FACULTY OF ENGINEERING

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DEVELOPMENT OF INNOVATIVE HEATING, COOLING AND AIR CONDITIONING (HVAC) TECHNOLOGIES FOR LOW-CARBON POULTRY HOUSES

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To my Parents

Hatice and Recep Gurler

To my siblings

Esra Gurler Sezer and Elif Kubra Gurler

ABSTRACT

The agri-food sector accounts directly or indirectly for around 30% of total energy consumption all over the world, with a particularly high contribution from the poultry industry, due to the high protein ratio associated with chicken meat and egg production. Over recent decades, chicken meat production has dramatically increased approximately tenfold, and this strong growth is projected to continue steadily based on the FAO 2015–2025 outlook. In this context, the potential of decreasing the energy usage of the poultry sector with efficient and cost-effective environmental control systems and various kinds of technologies is clearly important. This study evaluates potential energy saving using solar-based systems, heat pump units, and evaporative cooling applied to poultry house, tested based on correlative literature and laboratory investigation. The main aim of this study is to develop and adopt sustainable heating, ventilation, and air conditioning (HVAC) technologies integrated with novel components in poultry farms to increase the welfare of chickens and decrease the level of energy consumption in this form of agricultural production.

This study investigates novel technologies and their use in poultry farms to provide clean and cheap energy. There are three novel systems presented and investigated in this study: (1) a photovoltaic/thermal assisted ground source heat pump (PVT-GSHP) field study for poultry house sustainable heating; (2) a psychrometric energy core (PEC) direct cooling unit field study; and (3) a novel core-based dew point cooling unit for cooling demand (laboratory testing). PEC is a common cooling method worldwide, therefore in

addition to this PEC field trial on the poultry farm, a dew point evaporative cooler was designed and tested in laboratory conditions for application in poultry farms' HVAC systems in the UK, to decrease heat stress and death issues in poultry farms.

The PVT-GSHP system has novel Solar thin-tube Polyethylene heat exchangers (PHE) and copper vertical ground heat exchangers (VGHE) attached to a heat pump ducted with a buffer vessel. A full quantitative validation was made, and each innovative component was assessed under hourly (short-term) transient conditions. The system was also experimentally investigated under long-term transient condition (>1 year). The dynamic behaviours of the whole integrated system and each novel component were investigated. The COP of the whole system design was monitored over the long term and fluctuated between 2.05 and 5.17, with a maximum value of 5.17 in July, and a minimum of 2.05 in January. The seasonal COP of the system was 3.73. The simple payback period of the hybrid heating system is 8.14 years, with £4,890 total annual savings with government revenues under the condition of efficient heat pump running hours.

The novel PVT-GSHP design and components are relatively cheap compared to existing heat pump and solar systems, and the validated results indicate that the PEC cooling unit and the novel and cheap PVT-GSHP integrated designs are reliable systems for agricultural applications in terms of their dynamic behaviour.

PUBLISHED WORK

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.

TABLE OF CONTENTS

Abstract i

Published work iii

Acknowledgements iv

Table of contents v

List of tables xii

List of figures xiv

Chapter 1 Introduction and Research Summary 1

1.1. Introduction 1

1.2. Research background information 2

1.3. Aims and objectives of the study 5

1.4. Research methodology 7

1.5. Research innovation 10

1.6. Outline of the thesis 10

Chapter 2 Systematic Review of Proposed Thermal Regulation Systems and Recent Applications in Poultry Houses 21

2.1. Poultry houses 21

2.1.1. Internal air quality and lighting 23

2.1.2. Air movement and ventilation management 27

2.1.3. Air temperature and moisture content 31

2.1.4. Evaluation on poultry houses 33

2.1.5. Retrofit technologies on heating, cooling and lighting system 34

2.1.5.1. Evaporative cooling in poultry houses 35

2.1.5.2. Innovative Heating Systems in Poultry Houses 38

2.2. Systematic review of state-of-the-art evaporative cooling systems technologies 40
2.2.1. Direct evaporative cooling systems 43
2.2.2. Indirect evaporative cooling systems 44
2.2.3. Direct-indirect evaporative cooling systems 46
2.2.4. Dew-Point evaporative cooling mechanism – M Cycle 47
2.2.5. The focus of recent studies 54
2.2.5.1. Evaporative Cooling Pad Materials and Design 54
2.2.5.2. Dehumidification and Desiccant-Based Systems 59
2.3. Systematic review of hybrid PVT assisted with HP systems 63
2.3.1. PV panels and photovoltaic technology 66
2.3.2. Photovoltaic thermal (PVT) hybrid systems 69
2.3.3. Heat pump technology 72
2.3.4. The focus of recent studies 81
Chapter 3 Heating and Ventilation Performance of a UK Poultry house: air flow pattern 86
3.1. Introduction 86
3.2. Poultry house in Newark, UK 87
3.2.1. Set point temperature in the poultry house 90
3.2.2. Monitoring of the chicken production cycles (PCs) 91
3.3. Modelling the thermal regulation energy demand of the poultry house 93
3.3.1. Simulation of the poultry house in HAP software and Design Builder 95
3.3.1.1. Construction templates 95

3.3.1.2. HVAC templates 96
3.3.2. Mathematical model of the thermal regulation load of the poultry house in MATLAB 98
3.3.2.1. General parameters 98
3.3.2.2. Net heat gain and loss (Q_{gain}, Q_{loss}) 99
3.3.3. Results and discussion 104
3.3.3.1. HAP Software 104
3.3.3.2. Design Builder 106
3.3.3.3. MATLAB model 111
3.4. Poultry house ventilation management and airflow distribution 114
3.4.1. CFD analysis of airflow in poultry house during winter 114
3.4.2. CFD analysis of airflow in poultry house during summer 118
3.5. Summary 120
Chapter 4 Evaluation of an Innovative PVT-GSHP System in Poultry House 123
4.1. Introduction 123
4.2. System design and sub-model description 124
4.2.1. PVT–Ground Array model 128
4.2.1.1. PVT energy balance equations 131
4.2.1.2. Ground Array 137
4.2.2. Heat pump 140
4.2.3. Buffer vessel 142
4.3. Novel PVT-GSHP system description and poultry house set-up 142
4.3.1. System design 142
4.3.2. Photovoltaic thermal (PVT) array 145

4.3.3. Ground pipe array 146
4.3.4. Heat pump-buffer tank-fan coil unit 147
4.4. Novel PVT-GSHP system and daily monitoring 148
4.4.1. Instrumentation and Uncertainty 148
4.4.1.1. Instrumentation 148
4.4.1.2. Uncertainty Analysis 150
4.4.2. Daily monitoring of the novel PVT-GSHP 151
4.4.2.1. PVT collector and Ground array 152
4.4.2.2. Buffer Tank- Fan coil 153
4.5. Results and discussions 153
4.5.1. Temperature and energy from PVT and ground array 153
4.5.1.1. PVT Array 153
4.5.1.2. Ground array 155
4.5.2. Temperature and energy from novel PVT-GSHP and COPs . 157
4.6. Summary 158
Chapter 5 Performance Assessment of Novel PVT-GSHP: Long Term Monitoring 161
5.1. Introduction 161
5.2. Methodology 163
5.2.1. Chicken production cycles 164
5.2.2. PVT-GSHP system design 165
5.2.3. Experimental mathematical procedure 168
5.3. Results and discussions 170
5.3.1. Performance evaluation of the PVT-GSHP system during peak heating seasons 170

5.3.1.1. PVT collector 172
5.3.1.2. Heat pump-buffer tank-fan coil..... 174
5.3.2. Performance evaluation of the PVT-GSHP system during the heating season (middle of the heating) 176
5.3.2.1. PVT and Ground array..... 176
5.3.2.2. Heat pump-buffer tank-fan coil..... 177
5.4. Further experimental observation of the PVT-GSHP 178
5.5. Summary..... 179
Chapter 6 Techno-Economic and Environmental Assessment of Novel PVT-GSHP ……182
6.1. Introduction 182
6.2. Methodology 183
6.2.1. System design and measurements 183
6.2.2. Economic and environmental investigation of PVT-GSHP 185
6.2.2.1. Economic Assessment 185
6.2.2.2. Environmental assessment..... 187
6.3. Results and discussion 188
6.3.1. Temperature variations and energy from PVT -GSHP 188
6.3.1.1. PVT Array 188
6.3.1.2. Ground array..... 190
6.3.1.3. Heat pump-buffer tank-fan coil..... 192
6.3.2. Economic and environmental investigation of PVT-GSHP 199
6.3.2.1. Economic assessment..... 200
6.3.2.2. Environment assessment 202
6.4. Summary..... 203

Chapter 7 Performance Assessment of Air-Cooling Systems for Low Carbon Poultry Houses..... 206
7.1. Introduction 206
7.2. Methodology 207
7.2.1. System design and experimental setup..... 207
7.2.1.1. Direct evaporative cooling system design and setup 207
7.2.1.2. Dew point evaporative cooling system rig description and lab set-up ……209
7.2.2. Mathematical analysis 213
7.2.2.1. Mathematical analysis of direct evaporative cooler..... 213
7.2.2.2. Mathematical analysis of dew points evaporative cooler . 214
7.3. Results and discussion 216
7.3.1.1. Direct evaporative cooling..... 216
7.3.1.2. Dew point evaporative cooling system..... 219
7.4. Summary..... 226
Chapter 8 Conclusions and Future Research 228
8.1. Conclusions 228
8.1.1. Current HVAC systems' air distribution is inadequate 229
8.1.2. Thermal regulation energy requirements..... 230
8.1.3. Evaporative cooling 231
8.1.4. Renewable energy technology implications of PVT-GSHP 232
8.1.4.1. Heat pump 232
8.1.4.2. Application locations and installation 232
8.1.5. Academic contributions of validated experimental results 234
8.2. Future research directions 235

References..... 238

Appendix A 261

LIST OF TABLES

Table 2-1 The production and consumption of broilers in selected countries (USDA 2016)..... 22

Table 2-2:Chicken themal and air quality design conditions(Fawaz et al. 2014) 26

Table 2-3 Maximum ventilation rates (MVR)(Defra 2005)..... 31

Table 2-4 Case studies and model samples on evaporative cooling improvements in poultry houses 37

Table 2-5 Case studies and model samples on innovative heating systems in poultry houses..... 39

Table 3-1: House dimensions, floor area, and bird numbers 88

Table 3-2: Annual Production Cycles (PCs)..... 88

Table 3-3 The temperature and ventilation requirement in the house..... 90

Table 3-4 Basic structural component element..... 94

Table 3-5: Set points within the model (six cycles p.a.) 97

Table 3-6 The heating demand per week for each poultry production cycle 105

Table 4-1: Main loops in the innovative heating system on the cold source 126

Table 4-2: Solar optical parameters of the PV 129

Table 4-3: Specifications of PHE mat 129

Table 4-4 The properties of PV layer 130

Table 4-5 Physical properties of soil (Y. Cui and Zhu 2018)..... 139

Table 4-6 The heat pump standards..... 140

Table 4-7:Specifications of the heat pump 141

Table 4-8: Technical information of each component in the innovative heating system 144

Table 4-9: Instrumentation method 149

Table 4-10 Relative uncertainty of the components 150

Table 4-11: Flow rate specifications..... 152

Table 5-1: PC 1 and PC 2 tests 164

Table 6-1: Monthly averaged results from Solar 190

Table 6-2: Monthly averaged results from Ground Array 192

Table 6-3: Monthly energy from whole system..... 196

Table 6-4: Annual chicken production cycle, 2016-2019..... 196

Table 6-5: Annual heat production under real condition (-2017) 198

Table 6-6: Monthly averaged heat from Solar and Ground 199

Table 6-7: Heating system parts cost..... 201

Table 6-8: PVT-GSHP heating system saving cost (The current design condition) 202

Table 7-1: Maximum output with defined temperature increase and airflow (Eco-Cooling Manual) 216

Table 7-2: Testing results at steady state condition 220

Table 7-3: Testing results at steady state condition 222

Table 7-4: Testing results at steady state condition 224

LIST OF FIGURES

Figure 1-1(a) The optimum temperature (b) Relative humidity range for broilers and pullets (Jones, Donnelly, and Dawkins 2005)..... 4

Figure 1-2:Research methodology in the thesis..... 8

Figure 2-1: Air emissions from livestock 24

Figure 2-2:Effects on ventilation on egg production (Ruzal et al. 2011)..... 28

Figure 2-3: Chicken Chicken distribution in higher velocity zones under heat stress (Youssef, Exadaktylos, and Berckmans 2015) 30

Figure 2-4: Thermal Thermoneutral zone diagram (Defra 2005)..... 33

Figure 2-5 I Innovative applications and evaluations 34

Figure 2-6 CO_2 emissions aimed by region in 2040 (International Energy Agency 2015)..... 40

Figure 2-7 (a) Drip-type direct evaporative cooling system (b) Psychrometric graph of the system (Xuan et al. 2012a) 44

Figure 2-8 (a) Tubular IEC system, (b) Plate IEC system, (c) Heat pipe IEC system, (d) IEC system psychrometric chart (Xuan et al. 2012a) 46

Figure 2-9 Types of evaporative air conditioners (a) Direct, (b) Indirect, (c) Two-stage combined (Bruno 2009)..... 47

Figure 2-10 Psychrometric charts of (a) A one-stage DEC system and (b) A two-stage IEC + DEC system (Xuan et al. 2012b) 47

Figure 2-11 One--unit channel pair of the novel dew-point evaporative air cooler schemas (X. Cui et al. 2014) 50

Figure 2-12 (a) modified counter-flow HMX, (b) regenerative HMX, (c) regenerative HMX with perforation, (d) cross-flow HMX, (e) modified cross-flow HMX (Anisimov and Pandelidis 2014) 52

Figure 2-13 (a) 3D scheme of M Cycle: 1 – primary air stream; 2 – working air dry channel; 3 – working air wet channel. (b) Photograph of an HMX (Pandelidis and Anisimov 2015)..... 53

Figure 2-14 Fr uit weight, dry matter and yield (gm) in celdek pad (cp), straw pad (sp), sliced wood pad (swp) and outside (Ahmed et al. 2011)..... 56

Figure 2-15 The five pad materials (Xuan et al. 2012a) 57

Figure 2-16 (a) a palash fibre pad; (b) a coconut fibre pad (Jain and Hindoliya 2011)..... 58

Figure 2-17 A liquid-desiccant dehumidification air system schema (Mei and Dai 2008) 59

Figure 2-18 Desiccant dehumidification and regeneration process (Nayak 2005)..... 60

Figure 2-19 (a) Conventional cycle and system proposed configurations, (b) COP results (Elgendy, Mostafa, and Fatouh 2015)..... 62

Figure 2-20 The types of PV panel in accordance to its raw material (Gouda et al. 2019)..... 67

Figure 2-21 The variation of the Radiation on the PV panel (Ramanujam et al. 2016)..... 67

Figure 2-22 The variation of the temperature on the PV (Luque and Hegedus 2011)..... 68

Figure 2-23: Heat pipe sections (Zhang et al. 2013) 71

Figure 2-24 PHE based based PVT system(Buker, Mempouo, and Riffat 2015) 72
Figure 2-25 GSHP system (J. Luo et al. 2015) 75
Figure 2-26 UK-HP sales from 2005 to 2013 (Hannon 2015) 76
Figure 2-27 (a)New innovative type 3I heat exchanger mesh perspective its Borehole Heat Exchanger inside (X. Liu et al. 2015) 77
Figure 2-28 Water circulation in the hybrid system (Di Perna et al. 2015) ... 78
Figure 2-29Three different integrated Solar and heat pump concepts (Bertram, Pärisch, and Tepe 2012)..... 81
Figure 2-30 Integrated solar air composite heat source heat pump system (Y. Liu et al. 2016) 84
Figure 2-31 System configuration (Winteler et al. 2014) 85
Figure 3-1: Poultry house (S1) (a) Exterior, (b) Interior..... 89
Figure 3-2: Inside and outside house temperature (PC 6) 92
Figure 3-3: Inside and outside house temperatures (PC 1) 93
Figure 3-4: Ambient temperature of Nottingham, UK..... 97
Figure 3-5 Annual energy requirement in the poultry house 106
Figure 3-6: Digital model of the JWL poultry facility 107
Figure 3-7: Comparison of facility temperatures and energy loads over the winter day 24-hour peak period..... 109
Figure 3-8: Comparison of facility temperatures and energy loads over the summer day 24-hour peak period 110
Figure 3-9: Average temperatures and minimum ventilation rates (MVR) for the full 7-week PC..... 112
Figure 3-10 Peak load on the coldest day (including chicken occupancy) . 113

Figure 3-11: Direction of airflow into the facility..... 115

Figure 3-12: Horizontal representation of internal air-flow during winter peak period – heat and velocity 117

Figure 3-13: Vertical representation of internal air-flow during winter peak period – heat and velocity 117

Figure 3-14: Vertical representation of internal air-flow during summer peak period – heat and velocity 119

Figure 3-15: Vertical representation of internal air-flow during summer peak period – heat and velocity 119

Figure 4-1 System control flowchart of solar assisted heat pump (One operation mode)..... 127

Figure 4-2 Heat transfer coefficients in building integrated PV/T system.. 131

Figure 4-3 Flow chart of the Solar PHE based PVT mathematical model.. 137

Figure 4-4 PVT-GSHP heating system 143

Figure 4-5: Heating system plant room 145

Figure 4-6: Thin tube PHE mats and its set up 146

Figure 4-7: Ground loop sketches of flow direction and horizontal closed linear loop 147

Figure 4-8: Time step change of incident irradiance/temperature on the PVT module (18/12/2016, 9.00-17.00)..... 154

Figure 4-9: Collected Solar heat during day, 18/12/2016..... 155

Figure 4-10 Temperature variation of the flow and return fluid at inlet and outlet of the ground array fluid , 18/12/2016 156

Figure 4-11: Collected ground heat during day..... 156

Figure 4-12: The temperatures of the fluid at inlet and outlet of the heat pump and COP variation, 18/12/2016 157
Figure 5-1 (a) Heat pump stages (b) Heat pump working principle 165
Figure 5-2 PVT-GSHP heating system drawing (APPENDIX A) 167
Figure 5-3 Basic schematic diagram of System control: One operation mode 168
Figure 5-4 The temperature variation of the fluid at inlet and outlet of the Ground Array (Monitored data) 172
Figure 5-5: The temperature variation of the fluid at inlet and outlet of the Solar PVT Array (Monitored data) 173
Figure 5-6: Electrical HP input and PV output energy 173
Figure 5-7 The temperature variation of the fluid at inlet and outlet of the Fan coil unit (Monitored data) 174
Figure 5-8 The temperature variation of the fluid at inlet and outlet of the top, middle and bottom of the Buffer Vessel (Monitored data) 175
Figure 5-9: Heat output of the heat pump and COP_{sys} mean 176
Figure 5-10: Electricity and heat output 177
Figure 5-11: Heat pump output during middle of heating season 179
Figure 6-1: Solar PV output (Monitored data) 189
Figure 6-2: The temperature variation of the fluid at inlet and outlet of the Solar PVT Array (Whole year-Monitored data) 189
Figure 6-3: The temperature variation of the fluid at inlet and outlet of the Ground Array (Whole year-Monitored data) 191
Figure 6-4 The temperature variation of the fluid at inlet and outlet of the evaporator side of the heat pump (Whole year) 194

Figure 6-5 : Average daily COPs over the year (sampled monitored data) 194

Figure 6-6: Heat pump output during annual PCs 200

Figure 6-7: Net present values in UK 201

Figure 7-1: Direct evaporative cooler design: 2-fan attached (Ecocooling) 207

Figure 7-2: (a) Current 50” supply Alfan PEC cooler, (b) Front view, (c) Back view 208

Figure 7-3: PEC schematic and airflows 209

Figure 7-4: Working principle of M-Cycle (Wani, Ghodke, and Shrivastava 2012) 210

Figure 7-5 (a) System instruments, (b)System materials 211

Figure 7-6: Photograph of PEC with labelled airflows 212

Figure 7-7: Direct evaporative cooler working principle (Pistochini and Modera 2011) 213

Figure 7-8: Ambient and supply air during test period (a) June, (b) July PC 217

Figure 7-9: Ambient and supply air and cooling load during (a) June PC, (b) July PC 219

Figure 7-10: Inlet-outlet and working temperature sampling 221

Figure 7-11 Initial The values of effectiveness at steady state conditions.. 222

Figure 7-12: Inlet-Outlet and working temperature sampling 223

Figure 7-13: Initial effectiveness and EER values at steady state conditions 224

Figure 7-14: Inlet-outlet and working temperature sampling 225

Figure 7-15 Initial The values of effectiveness at steady state conditions.. 225

NOMENCLATURE

Symbol	Term	Unit
a	Length of the collector	m
A	Area	m^2
A_{TC}	Non-adjusted annual total costs	£
AA_{TC}	The adjusted annual total costs	£
b	Width of the collector	m
C	Specific heat capacity	J/kgK
COP	Coefficient of performance of heat pump	
D	Depth of the soil	m
D_{out}	Outer diameter of PHE mat	m
D_{in}	Inner diameter of PHE mat	m
F	Experimental slab heat transfer factor	
G	Irradiance	W/m^2
h	Heat transfer coefficient	W/m^2 K
i_r	discount rate	
i_{cc}	initial capital cost	£
if	Inflation rate	
k	Thermal Conductivity	W/mK
LHP	Latent heat production	
$\dot{m}_w$	Mass flow water in PHE	kg/s
M	Mass	kg
N	Number	
NPC	Net present cost	£
NPV	Net present value	£
Nu	Nusselt number	
P	Perimeter of poultry house	m
Pr	Prandtl number	
Q	Heat transfer rate	W
Re	Reynold number	
R_w	Rated wattage	W
SHP	Sensible heat production	

SPF	Seasonal performance factor	
t	Time	h
t_{year}	current time;	h
t_{shift}	the day of the minimum surface temperature. T(z)	h
T	Temperature	K
$T_{db,in}$	Dry bulb temperature of inlet	K
$T_{db,out}$	Dry bulb temperature of outlet	K
$T_{wb,in}$	Dry bulb temperature of inlet	K
$T_{wb,work}$	Dry bulb temperature of working fluid	K
$T_{dp,work}$	Dew point temperature of working fluid	K
T(z)	ground temperature of soil;	K
THP	Total heat production	
U	Overall building heat transfer coefficient	W/(m^2K)
VR	Ventilation rate	m^3/s
w_{out}	humidity ratio outlet air	%
w_{in}	humidity ratio of inlet air	%

Greek letter	**Term**	
α_z	Thermal diffusivity of soil	m^2/s
α_g	Absorptance of glass	
ρ	density	kg/m^3
ε	emittance of the surface	
ε_{wb}	Effectiveness wet bulb	
ε_{hum}	Effectiveness humidity	
ε_{wo}	Effectiveness working wet bulb	
ε_{dp}	Effectiveness dew point	
ϕ	The angle of inclination of the surface	degree
σ	Stefan Bolzmans constant	$Wm^{-2}K^{-4}$

Abbreviation	**Term**
CFD	Computational fluid dynamics
CIGR	International Commission of Agricultural and Biosystems Engineering

DEC	Direct evaporative cooling
DPIEC	Dew point indirect evaporative cooler
EER	Energy efficiency ratio
FAO	Food and Agriculture Organization
GHE	Ground heat exchanger
GSHP	Ground source heat pump
HAP	Hourly analysis programme
HDPE	High density polyethylene
HP	Heat pump
HVAC	Heating, ventilation, and air conditioning
IAQ	Indoor air quality
IAE	International Energy Agency
IEC	Indirect evaporative cooling
JWL	John Wright Ltd
MIS	Microgeneration Certification Scheme
MVR	Minimum ventilation rate
VGHE	Vertical ground heat exchanger

Subscript	**Term**
a	Ambient air
ac	Air condition
ag	Ambient air to glass cover of collector
air	Air
air,in	Inside air
b	Back sheet
bird	Total heat from the birds
c	Component
con	Overall conduction
ele	Fluorescent electrical source
ele	Electrical sources
floor	Floor
g	Glass cover

gain	Gain
gp	Glass cover to PV cell of collector
gr	Ground
gr,in	Gound inlet
gr,out	Ground outlet
HP	Heat pump
i	Inlet
in	Indoor
loss	Loss
m	PHE mat
mw	PHE mat to water in the PHE
o	Outlet
out	Outdoor
p	PV array
pb	PV cell to back sheet of the collector
pm	PV cell to PHE mat surface
r	Cement roof
rm	Cement roof to PHE mat
sa	Sol-air
sky	Sky
so	Gains
vent	Ventilation
w	Water
wall	Wall
walls	Walls

CHAPTER 1

INTRODUCTION AND RESEARCH SUMMARY

1.1. Introduction

All developed and developing countries seek to take action to avoid the increasingly evident catastrophic impacts of climate change and global warming. Consequently, national governments and international organisations are increasingly active and strident in their attempts to devise and agree upon solutions to mitigate the encroaching global climate crisis. This chapter evaluates the research gap regarding high energy consumption and heating requirement, related to the high carbon emissions of poultry houses, which are mainly due to the energy requirements of providing thermal comfort and welfare to commercially farmed chickens at all stages of their lives, and reducing heat stress in poultry production.

In terms of broilers and pullets, the optimum temperature ranges from around 33°C at birth to around 23°C at four weeks, and laying hens function best within the temperature range 26-28°C (as explained below). Temperature increase causes reduced feed consumption among chickens, which affects their weights and the quality of their eggs. Consequently, heating, cooling and air ventilation (HVAC) inside poultry houses is an essential welfare consideration. This study focuses on investigating the possibility of sustainable technologies for providing thermal comfort of chickens and the importance and decrease the effects of heat stress on poultry production.

This study purposes to investigate the integrated technologies in a novel and sustainable heating ventilation and air conditioning system using a polyethylene heat exchanger (PHE) with photovoltaic/thermal (PVT) and ground source heat pump

(GSHP) (PVT/GSHP) and evaporative cooling units. This is proposed for use in poultry farms to generate clean and cheap energy, such as a core-based dew point cooling unit for cooling demand in summer, and PVT/GSHP for heating demand in winter. This chapter summarises the research background, aim, objectives, design, and methods, and presents the thesis structure.

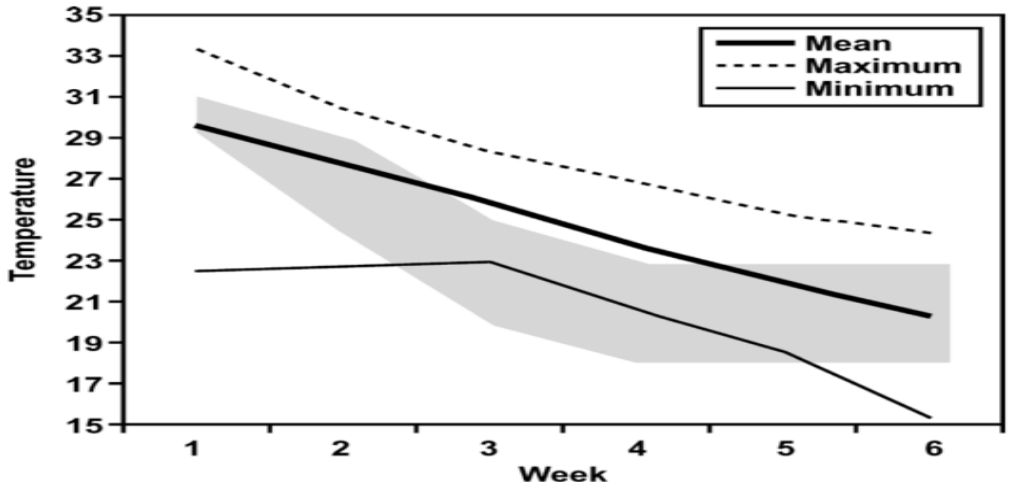
1.2. Research background information

The world had underwent an average temperature increase of 0.65°C during the 20th century, with the increase fluctuating between 0.2% and 0.3% in tropical regions (Borges et al. 2008). In the worst case scenario, the global temperature will potentially increase 1-3.5°C by the end of the 21st century (Sen 2008). NASA announced that the hottest December on record was 2015 (NASA 2015), and action to slow climate change has become an international concern for developed and developing countries alike attempting to mitigate the anticipated catastrophic results of global warming. Accordingly, 196 countries have been meeting to agree a new framework for energy saving and climate change (Willis et al. 2014).

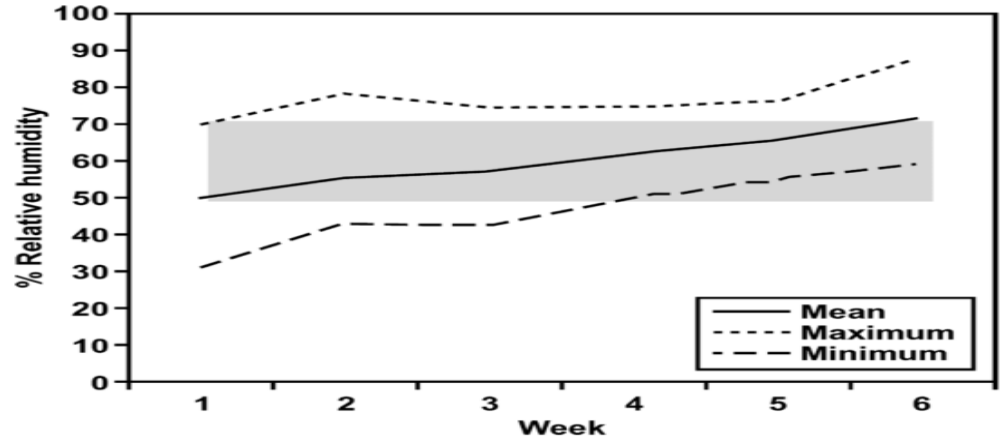
Current goals seek to limit the average global temperature increase to less than 2°C. It is necessary to emit less than 880 gigatons of carbon to achieve this goal. China makes the highest contribution to global warming, comprising 21.18% of all global GHGs emissions (T. Luo et al. 2014; Z. Wang et al. 2014). UN member states have reached an agreement to spend up to £90 billion ($100 billion) a year by 2020 as part of this plan (United Nations Chair 2014). The UK is a leading country to set a target to take precautions for climate change, and its 2008 Climate Change Action strategy saved an estimated £100 billion from early actions (Willis et al. 2014). The UK has

also aimed to reduce GHG emissions by approximately 80% by 2050, based on the carbon policy outlined from 2010 to 2015 (UK Committee on Climate Change 2016).

Environmental parameters such as ventilation rate, humidity, and temperature balance influence the production and welfare of the birds in poultry houses (Jones, Donnelly, and Dawkins 2005). The optimum ambient temperature for metabolic efficiency in broilers and pullets, as demonstrated in Figure 1-1a , ranges from around 33°C at birth to around 23°C at four weeks (Jim Donald 1999; Jones, Donnelly, and Dawkins 2005), while laying hens are most productive between 26°C and 28°C. Increasing temperature reduces feed consumption among chickens, which consequently affects the quality and weight of their meat and eggs (Jim Donald 1999). In general, relative humidity (RH) level is not a major concern in the poultry sector, but RH range, as shown in Figure 1-1b, is crucial for first-week chicks, because any illness caused by humidity at that age could influence the future welfare and health of the adult chickens (Jones, Donnelly, and Dawkins 2005).



(a)



(b)

Figure 1-1(a) The optimum temperature (b) Relative humidity range for broilers and pullets (Jones, Donnelly, and Dawkins 2005)

Review and simulation of building temperatures with regard to humidity and airflow management identifies the following problems: high fossil fuel consumption and insufficient welfare conditions with high flock density; high fossil fuel consumption and costs during winter; and high ventilation rates during summer. Poor mechanical ventilation management is common during summer months, causing heat stress to the birds.

1.3. Aims and objectives of the study

The aim of this project is to develop novel air heating, cooling, and air conditioning system design which can be integrated in poultry houses and adapted for other specialties, such as pig farming. Agricultural environment management is increasingly vital for food security, with diminishing arable land and increasing demand for food resources, and the academic field requires complex and multidisciplinary approaches to meet the severe challenges of agriculture in the 21st century. Developing a novel low carbon heating system for agricultural buildings can support the smart cities concept of the future as well as individual smart agriculture buildings. This thesis contributes to the economic development and sustainability of farming and the HVAC sector with a particular focus on efficiency and affordable design of the heat pump and evaporative cooler units applied in a poultry house under real site conditions.

This provides a practical and sustainable solution to the existing high fossil fuel consumption in the poultry-rearing industry and its corresponding adverse impacts on the environment. This novel HVAC system design (i.e. PHE based PVT assisted with HP and dew point evaporative cooling unit) is described in detail in order to provide thermal comfort and improved welfare for chickens. Consequently, the following are the principal objectives of this thesis:

1. Emphasising the research gap and reviewing recent studies in the literature regarding this subject.
2. Selecting and justifying the design to fill the gap and evaluating recent developments of the components of the proposed system.
3. Initial small-scale lab testing and a simplified model for evaluating the performance of the components.

4. Determination of thermal regulation energy requirement of a selected UK commercial poultry farm and building airflow simulation to size the proposed system.
5. Based on the simulations and laboratory testing described in objectives 3 and 4, sizing the whole system in the poultry house under real climatic conditions.
6. Setting up the whole system in the poultry house, evaluating the system performance to increase chickens' welfare and productivity, and validating the whole system modelling under real climatic conditions, based on experimental data monitoring in the poultry farm.
7. Carrying out experiments to characterise PVT collectors of integrated design and evaporative cooling units in terms of their dynamic behaviour, to provide reliable data for agriculture HVAC system designers over a wide range of time-varying operating conditions.
8. Carrying out techno-economic and environmental assessment before and after retrofitting, based on the outcomes of objectives 6 and 7, considering chicken production management and its fluctuating energy requirements.

This study provides the design, development, and testing of an innovative system, PVT-GSHP and evaporative cooling units, for efficient and cost-effective environmental control (i.e. for cooling, heating, ventilation, and humidity) to improve the air quality of poultry houses. The system integrates various technologies, including a direct and a novel core-based dew point cooling unit, a new type of inexpensive PHE, ready loop integrated solar roof collector, and a highly efficient GSHP integrated with novel vertical ground pipes. It is designed to provide good air quality throughout the year, leading to better poultry health and welfare, and subsequently enhanced productivity. The investigation of these novel components under real environmental

conditions contributes to understanding of the applications of technologies that can be adapted to the agricultural sector in terms of HVAC industry applications. To that end, the following investigations were conducted:

- Long-term heat pump test (running hours and COPs).
- Direct evaporative cooler unit investigation under real conditions.
- Dew point evaporative cooler lab test under UK weather conditions.
- Investigation of the novel technologies attached to the heat pump in a UK poultry farm under real conditions, with consideration of adoption for other agricultural buildings.
    - Solar Polyethylene heat exchanger (PHE)
    - Copper vertical ground heat exchanger (VGHE)

The outcomes include social impacts through improvement in animal welfare and reduced GHG consumption; and greater productivity in poultry farms, with a high potential for the system to have applications across other indoor livestock environments in the UK and global farms.

1.4. Research methodology

The methodology of this research utilises five approaches, as presented in Figure 1-2 literature review; small-scale lab testing of the technologies and their mathematical models; building simulations of a poultry house and system sizing of the technologies in the poultry house; the system set-up, long-term monitoring under real site conditions; and economic and environmental assessment of the systems.

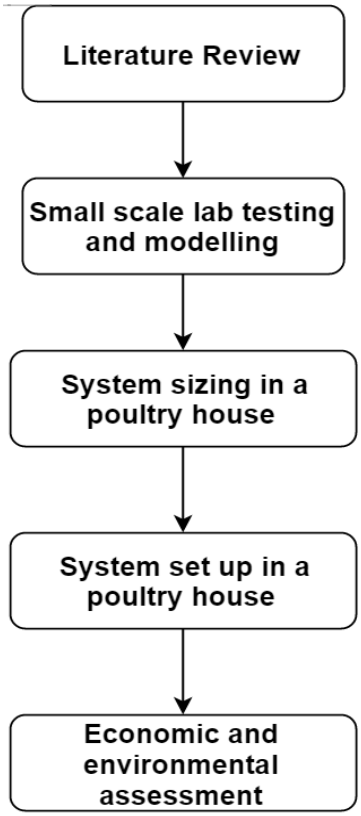


Figure 1-2:Research methodology in the thesis

The study follows the initial small-scale modelling, testing, and fieldwork procedures. In general, the research approaches to process these works are ordered as follows.

❖ *Approaches for Objective 1: Literature review*

- Technical and market review of poultry farms (highlighting the research gap and devising a sustainable HVAC systems design solution).
- State-of-the-art technologies review on the components of the novel HVAC design (i.e. evaporative cooling systems and PVT assisted HP).

❖ *Approaches for Objective 2: Small-scale lab testing and a simplified model of the PEC cooling unit*

- Novel PEC lab and on-site testing to determine efficiency in terms of measuring temperature, relative humidity, and airspeed measurements with regard to

cooling capacity, EER, wet-bulb effectiveness, and dew point effectiveness calculations and validation.

❖ *Approaches for Objective 3: System sizing in a poultry house*

- Thermal regulation load calculations of a poultry house under real climatic conditions, using MATLAB, HAP, and Design Builder design software.
- Based on the building heating load calculations, simple sizing of the whole system in a poultry house under real climatic conditions.
- Evaluation of airflow and its impact on the thermal regulation loads of the poultry house.
- Mathematical analysis of the novel components attached to the heating system.

❖ *Approaches for Objective 4: System design and set-up in a poultry house*

- Field data collection to observe the current energy consumption before retrofitting during field trial testing. Poultry house air conditions were monitored (temperature, humidity, ventilation rate, and CO_2 level).
- Based on the simulations and laboratory testing carried out, the proposed system was designed and set up in a poultry house under real climatic conditions for field trials.
- Case study, with data collection after retrofitting. Ambient indoor temperature and humidity were monitored throughout the trial period for whole-system model validation.

❖ *Approaches for Objective 5: Economic and environmental assessment*

- Long-term evaluation of the thermal and electrical performance of the whole proposed system.

- The performance of the innovative proposed system is compared with the older/previous emerging systems from the point of view of economic and environmental savings. This involves evaluating the capital cost, running cost, and payback period.

