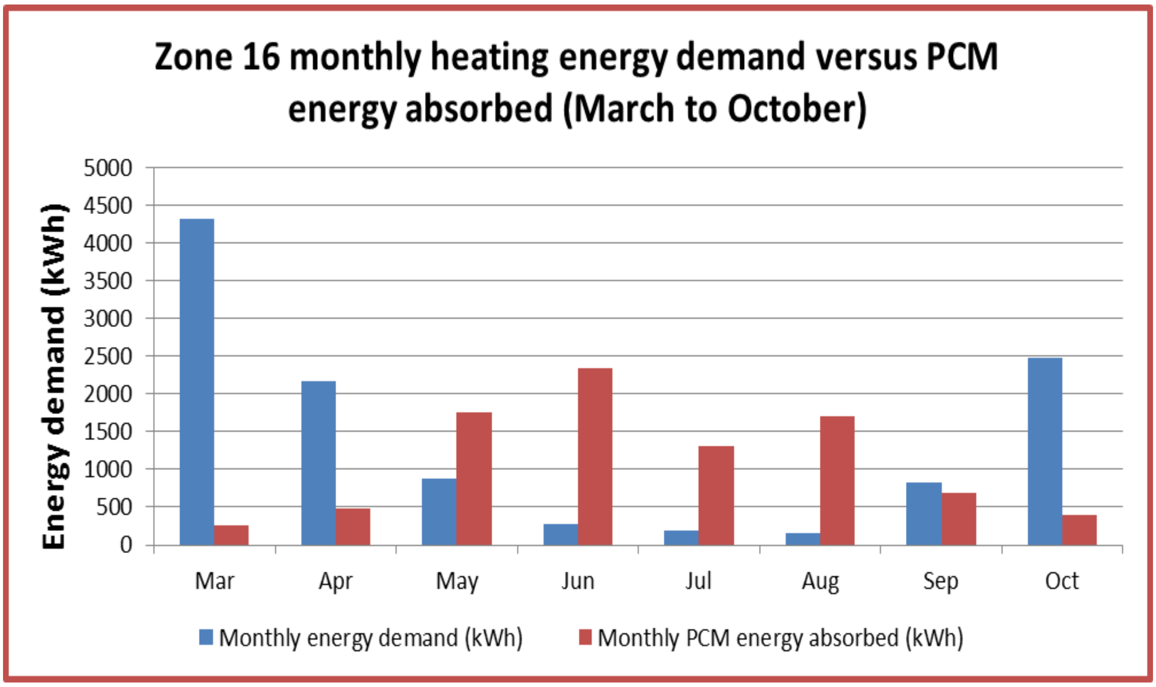
Appendix E6 Monthly heating energy demand versus PCM heating pipe absorbed energy



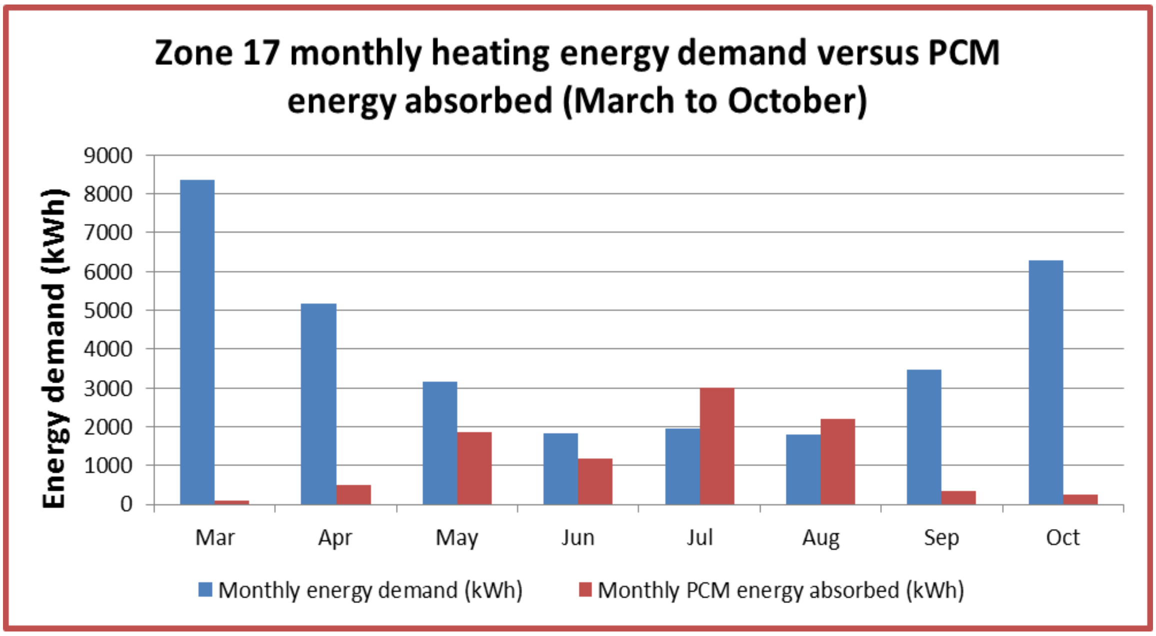
Appendix E7 Monthly heating energy demand versus PCM heating pipe absorbed energy