1.5. Research innovation

In addition to minimising energy consumption and GHG emissions from conventional livestock farming in the agriculture sector, this research is expected to offer applicable solutions for future PVT and HP technologies and novel PEC units. Innovative poultry farms are inevitably going to continue being a major aspect of food security and the agricultural sector in future, calling for a range of innovative systems to improve energy and cost efficiency worldwide.

Regarding the renewable energy sector in particular, this research focuses on some technical barriers faced by current sustainable technologies, especially the complex structure and high installation costs of existing sustainable components. This research incorporates more productive and affordable PHE for PVT applications and innovative and inexpensive copper vertical ground heat exchanger (VGHE) for heat pump (HP).

1.6. Outline of the thesis

❖ *Chapter 1: Introduction and research summary*

This chapter presents the scope of the thesis and its theoretical and experimental investigation of novel thermal regulation technologies adapted to poultry houses. It briefly describes the research background and research gaps addressed by this thesis and highlights the academic and industrial rationale for sustainable solutions to fill

these gaps. It explains the aim and objectives of this research, broadly outlines the novel HVAC design, and explains the research methodology and research innovations.

- *Chapter 2: Systematic review of proposed thermal regulation systems and recent applications in poultry houses*

This chapter reviews poultry production management and ventilation systems in poultry houses. Existential problems concerning chicken welfare are highlighted, which mandate solutions for the manifest failures of current HVAC systems in poultry houses. Recent studies and commercially available developments in HVAC systems for poultry houses are reviewed in detail, such as recent thermal regulation developments and existing sustainable applications. The main gap in the agricultural HVAC industry is described and the novel PVT-GSHP and evaporative cooling unit model assessed in this research is proposed as solution.

Moreover, this chapter includes reviews proposed systems for agricultural applications, particularly poultry houses. Efficient utilisation of renewable energy is becoming increasingly important because of the egregious negative environmental impacts of current agricultural inefficiency, and as a key part of food production the poultry sector is likely to be increasingly targeted by the political impetus to reduce GHG emissions and use of conventional energy sources. The obvious candidates to alleviate the heavy energy consumption of poultry farming are passive cooling techniques and renewable energy use.

In addition to the heat pump, evaporative cooler systems have drawn attention throughout the world for their potential use in the agricultural industry. These cooling technologies (solar and heat pump systems) mitigate GHG emissions and therefore

the main adverse impacts on the environment. In this chapter, a systematic review of and hybrid solar and PVT assisted HP systems and recent development on the evaporative cooling units is presented. A clear understanding of the types of these technologies and recent developments concerning them in the literature is summarized.

The main research directions relating to heating with solar based heat pumps in the literature target PV technologies, exploring heating problems of PV, utilisation of the heat PVT systems, existing types of the commercially available solar PV applications and solar based HPs, GSHPs, and the designs of the ground source heat exchangers. Numerous technologies have been explored with regard to evaporative cooling systems, and recent developments and applications have tested materials used to improve performance, along with innovative design concepts, M cycle, and dehumidification ideas.

- *Chapter 3: Heating and ventilation performance of a UK poultry house*

Poultry houses are buildings that provide the production of eggs and meat. Poultry meats are the most commonly consumed source of protein worldwide because of their relatively affordable price. However, there is increasing concern about unhealthy environmental conditions in poultry houses, in terms of both ethical and product quality issues (poor conditions decrease the quality of meat and eggs). While the environment in poultry houses and welfare conditions of chickens are concerning, current methods mainly rely on using conventional energy sources for HVAC systems, with high fossil fuel consumption to meet thermal regulation energy demands. Due to the continuous increase in scale of the poultry sector, set to continue for many years to meet the

increasing demand of an expanding global population, energy consumption in the poultry sector is a major issue for the coming decades.

This chapter evaluates the thermal regulation energy demand of a UK poultry house and environmental conditions like airflow under conditions of existing and peak ventilation rates. The thermal regulation energy load is determined under peak temperature conditions. The metabolism rate of the chickens based on their weight is defined, and sensible and latent heat release are considered in Design Builder simulations to analyse the peak load during the summer, to design the cooling unit. The heating load is 100 kW at a temperature of -4.7˚C in winter, while the cooling load is 40 kW at a temperature of 29˚C in summer. The existing ventilation rate management and setup temperatures are considered for whole-year energy simulations. MATLAB is used to develop the simulations to determine the whole year energy consumption in the building under the real UK weather conditions.

The whole system is sized theoretically based on the house's thermal regulation load. The maximum heating energy demand of the house is determined as 44 MWh/PC while the minimum is 11,1 MWh/PC. CFD analysis of the airflow under the UK weather conditions is presented in this chapter. Based on the results of the energy demands of the poultry house simulations, the whole system can be sized for the studied poultry house.

- *Chapter 4: Evaluation of an innovative PVT-GSHP system in poultry house*

Renewable energy systems, such as solar panel, GSHP, and cogeneration technologies are common systems integrated into buildings to improve their sustainability. GSHP systems are a very promising and common technology, transferring heat from a lower temperature source to one of a higher temperature for

a very efficient thermoregulation solution with widespread application potential. Although many studies have been carried out on ground heat pump systems and their ground heat exchanger design, the complexity of these designs causes restrictions on their use application in houses. Farmers are generally unwilling to set-up and run these heat pumps because of their expensive set-up, especially excavation costs to install vertical borehole heat exchangers, and/or limited places for horizontal heat exchangers.

This chapter evaluates the effects of the studied poultry house, integrating an innovative and cost-effective heating system design using PVT-GSHP system with novel ground vertical heat exchangers. Based on the theoretical calculations of the poultry houses and the models of proposed technologies explained in chapter 3, the proposed PVT-GSHP system was sized and set-up in the poultry house.

The efficiency and energy saving of each novel and commercially available heat exchanger attached to the heat pump was determined under dynamic, short-term (< 1 day) on-site conditions.

1. The potential energy saving of the novel design PVT-GSHP on agriculture under real outside weather conditions, and the inside air temperature variations of chicken production cycles.
2. Each of the following novel components' dynamic behaviour under on-site conditions:
    - Solar polyethylene heat exchanger (PHE)
    - Copper vertical ground heat exchanger (VGHE)
    - Horizontal ground heat exchanger (HGHE)

Humidity, temperature, and power consumption rates were monitored in the location for a short period daily in winter and summer conditions. The system involves a fan coil unit, a GSHP attached with a buffer vessel, a novel PVT unit, and a novel ground source array to use in the poultry house HVAC applications. In the system design, 52 PV panels were attached to four thin-tube (capillary tube) polyethylene mats (1 m × 12 m) and 50 underground vertical co-axial pipes (25 mm in diameter), connected in series.

- *Chapter 5: Performance assessment of novel PVT-GSHP: Long term monitoring*

The studied components comprise solar polyethylene heat exchanger (PHE), copper vertical ground heat exchanger (VGHE), and ground horizontal polyethylene ground heat exchanger. Humidity rate, inside ambient temperature, and inside air quality were measured during the time of the monitoring and analysis was carried out based on the hourly electricity (thermal regulation) loads from each component over a one-day period. The performance of COPs and the efficiency of the PVT-GSHP system was observed hourly during peak heating and start and end of the heating season.

The main purpose behind the use of ground heat exchangers is to enable the exploitation of geothermal energy for meeting the thermal regulation demands of buildings in an efficient and environment-friendly manner. However, the high cost of the soil digging, and space availability are common problems faced in the HVAC industry, as mentioned previously. Moreover, the long-term performance of ground heat exchangers and their actual environmental impact is a rising concern in the literature reviewed. Hybrid heat pump applications in poultry houses and their long-term analysis have not been substantively analysed in existing studies.

The mean weekly COP ranged between 2.3 and 2.43. The minimum daily COP fluctuated between 2.3 and 2.32 on the days when the mean outdoor air temperature dropped to 5°C during the daytime. During the peak heating season, the average heat transfer rate of 46 m^2 PHE equals to 2.3 kW, with corresponding heat fluxes of 50 W/m^2 per unit PHE area in continues operation.

- *Chapter 6: Techno- economic and environmental assessment of novel PVT-GSHP*

Recently agreed standards for testing and long-term monitoring specific to PVT-GSHP are not available for dynamic conditions. Therefore, on-site transient experimental investigation was undertaken to observe: The energy efficiency of each component was evaluated on the studied summer and winter days.

This chapter presents the results of a long-term performance assessment of the novel ground array integrated into a poultry house, taking into consideration the thermal regulation demands of the poultry house, and the intermittent operation of a GSHP. Furthermore, an economic assessment model is implemented to compare the performance of the novel PVT-GSHP system and existing gas coiler in the poultry farm. The economic enhancement provided by the adoption of the PVT-GSHP system is quantified with respect to a conventional thermal regulation system. The use of this novel ground array has a considerable reduction in economic and environmental impacts in the examined poultry house. The COP of the whole system design fluctuates over the long term between 2.05 and 5.17, with a maximum value of 5.17 in July, and a minimum of 2.05 in January. The total thermal output from the heat pump was approximately 15 MWh per annum, with a corresponding Seasonal COP of 3.73

The energy loads were validated from the monthly gas usage data from the poultry house, studying the impact of replacing gas heating with GSHP. Maximum heat demand occurs in winter mornings. The system was set to run from 9.00 to 17.00 during winter, and the runtime of the heat pump was slightly increased during the summer. The horizontal polyethylene heat exchanger (PHE) has an average 2.9 kW of heat transfer rate, while novel copper ground heat exchanger (VGHE) at 2.5 m length has 3.65 kW of heat transfer rate, with a corresponding heat flux of 64 W/m^2 through a year.

Economic and environmental analyses were conducted after retrofitting the poultry house. The annual savings and payback period were calculated to ascertain the economic and environmental benefit of the retrofits. The obtained results show that the simple payback period of the novel cost-effective heating system design is 8.14 years, with total annual savings of £4,890 and government support, alongside government support including export tariffs and RHI

- *Chapter 7: Performance assessment of air-cooling systems for low carbon poultry houses*

Climate control in the poultry house is essential for chickens' welfare, and thus economic productivity. In hot and humid conditions, chicken production productivity can be decimated by high mortality and disease among livestock. Heat stress in poultry farms is one of the main challenges in the agricultural industry. At temperatures above 30˚C chickens become uncomfortably hot; their surface temperature can tolerate between 35-40˚C, but such highs undermine their ability to lose heat from their bodies (i.e. impairing their physiological thermoregulation), which thus affects their tissues negatively and predisposes them to illnesses, including infections that flourish

in such humid conditions. Therefore, during summertime, it is essential to keep the air as cool as possible.

Traditionally, high summer temperatures in the UK rarely exceeded 30˚C, thus the main focus of thermal regulation in the country (as in northern Europe generally) has always been on heating for colder conditions rather than cooling, but this will increasingly change with rising temperatures due to global warming. Additionally, some cooling has always been necessary due to the heat caused by livestock and agricultural activities themselves (e.g. chickens' body heat and lighting etc.). Temperature requirements also vary by livestock age. The existing farm in Kirton uses an external fan unit for up to three weeks after the 4th week of the chickens' age, to keep the air temperature below 24^{o}C for chickens. Ventilation systems in the building are extremely important to prevent chilling the chickens and heat stress. The design and its control need to be considered if cooling is required daily and monthly.

Evaporative cooling systems presented in the literature commonly use cellulose and wood shaving pads, and high or low-pressure foggers are the most promising technologies to prevent the heat stress in poultry houses. In this chapter, an evaporative cooling unit is tested under the real site conditions. The unit has two integrated fans, run with or without cooling. The cooling demand was for a short time, from one to three weeks, mainly for up to four hours during daylight in June and July under UK climate conditions. Psychrometric Energy Core (PEC) is a common cooling method all over the world, therefore in addition to this field trial on the poultry farm, a dew point evaporative cooler design was evaluated in laboratory controlled, steady-state conditions to adapt to the agriculture HVAC industry. This method offers promise to decrease heat stress and death issues on poultry farms. The difference between

the lab test and field trial lies in the design and the method of the evaporative cooler and its impact on the chickens.

The PEC unit on-site investigation targeted direct evaporative cooling methods to increase the welfare condition of the chickens. The results show that under real site conditions from June and July, the cooling temperature was 5°C when the systems run with 22% fan speed capacity. The cooling time was a maximum of three weeks from June to July, with between 2.5 hours and 4.18 hours changing with daily cooling energy demand of the poultry house. The cooling load of the PEC unit was 6.5 kW on average. On the other hand, the results of the dew point cooler unit show promising results for the agricultural industry under steady-state conditions. The effectiveness was found to be 0.6 under the temperature of 29°C, with relative humidity of 27% and fan speed of 0.8 m/s. It is concluded that the direct evaporative cooler unit was effective with its control mechanism (including ventilation only or cooling and ventilating mode) with two fans. However, when the humidity increased in the poultry house, the unit could not provide the required temperature. These times are relatively uncommon, but they are crucial for chickens' wellbeing. On the other hand, the dew point evaporative cooler unit was found to be efficient, and it is thus recommended for use in poultry farms.

- *Chapter 8: Conclusions and future research*

This chapter presents the salient findings arising from the investigation undertaken in this thesis. It summarises the HVAC technologies and their theoretical and experimental investigations, and addresses achievement of the objectives, methods, and key outputs of the thesis. The overall system efficiency of each system is considered and discussed for the future purposes. The contribution of this work in the

literature and agriculture industry is highlighted, and some recommendations for future work are presented. The works carried out have mainly been based on poultry houses, thus further works are recommended to focus on different types of buildings such as pig farms, since the underlying thermoregulation aspects of this work are generally adaptable to all agricultural buildings, with wider implications for the agricultural HVAC industry. In this respect, more theoretical and complicated monitoring analysis can be conducted for particular variants of agricultural buildings.

In order to improve the welfare conditions of the chickens and provide sustainable energy solutions to address the shortcomings of existing HVAC solutions for poultry houses, a novel cost effective PVT-HP heating system with two different types of PRC cooling units is presented, designed, and monitored in this study. The focus of the study is not only investigating the novel design of these technologies but also their use in agriculture regarding optimal environments for chickens' welfare (and thus economic productivity). To provide improved welfare for chickens, the designed system is essentially cost effective and efficient, but the significantly higher heating requirements of UK poultry houses poses challenges in terms of soil charging and recharging processes over the course of time. Therefore, a smart-run of the design with novel heat exchangers is strongly recommended for future trials. The technologies for the increase of the soil conductivity will be studied to improve soil charging capacity in the future. Moreover, to increase the efficiency of the ventilation systems of poultry houses, novel air cleaning technologies, such as fibre filtering and cleaning with coating technologies, are recommended to improve the air quality inside houses, and the welfare conditions of the chickens.

CHAPTER 2

SYSTEMATIC REVIEW OF PROPOSED THERMAL REGULATION SYSTEMS AND RECENT APPLICATIONS IN POULTRY HOUSES

2.1. Poultry houses

Current poultry HVAC systems need cheap, clean, and consistent energy sources to meet global animal welfare laws protecting animal welfare on farms, in transport, at markets, and during slaughter. This chapter reviews poultry production, ventilation principles, and environmental parameters relating to poultry houses. It explains the importance and effects of heat stress on poultry production and outlines possible sustainable technologies for providing thermal comfort for chickens.

Chicken is a substantial source of proteins, vitamin B and minerals with a low rate of saturated fats (González-García et al. 2014). Over recent decades, chicken meat production has dramatically increased by approximately tenfold (Esquivel-Hernandez et al. 2016; Thornton 2010). The vast industrial scale of global poultry farming makes it a major consumer of fossil fuels in the form of energy consumption, with serious environmental impacts that increasingly concern consumers. In future, this trend will sustain a steady growth based on the (Food and Agriculture Organization) FAO 2015–2025 outlook (European Commission 2015). In the US, the consumption of poultry has significantly increased from an average per person 18 kg (40.2 lb) in the 1970s to approximately 40 kg (86.5 lb) in 2007 (American Meat Institute 2009; Eberle et al. 2013). A total of 8.5 billion broilers produced comprise 23.2 billion kilograms (51.3 billion pounds) of live weightiness in 2014 (Alali et al. 2016). Meat produce has consistently increased despite fluctuating value of meat products in the US over the years.

The global industry of poultry houses has a tendency to increase production volumes considerably every year. Table 2-1 presents the production and consumption of broiler in some selected countries. It is clear that the highest consumption rates are reported in industrialized countries (González-García et al. 2014; USDA 2016). Although livestock products are responsible for 33% protein with 17% of kilocalorie consumption, it is observed that there is a considerable gap in the consumption rate between the developed and developing countries (Thornton 2010).

Table 2-1 The production and consumption of broilers in selected countries (USDA 2016).

Ready to cook (1,000 metric tons)	2014	2015	2016 (Oct)
	Production		
Brazil	12,692	13,146	13,480
China	13,000	13,400	13,100
European Union	10,280	10,620	10,845
	Consumption		
China	12,830	13,267	12,985
European Union	9,859	10,171	10,375
Brazil	9,137	9,309	9,604

In the UK industrial area, total UK poultry meat production was 130.5 thousand tonnes in 2015, with broiler slaughtering increasing by 0.1 million birds (+0.1%) from 90.2 million to 90.3 million in October 2015 (Department for Environment food&Rural Affairs 2015). Syria has approximately 13,000 poultry farms with a capacity of 172,000 tonnes of meat production per year, employing 150,000 workers(Kharseh and Nordell 2008b).The poultry numbers in Kenya have increased from 25.8 million in 2003 to 30 million in 2006. Meat and egg production are 19,058 metric tons and 1.182 million

respectively per year in Kenya (Mutai et al. 2011). Pakistan poultry production was 554,000 tons in 2006; 601,000 tons in 2007 and 652,000 tons in 2008. Poultry meat accounts for 19% of total meat production. This sector has around 2,500 houses, with of 8-10% possible annual growth, and it provides employment opportunities for 1.5 million people in Pakistan (Nasir 2010). Poultry consumption accounts for 75% of total meat consumption, which is 30 kg $cap^{-1}.year^{-1}$ in Portugal (González-García et al. 2014). In a single year, 2.62 twelve billion of egg were produced in Brazil with a continuous production increase rate of 8.9% (Silva et al. 2013). In the European region, egg production was higher in Turkey than in Italy, France, the Netherlands, Greece and Germany, with 876,774 t, and was third after Germany and France in chicken meat production, with 737,037 t in 2004 (Dağtekin, Karaca, and Yildiz 2009). In the US, Tennessee alone increased production from 186.4 million broilers valued at around £185($268.4) million in 2002 to 213.5 million valued at around £285 ($413.7) million in 2006 (Bazen and Brown 2009). Clearly the increasing consumption of poultry worldwide necessitates careful study of the energy consumption and gas emission rates of poultry houses in order to evaluate either environmentally friendly improvement or cost and energy savings.

2.1.1. Internal air quality and lighting

There has been a lack of innovation addressing the range of health and welfare issues involved in livestock production, including (in the context of chickens) foot, clawpad, and joint damage (pododermatitis); pectoral muscle necrosis; and respiratory problems. They are the consequences of intensive production relating to HVAC, air quality, and stocking density factors. Temperature, relative humidity, air composition, air speed, air distribution, and light are the main factors to be measured at the animal level and are considered in more detail below (Figure 2-1).

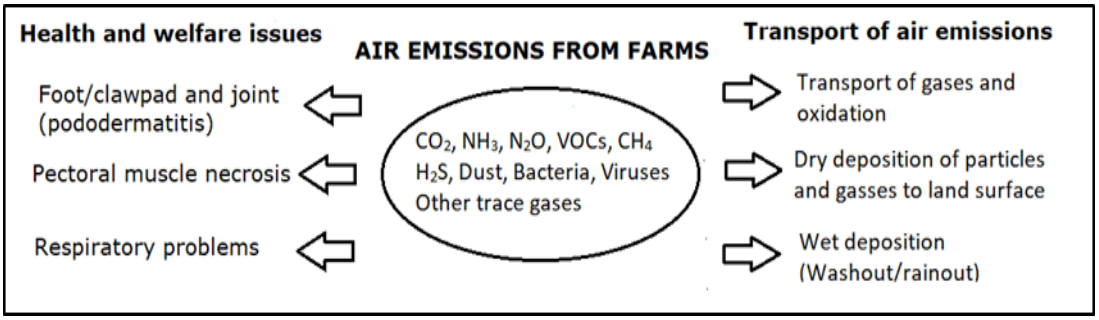


Figure 2-1: Air emissions from livestock

Classification and general rules in the livestock industry show that the most concerning air pollutants arising from poultry houses are ammonia and CO_2, therefore there is increasing standardisation in regulatory specifications for the emission of CO_2 and ammonia released from such buildings. An ammonia rate of < 25 ppm must be maintained in order to achieve optimum bird performance (i.e. growth and health to maturity). Over a long period of time, this harmful gas can become hard to detect by smell if the rate is lower than 50-60 ppm, but its impact on birds is significantly harmful. For example, in research conducted at the USDA laboratory in Mississippi, 0.31 lb weight loss was recorded per broiler within an environment having ammonia of between 25-50 ppm (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014). Studies of pollutants from poultry farming focus primarily on ammonia, hydrogen sulphide, and methane emissions. It is notable that there is no standardisation or information about emissions data for other contaminants, such as odour, nitrous oxide, non-methane volatile organic compounds, dust, endotoxins, and long-term health and environmental impacts, which remain unclear (Casey et al. 2006).

Moreover, deep litter poultry production brings a host of welfare-related problems such as pododermatitis, pectoral muscle ulceration from extended contact with wet corrosive litter, and respiratory distress from inadequate ventilation designed to

maintain room temperature at the expense of indoor air quality (IAQ), predisposing birds to a range of respiratory and systemic infections. Microorganisms, including pathogenic bacteria and viruses in airborne dust particles coming from dried droppings, feather, and skin scales (and some feed), represent the other concerns inside the house. Ventilation management however remains restricted in scope to heating, humidity, ammonia, and the CO_2 rate, with negligible consideration of the above factors. This highlights that poultry house management is an inherently complicated process, with lots of parameters to be considered (both inside and outside houses), with no easy fixes.

Solid particles of less than 500 microns in diameter are classified as 'dust' (Sector Guidance 2011). Less than 5 milligrams per cubic meter (mg/m^3) dust at broiler level, or a tolerable dust level of 8 mg/m^3 with regulated heating and ammonia level, is recommended. If this air quality is not provided, respiratory health problems will exist in the house (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014). Odour and gas emission rates are often optimised with the air emission rate and the number and weight of animals. Animal unit (AU) and animal live weight are common units used for emission rate comparison. However, it should be noted that the definitions of AU and livestock houses are not standardised. Additionally, there is a paucity of research on odour emissions from manure irrigation (Casey et al. 2006).

According to research conducted in Athens, dust with a Particular matter(PM) of 2.5 can quickly disperse over many miles away from a farm, given its small size and light weight (Worley et al. 2013). This makes the purification of the air inside the poultry building (before it is emitted into the surroundings during ventilation processes) much

more important. The most intuitive source of dust is bedding material, but bedding material management has been found to have no considerable impact on the PM emissions rate (J. van Harn et al. 2012). In 2008, the UK National Atmospheric Emissions Inventory announced that the poultry industry is responsible for 9 kilo tonnes of $PM^{1}0$ per annum (Sector Guidance 2011). Respirable particles with a diameter of 0.5 micron or smaller exist in poultry house airflow patterns, which are inhaled and frequently deposited in the lower lung (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014). Environmental management of poultry houses must therefore acknowledge the relationship between PM and airborne microorganisms and their changeable properties within the air in order to inform effective design techniques for reducing them in livestock buildings (Adell et al. 2015).

The optimisation of the temperature and air speed in poultry houses is crucial for the welfare of chickens and the quality of production. Table 2-2 highlights the required temperature, air quality, and air speed in broiler houses (Fawaz et al. 2014).

Table 2-2:Chicken themal and air quality design conditions(Fawaz et al. 2014)

Production cycle	Temp. (°C)	Air quality	Air speed
1	30-33	CO_2 ≤ 2,500 ppm	0.3 m/s (max)
2	27-30	NH3 ≤ 25 ppm	
3	24-27		Can be increased gradually to 2.5 m/s
4	21-24		
5	21		
6	21		

In addition to IAQ, livestock also require varying lighting regimes during different stages of production to maintain efficient growth. Effective light management has been seen to decrease mortality rates, reduce waste, improve metabolic efficiency, improve

animal welfare, and maximise the feed-conversion ratio. Poultry production is particularly influenced by the wavelength of the light source. Light colours with a long wavelength, such as red, may make birds bad-tempered (Lewis and Morris 2000). In addition, the intensity of the light may affect chicken growth. An experimental study shows that the inhomogeneous intensity level in the house may adversely affect laying performance and egg quality (Yildiz et al. 2006).

2.1.2. Air movement and ventilation management

Different climate conditions require different heating, cooling and air conditioning systems. The main parameters to estimate ventilation rate are inside temperature, humidity, and the amount of harmless gasses. Air quality inside houses decreases with ammonia released by wet litter (James Donald 2009). Therefore, it is important to manage the moisture rate of the air in the houses. In general, if the moisture rate is managed appropriately in the house then the dust, ammonia, and CO_2 rates are assumed to be under control, under normal circumstances (managed by professional breeders) (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014). Figure 2-2 shows that at high temperature (35°C), the provision of air movement at 2 or 3 m/s could reduce production loss after a few weeks of adaptation. Speeds below 1.5 m/s with elevated temperatures are detrimental to production (Ruzal et al. 2011). Good management of IAQ without adjustment of ventilation is needed to provide sufficient levels of oxygen, while eliminating damaging gases, volatile organic compounds (VOCs), particulate matter (PM), and microscopic bacterial airborne elements.

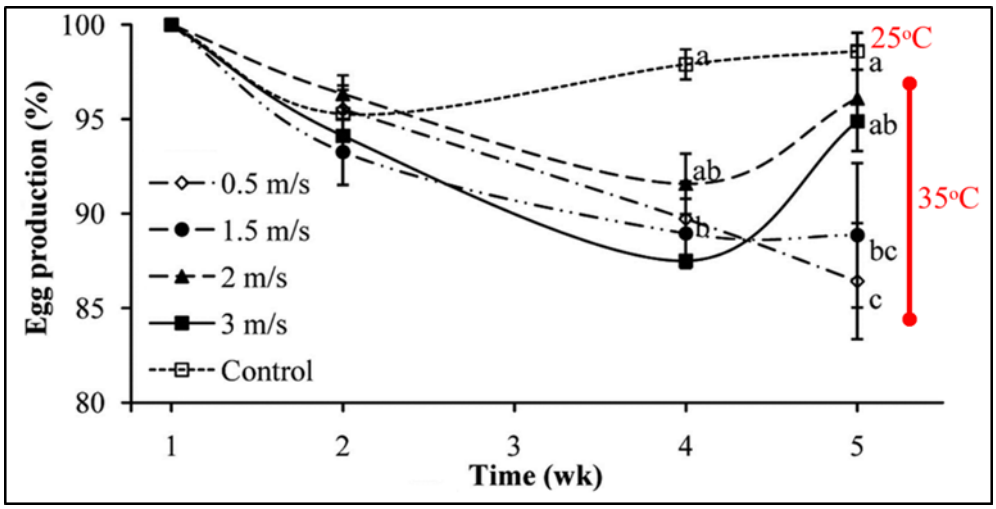


Figure 2-2:Effects on ventilation on egg production (Ruzal et al. 2011)

The determination of the building and manure management is not straightforward. Multiplying the rate of the fans by the gas concentration is inadequate, and assuming mass flux of a specific gas along the surface of litter on a floor is not straightforward. Different climate conditions require different heating, cooling, and air conditioning systems. The most common ventilation systems have minimum ventilation operated by a timer system, commonly with 0.5 m/s air velocity across the chickens in cold climates; transitional ventilation operating with temperature sensors and 0.875 m/s air velocity in average climates; and maximum ventilation operating in hot climates with 2.5 m/s air velocity (Abid et al. 2013; McQuiston, Parker, and Spitler 2010).

Air exchange and air distribution are the main two fundamental purposes of ventilation systems (Wheeler 2004). The two fundamental ventilation types are natural and mechanical (i.e. fan-powered). Natural ventilation is provided by raising or lowering sidewall curtains and relying on prevailing winds and/or convection heating, called curtain ventilation. Fan-powered ventilation systems create static pressure to manage air intake. Most poultry houses use negative-pressure-based ventilation systems,

which need to be tightly sealed (James Donald 2009). The efficiency of ventilation systems is directly proportional to tightness level of the items. Both fan performance and house tightness affect the intake air speed. Leaky houses can be protected with installation of mixing or circulating fans (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014).

The air velocity inside the house influences the broilers by creating a wind-chill effect to manage high temperature. During hot climate conditions, the maximum ventilation is provided by tunnel ventilation systems with wind-chill affect. Inside fans or tunnel ventilation can provide efficient wind-chill cooling for 4-7-week-old broilers in hot climate regions. Moreover, using hybrid ventilation systems is often a productive method in poultry houses. For example, four tunnel fans in the transitional montage can be as efficient as typical four tunnel ventilation systems (James Donald 2009).

Homogenisation of air velocities within the housing through recirculation of indoor air, thus reducing ventilation requirements, can reduce bird stress and postural behaviour from zonal migration. In a nonhomogeneous, distributed environment, the distribution of the chickens and their zonal occupation can be observed (Figure 2-3). Many studies have been conducted to homogenise the air inside poultry houses by the regulation of the ventilation rates. Observations indicate that under heat stress (>39°C), chickens move to the high air velocity region in the house, with a tendency to congregate in zones of lower air velocities, under cold stress (<29°C) (Youssef, Exadaktylos, and Berckmans 2015).

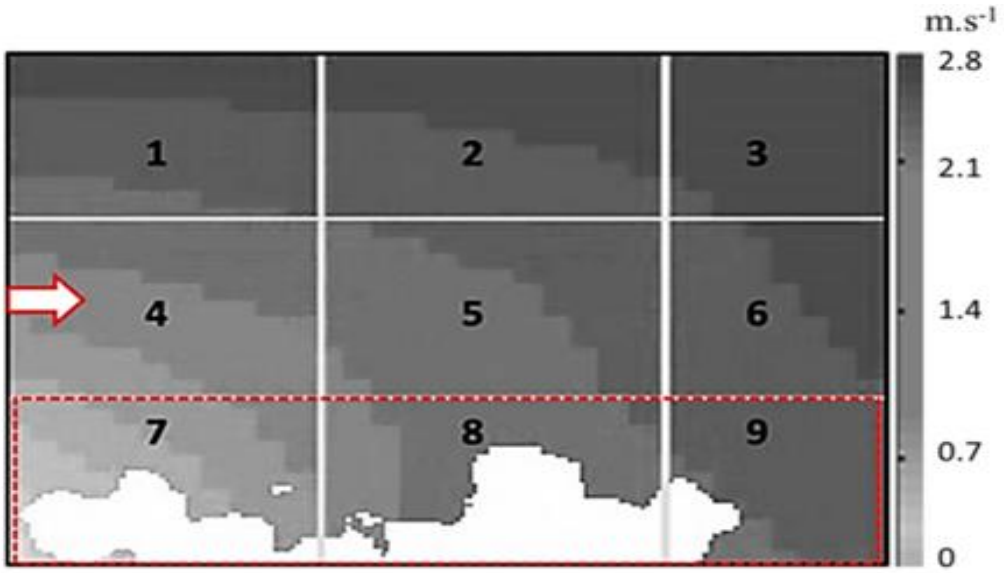


Figure 2-3: Chicken Chicken distribution in higher velocity zones under heat stress (Youssef, Exadaktylos, and Berckmans 2015)

As shown in Figure 2-3, the chickens were under heat stress with a 45°C inlet temperature, with zones 7-8-9 having a lower air velocity then the other zones. The chickens moved from the lower air velocity condition to the higher velocity condition under heat stress. This shows that the optimisation of the temperature and air speed in poultry houses is crucial for the welfare of chickens and the quality of production (Fawaz et al. 2014). As mentioned previously, the main factors affecting IAQ contamination in poultry houses are litter conditions and air movement (i.e. ventilation). Wet litter can cause illnesses in the houses and increase in ammonia. Ammonia gas results from the chemical decomposition of uric acid in the litter. The main parameters affecting the ammonia rate in houses are ventilation rate and litter conditions. An optimum moisture rate and litter are important for healthy and welfare chicken management (Gerber, Opio, and Steinfeld 2007).

The common method of minimising ammonia during chicken development is to optimise the ventilation rate, utilising HVAC systems universally used in the

maintenance of animal housing temperature and IAQ in professional poultry houses. HVAC systems consume substantial amounts of energy, primarily electricity, resulting in high CO_2 emissions associated with operation. Conventional poultry house HVAC systems unfortunately do not overcome such issues of effective air sterilisation and odour/particle removal. In such systems high ventilation rates are used to replace stale air with fresh outside air. High fresh air ventilation rates subsequently result in increased thermal regulation requirements, adding to annual energy consumption and CO_2 emissions. When no internal air purification is used, peak summertime ventilation rates can reach 13 m^3/hr per bird. Table 2-3 shows the maximum ventilation rates for classified birds (Defra 2005).

Table 2-3 Maximum ventilation rates (MVR)(Defra 2005)

Liveweight (kg)	Broilers (1,000)	Turkeys (1,000)	Laying birds (1,000)
2	9,700	9,700	9,000
2.5	11,500	11,500	10,800
3	13,000	13,000	12,250
5		19,000	
10		32,400	

2.1.3. Air temperature and moisture content

The moisture content is fairly important due to its impact on the sensible temperature and particulate generation. In high dry condition, the particles exist with bird movement in the litter. The interaction of these various compositions, temperature and wet/dry litter conditions together with the humidity rate is the main reason for the poor environment in the house. Moisture and temperature balance can protect animal from various diseases, inter alia, airsacculitis, dermatitis and systemic infections. In

addition, this balance and pH of the litter can decrease the rate of conversion of nitrogen into ammonia.

Heat stress is one of the most significant environmental stressors challenging poultry production, especially in summer. It not only causes suffering and death among chickens, but also results in decreased or lost production that adversely affects profits. In the worst case, heat stress causes death in birds subjected to more than 4°C above their standard body temperature of 41°C (Defra 2005; Moreki 2008).

The moisture content, pH and temperature of the poultry deep litter all contribute to its degradation. High concentrations of particles/pollutants, including ammonia, can cause irritation to mucous membranes of the respiratory tract and the conjunctivae and cornea in the eyes. Damage to the mucous membranes of the respiratory system increases the susceptibility of birds to respiratory infections. High levels of atmospheric pollutants and gases can also have a negative impact on the immune system, overall lifespan, weight gain, feed conversion, and carcass condemnation rate during processing.

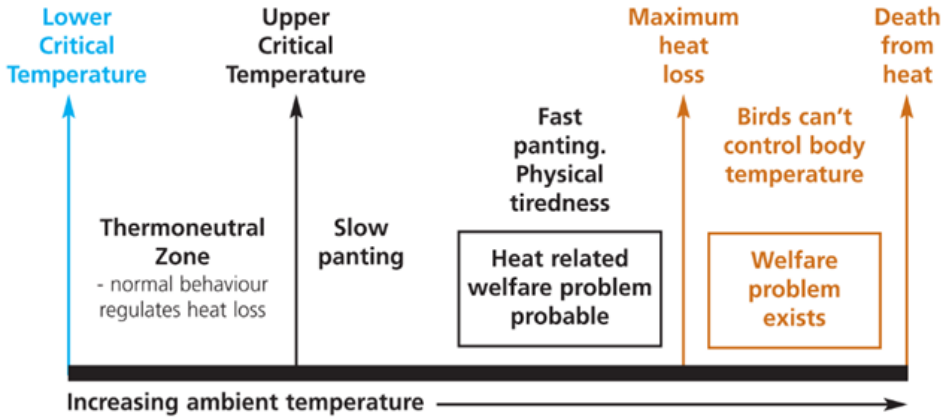


Figure 2-4: Thermal Thermoneutral zone diagram (Defra 2005)

Breeders should follow a correct management strategy to maintain the welfare of chickens. Figure 2.-4 presents the thermoneutral zone in the optimum temperature for chickens, which changes with their age (Defra 2005). The control of the environment of all animals, but particularly broiler chickens is a key factor in improving their welfare, particularly the ability to control indoor air quality (IAQ). While there is legislation on the density stock of the birds for European poultry producers, this rule does not, however, cover the welfare of the chickens in the building and such factors which can cause ineffective poultry management.

2.1.4. Evaluation on poultry houses

Recently, there has been increasing interest in providing thermal comfort by using innovative and energy efficient applications. There have been many case studies carried to identify and reduce excessive energy use in poultry houses and to suggest possible improvements. These studies have mostly focused on energy savings

associated with heating, cooling and lighting. Figure 2-5 demonstrates the evaluations and applications used for such innovations.

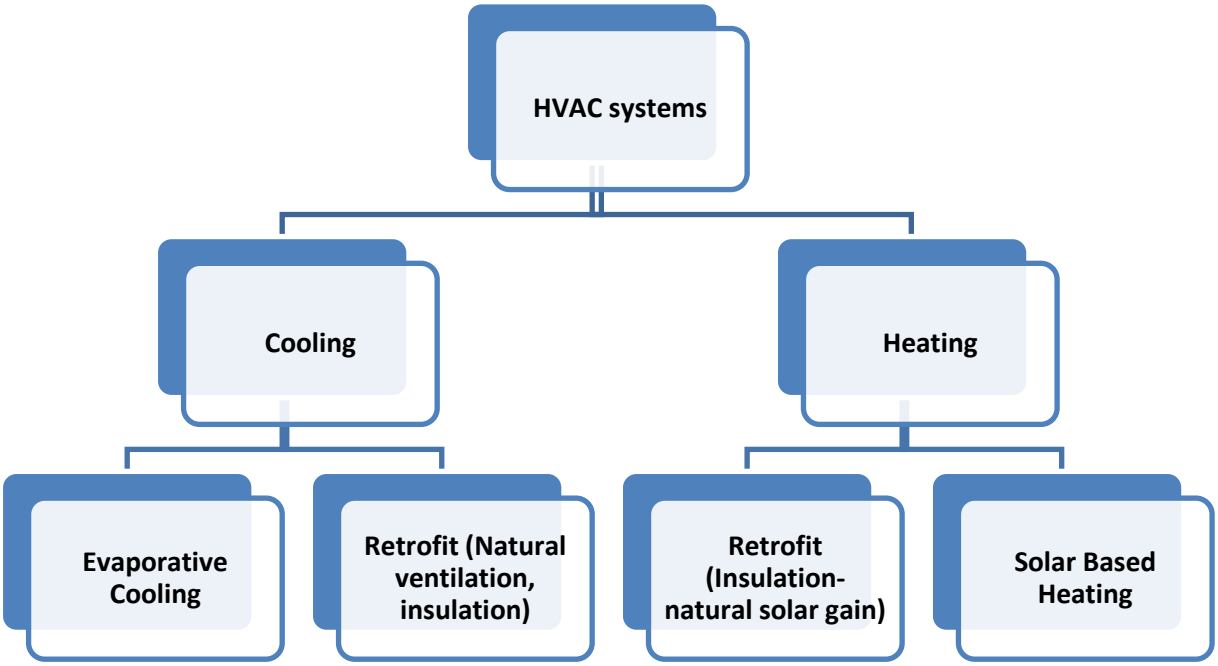


Figure 2-5 I Innovative applications and evaluations

2.1.5. Retrofit technologies on heating, cooling and lighting system

The improvement recommendations of the PHES at the University of Kentucky included passive heating cooling approach, such as sidewall curtain insulation, ceiling insulation and well-insulated tunnel doors; and more efficient active heating methods such as radiant tube heaters. Moreover, replacing incandescent light bulbs with energy-efficient compact fluorescents or dimmable cold cathode lights was estimated to be a cost-effective to save 8000 to 12000 kWh/house/year, representing a payback period of approximately 0.8 years (University of Kentucky College of Agriculture & Kentucky Poultry Federation 2014).

Attic inlets can also be added as a ventilation concept in order to recover readily available solar heat from the attic. One common problem about attic inlet is passing

cold air into the birds. It is important to know that attic inlets are gainful when useful heat exists in the attic, but it is better not to use attic air ventilation system if the attic air is within less than 10-15 degrees of house goal temperature (The National Poultry Team and University Auburn 2015). However, the hybrid applications in poultry houses in general show steady development in the use of attic ventilation (Love et al. 2014). This improvement can also reduce relative humidity in the house and cause letter moisture decrease with more thermal comfort. Although this retrofit strongly depends on the weather conditions, PHES evaluated that participating poultry houses could save between 400 and 800 LP/house/year (University of Kentucky College of Agriculture and Kentucky Poultry Federation 2014)

2.1.5.1. Evaporative cooling in poultry houses

Most countries do not have mechanical cooling systems in poultry houses, but because of global warming concerns, heat stress in farms is considered to be an alarming issue. A comprehensive literature review was conducted and it was found that evaporative cooling systems for agricultural applications, such as poultry houses, are well known in the art, and maintain the target temperature, humidity in buildings. Evaporative cooling is an effective method to protect broilers from heat stress during hot weather conditions. In terms of cooling performance, the main parameters are saturation efficiency, air temperature and relative humidity. With respect to chicken welfare and productivity, body weight, feed conversion and mortality rate are the measurement criteria (Petek, Dikmen, and Oğan 2012).

Improvements in evaporative cooling methods have mainly been considered with tunnel ventilation-based systems. This method provides rich cooling with the aid of wind-chill affect in very hot climates if the weather is not humid and relatively dry. For

example, four tunnel fans in the transitional montage achieve similar efficiency to typical four tunnel ventilation systems. For 35°C (95°F) and 50% relative humid outside air, 75% efficient EC systems can provide 7°C (12°F) of effective cooling to 28.3°C (83°F) and 6-7°C(10-12°F) of wind-chill cooling to 21.6-22.7°C (71-73°F) (James Donald 2009). Slower air velocities and thick pads provide more efficient results (Petek, Dikmen, and Oğan 2012).

There are various materials and designs for the development of evaporative cooling technology in industrial, residential and agricultural markets such as livestock (pigs, poultry), greenhouses and warehouses (Malli et al. 2011). Table 2-5 presents some case studies in order to evaluate the recent developments and studies on evaporative cooling and ventilation principled with their practical challenges in poultry houses. Almost all of the studies showed that evaporative cooling is very useful to protect flocks from heat stress, as expected. With two stage pad cooling the inlet air temperature can be decreased with the same relative humidity rate (Petek, Dikmen, and Oğan 2012). It is clear that in order to increase the efficiency of evaporative cooling systems, the air needs to have a higher temperature and lower relative humidity (Senawong, Suntivarakorn, and Radpukdee 2011). Morover, for cooling demand, an important study carried on by Kharseh et al. with solar assisted HP application used a system of ten boreholes with 120 m depth Integrated with 85 m^2 low temperature solar collector to alleviate the cooling load of a poultry house in Syria in hot, arid weather conditions (Kharseh and Nordell 2008b).

Table 2-4 Case studies and model samples on evaporative cooling improvements in poultry houses

Methodology	Location/ environmental condition	Aim	Application	Results	
Case study	Brazil hot weather (21°56'S & 50°44'W)	Air quality in three different climates	Negative ventilation Pad cooling & natural nebulization based ventilation & natural ventilation	Pad Cooling financial results : 12.04% production improvement & R$ 32.93 per month	(Silva et al. 2013)
SolidWorks & CFD modelling	36°C, 60% relative humidity	Optimizing opening of ventilation systems	Combination & evaporative cooling (closed vent.)	27.2°C internal effective temperature (32°C, 70% relative humidity)	(Abid et al. 2013)
Case study	~ 31°C to 32°C	Traditional pad & two stage pad cooling systems	Tunnel ventilated two stage evaporative cooling	Two stage pad cooling: 5.39°C heat reduction with 16.74% average cooling efficiency & +64.36 g broiler weight	(Petek, Dikmen, and Oğan 2012)
Case study	Cukurova, Turkey, ≥30°C between June and September	Long axis heat distribution & evaporative cooling	Tunnel ventilated cellulose based evaporative cooling (six fan)	26.91°C with 5.19°C temperature decrease with 69.35% cooling efficiency & 27.69°C in long axis distribution in poultry house (max 2°C increase)	(Dağtekin, Karaca, and Yildiz 2009)
Case study & mathematical model	Thailand, 33.41°C & 39.87°C RH condition	Internal temperature and relative humidity	Direct evaporative cooling	Higher temperature and lower relative humidity condition: 22.63°C internal temperature	(Senawong, Suntivarakorn, and Radpukdee 2011)

2.1.5.2. Innovative Heating Systems in Poultry Houses

Poultry houses consume diesel fuel, gasoline or coal for heating demand. For example, in Syria approximately 168.103 tons of coal (1170 GWh) is consumed for poultry heating(Kharseh and Nordell 2011)**.** Comprehensive worldwide case studies were carried out testing the use of biomass methods such as litter and manure combustion for either heating demand or electricity generation by conventional turbine methods(Park 2007), however because of the relatively low efficiency of biomass heating systems other renewable heating systems are assessed for poultry houses in this review. In terms of poultry production, solar applications are already widely used for chicken growing, egg incubation and chick brooding (Yohanna, State, and State 2010)**.**

In order to evaluate the recent developments and studies on heating applications in poultry houses, Table 2-6 shows experimental and numerical studies on HP applications and solar based technologies. It was clear that geothermal heat pump (GHP) technology is a potentially efficient replacement technology instead of conventional boilers in poultry houses and incubators (Choi et al. 2012; Kharseh and Nordell 2008a; Lin et al. 2012)**.** However, although heating costs significantly decrease with HP applications, it should be noted that actual electricity usage increases because of the power consumption of the HP (Choi et al. 2012)**.**

Table 2-5 Case studies and model samples on innovative heating systems in poultry houses

Methodology	Location/ environmental condition	Aim	Application	Ventilation/ flow rate	Results	
Case study	~−10.8°C outside temperature	Geothermal heat pump (GHP) assessment	A GHP-210.9 kW capacity & 3 oil heaters 116.3 kW capacity (if necessary)	720 m^3/min flow rate heat pump	1.73 vs. 1.62 kg/bird& CO_2 & NH_3 contents⇩;Heating cost ⇩; electricity cost ⇧	(Choi et al. 2012)
Case study & EED model	Syria (~15-18°C freezing for two months)	Heating & cooling demand	Solar coupled to ground source heat pump system (GSHP); floor heating	20 m^3/m^2h	Heating power ~40% ⇩ and heat demand ~48% ⇩ oil consumption ~0.72 Mbbl/year ⇩	(Kharseh and Nordell 2008b)
3-D Simulation	Winter, east Lebanon	Solar water heating system	Parabolic solar concentrator & biogas application	-	84% heat load flock of six-weeks provided 74% energy saving	(Fawaz et al. 2014)
Mathematical model	Alborz/ Iran (25° and 40° north latitudes)	Solar water heating	Solar & auxiliary heater fossil fuel based	Min: 690 m3/h	1944 W (20% of 9782 W heat load)	(Ghaleh et al. 2013)
CFD modelling	~−10.8°C outside temperature	Space heating	GHP	Windowless broiler	From 27,381.60/50,000 heads p.a. to 5,428.21 & 5.5 ppm GHP reduction & 5% body weight increase	(National Institute of Animal Science 2010)
Experiment	Thailand, 33.41°C & 39.87°C RH condition	Incubator	Medium-sized GSHP	Fan in 300 ~ 350 r/min	Hatchability > 91.4% & healthy rate >97.9%, 60% to 65% energy-saving rate & 37.8℃± 0.2℃	(Lin et al. 2012)

2.2. Systematic review of state-of-the-art evaporative cooling systems technologies

In particular, the cooling demand has been increasing continuously day by day because of the impact of global warming and the heating potential of the technological devices used for the systems needed to address it. In the last two years, conventional mechanical air conditioning systems based on a compression mechanism covered 95% of the air conditioning and ventilation market (Zhan et al. 2011). These conventional systems use electrical energy mostly in order to supply fresh and cool air. Electricity consumption of this type is the least efficient way to meet the energy saving and CO2 emission targets set to be achieved by 2040 by the International Energy Agency (IEA).The IEA's aim presented in Figure 2-6 is to decrease the amount of the CO2 emission by 2040 (International Energy Agency 2015).

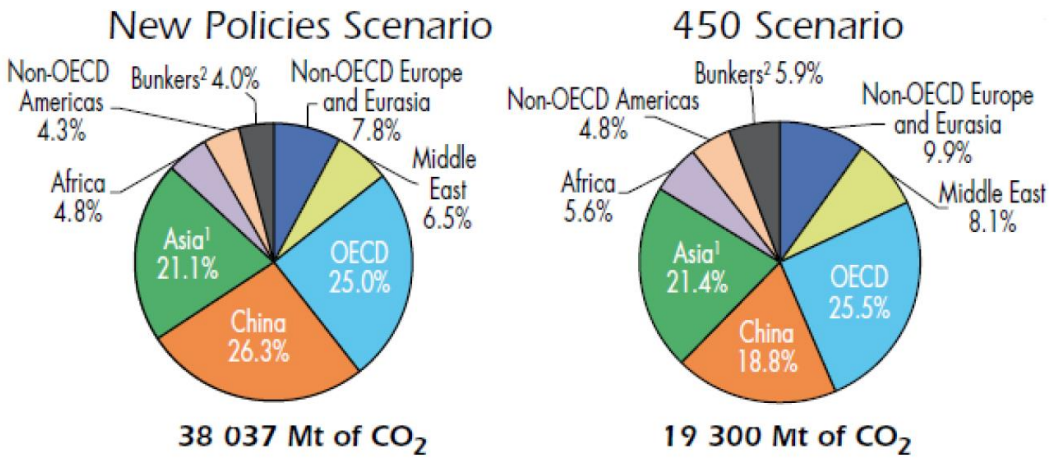


Figure 2-6 CO_2 emissions aimed by region in 2040 (International Energy Agency 2015).

In recent years, the green energy movement has gained momentum as a direct result of the clear and catastrophic consequences of the fuel-based economy. Several areas of research have been developed in an effort to reduce CO2 emissions and make

systems more sustainable. In this context, the aim of this paper is to investigate the current literature on Evaporative Cooling Technologies. There are three main configurations of evaporative cooling systems: Direct Evaporative cooling (DEC), indirect evaporative cooling (IEC) and Direct-Indirect Evaporative cooling (DIEC). All three configurations have some advantages and some disadvantages in their design. The main problems are the high humidity rate in the product air after DEC and the low efficiency rate in IEC because of the heat losses during heat transfers in the system. This paper covers the main idea of the evaporative cooling system configurations and the methods and inventions used in order to improve their efficiency. These methods have primarily focused on chemical solutions, namely desiccants, on material type, on the design of the evaporative pads and on heat exchanger design. Dew-point evaporative cooling system is a breakthrough in the evaporative cooling market in accordance with this purpose.

Researchers continue to study ways by which to meet this target and decrease the high energy consumption currently used for cooling. Recent studies have focused on the evaporative cooling method because of the environmentally friendly and non-compressive features of this innovative system. These methods were used in ancient times and evidence has been found that there was primitive use of evaporative cooling in 2500BC in Egypt, when porous clay jars were used for cooling the air in domestic buildings (X. Cui et al. 2014).

This method has been redeveloped in recent years and its use is widespread for providing air comfort in domestic residences and in agricultural applications involving plants and livestock. Not only have many poultry houses and greenhouses started to be restored with the integration of evaporative cooling systems, but also pig and cattle

farms. Temperature management in farms is a significant problem in hot and dry weather conditions. However, the evaporative cooling method is a reliable system for livestock management in such circumstances and its low energy consumption and eco-design make it attractive to farmers. In these hot regions, the rates of livestock and plant production are even more than in tropical and subtropical regions (Ahmed et al. 2011).

The latent heat in water is used as the air-cooling supplement. Water drops are vaporised under hot climate conditions and during the vaporisation of the water, hot air starts to be cooled by the absorption of the latent heat with hardly any energy change. However, the humidity rate of the air constantly increases because of the vapour coming from the water. Moreover, the enthalpy of the air slightly changes due to the heat and mass transfer between water and air in the system. In addition to the energy saving benefit of evaporative cooling systems, the other advantage of this system is that the working fluid of the system is water. Water is overwhelmingly more environmentally friendly compared with other possible working fluids in conventional refrigeration systems (Xuan et al. 2012a). This system is clearly more efficient in a hot and dry climate because of the greater potential vapour absorption capability.

There are two main types of evaporative cooling system: direct evaporative cooling (DEC) and indirect evaporative cooling (IEC). Some commercially available hybrid (direct-indirect) evaporative cooling designs are also available. There is a direct interaction between water and air in DEC systems, so the product is cold but humidified air in DEC. In IEC systems, however, the product air and water are separated by heat exchangers, and the product air is able to protect moisture during the vaporisation process (X. Cui et al. 2014).

2.2.1. Direct evaporative cooling systems

DEC systems are one of the most efficient systems used in the infrastructures of air conditioning systems, such as cooling towers, humidifiers, and evaporative coolers. Water and air are connected directly, usually with a crossflow condition; the channels are designed to be vertical for water and horizontal for product air. Hot air is circulated in the direction of evaporative cooling pads by the aid of fans in the system, and the heat is absorbed by the vaporised water from the pads, composed of porous, wetted materials (Malli et al. 2011). This method has become widespread in a short time because of its environmentally friendly structure and extremely cheap operating costs, with no compression process.

Figure 2-7 shows the drip-type DEC system. The water distributer sprays water through to the pad material using gravity and capillary pulling forces. The water is circulated by a pump and the air is circulated by a fan. The air and water interact directly in the system, and the latent heat of the water is used to cool the air. Therefore, the process in the surrounding environment is usually assumed to be adiabatic.

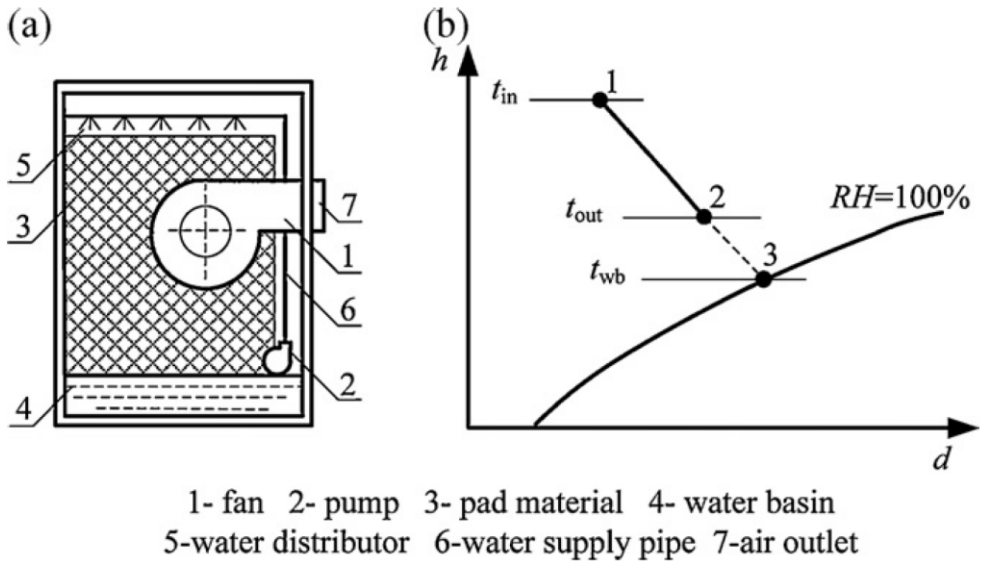


Figure 2-7 (a) Drip-type direct evaporative cooling system (b) Psychrometric graph of the system (Xuan et al. 2012a)

From the psychrometric point of view, the dry bulb temperature can be decreased from point 1 to point 3 by adding relative humidity to the air. However, the design of pads and the interaction process of the air and water have limited flexibility; put simply, heat and mass transfer is not enough, and 100% saturation of air is not possible in DEC systems. Commercially available systems have a range of between 70–95% saturation efficiency, depending on their design and the surrounding properties and application context (Xuan et al. 2012a).

2.2.2. Indirect evaporative cooling systems

Dry air can reach 60% relative humidity with 80% saturation efficiency in dry weather condition areas in DEC systems. However, if the weather is already humid, the efficiency can be significantly lower than in relatively dry weather conditions. Moreover, high humidity is not usually good for human, plant, or livestock life. To address this, Dr Willi Elfert developed the IEC system in 1903. This system is able to cool product air with the same level of humidity as inlet air. The air and water were

separated in this new design because of the moisture balance of the product air (Watt, Koral, and Crow 1986).

This system has two separate air and heat exchanger designs. Secondary or working air is cooled and humidified simultaneously with the latent heat energy of the water by a wet wall surface (X. Zhao, Liu, and Riffat 2008), and the primary/product air absorbs the heat from the working air with the aid of heat exchangers in separate channels. Most IEC systems are designed with crossflow heat exchangers. Because of the stepped processes and design limitations, IEC systems can only reach 50-60% of saturated efficiency (Zhan et al. 2011).

There are various types of IEC system, usually identified by their heat exchanger type. Figure 2-8 shows (a) the tubular, (b), plate, and (c) heat pipe types of IEC system. In a heat pipe IEC system, the condenser part of the pipe is run with working air and the evaporator part with product air. As a result, the product air is cooled with no added humidity, as shown in the psychrometric chart in Figure 2-8d (Xuan et al. 2012a). The efficiency of IEC systems varies between 50 and 70%. Consequently, IEC systems are usually relatively less efficient than DEC systems because of the indirect staged heat and mass transfer (Xuan et al. 2012a; Watt, Koral, and Crow 1986).

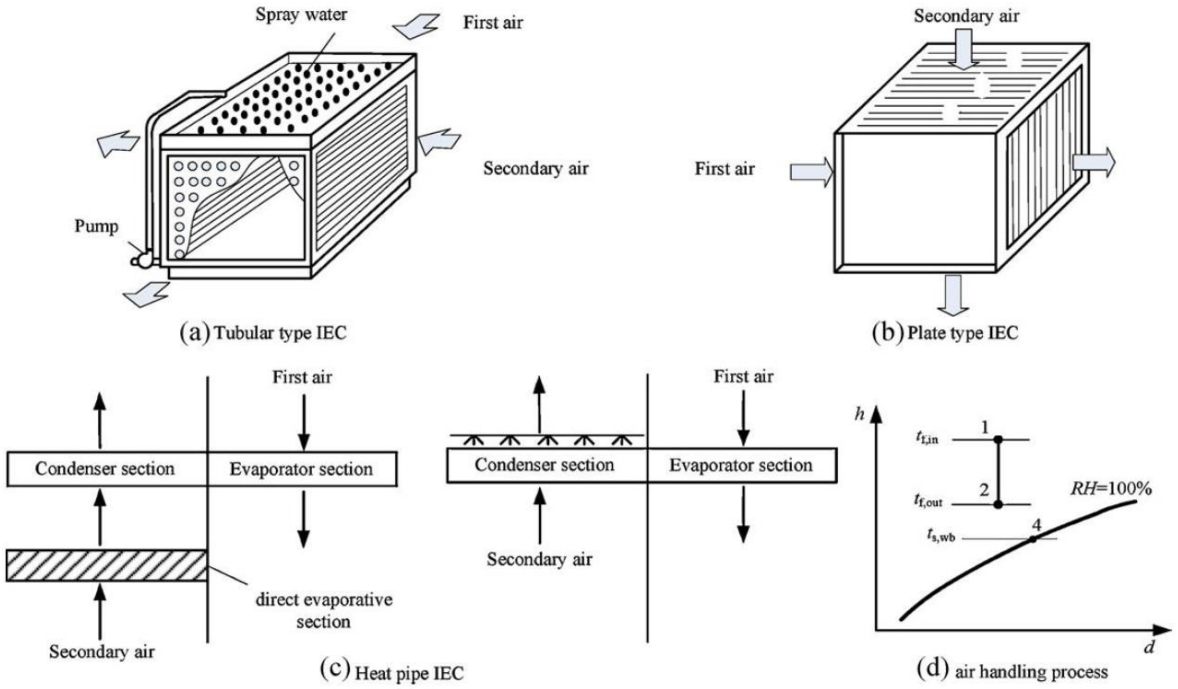


Figure 2-8 (a) Tubular IEC system, (b) Plate IEC system, (c) Heat pipe IEC system, (d) IEC system psychrometric chart (Xuan et al. 2012a)

2.2.3. Direct-indirect evaporative cooling systems

Because of the low efficiency of IEC and the high humidity rate in DEC systems, some studies have focused on the combination of the direct and indirect evaporative cooling methods. Figure 2-9 shows the constructions of the direct and indirect systems and of their combination. These systems have at least two stages. The psychrometric aspect is presented in Figure 2-10. These systems decrease, the negative impact of moisture and heat exchanger losses (Xuan et al. 2012b).

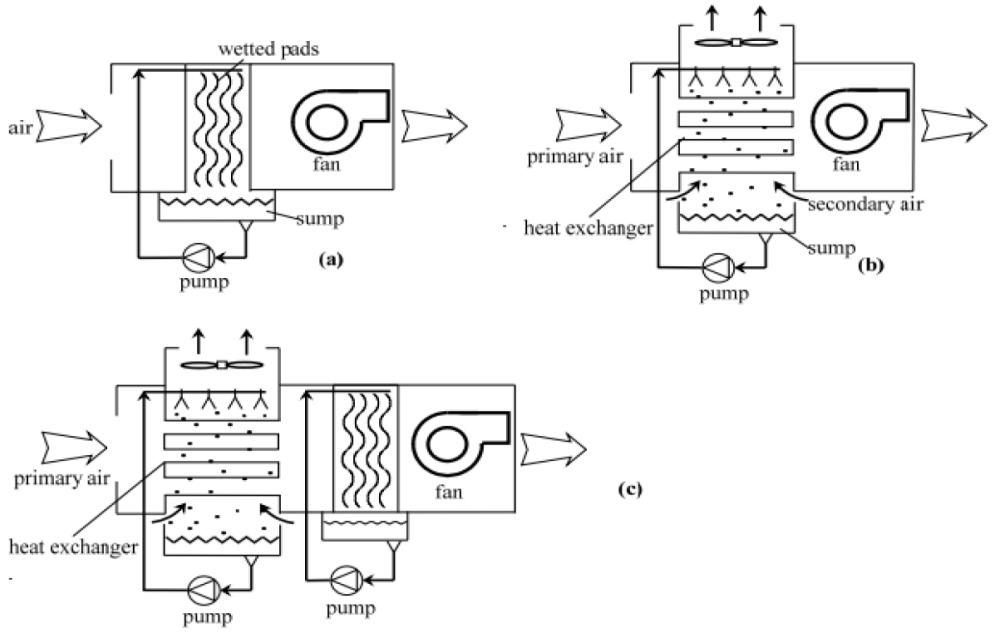


Figure 2-9 Types of evaporative air conditioners (a) Direct, (b) Indirect, (c) Two-stage combined (Bruno 2009)

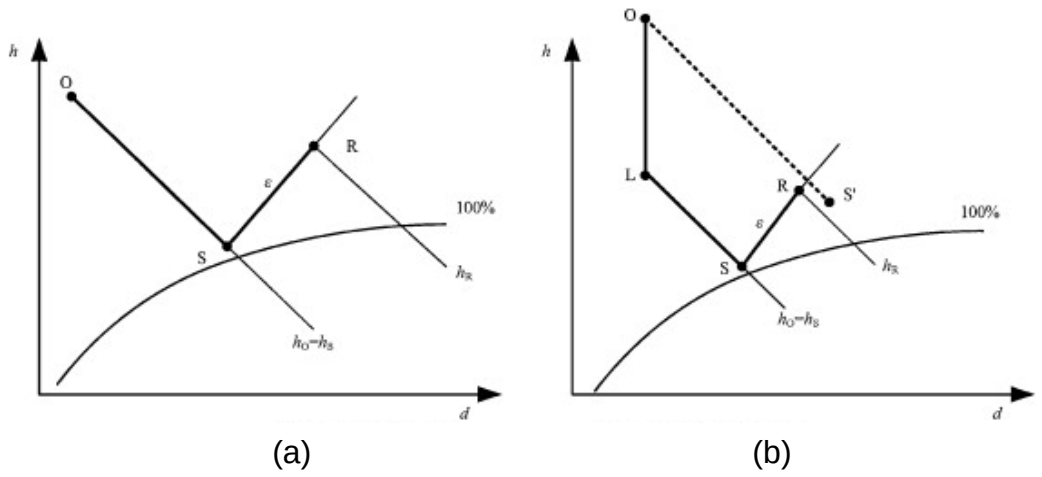


Figure 2-10 Psychrometric charts of (a) A one-stage DEC system and (b) A two-stage IEC + DEC system (Xuan et al. 2012b)

2.2.4. Dew-Point evaporative cooling mechanism – M Cycle

The conventional evaporative cooling systems have some disadvantages because of their cooler design and sudden moisture increase. The high rate of the humidity in direct evaporative systems makes them unhealthy and uncomfortable for people,

livestock and plants. Moreover, the efficiency of indirect evaporative cooling systems is not sufficient because of the losses during indirect heat transfer between channels. The other drawback is that the wet bulb temperature restricts the product air cooling (X. Cui et al. 2014). The cooling temperature cannot be more than the wet bulb temperature in conventional systems and is usually lower than the wet bulb temperature.

Researchers have carried out many studies to produce designs which eliminate the impact of the disadvantages. In Dew point evaporative cooling systems, the even distribution of the holes between the wet and dry working air channels is an important factor for the system efficiency. The design of the heat mass exchanger is the crucial factor in the system performance (Pandelidis and Anisimov 2015). In dew-point evaporative cooling systems, the main parameters are dry-wet bulb temperature, air flow rate, cooling capacity and water evaporation rate (Xuan et al. 2012a).

The M-cycle and the design of the dew-point cooler increase the heat flux of the system and decrease the pressure drops found in previous applications. The cooling (wet bulb) efficiency shown below is clearly regardless of inlet temperature and inlet humidity (Caliskan et al. 2011).

Zhan *et al*. carried out numerical and experimental studies in order to evaluate the thermal performance of an indirect evaporative cooler run with an innovative M-cycle crossflow heat exchanger. The finite element method was used for the analysis with EES (Engineering Equation Solver) software. The dry and wet channels were suggested with maximums of 1.77 m/s and 0.7 m/s respectively according to the iterations in this study. Moreover, the channel air velocity and inlet air relative humidity were better at a low rate with a high working-to-product air ratio in order to achieve

high cooling output. The flow ratio of working-to-product air was also recommended at 50%. A long and narrow channel was observed to be more efficient. This innovative heat and mass exchanger was able to achieve 16.7% higher cooling effectiveness than the conventional cross-flow heat and mass exchanger according to the analysis results. The results were 116.4% wet-bulb effectiveness and 456.2 W cooling capacity with a 20.7 C supply air in the new innovative neat mass exchanger (HMX). The reason was clearly that the pre-cooling process in M-cycle caused the high temperature difference between working and product air (Zhan et al. 2011).

On the other hand, the energy and exergy analysis method with sustainability assessment were implemented for the M-Cycle examination by Caliskan et al. unlike other studies in the literature. Heat and mass energy principles and the second law of thermodynamics were used for the methodology. Ti carry out an exergy analysis, exergy input rate, exergy output rate, exergy loss rate, exergy destruction rate, exergy efficiency, energetics COP, specific total flow exergy rate, specific total flow exergy rate of dry air, specific total flow exergy rate of water, entropy generation rate, primary exergy ratio and sustainability were assessed and compared in reference to temperature according to the exergy analysis in the study. The reference temperatures were selected as 0° C, 5° C, 10° C, 15° C, 20 ° C, and 23.88° C, and 82.9 kPa constant pressure. According to the definitions of the exergy analysis and performance equations above, the results were sufficient with 115% cooling efficiency, 78% dew-point effectiveness and 2.3 COP (Caliskan et al. 2011).

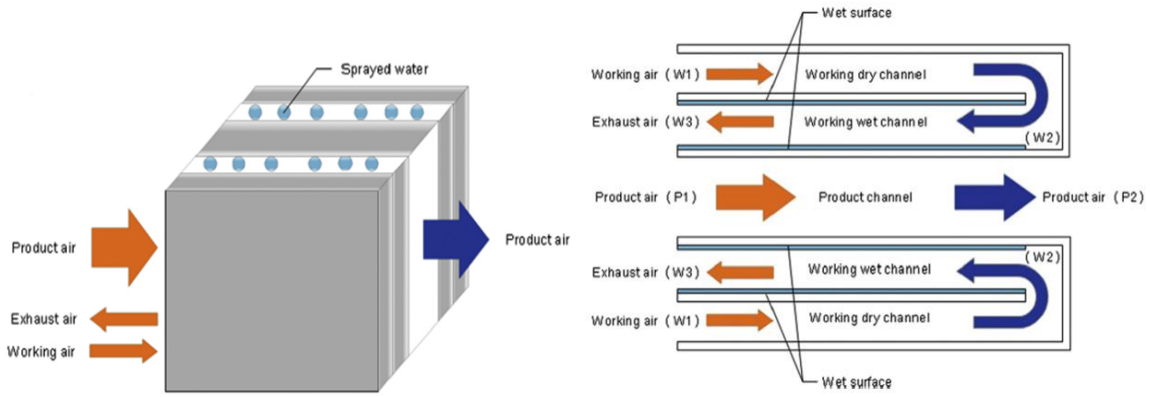


Figure 2-11 One--unit channel pair of the novel dew-point evaporative air cooler schemas (X. Cui et al. 2014)

Cui *et al*. analysed numerically a novel dew-point evaporative air cooler consisting of one product channel and two working channels (See in Figure 2-11) with a counter-flow closed-loop fluent using Ansys/CFD software. The data were taken from the previous experimental results in order to compare them with the results achieved from the mathematical model. According to the mathematical model results with a 7.5% similarity with the experimental findings from literature, the wet bulb effectiveness and dew-point effectiveness varied from 122% to 132% and from 81% to 93% respectively depending with the values of inlet air temperature and surrounding humidity. The study assumed two conditions regarding the fluent condition. First, laminar flow with a variation in Reynold number from 375 to 1250 and second, turbulence flow in case of any damage of the boundary layers (X. Cui et al. 2014).

Rogdakis *et al*. evaluated the M-cycle system experimentally and numerically. A mathematical model was composed using MathCAD. In the study, water consumption rate and efficiency determinations were carried out. The study found efficiency levels between 97% and 115% and water consumption between 2.5 kgwkWhc−1and 3.0 kgwkWhc−1. Greek weather conditions and the similar Mediterranean climate were

assumed to be efficient and good for integrating this innovative system in contrast to moister weather conditions (Rogdakis, Koronaki, and Tertipis 2014).

Anisimov and Pandelidis carried out many studies in order to improve the design of the dew-point evaporative cooler run with an M-Cycle. In their papers, a novel cross-flow HMX (heat and mass exchanger) and a novel combined parallel and regenerative counter-flow two-dimensional heat and mass transfer flow were developed separately (Anisimov et al. 2014; Pandelidis and Anisimov 2015; Anisimov and Pandelidis 2014; Pandelidis, Anisimov, and Worek 2015).

Anisimov and Pandelidis then carried out a numerical study with five different innovative designs, developed by them, of HMXs using the E-NTU method. The innovative designs are shown in Figure 2-12; a modified counter-flow (variant 1), HMX Regenerative (variant 2), perforated regenerative HMX with five holes (variant 3), an original cross-flow HMX with parallel flow of the dry primary and the secondary air stream (variant 4), and a modified cross-flow HMX with a counter flow of the dry primary and the secondary air stream (variant 5). The temperature level of the outlet primary air flow, the wet-bulb thermal effectiveness and the specific cooling capacity per cubic metre were evaluated in the study. The parameters and the sensitivity of the analysis were varied during the study in view of the commercial availability. HMX Regenerative (variant 2) had higher wet-bulb thermal effectiveness than the original and modified crossflow HMX (variants 4 and 5). However, the design was hard to modify to the applications (Anisimov and Pandelidis 2014).

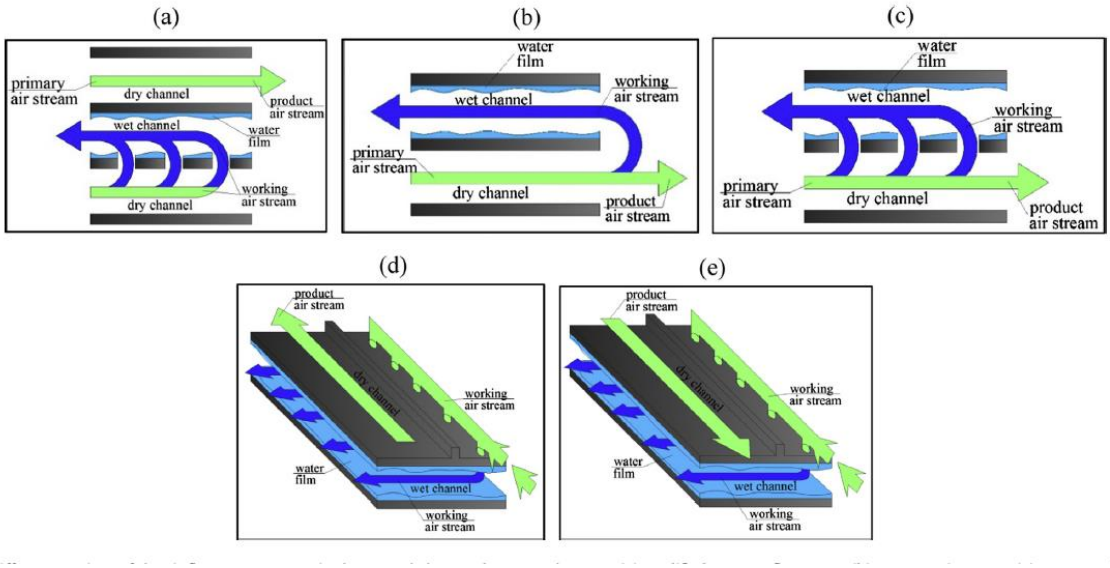


Figure 2-12 (a) modified counter-flow HMX, (b) regenerative HMX, (c) regenerative HMX with perforation, (d) cross-flow HMX, (e) modified cross-flow HMX (Anisimov and Pandelidis 2014)

Moreover, Anisimov and Pandelidis compared seven different types of HMX based on the M-Cycle. The E-NTU method was used for computational analysis with a two-dimensional heat and mass transfer model. The study aimed to improve the current structure of the M-cycle HMXs by analysing the precooling part of the working air. Figure 2-13 shows the working and product air in 3D aspect. The dependency of inlet air temperature parameters; the working to primary air ratio and the flow rate were considered in this comparative study. The results showed that significant improvements can be achieved by the editing of the holes and design size, and the study recommended a ration of at least 6:2 between primary and working parts in lower air flow ratios or 7:1 in lower or higher flow rates (Anisimov, Pandelidis, and Danielewicz 2014).

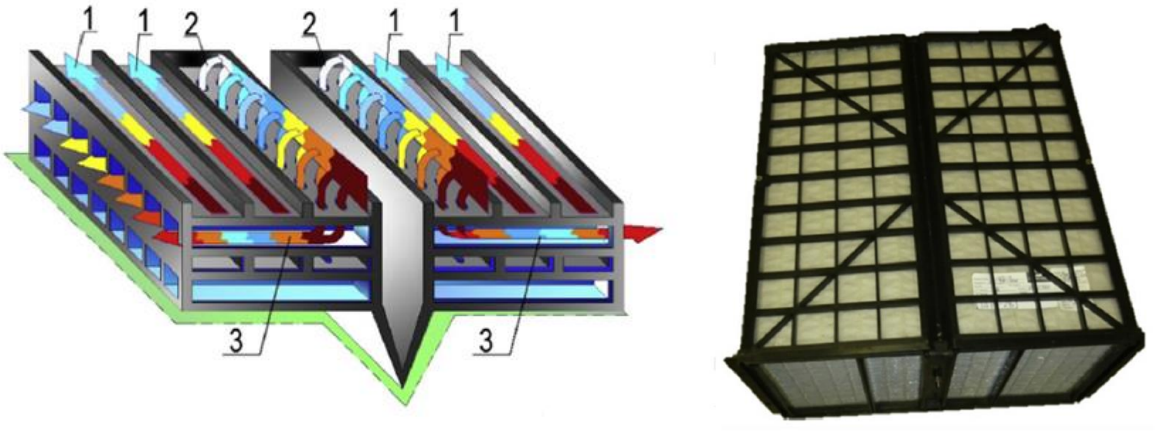


Figure 2-13 (a) 3D scheme of M Cycle: 1 – primary air stream; 2 – working air dry channel; 3 – working air wet channel. (b) Photograph of an HMX (Pandelidis and Anisimov 2015)

Jradi et al. described a proposal for a psychometric energy core (PEC) system, and tested a modified dew point cooling system, including a crossflow exchanger, using a numerical analysis method in MATLAB. The proposed system was investigated and the various parameters which could affect the system efficiency such as intake air temperature, velocity, ratio and channel height, and length were considered. The results of that study showed that this method was both efficient and economical in a variety of conditions (Jradi and Riffat 2014).