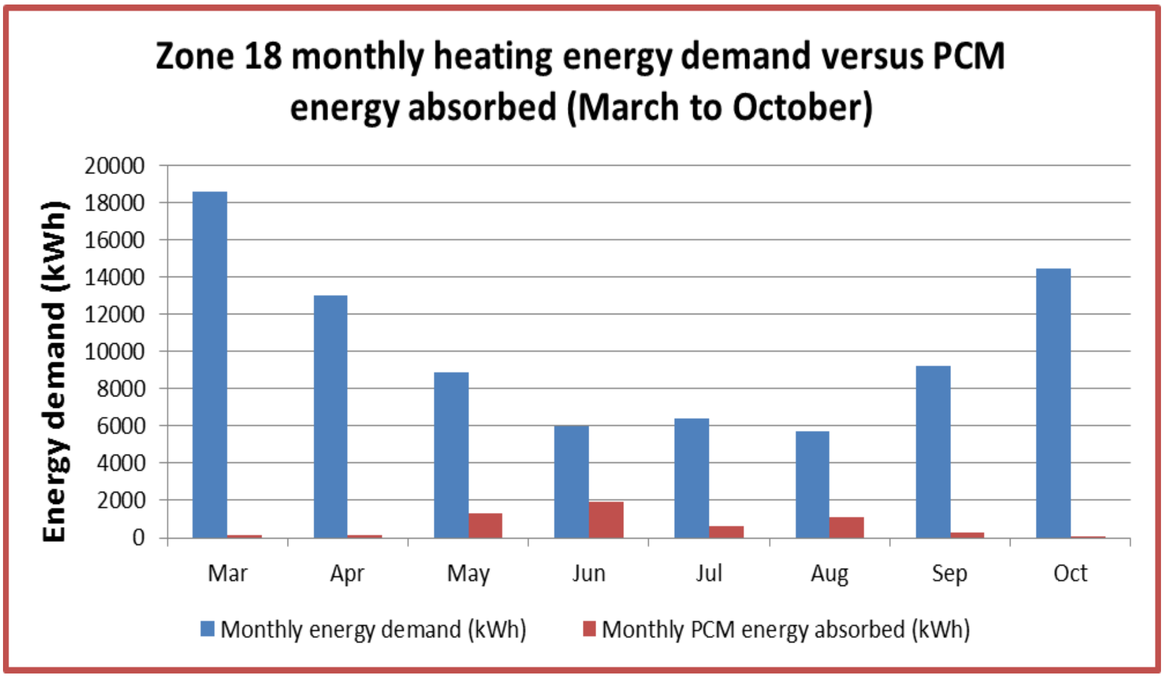
Appendix E8 Monthly heating energy demand versus PCM heating pipe absorbed energy