Another valuable study is the PEC system which uses wind catcher technology. In that study, the humidity issue was addressed and partly overcome. According to the test results, the PEC system was found to work productively in high humidity air conditions. It has also been reported that the effectiveness of the system is directly related to the yield of the heat exchangers (Elzaidabi, Omer, and Riffat 2010).

2.2.5. The focus of recent studies

The development of the existing technology is taking more time than had been anticipated. The first step for development is to understand the main parameters of the current technology. Surface and thickness size, material technology, penetration numbers and size, relative humidity, inlet/outlet air flow rate and volume of water are the main parameters of the system output. There are various materials and designs for the development of evaporative cooling technology in industrial, residential and agricultural markets such as livestock (pigs, poultry), greenhouses and warehouses (Malli et al. 2011).

Zhao et al. carried out a numerical study in order to determine the main factors of system efficiency in a novel counter-flow indirect system. The results of their study showed that the air velocity, pad design, materials and working-to-intake-air ratio are the main impacts on the system output (X. Zhao, Liu, and Riffat 2008).

This current review is going to present the findings of recent studies conducted with innovative pad materials and design, innovative heat and mass transfer exchangers called dew-point evaporative cooling systems, and dehumidification of the product air. Although there are various evaporative pad applications on the market, it is not easy to obtain the specifications of their materials (Malli et al. 2011).

2.2.5.1. Evaporative Cooling Pad Materials and Design

The conventional pad materials are metal, cement, wood, plastic and glass. Recent studies of producers have focused on cellulose paper as a pad material due to its relatively low price. Moreover, these types of pad are more environmentally friendly, energy conservative, small sized and light. Therefore, the materials currently used in pads are crucial for the system efficiency, particularly the rate of water absorption

(Malli et al. 2011).With regard to the size of this cellulose paper design, the thickness range varies from 10 to 30 cm per unit area (Ahmed et al. 2011).

Ahmed *et al.* experimentally analysed the three commercially available pad materials, CelDek, straw and sliced wood, in greenhouses in Sudan. The impacts of pad material on plant growth and the environment were examined at 8am, 1pm and 6pm in summer conditions by measurement of the temperature and relative humidity. Although no significant temperature differences were observed at 8am in the greenhouse, the temperature difference increased substantially in the zones which used different types of pad system at 1pm and 6pm. The zone which had sliced wood pads had the highest temperature. The lowest temperature with the highest relative humidity was found in the area constructed with straw pads. With regards to the growth rates with reference to the different pad materials, sliced wood pads produced the highest plant output efficiency presented in Figure 2-14, which was 820 g (Ahmed et al. 2011).

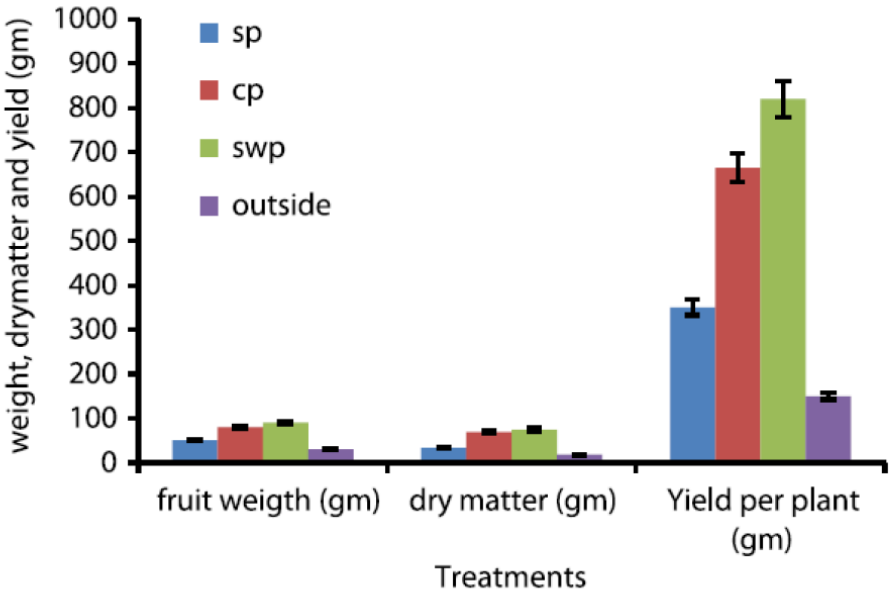


Figure 2-14 Fr uit weight, dry matter and yield (gm) in celdek pad (cp), straw pad (sp), sliced wood pad (swp) and outside (Ahmed et al. 2011)

Liu *et al.* also carried out an experimental study of various pad materials, GLASdek, CELdek, metal (aluminium foil), porous ceramic and PVC pads (See in Figure 2-15). Their dependence on air velocities and water spray rates were separately performed. Cooling efficiencies were observed from higher to lower rates respectively on GLASdek, CELdek, porous ceramic, PVC and aluminium foil in the 2.5 m s−1 air velocity and the 2400 kg m−2 h−1 spray water conditions (Xuan et al. 2012a).



Figure 2-15 The five pad materials (Xuan et al. 2012a)

The main parameters are a good heat transfer ratio, low pressure drop and steady water consumption. Using these parameters, Malli et al. carried out an experimental study comprising a performance comparison between 5090 and 7090 types of locally and commercially available cellulose material pads because of the benefits mentioned above. Steady state conditions were generated in the two systems comprising corrugated paper channels in a low moisture, low velocity and temperature-controlled polyethylene wind tunnel. The sizes of the proposed system were 0.5×0.5 m2 area with 75, 100- and 150-mm thicknesses. Thermal and hydrodynamic efficiencies were compared in various experiments. It was observed that the effectiveness and humidity variation inversely decreased with a rise in velocity. The maximum benefit was reached in the 1.8 m/s velocity and 150 mm thickness in 5090 type (Malli et al. 2011).

Aspen and Khus fibres have become widespread as evaporative cooling pad materials. Jain et al. conducted an experiment consisting of coconut fibres and palash fibres as alternatives to aspen and khus fibres. The palash fibre pads showed 13.2% and 26.31% greater efficiency than respectively aspen and khus fibre in constant air flow rate condition. Moreover, coconut fibres were found to be 8.15% more efficient than khus fibres. Khus pads showed the lowest pressure drop and aspen pads had the highest pressure drop. The results indicated that the proposed palash fibres showed lower pressure drops than the more popular material aspen. The pictures of thea palash fibre pas and a coconut fibre pad are presented in Figure 2-16. The recommendation of that study was that it is good to use coconut and palash fibres in the evaporative cooling market due to their lower prices and comparatively higher efficiencies (Jain and Hindoliya 2011).



(a) *(b)*

Figure 2-16 (a) a palash fibre pad; (b) a coconut fibre pad (Jain and Hindoliya 2011)

2.2.5.2. Dehumidification and Desiccant-Based Systems

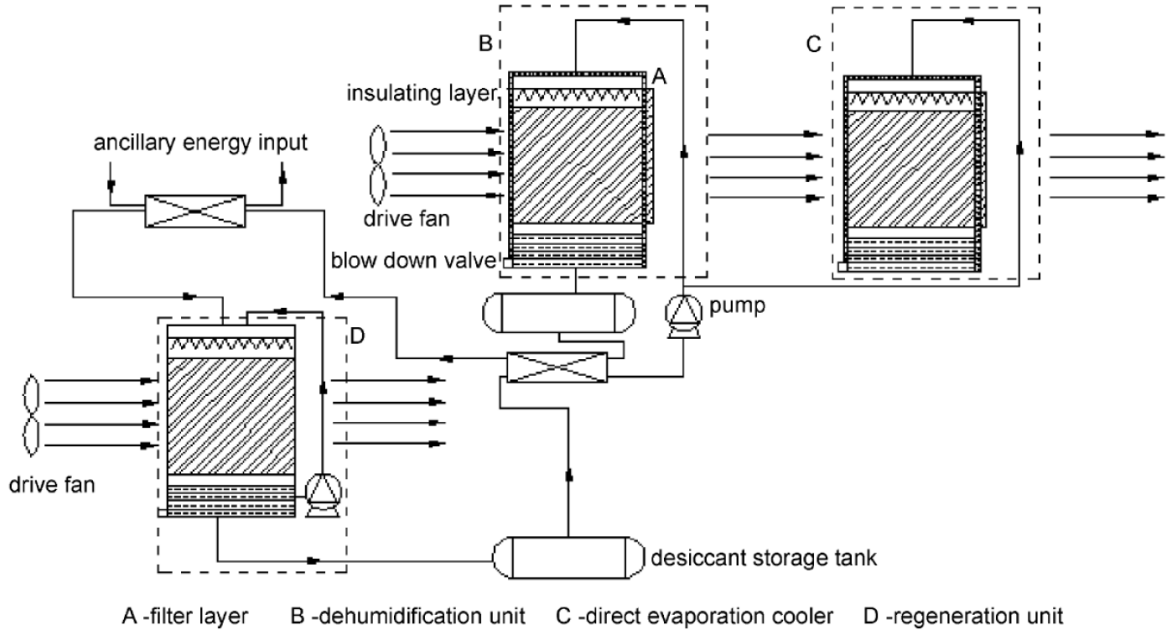


Figure 2-17 A liquid-desiccant dehumidification air system schema (Mei and Dai 2008)

In order to prevent a high humidity rate of the product air, another solution investigated was desiccant evaporative cooling systems. Basically, a desiccant solution is able to decrease the system humidity using its chemical properties. Latent heat loads can be eliminated with the aid of the desiccant solutions [26]. The structure of a liquid-desiccant dehumidification and air-conditioning system comprises a dehumidifier, a regenerator, liquid-desiccant storage and a filter layer (Figure 2-17). In these systems, the packing material type is one of the main impacts on the system performance. There are three main flow patterns in their applications: parallel flow, counter flow and the predominantly used cross flow (Mei and Dai 2008).

A solar-assisted desiccant cooling systems is one of main applications of dehumidification air conditioning systems. A project was carried out in Riesa with

various conditions based on the outside weather conditions in Germany. The size of the room was 330 m3. The condition air flow was 2700 m3/h. For the system, a 20 m^2 flat plate collector and a 2 m3 hot water storage tank were integrated. Active heating, heat recovery, free ventilation, adiabatic cooling and desiccant cooling (ventilation cycle) were applied according to the weather conditions, loads and profiles. The variable temperature of the regeneration unit was between 50 and 70° C. The output achieved was 18 kW with 0.6 COP and 76% solar fraction(La et al. 2010; Henning et al. 2001). In another study carried out in Sintra, Portugal, funded by the EU's THERMIE-program, ATECNIC ,the system was constructed with a heat recovery wheel and a nozzle type humidifier: 75 kW cooling power and 70% fraction were achieved in this unit (La et al. 2010).

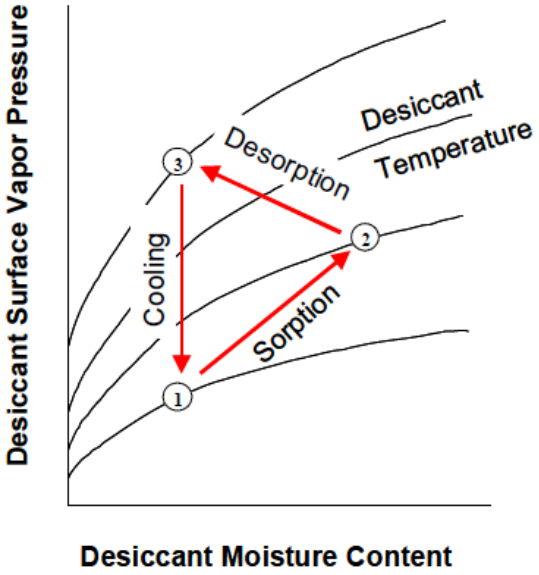
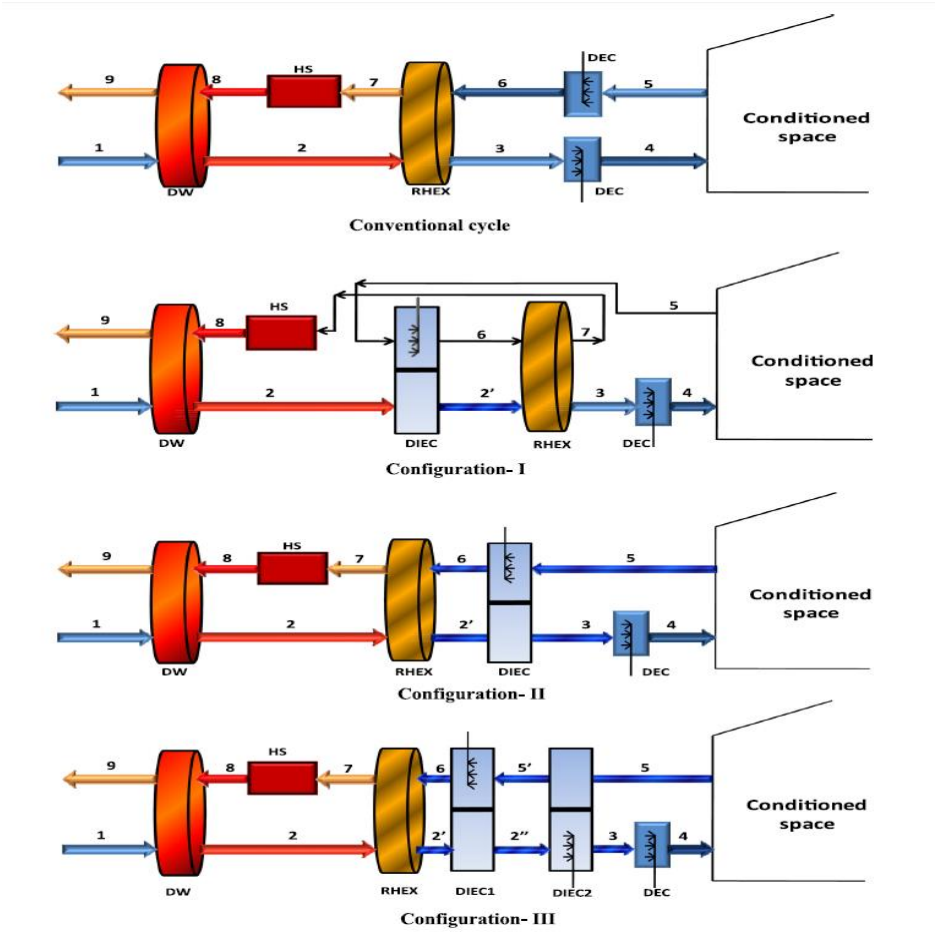


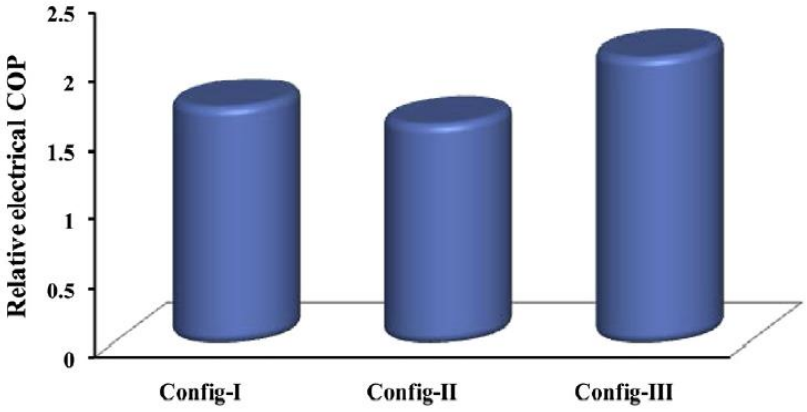
Figure 2-18 Desiccant dehumidification and regeneration process (Nayak 2005)

A desiccant works in three processes, sorption, desorption and desiccant cooling, as shown in Figure 2-18. An air conditioning system and a liquid desiccant dehumidification system are unified to achieve comfortable air conditions. With this integration, which is called a hybrid liquid desiccant system, the latent heat loads' impact in the evaporative cooling systems could be decreased. These innovative systems can be categorised as a hybrid liquid desiccant and direct evaporative cooler (DEC), a hybrid liquid desiccant and indirect evaporative cooler (IEC) and a hybrid liquid desiccant and indirect–direct evaporative cooler (DIEC). Experimental and numerical studies based on desiccant solutions of CaCL2, LiBr and LiCl evaporative cooling systems have been carried out in order to evaluate system effectiveness and COP (Mohammad et al. 2013).

Elgendy et al. compared three innovative desiccant evaporative cooling systems varying from 30° C to 40° C ambient air and from 0.01 to 0.02 humidity ratio conditions. The study aimed to compare the efficiencies of the innovative configurations and conventional air conditioning system as shown in Figure 2-19a. Conventional cycle comprises four elements, a desiccant wheel (DW), an evaporative cooler, a rotating heat exchanger (RHEX) and a regeneration heat exchanger (Elgendy, Mostafa, and Fatouh 2015). According to the results based on the simulations run with EES and TRNSYS, the DIEC unit integration after adding a rotating heat exchanger was the most effective method with respect to the system COP (Figure 2-19b).



(a)



(b)

Figure 2-19 (a) Conventional cycle and system proposed configurations, (b) COP results (Elgendy, Mostafa, and Fatouh 2015)

Woods and Kozubal carried out experimental and numerical studies with a liquid desiccant air conditioner integrated with a dew-point evaporative air cooler with counter-flow configuration (Woods and Kozubal 2013).

Gao et al. worked with solid desiccant evaporative cooling systems. A computational investigation was the aim of the study. The main performance parameters were calculated based on the Number of Transfer Units (NTU) methods. This method was used for finding optimum inlet temperature and humidity which were suggested to be 35° C and 18 g/kg respectively in the study. System COP and the air rate per unit cooling capacity, referred to as hybrid performance (Em), were calculated in order to investigate the main parameters of the system. According to the results, the inlet process air temperature, the inlet process air humidity ratio, the regeneration air temperature, the inlet process air face velocity, the NTU in heat exchangers, the NTU in the indirect evaporative cooler and the working air mass ratio are the main factors affecting the efficiency of solid desiccant cooling systems (Gao et al. 2015).

Armanasco et al. analysed the performance of an open-cycle solar cooling plant based on a liquid desiccant evaporative cooler in Northern Italy. This experimental study was carried out in summer weather conditions. In a variety of conditions, the system was found to be always more efficient than conventional systems and this innovative unit was claimed to offer an effective solution for solar cooling applications (Armanasco et al. 2015).

2.3. Systematic review of hybrid PVT assisted with HP systems

The UK government has targeted a decrease of around 80% in greenhouse gas emissions (GHGs). To achieve this, the government has created several incentives. For example, in 2014, the ‘Renewable Heat Incentive (RHI)’, which is a kind of

promotion law, was established in order to reach the carbon-free energy goal (Suleman, Dincer, and Agelin-Chaab 2014). In the modern world, one of the most significant issues regarding energy consumption is the use of heating systems in winter. With regards to carbon-free heating, heat pump systems are predominantly used not only because of their impact on carbon reduction, but also due to their energy efficient properties. By 2050, European countries hope to have reduced CO_2 emissions by 90%, meeting heating demand with a replacement for natural gases. Approximately 60% of these energy needs can be provided by heat pumps (Carvalho, Mendrinos, and De Almeida 2015).

The International Energy Agency (IES) has announced that fossil fuels are used to provide approximately 81% of the primary energy required worldwide. Although fossil fuels are in danger of depletion, energy demand has been increasing considerably with the development of various technologies. This has also caused dramatic environmental damage such as acid rain and global warming (Suleman, Dincer, and Agelin-Chaab 2014). Sustainable energy programs and targets are increasingly being set up all over Europe. Denmark, for example, has the ambitious target to get rid of the dependency on fossil fuels completely by 2050. These policies push other countries to go further. In order to achieve these targets, the efficiency rates of heating (space and domestic hot water) and electricity are crucial (Carvalho, Mendrinos, and De Almeida 2015).

In recent years, the green energy movement has gained momentum as a direct result of the clear and catastrophic consequences of the fuel-based economy. Several areas of research have been developed in an effort to reduce CO2 emissions and make systems more sustainable. In this context, the incorporation of green energy solutions

in architectural and building designs has become an important tendency which has brought to light the notion of ‘PV/Thermal systems. Waste heat is the main challenge in both conventional fuel-based and renewable energy technologies. Researchers work hard to find a solution for the huge thermal energy losses which occur. Developments have been made in combined heat and power systems in order to utilize the waste heat in these systems. With regard to solar power, this chapter discusses a hybrid system which has been developed to provide more efficient power and utilized heat for domestic hot water or any other heating demand. The waste heat which is produced can be utilized with the aid of a recently developed innovative system called a heat pump. The combination systems of a PVT system and a heat pump look to improve a PV panel’s efficiency, and subsequently to provide space heating or domestic hot water with various cost-effective environmental control systems.

More than one third of the total energy load of a building is comprised by the architecture used to meet various demands (Zhan et al. 2011; Caliskan, Dincer, and Hepbasli 2011). Around half of this demand originates from the ventilation, heating and cooling comfort systems in the building(Caliskan, Dincer, and Hepbasli 2011).

The aim of this section is to understand the basic idea of the technologies and integrated systems reported in studies carried out in recent years. From the basic initial research carried out, it is expected that increasing the speed at which the heat can be extracted from the PV will increase the efficiency of the PV. It is important to examine any research which has been carried out on all these innovative technologies with the aim of benefitting from their findings in this proposed project. The investigated solar heat pump systems comprise two main heating technologies: solar PVT and HP, the

latter of which is also considered as a cooling device due to its reverse impact on the system.

2.3.1. PV panels and photovoltaic technology

All renewable energies ultimately derive from the sun, including such as wind, wave, hydro, or biomass energy sources, which can be called indirect solar energy. For example, the earth's surface has an uneven distribution of the light coming from the sun; this causes differences in pressure which are the main reason for wind energy. Unbalanced evaporation and precipitation on the earth cause the hydrological cycle, and the process of the photosynthesis of biomass resources also depends on the level of light coming from the sun, Direct solar energy, harvested by PV cells to generate electricity (explained in more detail below), is a clean and inexhaustible form of energy which can make a significant contribution to global energy problems.

According to the IEA, a viable method of renewable electricity generation is solar photovoltaic generation, which was the most prolifically growing renewable energy technology during the 2000s. It is assumed a third of global energy demand can be met by solar energy by 2060(International Energy Agency 2017). Figure 2-20 presents the main types of the PV cell technology based on the material of PV panels.; Crystalline silicon, Organic/polymer, Brid PV cell, thin film technology and Dye-synthesized Panel (Gouda et al. 2019).

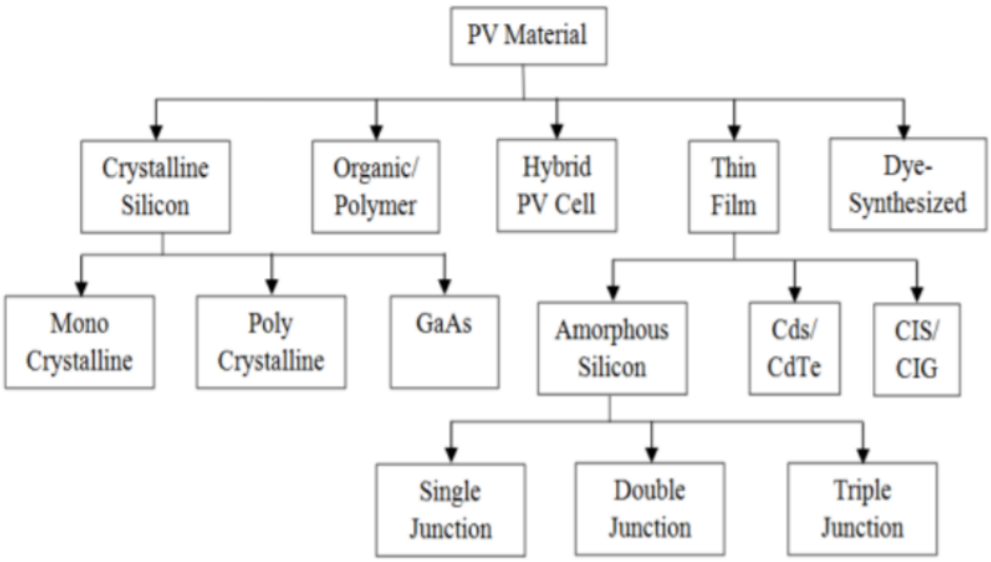


Figure 2-20 The types of PV panel in accordance to its raw material (Gouda et al. 2019)

There are four well known and developed types of PV cells on the market; (1) Monocrystalline (2) polycrystalline silicon (3) Thin-film and (4) Amorphous silicon(A-Si)(Gouda et al. 2019).

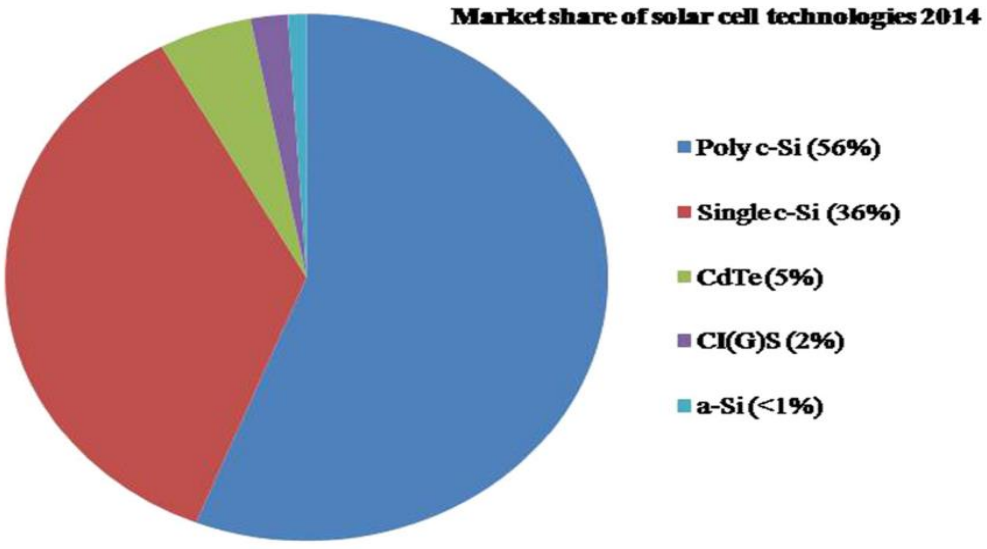


Figure 2-21 The variation of the Radiation on the PV panel (Ramanujam et al. 2016)

Although, monocrystalline PV cells have approximately 5% higher efficiencies than polycrystalline silicon, because of its complicated manufacturing process and therefore higher price, The PV market has been dominated by polycrystalline silicon with 56% market share (Figure 2-21). Graphene based solar cells, copper indium gallium selenide (CIGS), and cadmium telluride (CdTe) are also recent developed technologies on PV market (Ramanujam et al. 2016) (Mukanova et al. 2018) (Him et al. 2020).

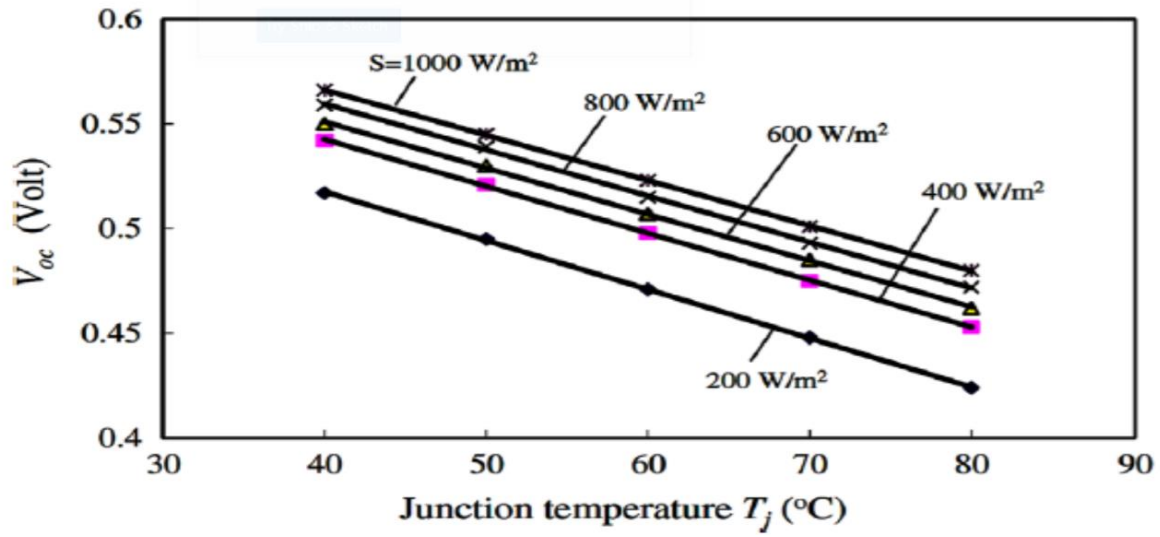


Figure 2-22 The variation of the temperature on the PV (Luque and Hegedus 2011)

There is an adverse effect on the temperature on the surface and the voltage of the Solar PV cell, therefore, Solar energy harvesting requires thermal collectors for heat management, PV cell technologies for electricity generation (Figure 2-22). The hybrid PVT systems is recent promising technology for both heat and power production. For utilising wind and hydro sources, wind and water turbines are used to convert the kinetic energy and potential energy into electricity. Moreover, compared with other renewable technologies, solar energy technology has some distinct advantages. For

example, because solar radiation energy is converted directly into electricity, there is no noise problem; also it does not require any specific additional construction to be built, because solar PV arrays can be retrofitted to most structures (usually roof-integrated), and solar cell systems are not as complex as those of other renewables.

However, there are many challenges to the use of solar energy sources, such as low radiation levels in certain parts of the globe and at certain times of year, high operating temperatures, soiling, and shading. The main parameters of PV technologies affecting their performance are cell temperature, the mass flow rate of the cooling fluid, insulation intensity, moisture rate, and dirt (Rahman, Hasanuzzaman, and Rahim 2015).

2.3.2. Photovoltaic thermal (PVT) hybrid systems

One of the most important problems in PV technology is high operational temperatures and their adverse impacts on PV performance. Waste heat energy needs to be extracted from PV panels in order to increase performance. The total energy efficiency can be much improved by utilising waste heat with the application of various technologies (e.g. using water as a coolant, which simultaneously functions as a source of heating for water for use, reducing the need for water heating using gas or electricity). Cooling PV systems is crucial for their working principles. A hybrid PV system seems to be a solution for this problem. This system can provide power and heat simultaneously and by the utilisation of the waste heat, reaching total efficiencies of around 70% (Zhang et al. 2012).

In order to understand the idea behind the cooling principle of PV and PVT systems, it is important to understand the working principle of solar thermal collectors. There are two main types of solar thermal system: the ‘flat plate collector’ and the ‘evacuated

tube solar collector'. Almost all kinds of hybrid PVT system are developments of either flat plate collectors or evacuated tube solar collectors. Flat plate collectors are commercially available for temperatures ranging between 20-80°C. Although flat plate collectors have a simple design and low maintenance costs, evacuated tube collectors are more resistant to (and efficient in) bad weather conditions, and operate in a temperature range from 50-200°C (Sabiha et al. 2015).

Air-based PV systems make use of naturally or forced ventilated air for the extraction of the waste heat from a PV. This heat is then used to heat a place or water. The important parameters in this system are air mass, inlet and outlet fluid temperature, irradiance level, and operating cell temperature (Xudong Zhao et al. 2011)**.** Forced heat makes the system more efficient because of the faster heat transfer between the air and the PV panel (Garg and Agarwal 1995). This benefit could probably compensate some of the energy consumed in the forced convection.

The working principle of a water-based system is close to that of forced-air PV panels, except that water instead of air acts as the fluid. The operating temperature decreases because of water heat absorption inside the pipes. Inlet and outlet temperatures, mass flow rate, the thermal conductance of water, and the length and depth of the pipes are the main parameters for these systems (Charalambous et al. 2007).

Moreover, heat pipe-based PV systems have many advantages in their working principles because they use both thermal conductivity and phase change simultaneously to achieve heat transfer. They consist of three parts: evaporator, adiabatic, and condenser sections (Zhang et al. 2013). Heat source increases the heat in the evaporator section, and heat sink takes the heat in the condenser section (Figure 2-23).

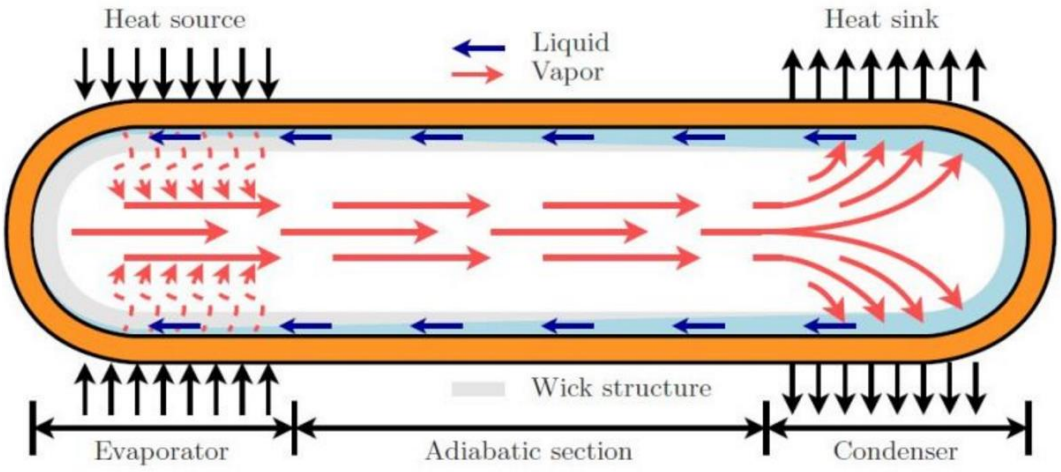


Figure 2-23: Heat pipe sections (Zhang et al. 2013)

This thesis thoroughly analyses each of PVT system based on heat pipes in two main steps:

- To determine the convection and conduction heat transfers between the surface of the PV panel and the covered glass on top of it.
- To calculate the heat transfer between the back surface of the PV panel and the evaporation part of the heat pipe (Duffie, Beckman, and Worek 2003).

There is an increasing trend of using heat pipe to increase the efficiency of PV panels. Because of the high conductivity of heat pipes, heat losses during the extraction of heat can be decreased, and the working fluid inside the heat pipe is allowed to pass through the inside tubes with hardly any risk of corrosion or freezing (Gang et al. 2011). Another impressive feature of this hybrid system is that heat pipe can convert the heat flux from a lower to a higher value by changing the condenser area.

The limited heat capacity of air and its low density make air-based PV panels less attractive solutions, because these parameters affect the performance of heat extraction by air (Royne, Dey, and Mills 2005)**.** Moreover, air-based PV panels have lower efficiency than other types because of the poor thermal mass of air inside the

panel. This poor efficiency led to the development of water-based PV panels, as supported by recently public house experimental research studies (Xu et al. 2012).

PHE based PVT module study is crucial to this field. A performance assessment and economical assessment by Buker et al. using this innovative application concluded that 10.3 MW sufficient annual energy savings are possible with £19,456.14 net present value (NPV) for a 25-year project life span (Figure 2-24) (Buker, Mempouo, and Riffat 2015).

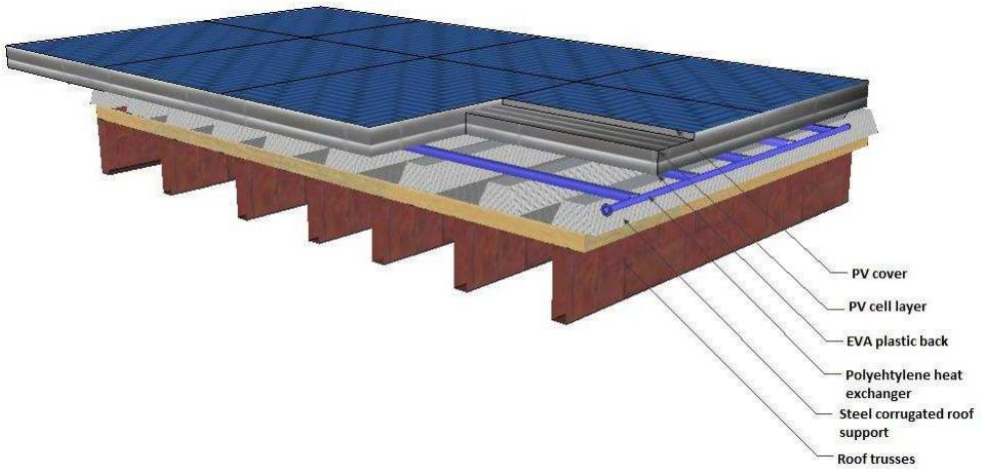


Figure 2-24 PHE based based PVT system(Buker, Mempouo, and Riffat 2015)

2.3.3. Heat pump technology

HP technology utilises low temperature source, converting it into higher temperature source. This device provides domestic hot water or space heating at a higher temperature. The heat pump functions through the absorption of low temperature heat in the evaporator, then the compressor starts to work with the consumption of energy, usually electricity. Finally, the condenser releases the high temperature heat with the condensation of the vaporised and compressed refrigerant in the evaporator and compressor, respectively (Miah et al. 2015).

Coefficient of performance (COP) and seasonal performance factor (SPF) are the main parameters of HPs, which are dependent mostly on the climate in the application zone, working conditions, heating load, the design of associated infrastructure (e.g. radiators and coils), and the fluid properties inside (Rahman, Hasanuzzaman, and Rahim 2015). COP can be calculated by the electrical energy consumed during working and the difference between heat sink and heat source, as shown in the following equations. SPF can be similarly calculated by the total thermal energy provided and the total electric energy consumed during the whole season. It is not wrong to say that the COPs are the average SPFs. The heat difference between the heat source and heat sink is the crucial factor that specifies the system's performance (J. Luo et al. 2015).

$$COP = \frac{Q_b}{W_{hp} + W_{cp}} \quad 2\text{-}1$$

$$SPF = \frac{Q_b}{W_{cp}} \quad 2\text{-}2$$

Proposal applications are found to be very efficient and cost-effective systems in comparison with alternative, conventional systems. Although the results show that these technologies have great potential to save energy, there are two main challenges for applications in poultry houses: the high humidity rate for evaporative cooling applications; and the high electricity consumption of HP. This thesis uses a case study with a dew point evaporative cooling system applied in poultry houses to maintain a stable humidity rate, particularly in hot and humid weather zones. Moreover, PVT technologies integrated with HP are proposed in order to resolve high electricity costs and utilise renewable electricity generated by the PV system.

It is concluded that ventilation rates, with their significant energy consumption, low production performance, and feed conversion ratio due to airborne pathogens and high veterinary costs are the primary environmental and economic factors of concern in the sector. Moreover, swine and poultry viruses, such as PRRSV, PEDV, and HPAIV, are economically important pathogens that can spread via aerosols. The poultry-rearing industry clearly needs practical and sustainable solutions to its current impasse of egregiously high fossil fuel consumption and associated GHG emissions, as well as its direct environmental impacts from harmful gaseous emissions arising from poultry production. Recent studies on air purification reviewed in this chapter demonstrate promising routes in sustainable mitigation strategies for odour, VOCs, NH_3, and H_2S. Therefore, in conclusion, an innovative TiO_2 coated air purification and lighting system design is strongly recommended as a solution to the market failure detected by the literature.

In terms of cooling capacity demand, almost all previous studies focused on investigating evaporative cooling systems to address the heat stress problem among poultry, and to reduce the build-up of harmful ammonia gas in the poultry houses in hot and dry climatic zones. The results indicate that these systems have great potential to overcome the challenges in poultry houses. Meanwhile, regarding high fossil fuel consumption, heat pump and solar energy-based technologies are also applicable for poultry houses, particularly in environments that can benefit from high solar radiation rates in the summer, and low and intermittent solar radiation in the winter, such as the UK.The other benefit of these innovative systems is cooling support within the integrated cooling and heating heat pumps. These can either cool the refrigerant in summer conditions or warm it in winter conditions. Based on research in Portugal, the replacement of natural gas by large scale heat pumps can significantly reduce carbon

emissions. Air source heat pump (ASHP) systems have very environmentally friendly benefits. However, the risk of freezing in the coil surfaces outside causes their efficiency to decrease in the cold weather. A lot of research has been conducted with the aim of finding a solution to the risk of surface freezing, and it has been shown that relative humidity can affect the duration of the freezing and the COP of a system (J. Luo et al. 2015).

Luo *et. al* has examined the performance of GSHPs in winter and summer conditions, with its heating and cooling supply (Figure 2-25). The GSHP COP and EER were calculated at 3.0 and 8.0 on a winter day and summer day respectively. It was clear that, according to this study, GSHPs have much higher efficiency in the summer. From a seasonal point of view, over a longer term of 4 years, the EER calculated increased by 8.7% for cooling, but in contrast COP decreased by over 4.0% for heating. Therefore, it is important to consider the system's profit over a long term period. (J. Luo et al. 2015).

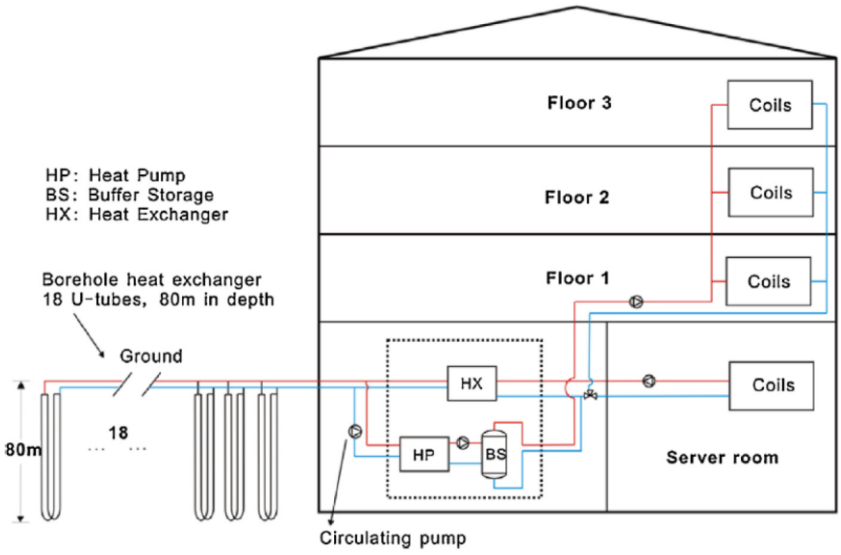


Figure 2-25 GSHP system (J. Luo et al. 2015)

The types of heat pumps and their use rates are presented in Figure 2-26 (Hannon 2015). According to this figure, the dominant type in the HP market is the air source heat pump (ASHP). Recent innovative projects have focused on GSHPs, which not only have higher efficiency (with one unit of electricity creating 4 to 6 units of heating or cooling energy) but are also resistant to cold weather conditions. The heat difference in the ground is stable because of the temperature existing in the ground. Moreover, GSHPs have extra parameters that can influence a system's efficiency, such as borehole depth, ground and soil properties (thermal conductivity), heat exchanger rates, and fluent velocity inside the ducts. These systems utilise a special kind of heat exchanger called a Borehole Heat Exchanger.

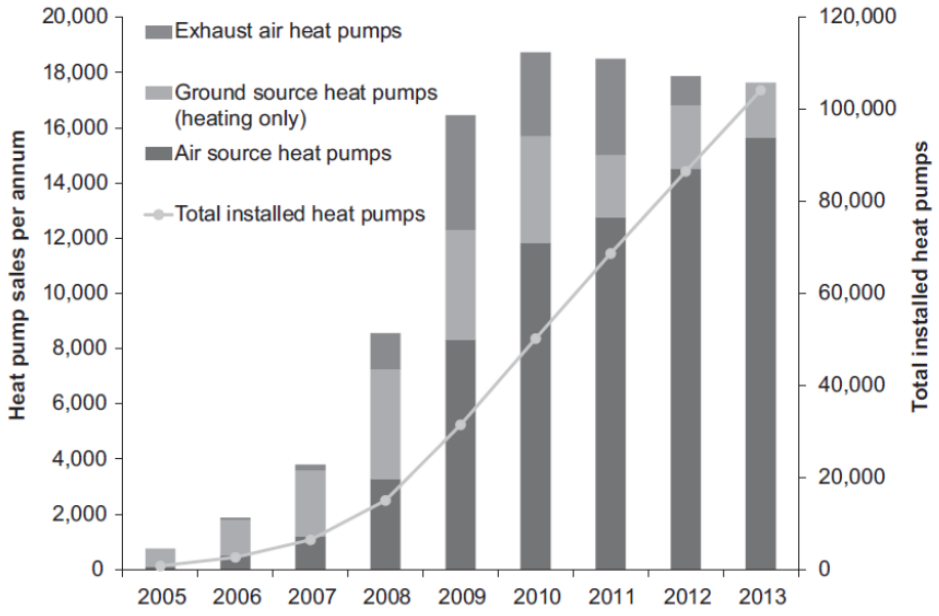


Figure 2-26 UK-HP sales from 2005 to 2013 (Hannon 2015)

This increase in the use of HPs is a common area for recent study, with both analytical and experimental analysis being provided. One of the more common methods used is to increase the efficiency of the heat exchanger inside. In this context, Liu at al. designed a ground heat exchanger system including one outlet and three inlet pipes (single-U, double-U, and the new design 3I). Experimental results showed that circulating water temperature in the new design 3I (Figure 2-27) led to lower temperatures than the single-U and double-U HXs. Based on these results, the temperature was 15.8% and 31.1% lower in the new design 3I. For the long term condition a numerical model was also carried out in this study(X. Liu et al. 2015).

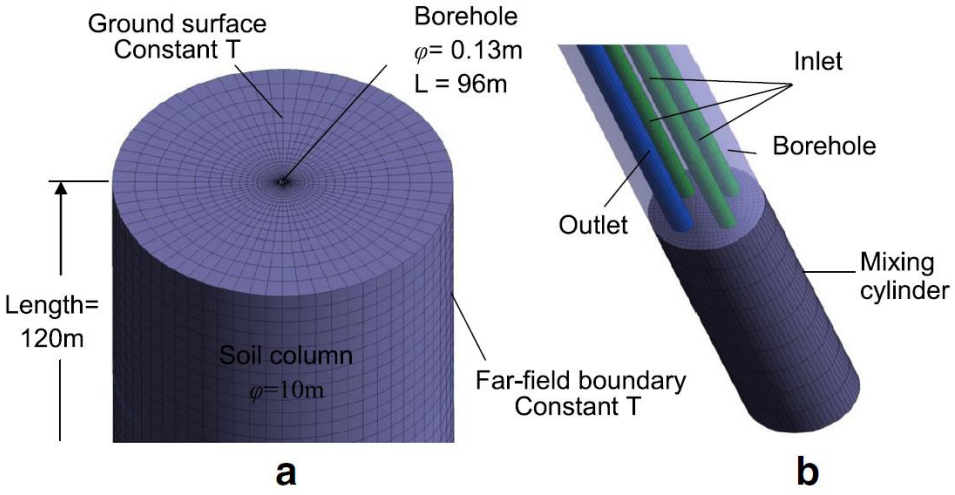


Figure 2-27 (a)New innovative type 3I heat exchanger mesh perspective its Borehole Heat Exchanger inside (X. Liu et al. 2015)

The common conflict for HPs is the high Global Warming Potential (GWP) of the refrigerant used in the heat exchangers. 11, therefore, carried out a study into the development of low-GWP refrigerants. The low-GWP R32 and an HFO were replaced with the high-GWP R-410A. R32 and HFO showed 20% and 10% lower refrigerant

charges respectively. There were also many comparable refrigerants used in this study which showed higher capacity (Bony and Citherlet 2009). Moreover, oversizing could cause a less efficient system with a longer lifespan. The new combination system including an electric heat pump and a condensing gas boiler for back-up were assessed in the study of Perna et al. (Figure 2-28).

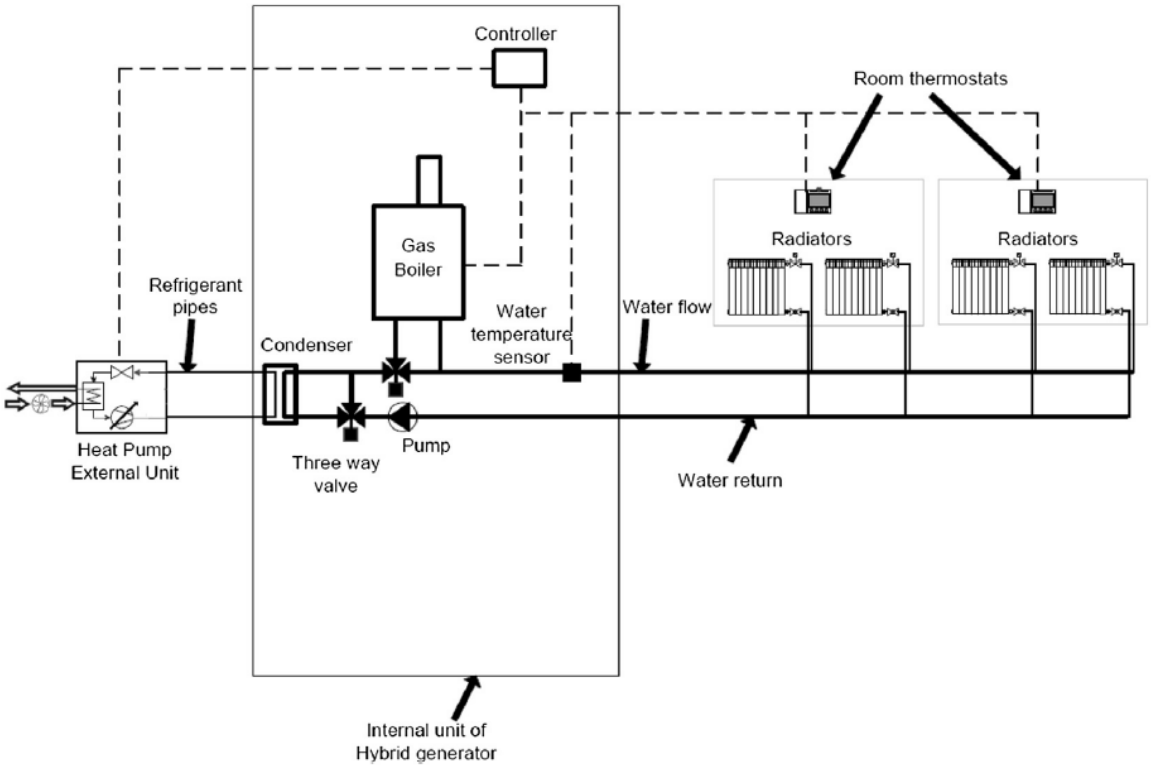


Figure 2-28 Water circulation in the hybrid system (Di Perna et al. 2015)

Solar Heat Pump Systems

These PV systems are supported by heat pumps and can be described as Photovoltaic-Thermal Solar-Assisted Heat Pump (PVT-SAHP). There are PVT-SAHP systems using a refrigerant as a fluent (Ji et al. 2008). The working principle of this innovative system is to pass the refrigerant through using a heat pump.

There have been many studies which have sought to develop system efficiencies using heat pumps. For example, an experimental study carried out in the University of Nottingham was found to substantially increase the efficiency of the system (Zhang et al. 2012). However, in these systems, the intensity of the dispersion of the refrigerant inside the pipes is an issue which needs to be considered carefully. Uneven distribution inside the pipes causes a drop in pressure which can lead to possible cracks and leakage in the pipes (Zhang et al. 2013)**.** Almost all studies were carried out using a TRaNsient SYstems Simulation Program (TRNSYS), according to the literature review carried out in this study.

Aste et al. assessed a proposed PVT-SAHP system for energy demand (heating, cooling and domestic hot water), with comparison to a 'water to air' heat pump and separate PVT. The proposed systems were simulated in two different configurations in TRNSYS, which were a 'water to water' heat pump and PVT, and additionally a 'water to air' heat exchanger, storage tank, circulation pump, heating floor, iterative feedback controller and three-stage aquastat controller. The separate PV showed approximately 6% lower efficiency than the integrated one. Moreover, the COP of the proposed HPs was calculated to be 30% higher than the most effective HP commercially available (Aste et al. 2015)**.**

Suleman et al. carried out an energy and exergy analysis of a proposed integrated solar and heat pump system. The energy and exergy efficiency were calculated at 58% and 75% respectively. With regards to the HP, 42.5% exergy efficiency with 3.54 COP and 35.7% exergy efficiency with 2.97 COP were found in the analysis of the innovative system, using some assumptions for making the system analysis easier,

such as almost adiabatic 90% isentropic efficiency of compressors and pumps (Suleman, Dincer, and Agelin-Chaab 2014).

Wang et al assessed in TRNSYS the long-term performance of a proposed hybrid solar GSHP used for the heating and cooling load in an office. The borehole heat exchanger (BHE) and borehole thermal energy storage (BTES) are the main extra components in GSHPs. According to this long term assessment, it was shown that 32% energy can be achieved with a closed load circulation pump during no fan-coil (E. Wang et al. 2012).

Three different integrated-system solar and heat pump concepts (Figure 2-29) were simulated using TRNSYS in the study of Bertram et al In one concept, the heat extracted from the solar panels was connected to the hot side of the HP. The SPF achieved an increase of 1.2 in HP. In another concept, the connection of the cold side of the HP with the panels affected the system by only 0.2 with regards to the SPF, with dependency on the collector area and the HXs length. The best solution was found to be the direct use of the hot side for the hot water (Bertram, Pärisch, and Tepe 2012).

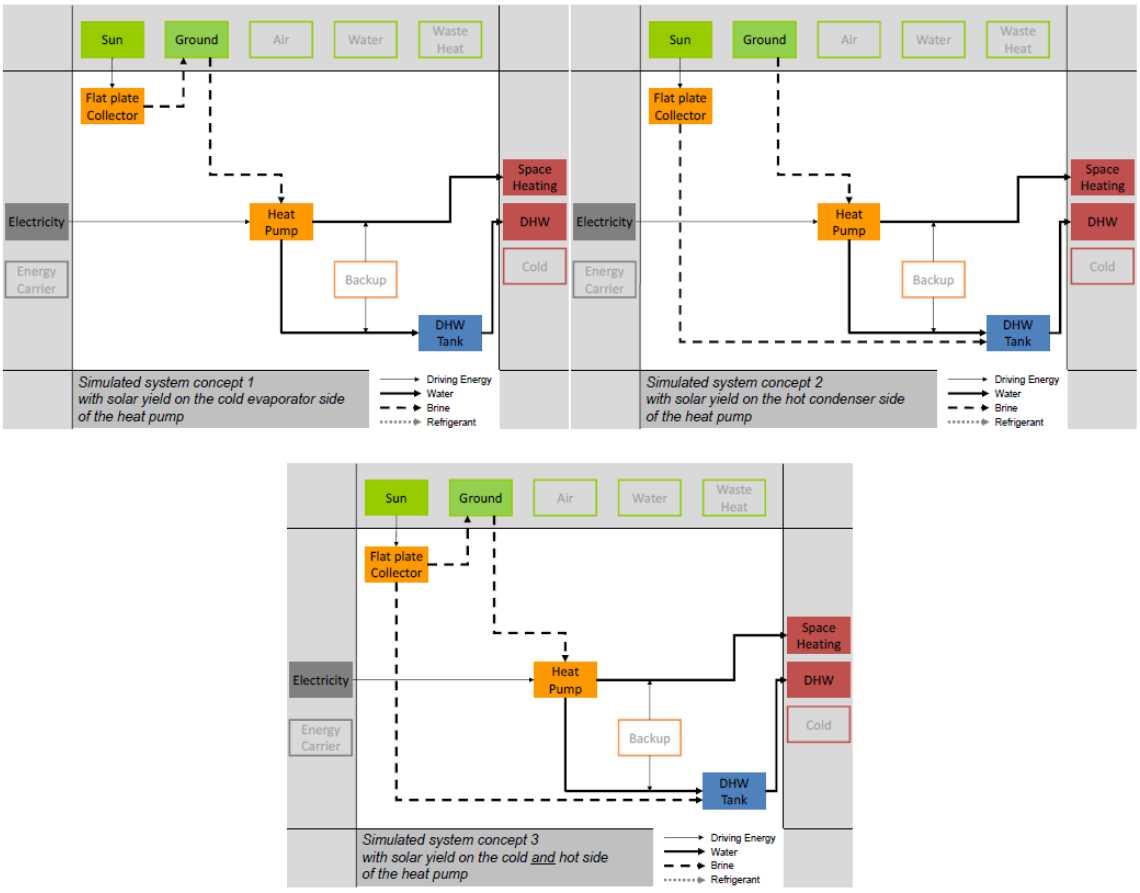


Figure 2-29Three different integrated Solar and heat pump concepts (Bertram, Pärisch, and Tepe 2012)

2.3.4. The focus of recent studies

One of the impressive studies was a proposed solar-assisted ground-coupled heat pump (SAGCHP) system with the aim of providing space and water heating by Chen et al. on the TRNSYS. The solar collector area and borehole length impact on the system's efficiency were investigated in this simulation study. Moreover, this study was experimentally repeated in various weather conditions in northern China. With the determined weather conditions in simulation, a solar area of 40m² and a borehole length of 264m in the proposed system were chosen(Chen and Yang 2012).

Another substantial study was carried out with an innovative solar heat-pump heating system (SHPHS), including dual-tank latent heat storage (DLHS), by Qu *et al*. The basic rule of heat storage systems compromising Phase-change materials (PCMs) is the storage capacity of the latent heat energy when they change their phases. SHPHS–DLHS systems were experimentally assessed in China, and according to the results new parameters were found for the system COP because of the integration of the storage. The new COP was almost 3.5 times higher than the general heat pump systems. The PCMs significantly improvement the HP's efficiency (Qu et al. 2015)**.** In a similar manner, Yumrutas *et al*. have designed a computational model for a proposed integrated system including a HP and an underground thermal energy storage tank (TES). This study assessed the various impacts on system performance such as thermophysical properties, HP and TES features, system operation years etc. The study was concluded with a lifespan assumption of 5–7 years(Yumrutaş and Ünsal 2012)**.**

An innovative solar-heat pump dryer system has been investigated in various weather conditions by Sevik *et al*. They observed the different types of agricultural products and tried to keep constant drying air with a PID controller. The energy utilization ratio (EUR) was variated from 0.19 to 0.48. These observations showed that the dried product had good quality appearance (Sevik 2014)**.**

Unglazed PV collectors integrated with heat pumps including a borehole heat exchanger were assessed by Bertram et *al*.. In this study, the effectiveness of these systems was expected to be double that of conventional heat pumps and PV systems separately. A 39m² unglazed PVT collectors was used for the system. The yearly additional PV profit was calculated at 4% in general, with the capability to reach 10%

in extreme conditions. In the first year, the SPF of the HP was achieved to increase 0.36, and 0.51 for the 20th year (Bertram, Glembin, and Rockendorf 2012).

Zhang *et al*. has examined theoretically and experimentally an enhanced solar photovoltaic/loop-heat-pipe (PV/LHP) heat pump water heating system. The electrical, thermal and overall efficiency was found to be around 10%, 40% and 50% under specified experimental conditions, with 8.7 overall system COP. In contrast, the fluid rate impacted the exergy efficiency (Zhang et al. 2013).

A ground-coupled heat pump combined with solar collectors was assessed in order to find out the main operating conditions affecting the system's performance by Parisch *et al*. Steady-state and dynamic tests were allocated with the brine/water heat pump. It was observed that when the source temperature rises constantly, it causes considerable increase in the COP (Pärisch et al. 2014).

A Solar Air Composite Heat Source Heat Pump System (SACHP) integrated with a fin tube and a tube heat exchanger was examined by Liu et al. (Figure 2-30). The dual heat source mode showed 62% higher heat capacity and 59% higher COP than a single heat source mode in the condition of -15°C ambient temperature. Moreover, the dual heat source mode showed 51% higher heat capacity and 49% higher COP than the single heat source mode in the heat transfer difference of 5°C(Y. Liu et al. 2016).

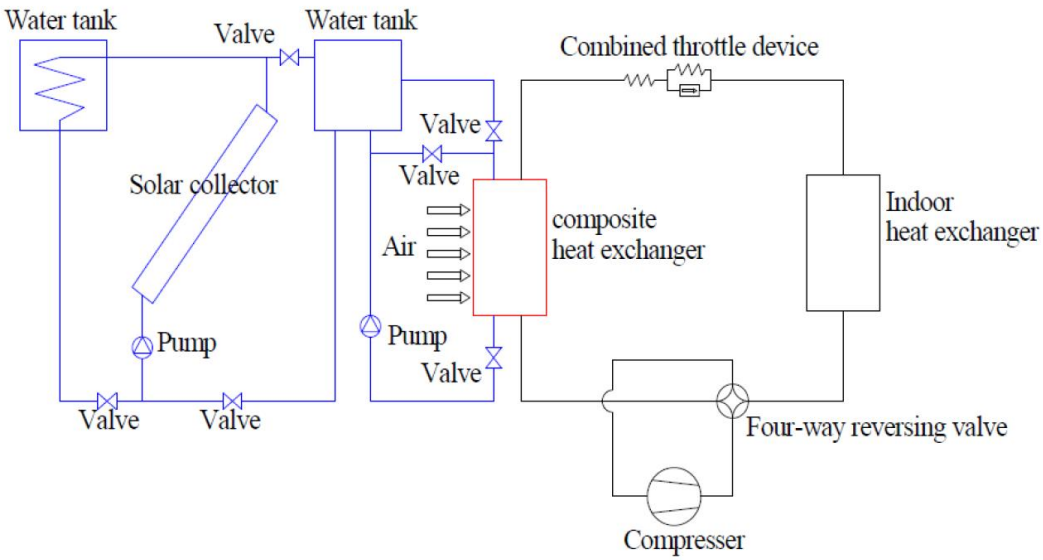


Figure 2-30 Integrated solar air composite heat source heat pump system (Y. Liu et al. 2016)

Winteler *et al*. has conducted a study into an integrated Solar, HP and ice storage system for residential buildings (Figure 2-31). MATLAB®/SIMULINK® was used for all analysis. This study has taken the building types and information from a similar project, namely IEA HPP Annex 38 / SHC Task 44 "Solar and heat pump systems" specified SFH15, SFH45 and SFH100. According to the simulations, the specified building was able to achieve more than 4 SPF with various heat loads and weather conditions in SFH25 to high SFH100 residents. And in SFH25, SPF was also just under 4 (Winteler et al. 2014).

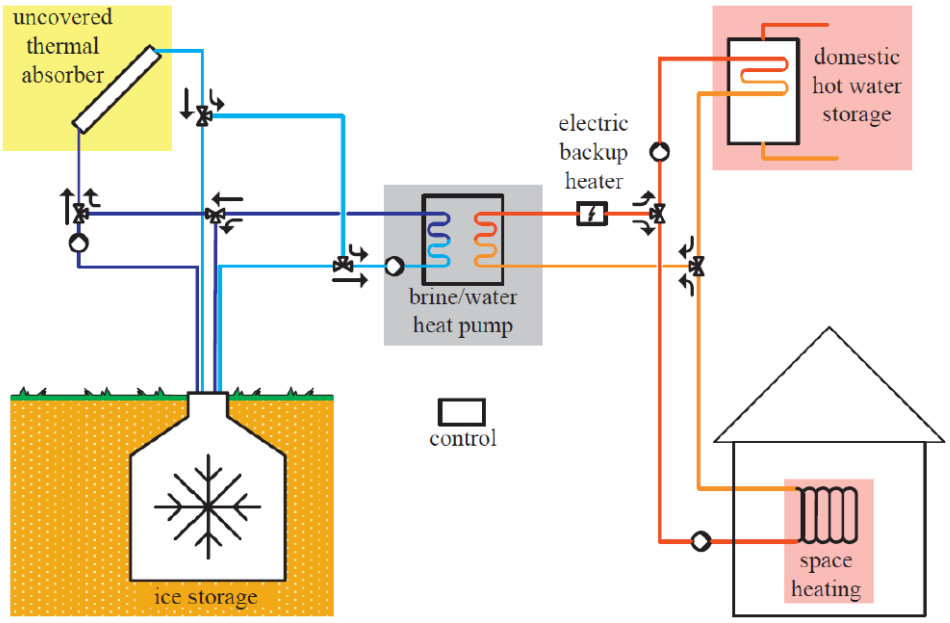


Figure 2-31 System configuration (Winteler et al. 2014)

CHAPTER 3

HEATING AND VENTILATION PERFORMANCE OF A UK POULTRY HOUSE: AIR FLOW PATTERN

3.1. Introduction

The chapter presents the design process based on the assessment of the poultry house's energy requirements. The optimum temperature for live poultry ranges from around 33°C at birth to around 23°C at four weeks. The most important and initial step is the determination of the building's thermal loads to size the required air conditioning systems. The aims of this chapter are to: (1) detail the energy simulation and airflow analysis of the existing building; and (2) assess the potential impacts on the environment on the production cycles (PCs, referred to as PC ss 1, PC 2 etc.). The evaluation of heating load and air distribution in the building is premised on the following considerations:

- All content relating to the design considerations of the design process undertaken to complete the initial assessment of the poultry house's energy requirements is based on the existing building in the UK.
- Determination of the thermal regulation energy demand for the air conditioning system depends on the heat provided by the building heat pump (HP) being equal to the heat removed from the building. This informs the selection of the air conditioning unit, and the size and design of the HP and evaporative cooler, with particular consideration of the size of the available land and the building’s roof.

- Discussions on the airflow and ventilation management on the building pertain to a psychrometric energy core (PEC) cooling design and operational conditions, which affect the performance of the PEC unit and its design.

The chapter covers all content relating to the poultry house assessment. It details the energy simulation and airflow analysis of the existing building in the following three sections. Section 3.2 presents the description of the real site and the facility management in terms of heating and ventilating set points, and the temperature and humidity under transient conditions. Section 3.3 presents the thermal regulation simulations in Design Builder. MATLAB and Hourly Analysis Program (HAP) software and their development under the energy balance equations. Section 3.4 provides the airflow and ventilation management and its Computational fluid dynamics (CFD) model under the winter and summer conditions in Design Builder.

3.2. Poultry house in Newark, UK

A selected poultry farm located in UK is theoretically investigated in terms of the thermal regulation energy demand and internal air quality (Figure 3-1). Broilers are grown on a 7-week cycle in this farm. The pullets leave at 4 weeks old, and the cocks at 7 weeks. The building consumes gas with hot air blowers for heating. Regarding environmental conditions, the houses have sufficient levels of insolation and the area has a sufficient amount of sun hours throughout the whole year. The environmental condition in UK and the energy demand of the poultry farm is assessed in detail in the following section. The four houses (S1-S4) vary slightly in size, including approximately 40,000 birds at any one time, as shown in Table 3-1.

Table 3-1: House dimensions, floor area, and bird numbers

Building name	*Floor area ($m^{2)}$*	*Bird numbers*
S1 (Control house)	502	6700
S2 (House 4)	636	8500
S3 (House 6)	818	10400
S4 (House 5)	1034	14400

Based on the observation, following an exhaustive analysis, the most representative periods for the heat pump operation seasons based on the outside and inside weather conditions were extracted for the poultry houses from the total monitoring period: the peak heating season (December-January, February-March), heating season (April-May, October-November), the minimum heating season (June-July, August-September) under UK weather conditions. Annual production cycles are defined in the Table 3-2 based on these observed seasons.

Table 3-2: Annual Production Cycles (PCs)

Production Cycles (PCs)	**Time**
Production Cycle 1 (PC1)	December
Peak Heating (kWh)	January
Production Cycle 2 (PC2)	February
Peak Heating (kWh)	March
Production Cycle 3 (PC3)	April
Heating (kWh)	May
Production Cycle 4 (PC4)	June
Minimum heating (kWh)	July
Production Cycle 5 (PC5)	August
Minimum heating (kWh)	September
Production Cycle 6 (PC6)	October
Heating (kWh)	November



(a)



(b)

Figure 3-1: Poultry house (S1) (a) Exterior, (b) Interior

One of the poultry houses, S1 was selected for assessment as it is the smallest house in the site (Figure 3-1), with the lowest heating requirement. This house has 6700 chickens

The PC of the chicks is 7 weeks. Each PC requires around 32˚C in the first two days, and every day their heat demand decreases by around 0.5˚C.

3.2.1. Set point temperature in the poultry house

The internal temperature of S1 was required to be maintained between 21-32˚C. The set points of the heating system within the model were assigned based on the direction of the standards defined in Table 3-3. The set point of the fan coil unit was set 0.5°C higher than the gas burner set point. This house has 6700 chicks inside, with a PC of 7-weeks. The set-up temperature is around 32°C in the first week, and every day the set points decreases by around 0.5°C. The distribution of the energy consumption among different production cycles for each week in the house was assessed in this section.

Table 3-3 The temperature and ventilation requirement in the house

7-week production cycle	Chicken weight (kg)	Fan coil heating set point temp. (˚C)	Gas boiler heating set point temp. (˚C)	Air Flow	*Minimum ventilation rate (MVR) (m^3/h-chicken)*	Ventilation Requirement (l/s)	Sensible latent heat (W/Chic
1st week	0.18	31.0	31	0.17~0.47	0.414	463	0.7/0.4
2nd week	0.5	29.0	29	0.58~0.94	1.15	1152	2.0/1.1
3rd week	0.68	27.0	27	0.94~1.52	1.564	1463	2.7-1.4
4th week	1.09	24.0	24	1.52~2.15	2.507	1871	4.4-2.3
5th week	1.41	23.0	23	2.20~2.76	3.243	2252	5.6-3.0
6th week	1.74	22.5	22	2.86~3.15	4.002	2596	7.0/3.7
7th week	2.2	21.5	21	>3.15	5.06	3005	8.8-4.6

The sensible and latent heat production per chicken were presented in Table 3-3. Using a chicken emits excess data, one chicken generates heat and water of 3 W/hr and 0.1 l/day, respectively (Genç and Portier 2005). Sensible and latent heat

(moisture) production data were obtained and defined in the HAP software to give correct information about the distribution of the energy gains among chickens and determine annual thermal regulation energy demand in the poultry house.

The building air conditioning is modelled by using minimum ventilation rates (MVR) method. The building air conditioning is modelled using minimum ventilation rates (MVR) methods. Based on the closed chicken house and weather conditions, the minimum required ventilation rate for winter is 2.3 m^3/hr per 1 kg of broiler mass. Based on the closed poultry house and weather conditions, minimum ventilation is operated on a five-minute timer with MVR needed for starting chickens is about 0.047-0.094 m^3 /s per 1,000 chickens, depending on outside air temperature. (ROSS 2010).

The minimum ventilation requirement for each production cycle in the existing building is presented in Table 3-3. The season has an important impact on ventilation management. During summer the minimum rate is kept open longer, although ideally the ventilation requirement is mainly the same during each cycle.

3.2.2. Monitoring of the chicken production cycles (PCs)

This section presents the monitored data of the critical temperatures in the poultry house under autumn and winter weather conditions. The weather data was collected on 1st October 2016 from the poultry house for experimental assessment. The monitoring started on 1st October 2016, which is the start date of the first PC monitored (PC 6) under autumn weather conditions. The internal temperature of the PC started at 32°C and dropped to around 19°C by the end of the period. Figure 3-2 shows the monitored data of the internal temperatures of the houses and ambient temperatures during PC 6.

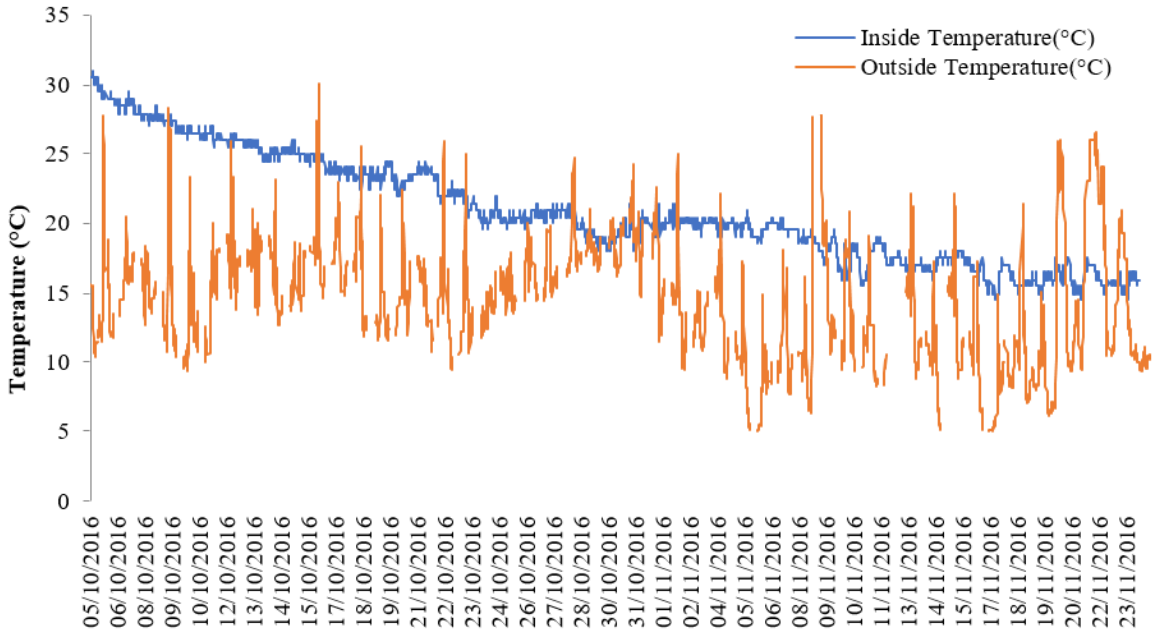


Figure 3-2: Inside and outside house temperature (PC 6)

Moreover, monitoring of the critical temperatures in the poultry house under the winter weather conditions is presented in this section. Monitoring in PC 1 began on 5th December 2016, the start date of this PC. Figure 3-3 shows the external and internal house temperatures over the period from 5th December 2016 to 24th January 2017. The external temperature ranged between -3°C and 20°C during the second PC test. The day and night temperature differences were observed, with a minimum value of 4˚C and a maximum of 10˚C. The minimum temperature was collected as -3.6˚C during January. Regarding the minimum temperature value for the site, it is important to determine how frequently it falls to such levels in order to determine the heating load of the building.