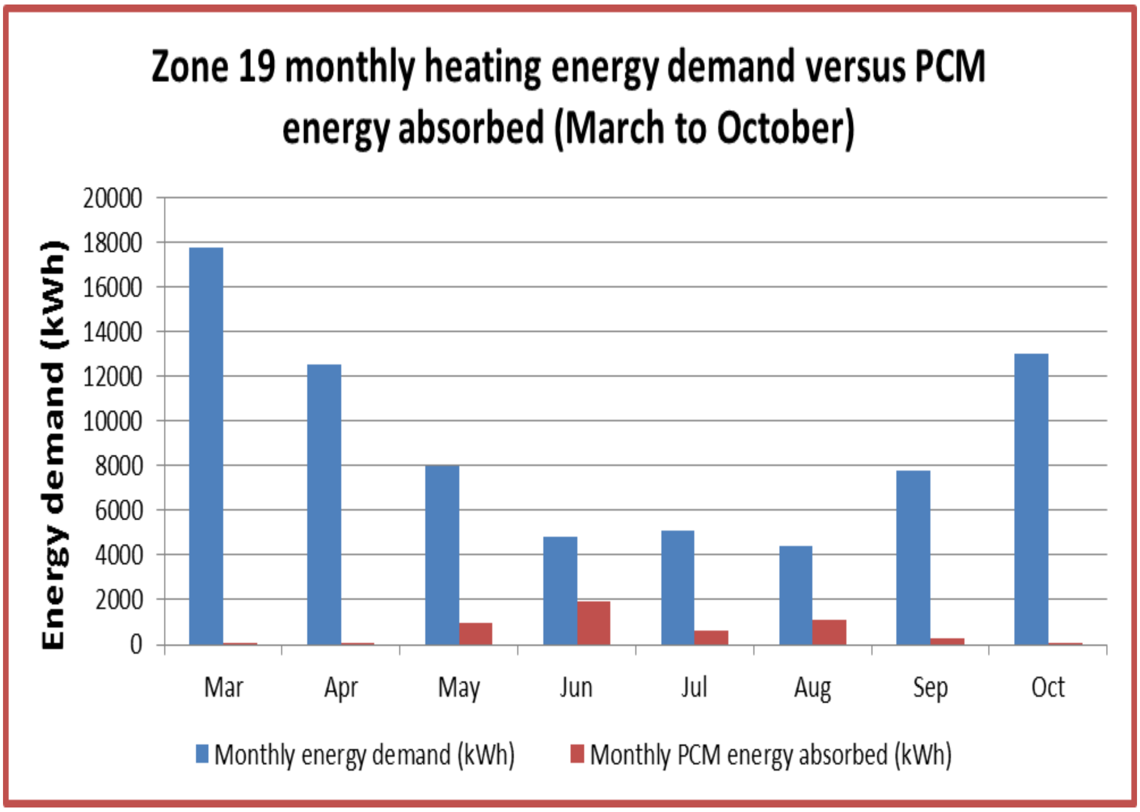
Appendix E9 Monthly heating energy demand versus PCM heating pipe absorbed energy