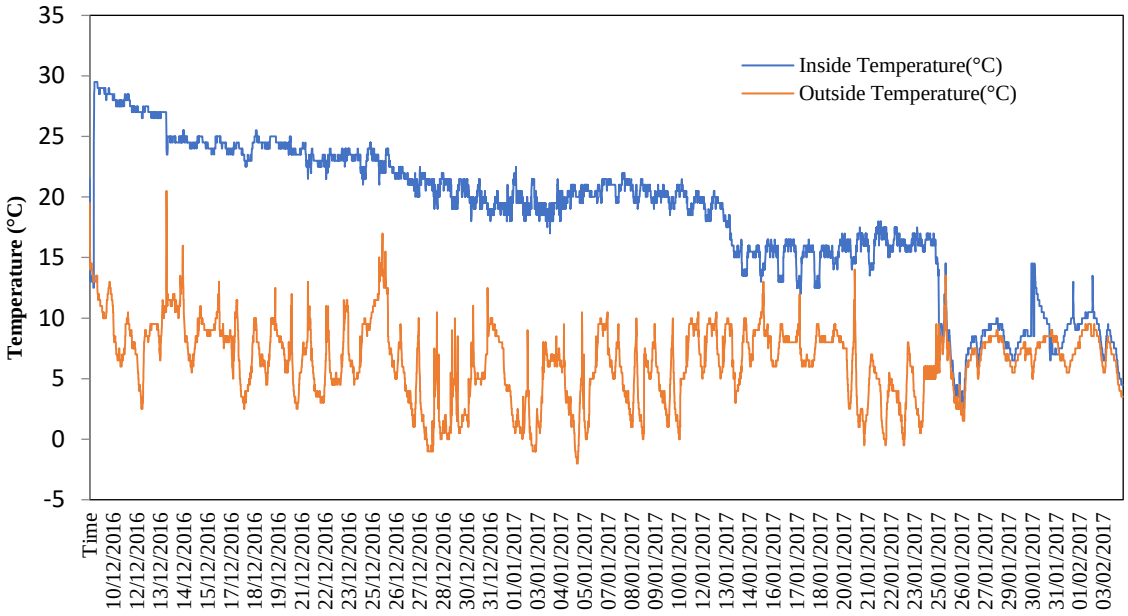


Figure 3-3: Inside and outside house temperatures (PC 1)

3.3. Modelling the thermal regulation energy demand of the poultry house

Estimations about the consumption and distribution of energy according in the poultry house entail energy audit. This involves estimations about the consumption and distribution, covering the work relating to the analysis of the building energy consumption in Design Builder and HAP software. Energy consumption is determined using data from Energy Plus software. The sensible and latent heat (moisture) production data were obtained and defined in each software to determine annual thermal regulation energy demand in the poultry house. Moreover, the distribution of the energy consumption among different PCs in the house was assessed among different applications.

The second approach is the heating load determined in MATLAB software by considering the data collected from the site, which are the data of Inside (T_{in}) and outside (T_{out}) variations in the winter chicken PC, and validated gas use from the site

during the time of data collection. The peak load is calculated based on the recommended design temperature for Nottingham, UK, which is -4.7°C for the system sizing in Design Builder (Change 2008). The main calculations on the peak load are the heat losses through the walls and roof, heat exchange with the slab floor, ventilation heat losses, heat gains from the birds, and heat gains from electrical sources.

The temperature and ventilation changes were monitored over the 7-week PC. The environmental management strategy relevant to the heating and ventilation change during PC acknowledges that an increase in body mass is associated with increased food consumption, with chickens' internal heat, air emission, and pollutants commensurately increasingly, along with the ventilation rate. The building simulations take these changes into account, and the assessment of the poultry house heating and ventilation requirements was assessed to adequately size the air conditioning system.

Details on the basic structural component elements, including dimensions and constructions, are provided in Table 3-4 These were used to develop the assumptions in the model

Table 3-4 Basic structural component element

Building element	Material	Area (m2)	Thickness (mm)
Roof sheet	Cement.	580	12
Attic insulation	Fibre glass	500	150
Wall sides	Wooden sides	224	20
Wall insulation	Rockwool insulation	224	75
Wall openings	-	80	
Floor	Concrete	504	100
Floor covering	Sawdust base	504	15

3.3.1. Simulation of the poultry house in HAP software and Design Builder

As stated, the building is located in the county of Kirton, near the town of Newark, UK. In order to estimate the heating load, the well-known building energy simulation software Hourly Analysis Programme (HAP) was used to simulate the poultry house. This involves estimations about the consumption and distribution, covering the work relating to the analysis of the building energy consumption. Moreover, In Design Builder, the theoretical model is constructed based on six main templates considered in the simulations, as explained below. Design Builder software uses the Energy Plus database for the simulations, and the chicken thermal production data was added manually in the software database based on ASHRAE guidance (H. J. Chepete and Gates 2004).

Activity templates model the typical activities that occur in the building. These contain the cooling/heating set points of the temperature required and information of occupancy, which concerns chickens in this study. All activity templates are publicly available; whereby new activity templates were produced based on the chickens' properties. The temperature of the set points and metabolic rate of the chicken's changes with chickens' age and the high volume of stocking density in the building, and these parameters were considered and adapted to the software in a simple way.

3.3.1.1. Construction templates

Construction templates contain information on the materials used in the building elements, such as floors, walls, ceilings, and roofs etc. In the model a new construction template was defined with the wall material of 75 mm Rockwool insulation; 100 mm

rock wool insulation in the ceiling; wooden sides/fibre; and cement roof sheets.

Airtightness templates

Airtightness templates were applied to model in design builder the infiltration rate for the chicken building. The assumptions are made for the airtightness template in design builder software based on measurements taken in similar poultry farms reviewed in the literature (Lopes, Morello, and Earnest 2013). On the other hand, in HAP software and MATLAB software, the ventilation includes infiltration.

- **Opening templates**

Opening templates contain information on windows, vents, and doors of the building simulation.

- **Lighting templates**

Lighting templates contain information on lighting used, efficiencies, and loads.

3.3.1.2. HVAC templates

HVAC templates contain data on the type of HVAC systems used. Active items such as HVAC, frequency of occupancy (chickens), lighting, and window operations (during winter and summer) were controlled by the domestic schedule template of Design Builder and HAP software.

Nottingham, UK, was selected for the simulations of the poultry house since it is the nearest place defined in the software and has similar weather properties to Kirton. The digital model from Design Builder analysis and the weather file for Nottingham presented in Figure 3-5 as environmental data.

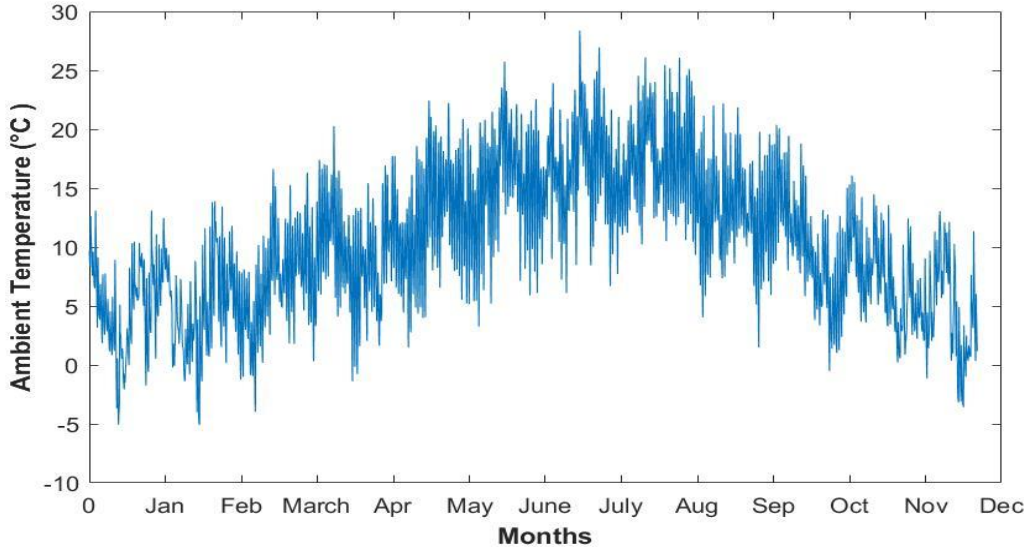


Figure 3-4: Ambient temperature of Nottingham, UK

Building services were designed to react to provide the internal conditions required for the chickens during different phases of their growth cycles. As mentioned preciously, at the start of the process the internal temperature must be maintained at 32˚C; this is gradually lowered to 22-24˚C once the chickens reach maturity, towards the end of the 6 weeks. The temperatures of the set points within the model were assigned based on the standards defined in Table 3-5.

Table 3-5: Set points within the model (six cycles p.a.)

Week cycle	*Chicken weight (kg)*	*Heating set point temp. (˚C)*	*Heating set back temp. (˚C)*	*Minimum ventilation rate (MVR) (m^3/h-chicken)*
1st week	0.18	32.5	32	0.414
2nd week	0.5	29.5	29	1.15
3rd week	0.68	27.5	27	1.564
4th week	1.09	24.5	24	2.507
5th week	1.41	23.5	23	3.243
6th week	1.74	22.5	22	4.002
7th week	2.2	21.5	21	5.06

3.3.2. Mathematical model of the thermal regulation load of the poultry house in MATLAB

3.3.2.1. General parameters

The heating season is up to ten months, while the non-heating season is up to two, depending on the 7-week cycle of the chickens. Six hours of cooling and 24 hours of heating are required per day during summer and winter, respectively. In this section, two chicken cycles were highlighted and validated with the data monitored during autumn and winter.

For the seasonal energy demand analysis in MATLAB, the data collected from the site, adapted to the mathematical model in the software. The first monitoring period was from 05/10/2016 to 31/10/2016, and the second was from 05/12/2016 to 31/12/2016. Selected chicken house and weather details were collected from the field study in Kirton, UK. The house has the size of 8 m x 63 m x 2 m. Data logging of the site's temperature, relative humidity, and CO_2 concentrations was processed from October 2016. The minimum solar irradiation was observed in December during the monitoring period.

The ventilation and heating requirements of the building are directly affected by the poultry fresh air requirements. Mature chickens ideally need an air temperature of 20-24°C, with a relative humidity of 50-70%. However, this temperature and relative humidity will vary depending on the mass and age of the chickens. The poultry house is considered as a control volume connected with the heat sources and sinks, and the principles of mass and energy conservation are applied in the mathematical model in MATLAB. The main energy interactions in the modelling are siding walls; external and internal temperatures; the mass of air exchange through the building; conduction heat

exchange; solar heat absorption through the roof; and sensible and latent heat generated in the building (J. Chepete, Resources, and Tsheko 2014). The heat distribution on the surface and walls are not considered in the models.

As an initial modelling step these loads were simply defined and the model was developed with seasonal energy requirements depending on the chicken PCs and the age and the size of the chickens.

Regarding environmental conditions, Kirton has sufficient levels of insulation and sun hours throughout the whole year. The prevailing environmental conditions and energy demand of the poultry house were assessed. Selected chicken house and weather details were collected from the field study in Kirton. There are seven ventilation fans, each of which has a maximum output of 5,800 m^3/hr, and two units of 66 kW gas heaters are used for the house auxiliary heating. The relations between the discussed general parameters can be given as:

$$M_{air,in} C_{air,in} \frac{dT_i}{dt} = \left(Q_{gain} - Q_{loss}\right) \quad 3\text{-}1$$

Where $M_{air,in}$ is mass of air inside the house, $C_{air,in}$ is specific heat capacity of air, T_i is indoor temperature, t is time, Q_{gain} is the total heat gain , Q_{loss} is the total heat loss. The following sections explain the formulate used to determine other studied variables.

3.3.2.2. Net heat gain and loss (Q_{gain}, Q_{loss})

Based on the Equations 3-1, the total heating capacity requirement is given by:

$$\mathrm{Q_{ac}} = \mathrm{Q_{gain}} - \mathrm{Q}_{loss} \quad 3\text{-}2$$

There are three energy interactions in the model considered in the poultry house to determine total heat gains in the poultry house (Mutai et al. 2011).:

$$Q_{gain} = Q_{bird} + Q_{light} + Q_{so} \qquad 3\text{-}3$$

- Total heat production (Sensible and latent heat gain) from the birds (Q_{bird}).
- Heat gain from electrical sources (Q_{ele}).
- Solar heat gains (Q_{so}).

On the other hand, ventilation losses have the most impact on the total heat losses.

$$Q_{loss} = Q_{walls} + Q_{floor} + Q_{vent} \qquad 3\text{-}4$$

Where Q_{walls} is the heat loss from the walls, Q_{floor} is the heat loss from floor, and Q_{vent} is total heat loss from ventilation.

❖ **Heat gain from chickens (Q_{bird})**

A chicken emits excess heat in two forms: (1) losses to the surrounding air (sensible heat loss), and (2) losses through evaporation of water through respiration (latent heat loss) (Genç and Portier 2005). Total heat production (THP, W/bird) of the bird is calculated by Equation 3-5.

$$Q_{THP} = 10.M_{bird}^{0.75} \qquad 3\text{-}5$$

Where M_{bird} is the mass of birds (kg), and Q_{THP} is produced heat by birds (W/bird). 6700 birds are raised in the test house with a live body mass (M) of 2 kg per bird.

To obtain sensible and latent heat production (SHP and LHP, W/bird) of the birds individually, data on total heat production (THP) and SHP as functions of indoor temperature for broilers are used (CIGR Handbook 2000), and SHP is expressed as a percentage of THP (% SHP of THP) at various temperatures. Proposed models are developed using Equations 3-7 and 3-8.

$$\%\, SHP\ of\ THP = -0.0202 t_i^2 + 0.4464 t_i + 55.255 \qquad 3\text{-}6$$

$$SHP = \frac{\%\, SHP\ of\ THP}{100} \times THP \qquad 3\text{-}7$$

$$LHP = THP - \left(\frac{\%\, SHP\ of\ THP}{100} \times THP\right) \qquad 3\text{-}8$$

- **Heat gain from electrical source, Q_{ele}**

Suspended lamps are the sources of electrical heat. Therefore, it is assumed that 20 lamps (8 W each) are installed in the new poultry house. Equation 3-9 is used to quantify the heat flux.

$$Q_{ele} = \frac{1.2\ R_w\ N_{ele}}{A_{floor}} \qquad 3\text{-}9$$

Where R_w and N_{ele} are the rated wattage and total number of fluorescent light bulbs, respectively, and A_{floor} is the floor area, m^2.

❖ Solar heat gain, Q_{so}

Due to the impact of solar radiation intensity in Newark, the sol-air temperature approach was used to quantify heat gains via the roof and wall using Equation 3-10 (J. Chepete, Resources, and Tsheko 2014).

$$Q_{so} = \frac{h_s A_s (T_{sa} - T_s)}{A_f} \quad \textit{3-10}$$

Where Q_{so} is the solar heat gain through the components (walls or roof) (W/m^2); A_s is the surface area of component, m^2; T_{sa} is the sol-air temperature of component; T_s is the surface temperature of component, °C; and h_s is the convective heat transfer coefficient of component, W/m^2·K. The h_s factor is 0.6 (J. Chepete, Resources, and Tsheko 2014). The sol-air temperature or effective outside temperature of the various surfaces are calculated using the following (Beauttah et al. 2011).

$$T_{sa} = T_{oa} \frac{I\alpha - 6\varepsilon \cos \phi)}{A_f} \quad \textit{3-11}$$

Where T_{oa} is the outside air temperature, °C; α is the absorption coefficient for solar radiation; I is the solar radiation intensity, W/m^2; ε is the thermal radiation emittance of the surface; and ϕ is the angle of inclination of the surface. Therefore, the overall heat gain is the sum of Equations 3-5, 3-10, and 3-11, given by:

$$Q_{Total,gain} = Q_{sens} + Q_{ele} + Q_{so} \quad \textit{3-12}$$

❖ **Heat loss through the building elements**

The heat losses were considered through the wall and roof, slab floor, and ventilation air. Heat loss through the walls or roofs, Q_c, and heat conducted via the walls or roofs, is calculated using Equation 3-13 (J. Chepete, Resources, and Tsheko 2014).

$$Q_{con} = \sum (\mathrm{U\ A})\ (T_{in} - T_{oot}) \qquad 3\text{-}13$$

Where Q_{con} is overall conduction heat transfer (W), A is wall or roof area (m^2), U is overall building heat transfer coefficient ($W/m^2 \cdot K$), and T_{in} and T_{out} are indoor and outdoor temperature, respectively (°C). Heat exchange with the slab floor, Q_f, is determined using Equation 3-14:

$$Q_{floor} = F\ P\ (T_{in} - T_{out}) \qquad 3\text{-}14$$

Where P is the perimeter of poultry house; and F is an experimental slab heat loss /transfer factor , W/m·K. Values of F, for an unheated slab floor on grade range between 1.4 and 1.6 W/m·K, depending on the severity of the winter. In this study, an F value of 1.5 W/m·K was used. Heat exchange of the slab floor is determined by using the unit temperature difference between the outside and inside air temperature on the per linear length of the exposed perimeter of the floor (J. Chepete, Resources, and Tsheko 2014).

Heat loss through transfer via ventilation (Q_{vent}) is given using Equation 3-15:

$$\mathrm{Q_{vent}} = \frac{\mathrm{C}_{air}\ \mathrm{V_R}\ \rho_{\mathrm{air}}\ (T_{in} - T_{out})}{\mathrm{A_{floor}}} \qquad 3\text{-}15$$

Where C_{air} is specific heat of the air, J/kg·K; ρ is air density, kg/m^3; and V_R is ventilation rates, m^3/s. Generally, enough and continuous ventilation is required for supplying clean air and reducing humidity in the poultry house. The amount of ventilation depends on the age (mass) of broilers, their number, and the season of production.

The total heat losses are calculated using Equation 3-16:

$$Q_{loss} = Q_{con} + Q_{floor} + Q_{vent} \qquad 3\text{-}16$$

3.3.3. Results and discussion

The welfare of the chickens depends to a large extent (among other factors) on thermal comfort. The main parameters affecting thermal comfort include the internal and external temperature, air velocity, and relative humidity. The average ambient temperature from November to February is 5.25°C, with average wind velocity over the period of 4.85 m/s and mean solar irradiation of 34.3 W/m^2.

3.3.3.1. HAP Software

The heating demand for the production cycles and seasons observed from December to January has been determined in HAP software. Table 3-6 shows the heating demand per week for each poultry production cycle under these heating seasons in UK. Although the building temperature decreases each week, half of the chickens leaves the house in the fourth week of the production. That causes a noticeable decrease on the chicken heat gains in the building and increases the heating demand of the building.

Table 3-6 The heating demand per week for each poultry production cycle

		1st week	2nd week	3rd week	4th week	5th week	6th week	7th week
December	PC1	4,806	6,700	4,680	4,105	-	-	
January	Peak Heating (kWh)	-	-	-	-	3,607	5,023	5,507
February	PC2	4,865	7,298	5,333	1,922	-	-	-
March	Peak Heating (kWh)	-	-	-	-	4,112	5,387	3,751
April	PC3	3,872	4,089	3,548	2,486	-	-	-
May	Heating (kWh)	-	-	-	-	3,229	3,121	1,811
June	PC4	3,080	3,365	1,407	491			
July	Minimum/non-heating (kWh)	-	-	-	-	2,081	951	658
August	PC5	2,656	2,179	1,225	323			
September	Minimum heating (kWh)					1,267	2,120	1,350
October	PC6	3,462	3,483	2,998	2,527			
November	Heating (kWh)	-	-	-	-	4,111	3,285	3,434
Annual	Inside temperature (^{o}C)	31	29	27	25	23	22	21

Most of the energy loss is accounted for through the ventilation loss to prevent health issues because of the air pollutants and ammonia inside the building. This causes more energy consumption in the building to maintain required internal temperatures, given the relatively large continual influx of external, unconditioned air. Figure 3-5 shows the heating demand of the building for each production cycle (PC). The maximum and minimum heat required were determined 34.4 MWh in PC1 during the peak heating season and 11,1 MWh in PC4 during minimum heating season. The heater of the poultry house is sized as 100 kW under UK weather condition.

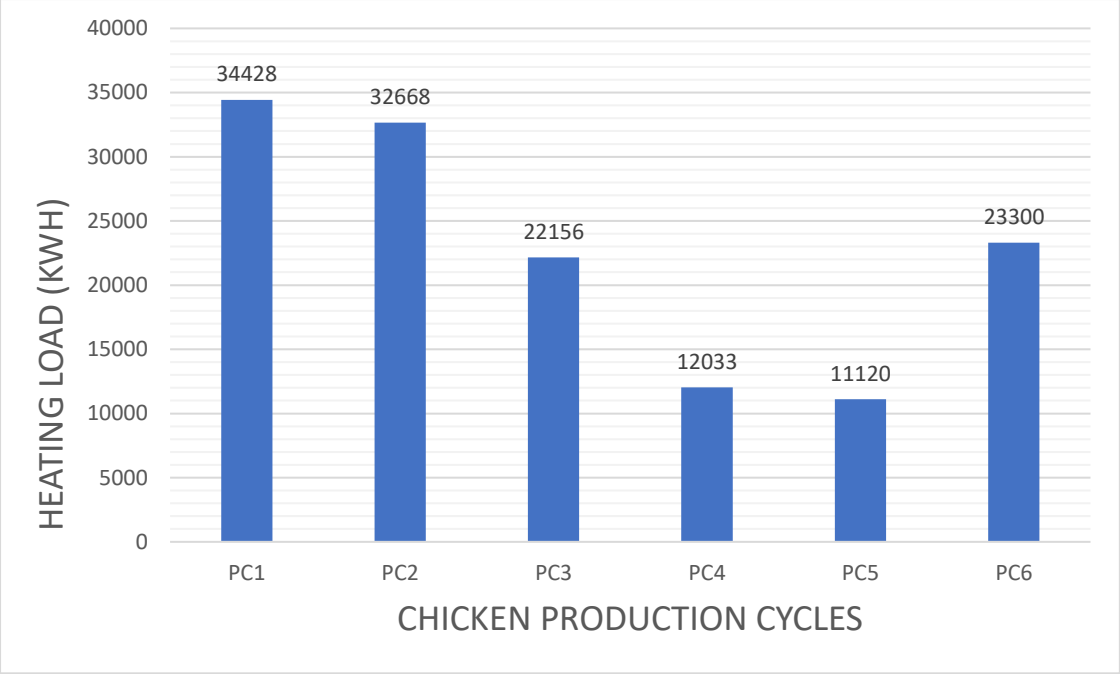


Figure 3-5 Annual energy requirement in the poultry house

3.3.3.2. Design Builder

Simulations were run in two scenario types. The first scenario was to analyse the proportion of energy used in a 24-hour period to assess when the main energy loads are observed within the extreme conditions described. The second scenario involved the simulation of a standard 7-week PC, with the 4th week condition, which has the highest rate of the ventilation in both the winter and summer periods.

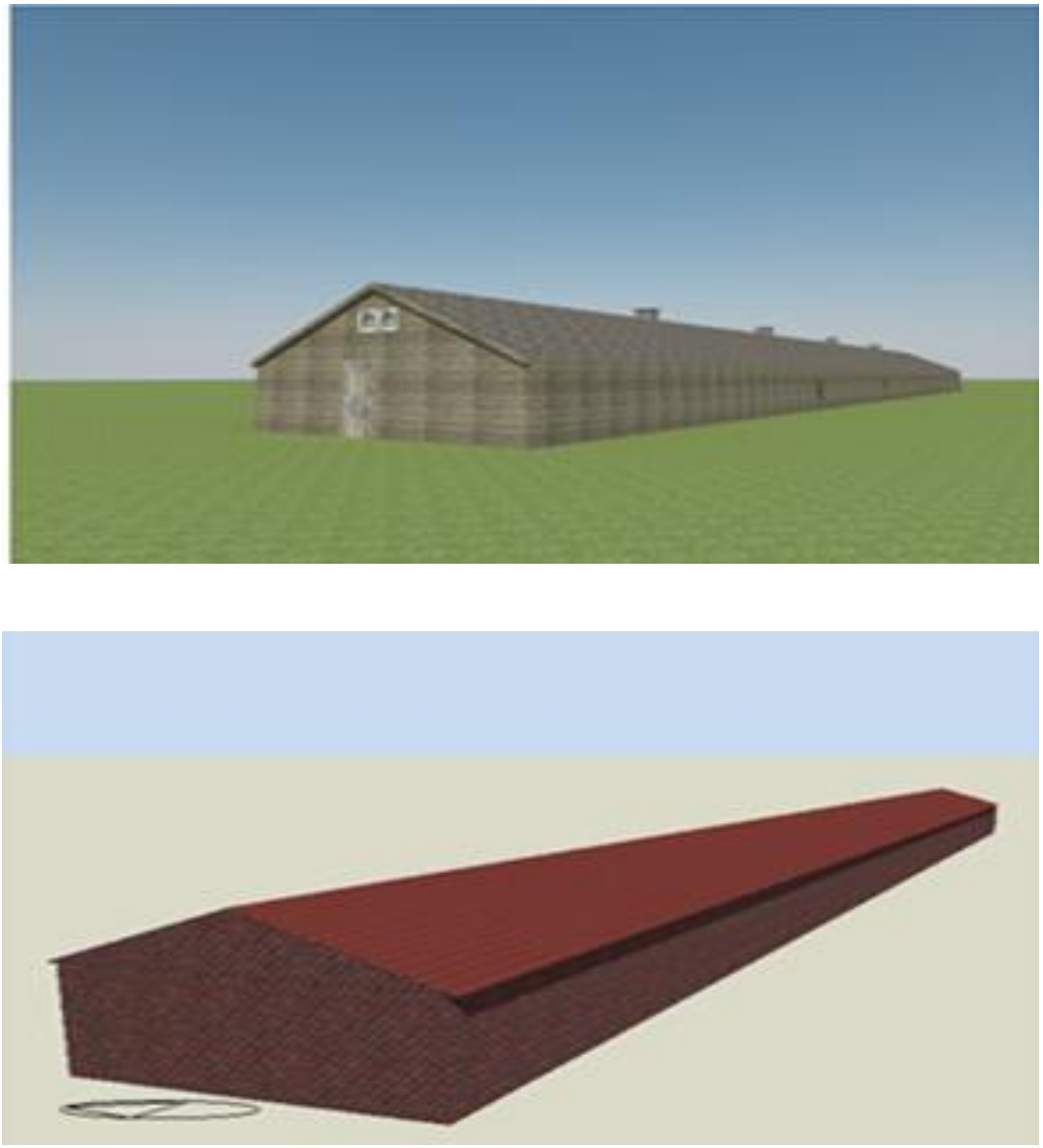


Figure 3-6: Digital model of the JWL poultry facility

Figure 3-6 presents the model constructed and analysed using Design Builder. The purpose of the simulation was to determine the energy requirements of the building that were expected to occur during the 7-week PC within extreme zcweather conditions (winter and summer seasons).

The heating load varies from 60 kW to 120 kW from the first week of the PC to the last week of the PC under the temperature of -4.7^{o}C. The ventilation rate tends to stay constant during the winter, based on the minimum ventilation management with the

seven fans inside the building. Figures 3-7 and Figure 3-8 present the winter and summer peak, for comparison of internal and external temperatures. The air temperature inside the poultry house is observably between radiant and operative temperature, as expected.

The methodology applied on this process was based on the highest energy requirements: (a) high ventilation coupled with the heating requirement in the winter, and (b) supplementary ventilation requirement to account for potential overheating in the summer.

The hourly energy consumption reflects the energy requirement demand from the ventilation systems employed to provide required fresh air for the chickens during peak times. The strategy employed in the simulations reflects the existing (legacy) HVAC system in the poultry house, which (as discussed at length in previous chapters) does not provide the optimum conditions required for the birds' welfare and development.

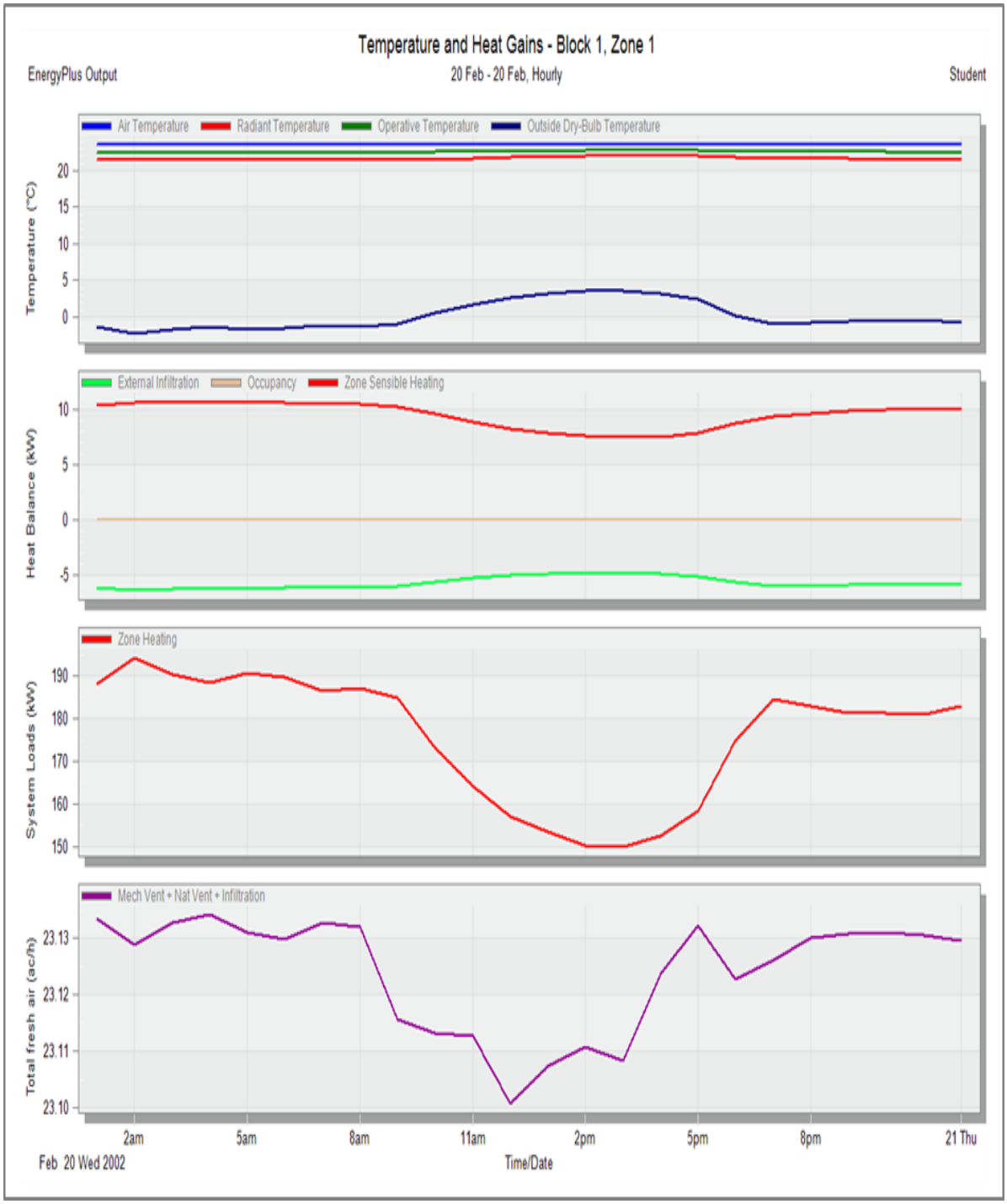


Figure 3-7: Comparison of facility temperatures and energy loads over the winter day 24-hour peak period

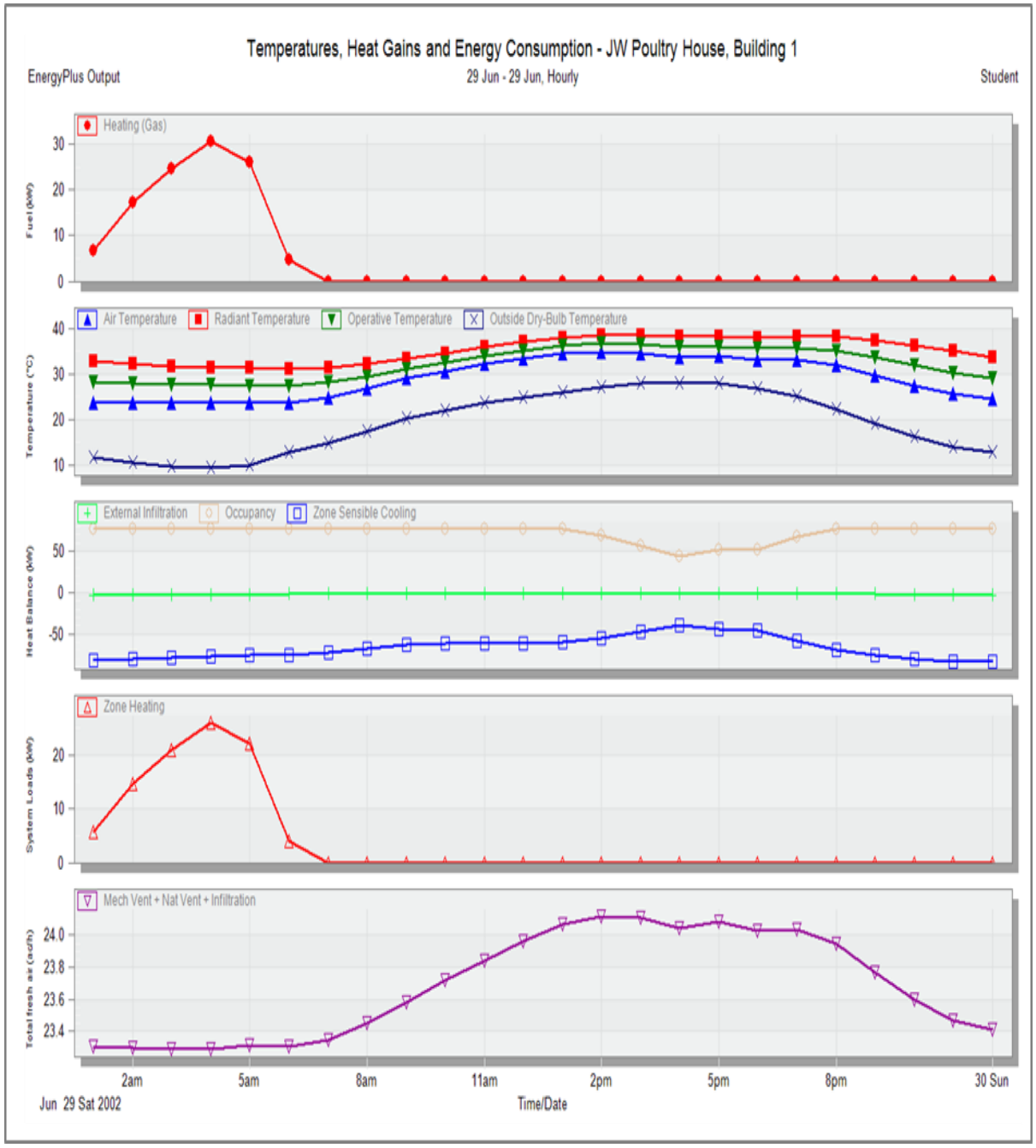


Figure 3-8: Comparison of facility temperatures and energy loads over the summer day 24-hour peak period

In terms of the summer ventilation management, one external fan was used to keep the chickens cool during the peak time of the summer under the summer weather conditions of the UK. The only cooling strategy available at this time was enhanced ventilation with this external fan, which was used to cause forced convention on the surface of the birds to provide comfort. Having stated this, the impact of high ventilation rate coupled with insulated walls caused the internal temperature to fail to fall into comfortable regions. The external peak temperature observed was 29˚C from the data monitored in the poultry house, and the internal temperature of the building failed to fall below this. This heat region is dangerous for the health of the birds and would likely lead to loss of the chickens because of heat stress. The poultry house was mathematically modelled in MATLAB in order to determine the detailed thermal regulation energy demand.

3.3.3.3. MATLAB model

Most of the energy loss is accounted for through the ventilation loss from the total of 125.1 MWh/PC to prevent health issues because of the air pollutants and ammonia inside the building. This causes more energy consumption in the building to maintain required internal temperatures, given the relatively large continual influx of external, unconditioned air. The energy losses due to heat transfer through the building elements are relatively low at 25.2 kWh/ PC, because of the high insulation of the walls in the existing building.

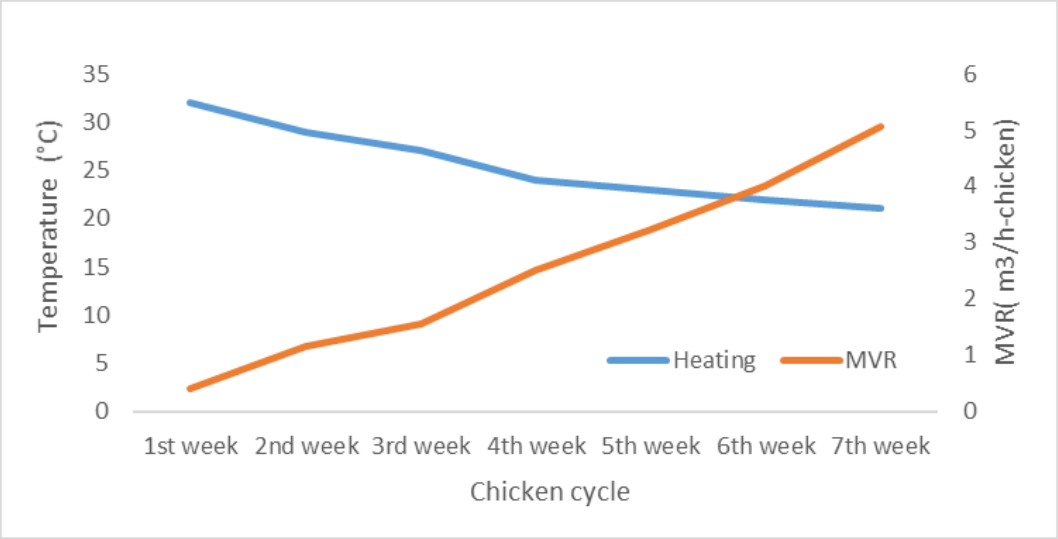


Figure 3-9: Average temperatures and minimum ventilation rates (MVR) for the full 7-week PC

The minimum ventilation management is the key of the HVAC failure and the uncomfortable environmental conditions of the chickens. The existing minimum ventilation rate management and the temperature required depending on the chicken age is applied on the MATLAB model of the poultry house. The maximum heating load is 76 kW for the 4th week of the PC conditions including the occupancy load, with an inside temperature of 24°C and 2.5 m^3/h minimum ventilation rate per chicken. Figure 3-9 presents the heating load of the poultry house under the various temperatures and airflow rate conditions for the full winter PC. The model involves the various internal gains from the chickens, changing with their mass and age.

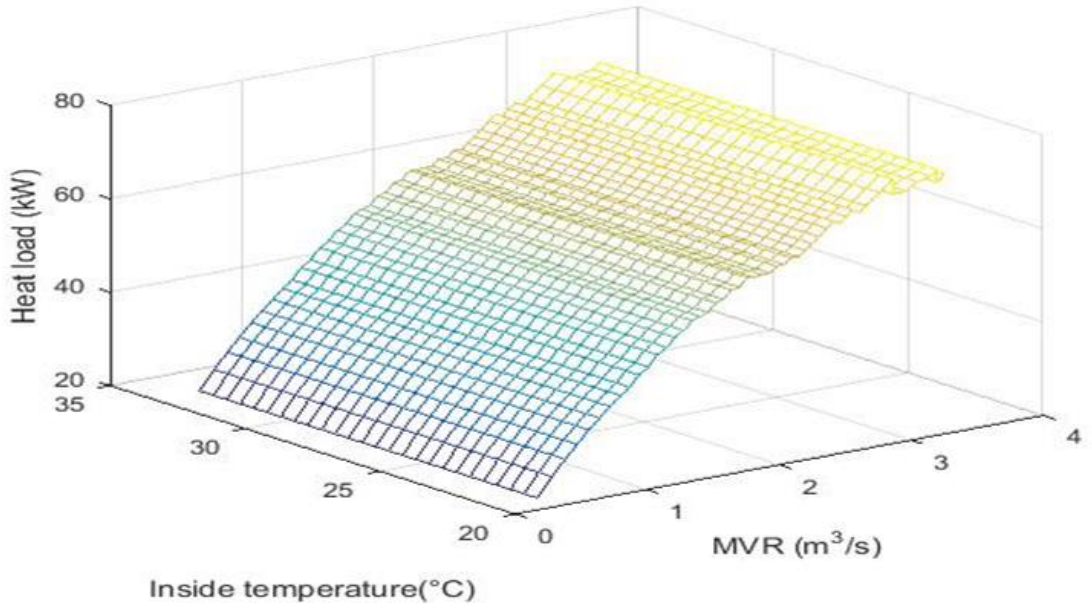


Figure 3-10 Peak load on the coldest day (including chicken occupancy)

The heating load in the building during the year is presented in Figure 3-10. The PCs were analysed to estimate the annual energy requirements of the facility by estimating the total energy requirement based on the number of PCs completed annually. The results are subsequently used to size the output of the HVAC systems ready to be designed and installed for validation in the poultry facility during the whole year.

The simulation results are validated with the energy consumption of the chicken house real consumption data and the bills collected from the farmer. The existing gas boiler is assumed to be run at 85% efficiency on average. The maximum heat demand is during the chicken PC from February to March, despite the coldest term being in January, because of the chicken cycle heating start time (32 ^{o}C in February) and cleaning process, which means that there is no heat requirement for around 15 days during January. This is probably a deliberately scheduled lull in conventional heating during the coldest period of the year, utilising this as a window for downtime and maintenance activities, to avoid the most intensive heating costs.

3.4. Poultry house ventilation management and airflow distribution

Equivalent airflow is fundamental to provide welfare conditions for chickens in poultry houses. Broiler welfare and production is highly affected by air and temperature distribution around the chicken level of the building (usually at or near ground level). In this respect, air movement analysis using CFD was performed for both winter and summer, to analyse the chickens' welfare in the existing building as part of the theoretical investigations of the poultry house. Two approaches are assessed as winter and summer scenarios under the peak temperature conditions. The poultry house has 7 fans with 5,800 m^3/hr (1.6 m^3/s) flow rate, and the volume of the poultry house is around 1500 m^3, with positively pressured ventilation management. The internal conditions vary, with temperatures from 0-40°C, with 10-80% relative humidity. In summer the farmer currently uses very high ventilation rates to achieve thermal comfort, yet it is reported that it can still become uncomfortably warm for the chickens.

3.4.1. CFD analysis of airflow in poultry house during winter

HVAC is currently the key tool in IAQ and temperature maintenance of animal housing, but farmers tend to use minimum ventilation rates (MVR) because of the high costs of ventilation management to provide optimum animal welfare. In extreme winter conditions, ventilation can often be kept at a minimum rate to maintain the building within an acceptable internal temperature range. In the summer, correct ventilation management is crucial to prevent the adverse effects of high ambient temperature on chicken welfare. Figure 3-11 presents the airflow direction in the poultry house. The fans blow air into the volume of the building and create a positive pressure to push air through scheduled outlets, which are notably at the base of the walls (contrary to passive convection airflows). The flow was directed in via the top inlets, as shown in Figure 3-11.

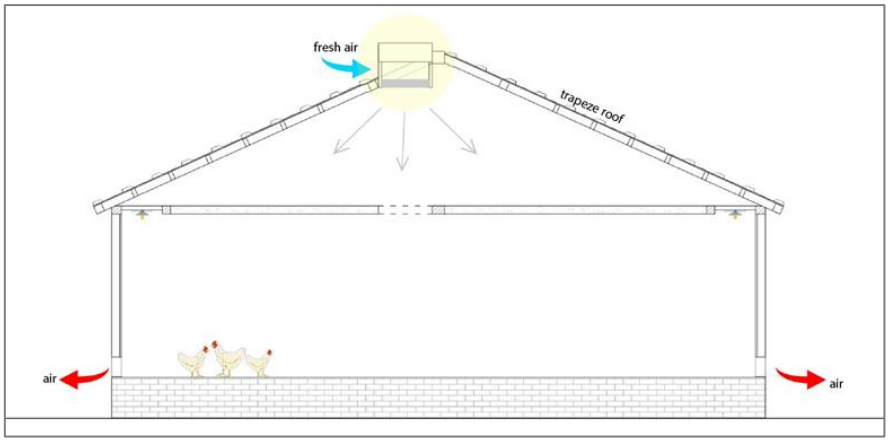


Figure 3-11: Direction of airflow into the facility

A CFD analysis completed under winter conditions is presented in Figures 3-12 and Figure 3-13 for 2 am in February, when the outside air temperature reached an annual low of -4.7ºC. During this time the energy requirements of the building are maximum, with the highest heating energy temperature setting of 32ºC as the inside air temperature condition for the first week of the chickens' lifecycle. The airflow movement and temperature regulation is critical for chickens because of the risk of freezing, and sudden cold and related illnesses.

The results are presented in the following figures as 2D cross-sections and 3D representations of the model. In terms of the distribution of air vertically in the building, the ventilation in the roof zone provides low temperatures from the outside air, although it cannot affect the internal temperature of the occupied area because of the insulated ceiling (an insulation sheet between the occupied area and the fans attached on the trapeze roof, with air vents to the area below, as shown in Figure 3-11). Based on the CFD results shown in Figure 3-12 and Figure 3-13, the current use of positive pressurised ventilation is not recommended because of the chicken level and the height of the scheduled outputs. Due to the air movement provided by the fans in the occupied area, the vertical air and heat circulation was evenly distributed. Hotter

regions tended to be closer to the turbulent areas on the ground due to the high density of level of the chickens and flow per square meter. Air distribution using fans proved relatively effective in the CFD analysis, with the presence of cool spots being relatively low. The maximum airflow on the chicken level is 2.3 m/s, observed in the middle of the occupied area, because there is no insulated ceiling between fans and occupied area in the simulations. The air from the fan strikes the floor and then spreads laterally (with some bounce-back) through the room. The results show that there is no significant temperature difference between the hotter and colder regions, with a maximum of 2˚C.

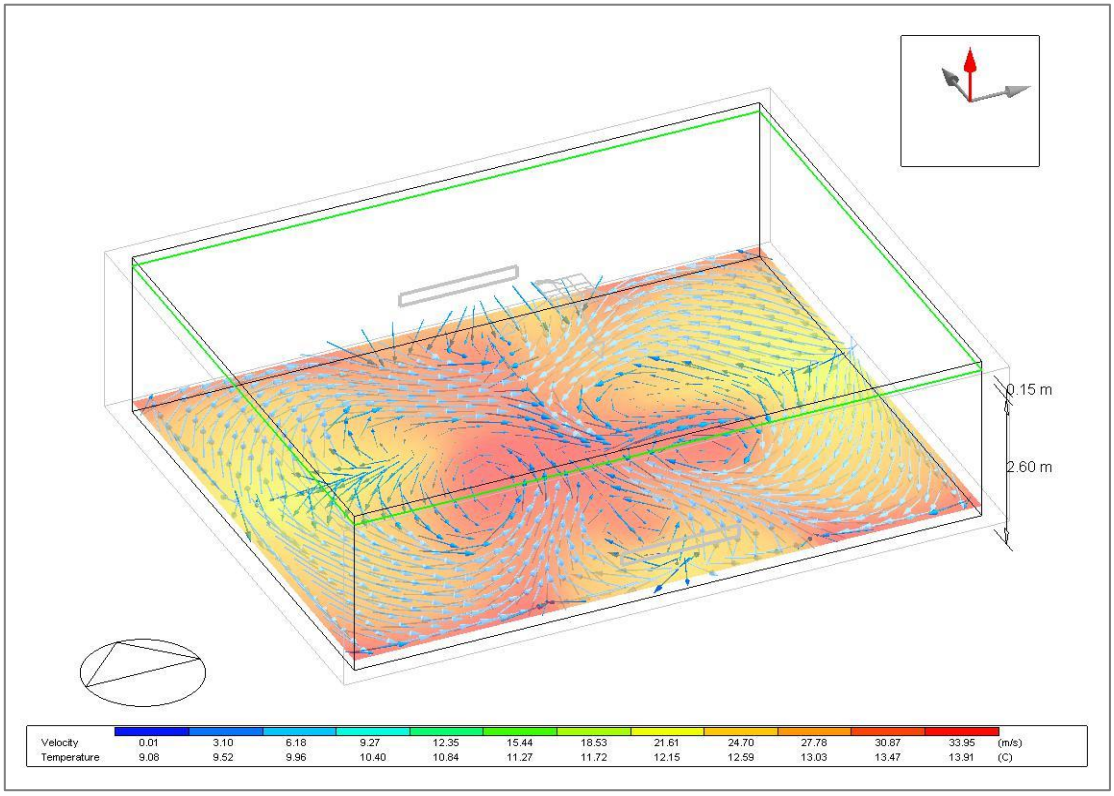


Figure 3-12: Horizontal representation of internal air-flow during winter peak period – heat and velocity

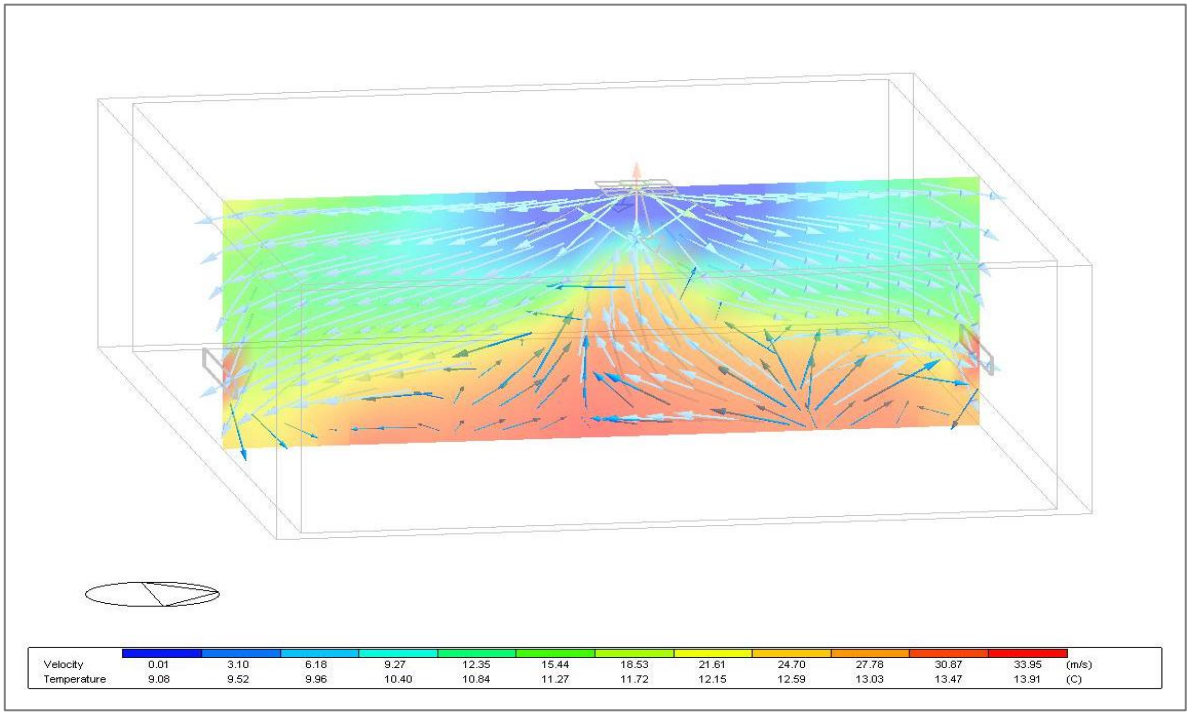


Figure 3-13: Vertical representation of internal air-flow during winter peak period – heat and velocity

3.4.2. CFD analysis of airflow in poultry house during summer

Because of the heat stress issue on poultry farms during the summer, airflow assessment is the very important. If the air is not evenly distributed, hot air causes health issues for chickens due to intrinsic physiological thermoregulation issues, and airborne infections. The CFD analysis was run for conditions with outside air temperature reaching an annual high of 29˚C. During this time the cooling energy requirements were at a peak for the summer period, to supply cold and fresh air to the poultry house.

Under the summer ventilation management conditions, the maximum airflow is observed in the area behind the back door on the chicken level. Bearing in mind there is no insulated ceiling between the fans and the occupied area in the simulations, as explained previously, there is an external fan to keep the chickens cool under hot weather conditions. The only cooling methodology currently available in the poultry house is the provision of high air movement on the birds. On the other hand, the existing fans on the roof create airflow that strikes the floor and then spreads into the room, as explained above. The results show that there is no significant temperature difference between the hotter and colder regions, but the temperature on the chicken level is remarkably uncomfortable for chickens, at 25˚C, as shown in Figure 3-14 and Figure 3-15. This poses serious issues pertaining to airflow distribution and temperature regulation to avoid overheating and environment-related illnesses.

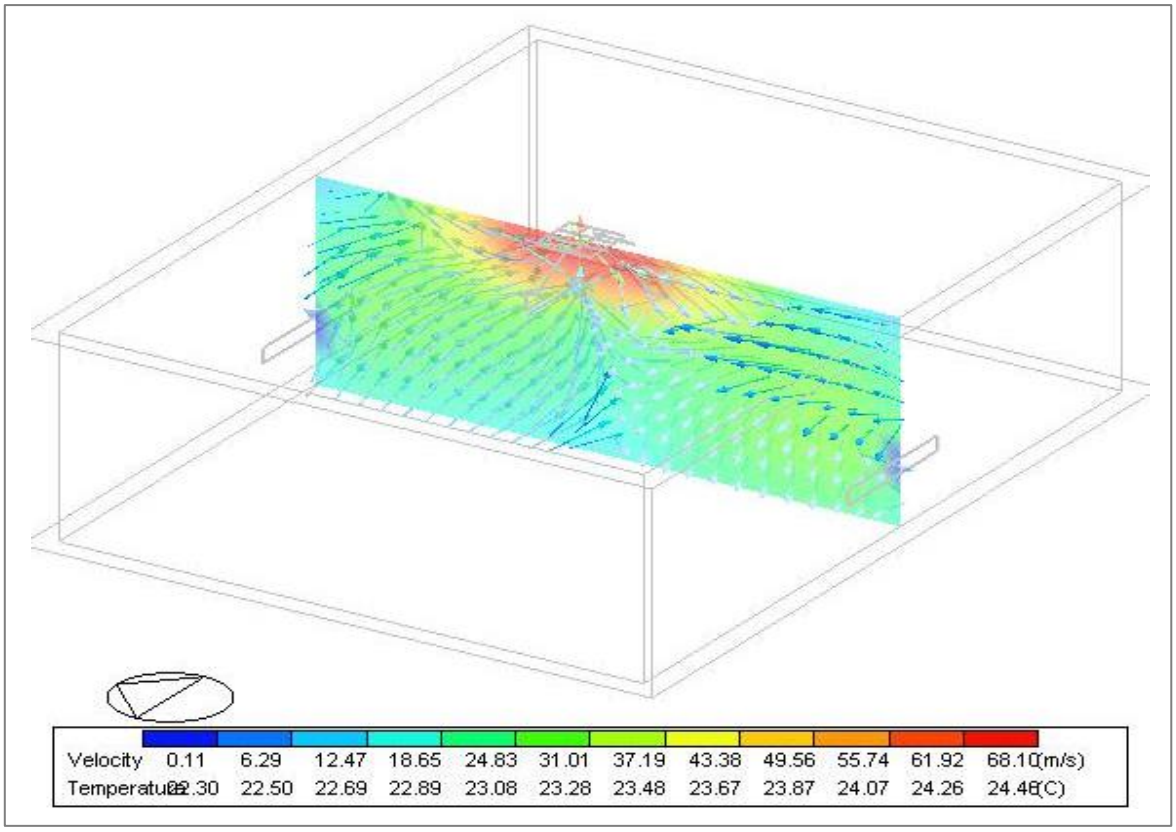


Figure 3-14: Vertical representation of internal air-flow during summer peak period – heat and velocity

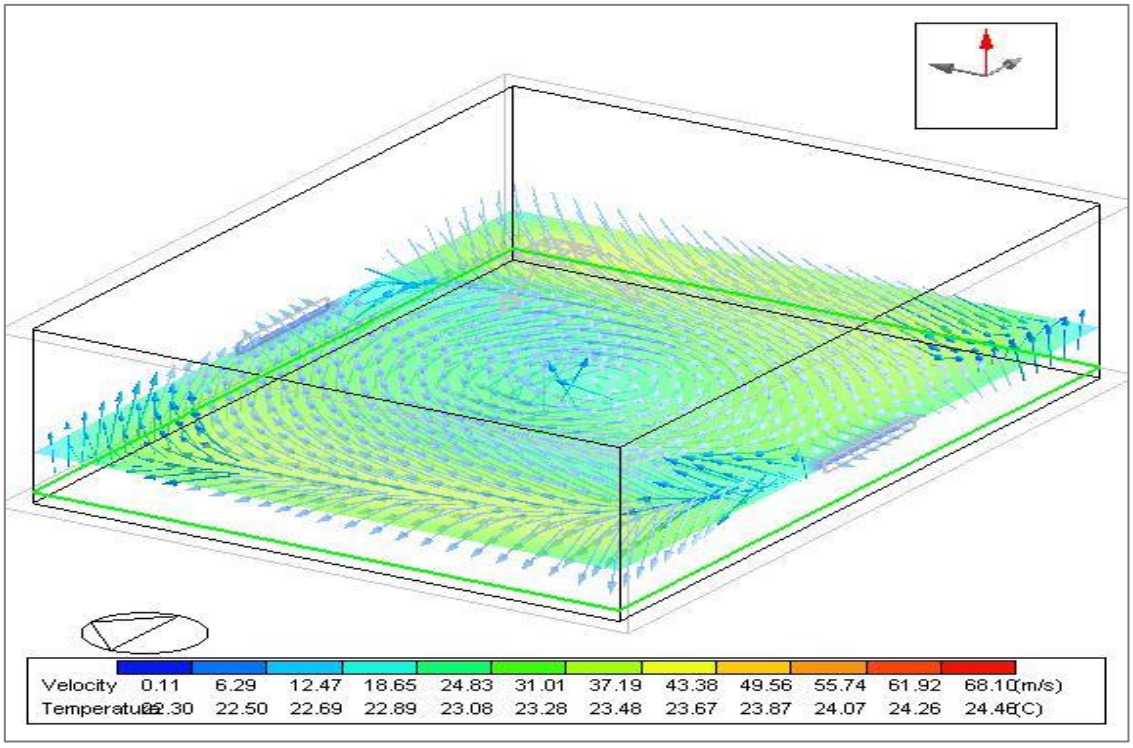


Figure 3-15: Vertical representation of internal air-flow during summer peak period – heat and velocity

Moreover, the velocity of air movement was observed to be higher in the centre of the house, which is an expected result of air drags on the facility walls. The birds would likely occupy the higher velocity region as a consequence of high-density stocks in the building. During summer condition, in terms of vertical distribution, ventilation in the roof space was observed to provide hot outside air during the period. Although this would improve air quality, it would likely cause heat stress to the birds; however, the impact of this limited, due to the high horizontal velocity profiles caused by the large fans operated at the end of the house. Due to sheet/drag on the floor of the building, air velocity in this occupied region was lower than that observed in the centre, based on the measurement of the airflow in the building. The airflow was observed to be 4 m/s from the holes on the side walls during the 4^{th} week of the chickens' Lifecyle, with 2.5 m^3/h minimum ventilation rate per chicken.

3.5. Summary

Variations in hourly energy requirements for the poultry house are affected by external temperatures and the mass and age of the chickens, which is expected given the continual ventilation requirements. The HP size is highly dependent upon the energy consumption profile; therefore the optimum level of insulation and proper management of ventilation are essential. Because of the relatively high capital cost of the renewable HVAC systems (i.e. heat pumps, evaporative coolers, and solar powered systems), adequate sizing is crucial for the renewable thermal regulation on the poultry farms. If a system is oversized it will incur higher installation costs than required, while an undersized configuration will have inefficient periods of operation. In other words, appropriate sizing is essential to make the system economically and environmentally feasible.

The heating load is determined using Design Builder software. The results from the building simulations show that the thermal regulation loads are 100 kW for heating and 40 kW for cooling, respectively. Based on the heating load calculations, the poultry house requires ventilation and heating during all PCs. A mathematical model in MATLAB was developed using combined data collected from the site in Kirton and recorded in Nottingham from the Design Builder library. The model involves the heating loads arising from chicken occupancy and their sensible and latent heat, which fluctuates with the age and mass of chickens. The cooling requirement was determined theoretically and observed experimentally only for the June-July PC (PC 4). Moreover, the observed internal air temperature distribution in the peak period was relatively uniform under the summer and winter conditions. If the air was not well distributed, this would lead to a rapid loss of the stock due to overheating or sudden freezing.

This paper presents the results of a field trial of an innovative and cost-effective HVAC system design applied to a poultry house from the monitored between 2016 and 2018. The observation from the long-term site investigation indicates that UK poultry houses mainly require heating for about 10 months, particularly when growing chickens. Buildings occupied by young chickens require heat in every season and production cycle (PC). The heat demand of the building varied, depending upon the time of day, season and PC. The system provides a highly efficient solution to improve thermal regulation and air quality in poultry houses. It is fully adaptable and customisable by its nature to also address other markets.

The study concludes that:

- The maximum heating demand od the poultry house is determined 34.4 MWh/PC while the minimum is 11,1 MWh/PC.
- The heater of the poultry house is sized as 100 kW under UK weather condition.

CHAPTER 4

EVALUATION OF AN INNOVATIVE PVT-GSHP SYSTEM IN POULTRY HOUSE

4.1. Introduction

This chapter presents component and integrated evaluation, based on transient conditions, of an innovative and cost-effective heating system design for energy-efficient poultry houses. Although extensive work has been carried out on ground heat pump systems and their ground heat exchanger design, the complexity of existing systems and designs has inhibited their wider application in poultry houses. Farmers tend to not to apply GSHPs because of the high prices of excavation and time consumption. There is therefore a need for simpler, more compact heat exchanger designs for heat pumps, particularly for poultry farms, but which can also be adaptable to other agricultural building applications. This study focuses on investigating the possibility of novel sustainable technologies for providing thermal comfort of chickens.

The key design innovations of the hybrid heating system here include the use of a combined design with novel solar polyethylene heat exchanger (PHE) ready loop modules attached between the roof structure of the poultry houses and PV panels, and innovative copper vertical ground heat exchangers (VGHE) attached to the heat pump. PHE harnesses the solar energy to provide a heat source for the heat pump evaporator and a new low-cost ground pipe array which provides the heat required when the thermal output from the PVT array is low. The brine fluid passes down and then back up each VGHE in the ground array loop to extract heat from the soil. The heat pumps combined with these two affordable and easily applicable solar and ground heat exchangers can achieve significant energy, money, and time savings.

This chapter presents the results of a field trial of an innovative and cost-effective HVAC system design applied to a poultry house. This system comprises two integrated novel components: a PHE based a PVT array, and an efficient and innovative ground pipe array coupled to a heat pump driven HVAC system. The poultry farm has four houses that vary slightly in size, with 40,000 birds in total. The selected house (S1) has 6700 chickens, and an area of 502 m^2.

The objectives of the study are summarized below:

- To operate the systems in more sustainable way in agriculture industry, the horizontal ground array is implemented in the heat pump unit in the study.
- In order to reduce energy consumption in heat pump applications, novel VGHE ground heat exchangers and PHE solar heat exchangers are presented. Each of the following novel components' dynamic behaviour under dynamic conditions:
  - Solar polyethylene heat exchanger (PHE)
  - Copper vertical ground heat exchanger (VGHE)
- The novel PVT-GSHP system are experimentally investigated and validated for specified day in winter.

4.2. System design and sub-model description

The thermal interaction of the VGHEs and PHEs with the ground and ambient makes the design process difficult and increase the risk of the under sizing the system. Many studies focus on monthly simulation, and, is enough for most systems. The handicap of monthly simulation is to neglect the effects of extreme hourly or daily variations in the loads, which can cause unrecoverable errors in the system sizing process. Especially if a novel system is aimed to investigate, a simulation is recommended

based on an hourly or smaller time step to provide the best possible design of the actual system.

The heating consumption pattern and hourly average heating load was identified in Chapter 3. A heat pump was designed to provide part load of the poultry house's annual energy demand to evaluate the novel hybrid system performance under the real site condition. 15 kW HP was selected based on the thermal energy collecting from solar and soil on the available spaces in the farm. The heating system design is a hybrid system including novel PVT modules attached to a novel GSHP. The heat pump is sized as 15 kW to provide part load in winter (considering the parameters affecting heat pump efficiency, including return brine temperature, heat sink, and source temperature). 15 kW heat pump was also the largest capacity the farmer could afford whilst considering adequate thermal input from the spaces available for renewable system components in the chicken house. The PVT array size was selected based on the constraints of a suitable building to mount the array on the poultry house in Kirton. A workshop next to the poultry shed is the only suitable location and 13.5 kW is the maximum power output from the largest PV array the roof can support.

A detailed analysis has been conducted for a chosen day, 18th December, for peak heating season with the data taken from the databased of the Design builder and/or the validated measured data**.** The temperature and radiation values have a good match with the data measured on 18th December.

The PVT-GSHP system consists of five subsystems. The first subsystem is PHE – ground array loop for heating the circulating water. Heated water by the sources is mixed and used as a heat source for the heat pump unit. The second subsystem is heat pump device. It is a water source heat pump unit which boosts the collected heat

from the sources and increases the buffer vessel temperature until top temperature reaches to a desired level. As a third subsystem, buffer vessel is used and attached to a fan coil. The fan coil provides the heating required in the poultry house as a fourth subsystem. The buffer vessel stores heat to control the circulating water temperature on the heating loop and make the heat8ing system more stable. Finally, transient heat load profile is simulated in the poultry house under real conditions.

The system's operation mode is on between 9.00 am to 5.00 pm during the day, therefore the system is simulated from 9.00 am to 5.00 pm under the selected winter day conditions. Table 4-1 shows the main loops in the innovative heating system on the cold source

Table 4-1: Main loops in the innovative heating system on the cold source

PVT-ground array circuit	*Component*	*Boundary condition-Flow properties*
PHE loop	SU10 thin-tube PHE PV: 86 m2 PHE: 48 m2	Fluid inlet temp. = 0°C Fluid inlet flow rate range = 1.5-7.5 L/min Mean flow rate 4.5 L/min
VGHE	Vertical Copper ground heat exchanger(VGHE) 50 and 25 mm in diameter 2.5 m in length	Fluid inlet temp = = 0°C Fluid inlet flow rate range = 3.5-8.5 L/min Mean flow rate= 6 L/min
Horizontal ground heat exchangers	Horizontal Polyethylene Pipes, 32 mm in diameter	Fluid inlet temp. range= 0°C Fluid inlet flow rate range=10-15 L/min Mean flow rate= 12.5 L/min
PVT –Ground array	a mixture of PHE return-VGHE return-HGHE return	Flow rate Initial temp.: T brine flow =T mix return
PV power		Peak power output of 13.5 kW

Energy balance equations and heat transfer equations are written in MATLAB software with queasy steady state approach. The temperature of the fluid at inlet and outlet of each component and available solar and ground energy are determined to assess whether the novel component provides enough energy for heat pump. The novel components, solar and ground heat exchangers (PHE and VGHE), are evaluated with one operating mode system running on daily basis under peak heating season condition. The simple control flowchart of the heating system is presented in Figure 4-1.

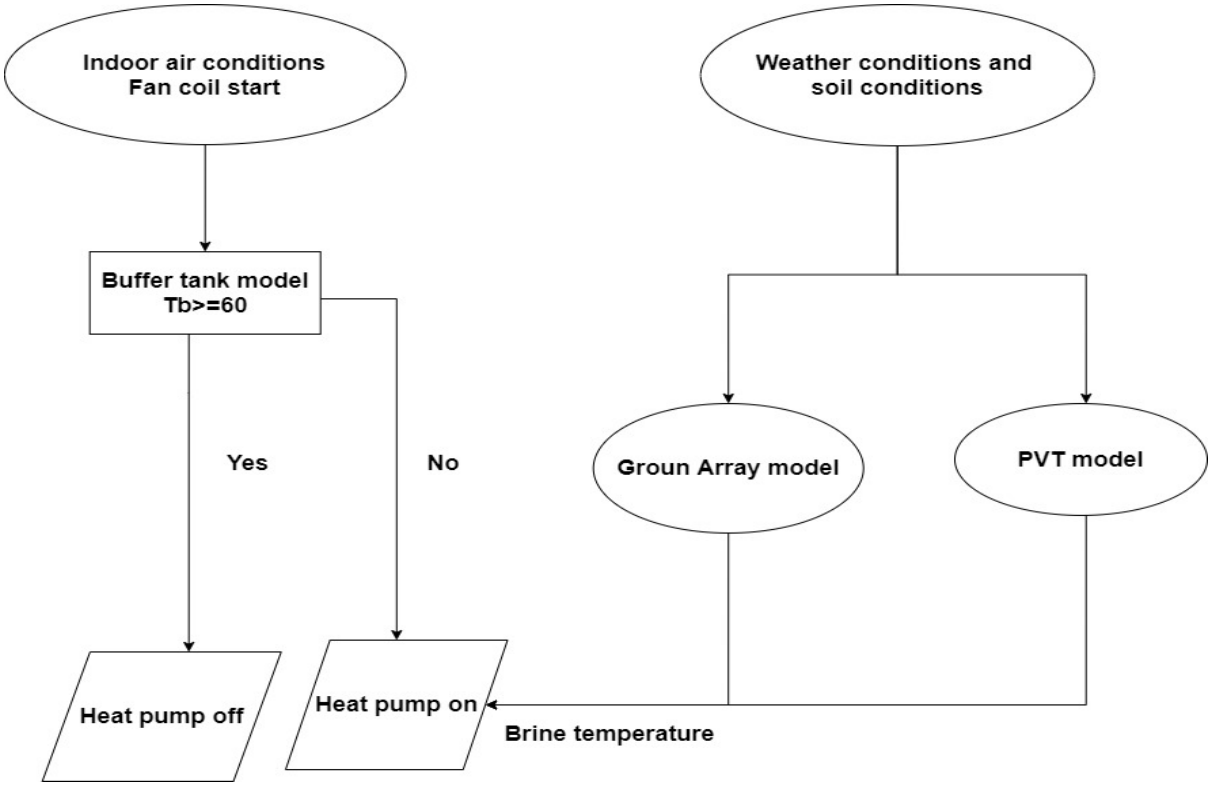


Figure 4-1 System control flowchart of solar assisted heat pump (One operation mode)

The analysis using the basic operating condition presented in Figure 4-1 has been conducted for a chosen day, 18th December, from peak heating season. Fan coil delivery temperatures are designed to a set temperature of 60 ^{o}C in the fan coil the system. The heat pump stops running at 17.00 during peak heating season. The buffer vessel is cooled down during the night-time to 30 ^{o}C, energy consumption from the heat pump will be maximum in the morning and reaches the peak level of the electricity consumption. The heat pump starts running one hour before the fan coil unit switches on to make the system ready to release the hot air inside the poultry house. When temperature reaches the set temperature, 60^{o}C, the heat pump is switched off. The system is not switched on until the temperature at the top of the buffer vessel falls below 55 ^{o}C between 9.00 am and 5.00 pm. The theoretical assessment is identified in this section.

4.2.1. PVT–Ground Array model

Main objective of the section is to show advantages of using novel solar PHE and copper VGHEs attached to the heat pump. Therefore, each PVT and ground array circuit has been modelled by using energy balance and heat transfer equations.

MATLAB software is used to perform the simulation and the simulation parameters are given in Table 4-3 and Table 4-4. The PVT system comprised 52 PV panels, each with an estimated peak power output of 260 W. They were installed on the 15° tilted south-facing roof. The solar optical parameters of the PV module are presented in Table 4-2. The electrical efficiency is 16.16% in the condition of nominal operational temperature.

Table 4-2: Solar optical parameters of the PV

PV module type	*Absorbance (-)*	*Emissivity (-)*	*Thickness, (m)*	*Reference efficiency (%)*	*Temp. coefficient (%/˚C)*
Polycrystalline	0.9	0.96	0.04	16.16	0.053

The specifications of the PHE are given in Table 4-3. Each PHE mat comprises thin tubes, with a distributor and a collector tube. The flowrate of the PHE was between 1.5 L/min and 7.5 L/m.

Table 4-3: Specifications of PHE mat

PHE type	SU10
PHE mat number	4
Thin tubes diameter (mm)	4.3
Distributer tube diameter (mm)	20
Spacing between tubes (mm)	10
Maximum pressure (MPa)	1
Maximum temp. (˚C)	60

The collector system is modelled as operating under transient conditions and used energy balance equations for each layer of the PVT collector presented in Figure 4-2 Glass cover, PV cell (absorber), PHE mat, back sheet and cement roof.

The model is modified from the method used by Bhattarai *et al* (Bhattarai et al. 2012; Chow 2003)The properties of each layer, which means the data used, are presented in Table 4-4(“Canadian Solar Cs6p-260” 2001; “PHE Capillary Tube Mat,” n.d.)

Table 4-4 The properties of PV layer

Glass cover	A	Area of the glass cover	86 m2
	ρ_g	Density of the glass cover	2200 kg/m3
	ε_g	Emissivity of the glass cover	0.8
	c_g	Specific heat capacity	670 J/kg
	δ_g	Thickness of the glass cover	0.003 m
PV cell	A_p	Area of PV cell	78 m2
	δ_P	Thickness of the PV cell	0.0005 m
	k_p	Thermal conductivity of PV cell	148 W/m K
Back sheet	A_b	Area of back sheet	86 m2
	δ_b	Thickness of back sheet	0.0005 m
	k_b	Thermal conductivity of back sheet	0.34 W/m K
PHE mat	$D_{m,o}$	Outer diameter of PHE mat	0.004 m
	$D_{m,i}$	Inner diameter of PHE mat	0.0027 m
	ρ_m	Density of PHE mat	960 kg/m3
	c_m	Specific heat capacity of PHE mat	2260 J/kg.K
	A_m	Area of PHE mat	48 m2
	k_m	Thermal conductivity of PHE mat	0.25 W/m K
Cement roof sheet	ρ_r	Density of cement roof	700 kg/m3
	c_r	Specific heat capacity of cement roof	1000 J/kg.K
	δ_r	Thickness of cement roof	0.18 m
	k_r	Thermal conductivity of cement roof	1.13 W/m K

4.2.1.1. PVT energy balance equations

Solutions to energy balance equation for each layer (Equation 4-1, Equation 4-8, Equation 4-13, Equation 4-16 and Equation 4-20), are obtained by using finite-difference method to discretize the defined parameters. The time step and the space step in the flow direction are dt and dz respectively. Figure 4-2 presents the heat transfer coefficients of the building integrated PV/T system in the model.