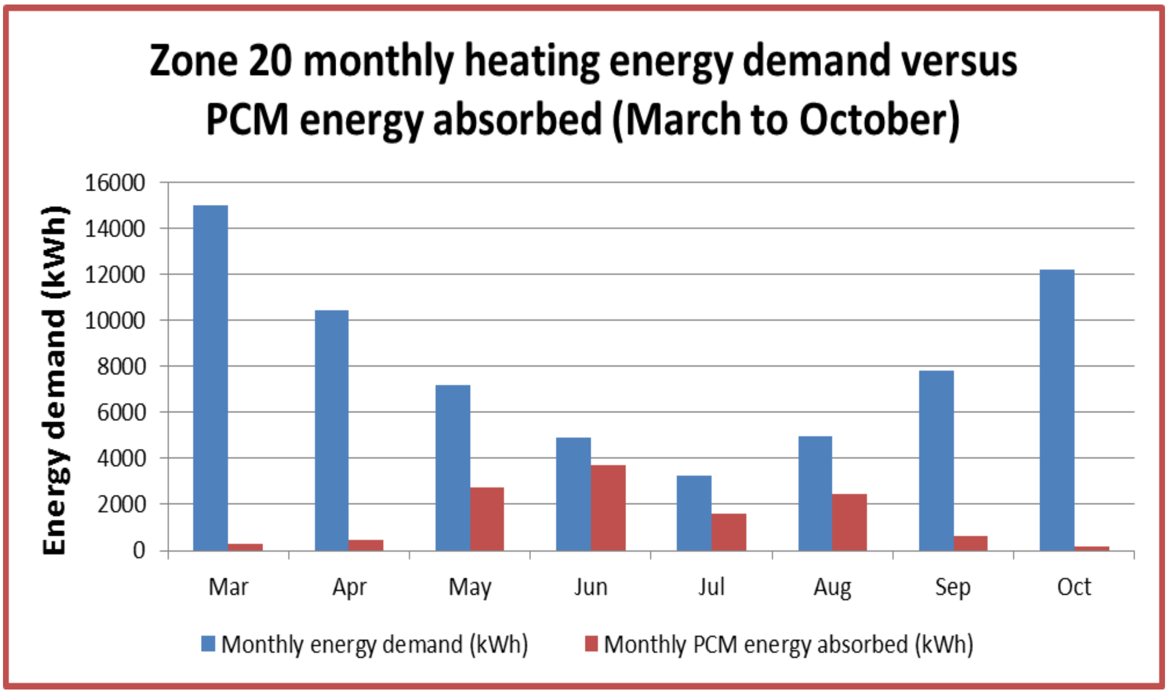
Appendix E10 Monthly heating energy demand versus PCM heating pipe absorbed energy



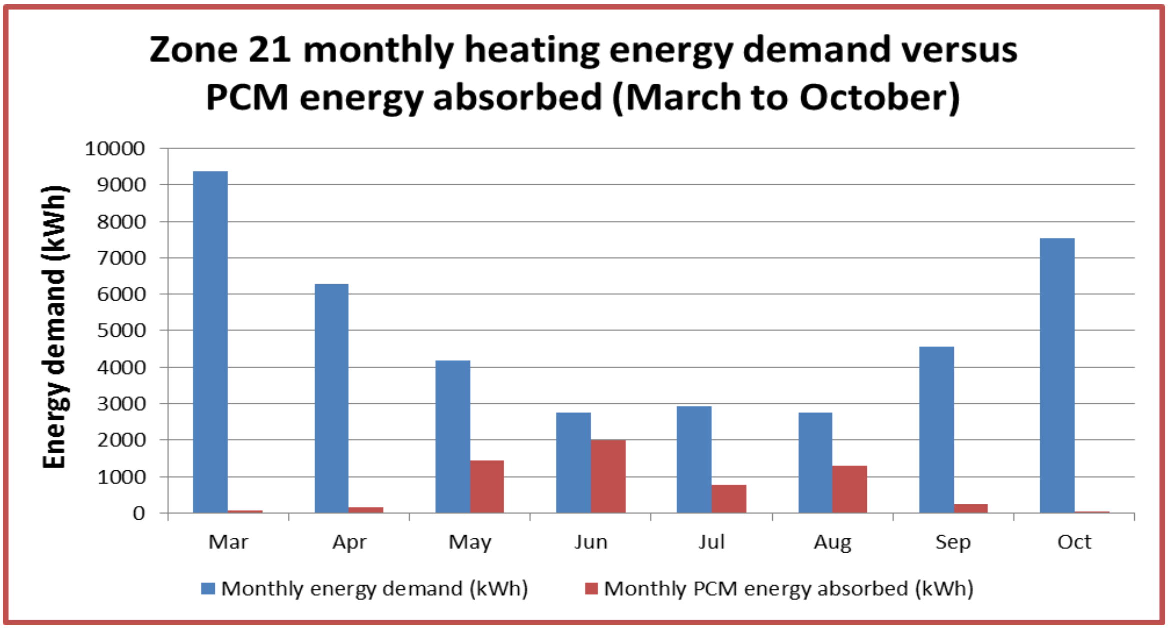
Appendix E11 Monthly heating energy demand versus PCM heating pipe absorbed energy



Appendix E12 Monthly heating energy demand versus PCM heating pipe absorbed energy



Appendix E13 Monthly heating energy demand versus PCM heating pipe absorbed energy



Appendix E14 Monthly heating energy demand versus PCM heating pipe absorbed energy

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