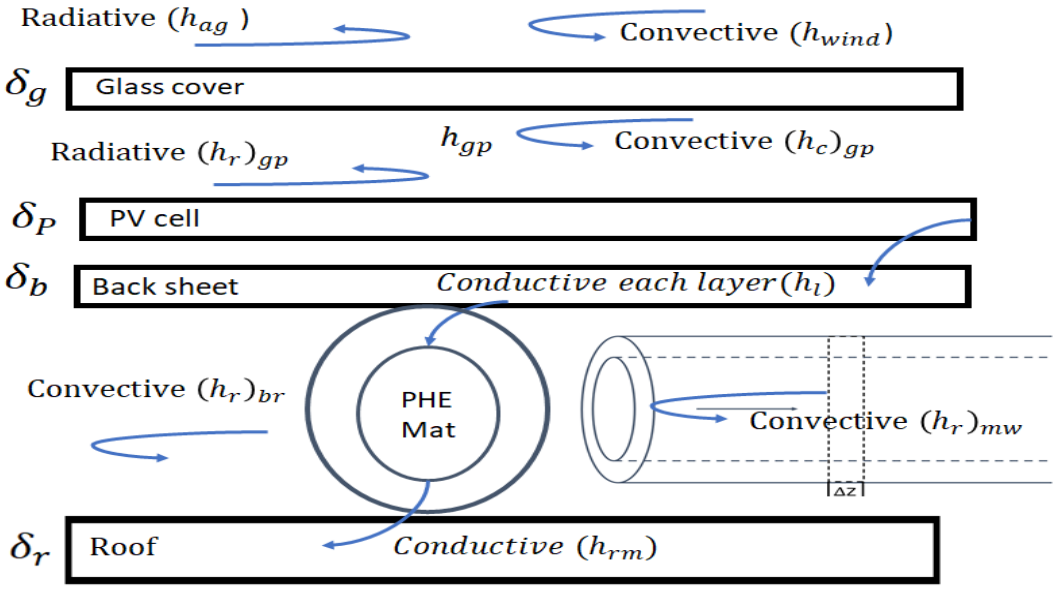


Figure 4-2 Heat transfer coefficients in building integrated PV/T system

❖ **Energy balance equations of Glass Cover**

$$M_g C_g \frac{dT_g}{dt} = G\alpha_g + h_{wind}A_{ag}(T_{am} - T_g) + h_{ag}A_{ag}(T_{sky} - T_g) + h_{gp}A_{gp}(T_p - T_g) \quad 4\text{-}1$$

Where T_g, M_g and $C_g$are the temperature, mass and specific heat of the glass cover, respectively. G is the irradiation in the solar glass. α_g is absorptance of glass. h_{ag} is

the heat transfer between ambient air and glass cover, respectively. The convective heat transfer between ambient air and glass cover, h_{wind}, is (Ji et al. 2008);

$$h_{wind} = (h_c)_{ag} = 3 \cdot u_a + 2.8 \qquad 4\text{-}2$$

h_{gp} , the heat transfer coefficient between PV array and the glass involving air gap, is sum of the convective heat transfer, $(h_c)_{gp}$, and radiative heat transfer, $(h_r)_{gp}$, of the PV array and the glass cover (Chow 2003).

$$h_{gp} = (h_r)_{gp} + (h_c)_{gp} = \frac{\sigma\left({T_g}^2 + {T_p}^2\right)\left(T_g + T_p\right)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g} - 1} + \frac{Nu_a \cdot k_a}{\delta_a} \qquad 4\text{-}3$$

Where h_{ag} is the radiative heat transfer coefficient between ambient and glass cover

$$h_{ag} = (h_r)_{ag} = \varepsilon_g \sigma\left({T_g}^2 + {T_{sky}}^2\right)\left(T_g + T_{sky}\right) \qquad 4\text{-}4$$

T_{sky} is sky temperature, given with a relationship with ambient temperature, and σ is Stefan Bolzmans constant , $5.67*10^{-8}(W/m^2 K^4)$ (Ji et al. 2008).

$$T_{sky} = 0.0522 \cdot {T_a}^{1.5} \qquad 4\text{-}5$$

The Nusselt number of the free convective heat transfer on the PV surfaces is determined by the equation given by Duffie and Beckman (Duffie, Beckman, and Worek 2003). And The Nusselt number of the air gap between PV cell and glass cover is determined by the equation given by Hollands *et al* (K. G. T. Hollands , T. E. Unny , G. D. Raithby 1976)*;*

$$Nu_a = 0.86\, Re_a^{0.5}\ \ Pr_a^{1/3} \qquad 4\text{-}6$$

$$Nu_{ag} = 1 + 1.44\left[1 - \frac{1708}{Ra\delta_{air}cos\theta}\right]^{*} \cdot \left\{1 - \frac{1708[sin(1.8\theta)]^{1.6}}{Ra\delta_{air}cos\theta}\right\} + \left[\left(\frac{Ra\delta_{air}cos\theta}{5830}\right)^{1/3} - 1\right]^{*} \quad 4\text{-}7$$

where Re is Reynolds number; Pr is Prandtl number and Ra is the Rayleigh number. Where 'θ' is the critical angle of the collector slope. The notation $[\ ]*$ in Equation 4.7 implies that, if the determined value is negative, it set equal to zero *al* (K. G. T. Hollands , T. E. Unny , G. D. Raithby 1976).

❖ Energy balance equations of PV array

$$M_p C_p \frac{dT_p}{dt} = G(\tau\alpha)_p - E + h_{gp}A_{gp}\left(T_g - T_p\right) + h_{pb}A_{pb}\left(T_b - T_p\right) \quad 4\text{-}8$$

Where M_p, C_p and T_p are mass, specific heat and temperature of PV plate. $(\tau\alpha)_p$

Is effective absorbance of PV array(Chow 2003).

$$(\tau\alpha)_p = \frac{\tau_a\tau_b\alpha_p}{1 - (1 - \alpha_p)r} \quad 4\text{-}9$$

τ_a is the transmittance of the glass considering only absorption loss, τ_b the transmittance of the glass considering only reflection loss, and r is the reflectance of glass for diffuse radiation. and r is the reflectance of glass for diffuse radiation. $h_{pb}\ is$ conductive heat transfer heat transfer coefficient between back sheet and PV array and between PHE mat and PV array.

The conductive heat transfer equation is given in. Equation 4-10 and applied for each layer which has conductive heat transfer.

$$h_l = \frac{k_l}{\delta_l} \quad 4\text{-}10$$

Where k_l is the conductive heat transfer coefficient between the layers assessed and δ_l is the thickness of the layers assessed (Chow 2003).The values of k_l and δ_l for each layer has been given in Table 4-4.

The electrical efficiency of the solar cell (η_e).

$$\eta_e = \eta_{re}\left[1 - \beta_{re}\left(T_{pv} - T_{re}\right)\right] \quad 4\text{-}11$$

$$E = GA_p\eta_e \quad 4\text{-}12$$

E is the electrical energy. T_r is the reference temperature of the PV array. The main parameters affecting the electrical efficiency of the solar cell are the reference cell efficiency (η_{re}) at a reference operating temperature (T_{re}) and temperature coefficient (β_{re}).

❖ **Energy balance equations of back sheet**

$$M_b C_b \frac{dT_b}{dt} = h_{pb}A_{pb}\left(T_p - T_b\right) + h_{mb}A_{mb}(T_m - T_b) + h_{br}A_{br}(T_r - T_b) \quad 4\text{-}13$$

Where M_b, C_b and T_b are mass, specific heat and temperature of back sheet. h_{mb} and h_{ar} are the heat transfer heat transfer coefficient between back sheet and PHE mat and between cement roof sheet and back sheet.

The radiation can be neglected in the air between back sheet and cement roof because of the shading (Chow 2003).The area of the air gap between back sheet and cement roof is A_{br}.

$$h_{br} = (h_r)_{br} = \frac{Nu_a \cdot k_a}{\delta_a} \quad 4\text{-}14$$

$$A_{br} = \left(\frac{\pi}{2} + 1\right) D_o dz \quad 4\text{-}15$$

- **Energy balance equations of PHE mat**

$$M_m C_m \frac{dT_m}{dt} = h_{bm} A_{bm} (T_b - T_m) + h_{rm} A_{rm} (T_r - T_m) + h_{mw} A_{mw} (T_w - T_m) \quad 4\text{-}16$$

Where M_m, C_m and T_m are mass, specific heat and temperature of back sheet. h_{mw} is the heat transfer coefficient between PHE mat and Ethylene glycol water.

Heat flow from the PHE surface to the water in tube is by means of conduction and convection is determined; The convective heat transfer, $(h_r)_{mw}$, can be obtained from the Dittus–Boelter equation for fully developed turbulent flow and Equation 4-18 is used if the flow is fully developed convective laminar flow (Chow 2003)

$$(h_r)_{mw} = 4.364 \frac{k_w}{D_i} \quad 4\text{-}17$$

$$A_{mw} = \pi D_i dz \quad 4\text{-}18$$

Ethylene glycol water (brine) in the PHE

Heat flow from the PHE mat to the water in tube is (conduction and convection) is required adapted equation used by Bhattarai *et al.* (Bhattarai et al. 2012).

$$T_w = \frac{T_i + T_o}{2} \qquad 4\text{-}19$$

$$M_w C_w \frac{dT_w}{dt} = h_{mw} A_{mw} (T_m - T_w) + \dot{m}_w C_w \frac{dT_w}{dz} \qquad 4\text{-}20$$

T_i and T_o are the temperature of inlet and outlet of the mixed water.

❖ **Mathematical model flowchart**

The flow chart of the numerical solutions of the derived model is presented in Figure 4-3.

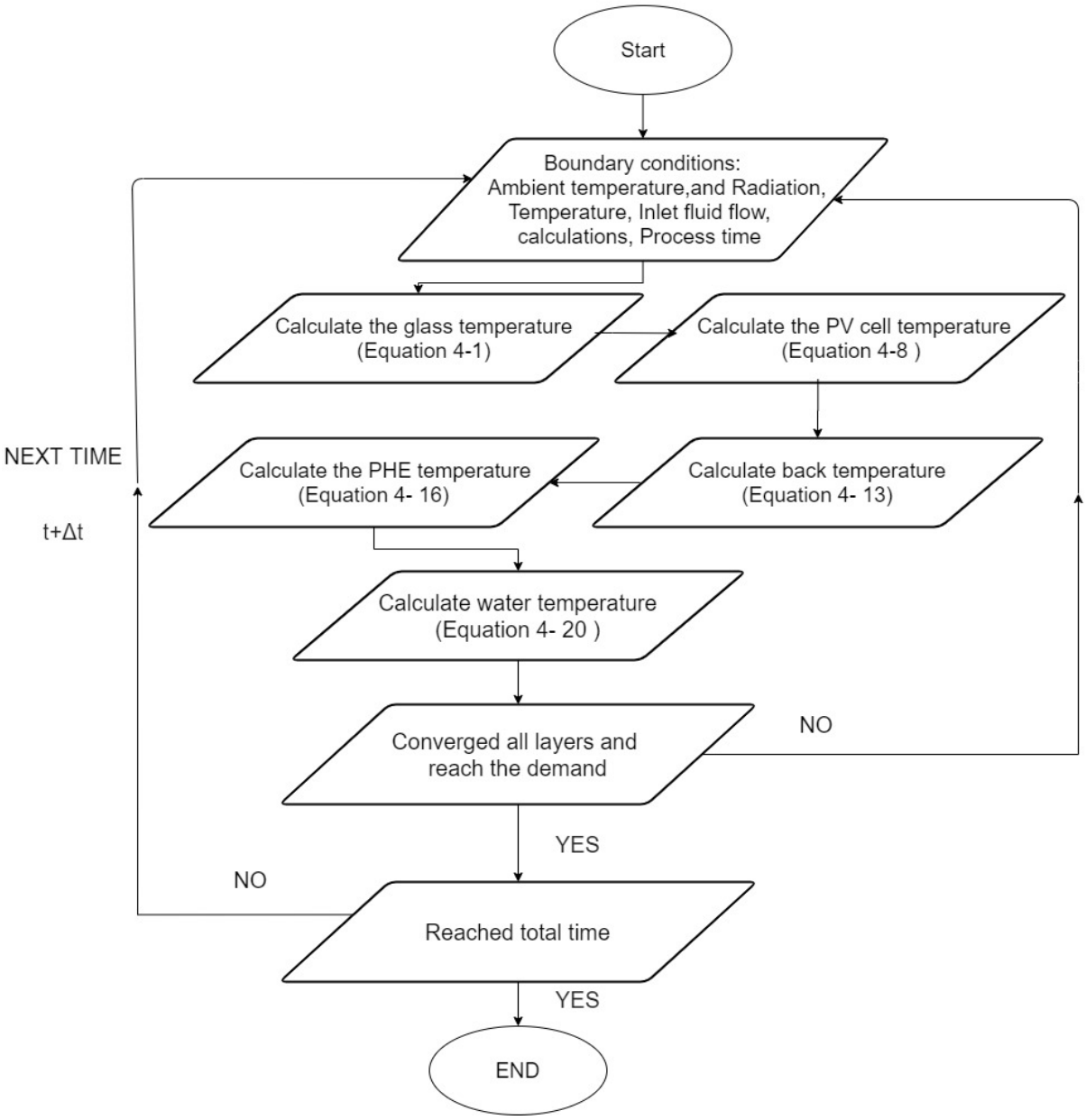


Figure 4-3 Flow chart of the Solar PHE based PVT mathematical model

4.2.1.2. Ground Array

The productivity of the ground array and heat pump is influenced by soil depth, length, and diameters of thermal ground pipes, the thermal conductivity of the materials in the components, thermal diffusivity of soil, ground temperature values and their distribution. The useful energy which can be extracted from the soil is influenced by two main parameters: soil temperature at the depth of the borehole.

The physical properties of soil are presented in Table 4-5. A method developed by Kusuda has been used to determine the temperature of the soil (T. Kusuda and R. P. Achenbach 1965) and the seasonal swing of ground temperatures has to be taken into consideration to investigate UK shallow ground temperatures (Jon Busby 2011).

$$T(z) = T_{mean} - T_{amp} \times \exp\left(-D \times \sqrt{\frac{\pi}{365 \times \alpha_z}}\right) \times \cos\left(\sqrt{\frac{2\pi}{365}} \times \left[t_{year} - t_{shift} - \frac{D}{2} \times \sqrt{\frac{365}{\pi \times \alpha_z}}\right]\right) \quad 4\text{-}21$$

Where is $T(z)$ the ground temperature of soil; is the mean surface which is the assumed as the average air temperature is the amplitude of surface temperature; D is the depth below the surface; α_z is the thermal diffusivity of soil; t_{year} is the current time; t_{shift} is the day of the minimum surface temperature. T(z) and the ground thermal diffusivity (α_z). Thermal diffusivity depends on soil type, density and water content.

Available heat, T(z) in the homogenous soil is calculated in simple terms to determine the heat extraction limit of the new innovative ground pipes. The ground surface is exposed to a natural environment thereby influenced by seasonal temperature fluctuations. Above a depth of around 15 m, soil temperatures are affected by outdoor temperatures with daily and seasonal terms (Márquez, Bohórquez, and Melgar 2016; J Busby et al. 2009)**.** The method developed by Kusuda has been used to determine the temperature of the soil (T. Kusuda and R. P. Achenbach 1965)**.**

Table 4-5 Physical properties of soil (Y. Cui and Zhu 2018)

Depth of the soil (m)	**TMaterial** (W/mK)	**Thermal conductivity** (W/mK)	**Density** (kg/m^3)
0-2.22	Mixed gravel and coarse sand	1.30	2277
2.2-3.3	Sand gravel	1.15	2094
3.3-5.5	Slightly gravel	1.68	2223
5.5-10	Slightly gravel clay	1.75	2392

There are various numerical methods that use the finite difference method, finite element method, and finite volume method that can be used to model geothermal boreholes. These methods can be used to generate g-functions but are not typically used directly for design purposes (Spitler and Bernier 2016).

Most of the currently available models are focused on long-term response simulations, while models able to predict the BHE short-term behaviour are usually based on numerical techniques like such as finite difference method, finite element method, and finite volume method, with consequent high computational costs. There are valuable analytical and numerical developing models to decrease the time and the costs of the ground heat exchanger design. Horizontal Ground heat Exchangers (GHE) have a lower cost in comparison to vertical ones; however, they require more land area, and they are relatively less efficient. For the sake of reducing the required land area for horizontal GHEs, flat panel GHEs have recently been introduced and tested (M Bottarelli, Bortoloni, and Su 2019; Michele Bottarelli and Ferrara 2019) and a copper tube with an outer surface protected with low-density polyethylene HGHE have been developed (Kariya and Miyara, n.d.). It is beneficial to implement simplified mathematical models & energy balance equations of heat transfer for this

experimentally focus and techno economical investigation study for the agriculture industry. To simple analyse temperature gradients with heat transfer process of the ground array some assumptions have been done and the temperature variation has been given in

4.2.2. Heat pump

Heat pump models are often categorized depending on the physical approach: (1) Thermodynamic/physical approach, based on the geometry of the heat pump and general physics laws (heat and mass transfer); (2) Black box approach, in which empirical correlations determine heat pump performances as a function of the conditions of use (exterior and interior temperatures, power demands, etc.) (3) Grey box approach, which is at the intersection of the two above-mentioned approaches. The model consists of a quasi-steady state distributed parameter model of a ground source heat pump and a fan coil unit.

Table 4-6 The heat pump standards

T out [°C]	Part load capacity	COP
-10	14,6**	2,96
-7°C	14,6**	3,16
-5.1 °C	14,6	3,27
+2 °C	14,8	3,72
+7 °C	15,1	4,01
+12 °C	15,4	4,27

* Max supply temperature= 55 [°C
** Supplementary heater(3.4 kW)

Moreover, a simplified thermodynamic/physical approach has been taken to characterize the operation of the brine-to-water heat pump. The model has a good

match with performance data described by the manufacturer of the heat pump (NIBE) presented in Table 4-6.

The heat pump is a water source heat pump attached to the fan coil unit for space heating. HP specifications are shown in Table 4-7. The heat pump has four main components as mentioned; Evaporator, compressor, expansion valve and condenser. The evaporator is a heat exchanger which transfers the collected heat from the sources, which are solar and soil in this study, into the heat pump. The evaporation is an isobaric process.

Table 4-7:Specifications of the heat pump

Component	*Value*
Rated thermal output	15.4 kW
Immersion heater	4.5 kW
Operating voltage	400 V
Maximum temp. heating medium	60/58°C

The electricity consumed by the compressor including the internal water pump of the HP is W_{HP} was measured by using an electricity meter and heat Q_{HP} provided by HP was measured by a heat meter 10 minutes intervals. The COP of the HP is then given by the following equation;

$$COP = \frac{Q_{HP}}{W_{HP}} \qquad 4\text{-}22$$

4.2.3. Buffer vessel

In this system, collected heat from the heat pump is stored in a buffer tank. Sizing the buffer vessel volume is related with charging flow rates. To simplify the model, it is assumed that the water in the insulated water storage tank is completely mixed with the water (Sameti et al. 2014).

In this study, part load of the space heating for the poultry house is considered and a 300L buffer vessel is chosen for the analysis. In control volume, an energy balance equation can be written considering the heat loss to the environment. By solving the energy balance equations, temperature on average inside the tank can be determined. Space temperature is assumed as 20ºC. There is a surface area between water and the ambient. U is overall tank heat loss coefficient.

$$m_{tan} \cdot c_{w,tank} \cdot \frac{dT_{tank}}{dt} = Q_{HP} - Q_{store} - UA \cdot (T_{tank} - T_{plant}) \qquad 4\text{-}23$$

Where Q_{HP} is the heat gain from HP to the storage, Q_{store} is the heat remained heat after Fan Coil use, U is the heat transfer coefficient and A is the surface area of the tank.

4.3. Novel PVT-GSHP system description and poultry house set-up

4.3.1. System design

The novel solar-based GSHP design has three design characteristics that aim to address the issues of size of the PHE and ground pipe arrays (novel vertical and commercial horizontal) and thermal and electrical energy output of each component; the whole heating system's thermal energy outputs and its runtime; and the daytime

energy needs of the poultry house. During winter conditions, an auxiliary or back-up heating system provides the additional heat for adequate heating to maintain the required indoor temperature.

The schematic diagram of the novel PVT-GSHP system is presented in Figure 4-4. This productive hybrid design leads the PV cell to operate more efficiently at relatively high temperatures. Meanwhile, the heat absorbed by PHE is used for meeting the heat demand in the evaporative section of the HP for space heating in the chicken house. A summary of the main circuits of the heating system is given in Table 4-8. The hot water in the buffer tank was sent to the fan coil unit. The fan coil thermostat was set based on the required house internal temperature. Data were recorded at 10-minute intervals during winter weather conditions.

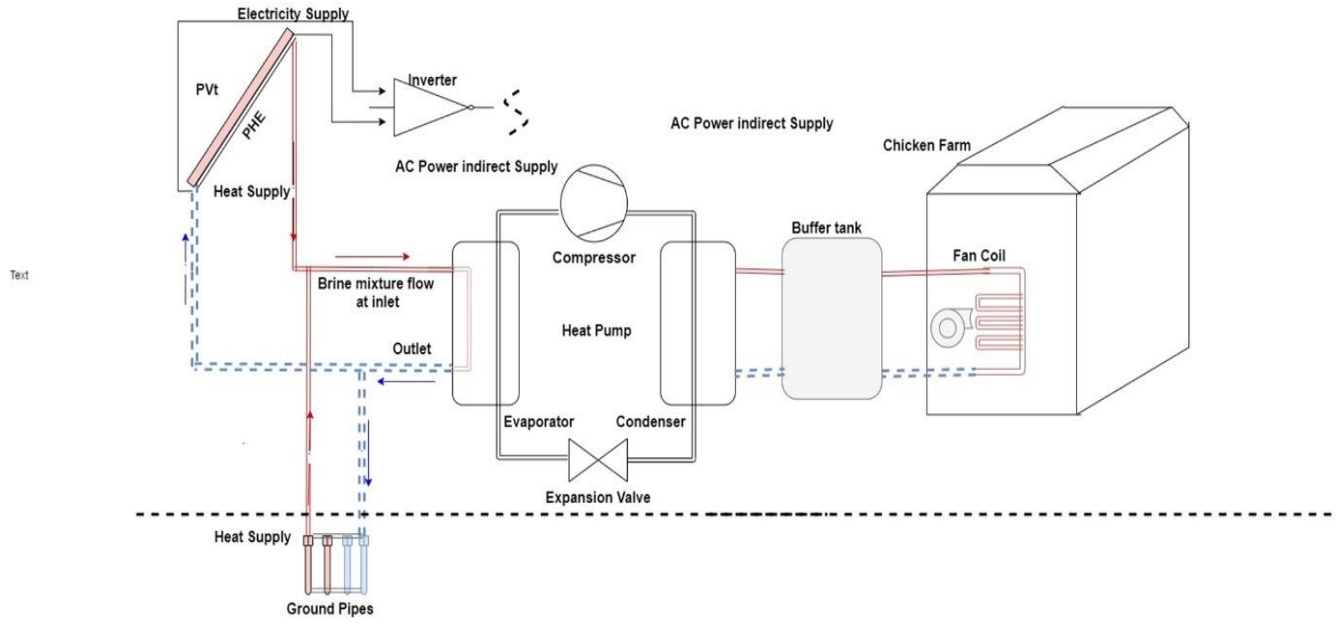


Figure 4-4 PVT-GSHP heating system

Table 4-8: Technical information of each component in the innovative heating system

Air-brine-refrigerant loops	Component	Technical information
Fan coil brine loop	Central heating circulating pump Fan coil unit Poultry house	Grundfos UPS2 15-50/60 Daikin FW12DAT fan coil (3/4" BSP)
HP refrigerant loop	Evaporator Expansion valve Condenser Compressor	R407C (0.085 kg/s mass flow rate) NIBE F1145 15 kW 9 kW electrical immersion heater
PVT ground array brine loop	SU10 thin-tube PHE Ground thermal pipes Central heating Circulating pump	PV: 86 m2 PHE: 48 m2 4 PHE module (12 m1 m): 20 mm in diameter main manifold tube, 4.3 mm diameter capillary tubes (10 mm gap) 4 Grundfos UPS2 15-50/60 -240 V; CEM20 expansion pack 50 25 mm in diameter vertical copper pipes (2.5 m length) m in diameter horizontal polyethylene pipes

The system comprises a fan coil unit, a GSHP ducted with a buffer vessel, and a PVT unit and ground source array (Figure 4-5). Furthermore, a grid-connected PVT power system is used to supply the electrical demand of the HP indirectly. Electricity is imported or exported easily, depending on power demand. The commercial 15 kW F1145 NIBE HP is integrated with innovative thin-tube PHE installed behind commercial PV panels and ground thermal pipes to provide heating to S1 during winter months and most summer months, depending on the required temperature. The innovative thermal pipes can range between 2.5-10 m in length. Excess daytime thermal energy from the heat pump is directed to the 300 L NIBE buffer tank, and then used for periods of insufficient thermal input during the day. This commercial buffer

tank is connected to a Daikin FWD12AT fan coil unit located in the middle of the south-facing wall of the house.

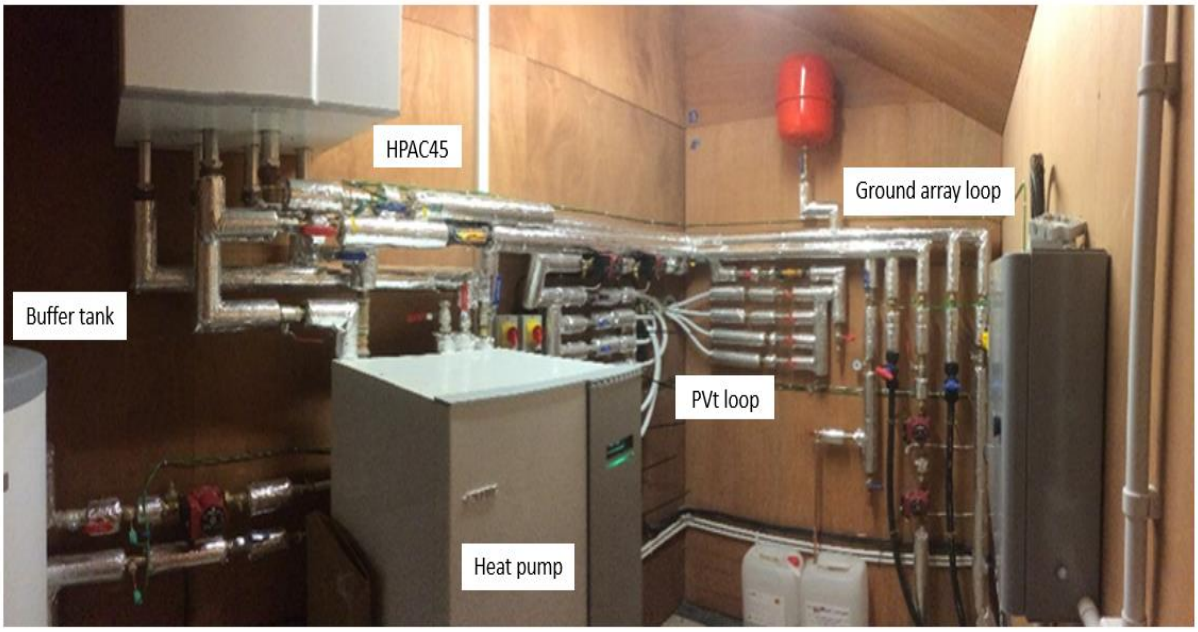


Figure 4-5: Heating system plant room

4.3.2. Photovoltaic thermal (PVT) array

Four thin-tube (capillary tube) PHE mats (1 m × 12 m) were placed underneath PV panels to extract heat from the PV cells and supply heat to the HP Capillary tubes have the dimensions of 4.3 mm diameter and 12 m length, with 10 mm distances between tubes. PVT collectors supply heat for the ground during the peak time of the summer to charge the soil. PHE was located and connected in a series, based on the Tichelman reverse return impact, to distribute the brine equally (Figure 4-6).

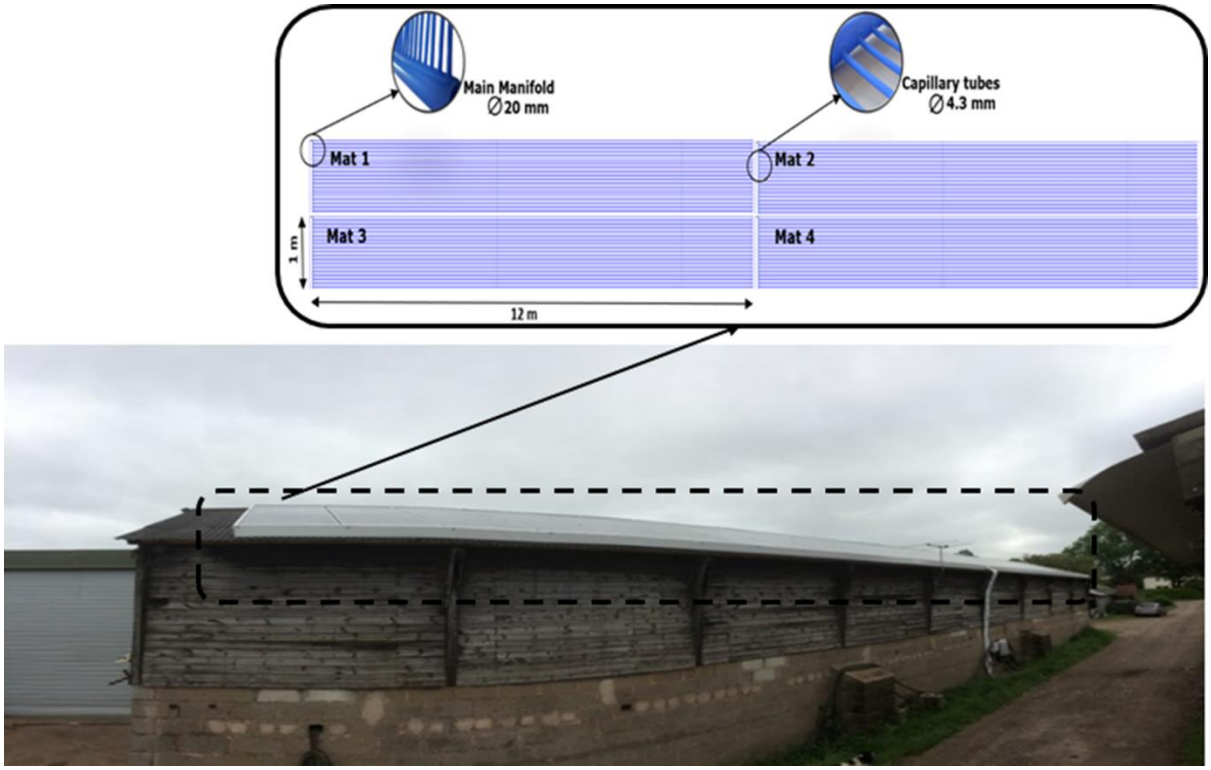


Figure 4-6: Thin tube PHE mats and its set up

4.3.3. Ground pipe array

The ground pipe array system comprised 50 vertical co-axial pipes (25 mm in diameter) connected in serial and a horizontal pipe (32 mm in diameter). The pipes were attached to a simple set of units of copper vertical pipes and polyethylene horizontal pipes to extract heat from the available ground area. Figure 4-7 shows the pictures and the sketches of vertical and horizontal thermal pipes and their flow direction through pipes within their inner and outer channels, respectively. In the heating system, the innovative vertical thermal pipe of 2.5 m in length was used to extract heat from the soil, which was easily installed in the ground array of 100 m^2 (10 m x 10 m) surface area at low-cost.

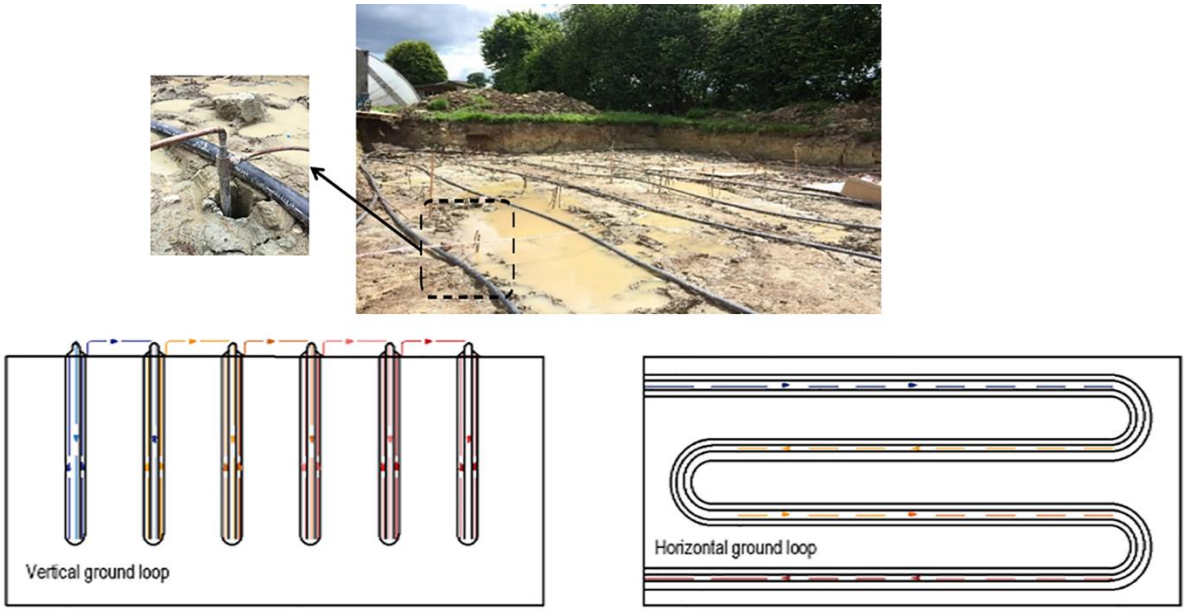


Figure 4-7: Ground loop sketches of flow direction and horizontal closed linear loop

The insulation inside the trenches is 1.50 m in depth and the ends of the copper tubes are sealed properly. The main challenge of the ground heat pump is the time-consuming and expensive process of vertical borehole drilling. The novel vertical ground thermal pipes can be placed into the soil easily, in 10 minutes per pipe, via a portable petrol hydraulic vibrating hand piling unit.

4.3.4. Heat pump-buffer tank-fan coil unit

NIBE F1145 HP was used to maintain a temperature of 21-32°C, depending on the chicken age. The commercial 15 kW HP circulated R407 C comprised a compressor, condenser, evaporator, and expansion valve. The fan coil fed into a highly-insulated A 300 L NIBE UKV buffer located between HP and fan coil unit. This allowed more brine to be heated and stored at the sunniest peak time, to be used for required heat during the least sunny hours. In the heating mode, the heat coming from the condenser section of the HP was stored in the 300 L buffer tank to both prevent the interaction of the heat losses due to the fluctuated ambient air and make the system more efficient. The heat stored in the buffer tank was between 55-60˚C when the HP was run between

the hours of 9.00 and 17.00. The fan coil provided heat for the poultry house, with hot brine coming from the buffer vessel. The unit was mounted on the south-facing wall of the building.

The heat pump has water circulation pumps on both its evaporator and condenser side. The evaporator side water pump was used to circulate the water in the ground pipe array. Similarly, the water pump on the condenser side of the heat pump was used to circulate the water to the buffer tank. The fan coil unit control was provided by an air thermostat, monitoring return air against a set point. This then operated the valves on the water flow from the heat pump

NIBE F1145 can be connected to various accessories, e.g. solar panels, extra hot water heaters, and ventilation recovery and heating systems with different temperatures. Because the temperature of the brine system can fall below 0°C, it must be protected against freezing down to -15°C, using ready mixed brine (Thermox ITF20) as system flow. Because the inlet temperature ranges from 20-32°C and the climate system volume is considered too small for the heat pump output, the fan-coil system is supplemented with a buffer vessel, A 300 L NIBE UKV. This volume was simply determined to provide adequate thermal storage capacity to ameliorate variations in heating supply and demand.

4.4. Novel PVT-GSHP system and daily monitoring

4.4.1. Instrumentation and Uncertainty

4.4.1.1. Instrumentation

The system was equipped with instrumentation for inside-outside temperature radiation and energy monitoring (stored, used, and provided). Flow and return temperatures, flow rates, and electrical and heat energy were measured by the

instrumentation method below (Table 4-9). All data were measured at 10-minute intervals.

Table 4-9: Instrumentation method

Instrument	*Location-measured parameters*
Lascar sensors	Outside and inside house temp.
Vasalia sensors	Outside and inside house temp.
Pyrometer	Incident solar radiation
Sharky 775 heat meter	The thermal output of the heat pump system
	Brine entering and leaving temperatures in the PVT array, the ground horizontal and vertical array
	Hot or cold brine in the fan coil
	On the top mid and endpoints in the buffer tank
DT85 Data Taker	All connected to a DT85 Data Taker, Vasalia sensors (Pyrometer, A Sharky 775 heat meter, K type thermocouples)
A GSM energy meter	PV electricity output
Electricity meter	Electrical input of the compressor of heat pump
TinyTag sensor	Internal CO_2 concentrations
Portaflow 330	Flow rate measurements

Portaflow 330 was clamped on the outside of the pipe. The pipe details were entered, including material, outside diameter, and either wall thickness or inside diameter. This was used to calculate the distance between the sensors. The types of pipes are copper tube 25-50 mm and 20-32 mm polyethylene. Flow and return directions were considered, however downwards measurements were not taken, because downward

flow is not advised for any form of flow measurement due to the interference of cavitation and swirling etc. with readings.

4.4.1.2. Uncertainty Analysis

Long-term monitoring and numerical computations have been carried out in terms of climate conditions of Newark, UK. The system was equipped with instrumentation for inside-outside temperature radiation and energy monitoring (stored, used, and provided). Inlet flow and outlet return temperatures, flow rates, and electrical and heat energy were measured by the instrumentation method below (Table 4-10).

Table 4-10 Relative uncertainty of the components

Instrument	Location-measured parameters	Relative uncertainty (Max)
DT85 Data Taker	All connected to a DT85 Data Taker	
	Pyrometer (Incident solar radiation)	1%
	K type thermocouples (Flow temperatures)	±0.3◦C
	Sharky 775 heat meter	
Portaflow 330	Flow rate	2%

Moreover, an Electricity meter and Sharky 775 heat meter were used to measure electrical input of the compressor of heat pump and the thermal output of the heat pump, respectively. PV electricity output was monitored by A GSM energy meter. All data were measured at 10-minute intervals.

Flow and return directions were considered, however downwards measurements were not taken, because downward flow is not advised for any form of flow measurement due to the interference of cavitation and swirling etc. with readings. Uncertainty analysis is important for more reliable energetic and economic analysis. The following formula were used for this purpose:

$$Y = f(X_1, X_2 \dots X_n) \quad \textit{4-24}$$

$$\delta \mathrm{Y} = \sqrt{(\frac{\partial Y}{\partial X_1}\delta \mathrm{X}_1)^2 + (\frac{\partial Y}{\partial X_2}\delta \mathrm{X}_2)^2 + \cdots + (\frac{\partial Y}{\partial X_n}\delta \mathrm{X}_\mathrm{n})^2} \quad \textit{4-25}$$

The overall relative errors of the performance indicators, which are W_{hp} and COP are determined by using the relative errors of the water flow rate and temperature measurements on the measurements of the brine(glycol-water) entering and leaving temperatures in the PVT array, the ground horizontal and vertical array and relatively hot or cold water in the fan coil (DAIKIN, n.d.).

4.4.2. Daily monitoring of the novel PVT-GSHP

This section presents the validated results of the daily monitoring of each component and discusses their interactions. During a selected winter day, the attached components operated at a variable flow rate while energy was accumulated in the buffer vessel; at the end of the monitoring period of daily electrical efficiency, the electricity generated, daily average thermal efficiency, and thermal energy accumulated in the tank were evaluated. As explained previously, the first two days require an internal temperature of 32°C, which drops by 0.5°C every other day until 20°C is reached. Full-day monitoring of the novel components and whole heating system was conducted in winter 2016. The heat pump has on-off cycles, therefore only data with positive thermal differences in brine were considered in the analysis (thus the data collected in the time heat pump off-cycle were not considered). The short-term response is important for design considerations that limit the minimum and maximum entering fluid temperature to the PHE and ground arrays.

4.4.2.1. PVT collector and Ground array

In this section, the effective heat capacity is evaluated from data collected during a winter day over an 8-hour monitoring period around solar noon. There are two main points in this section:

1. Dynamic response of PVT collectors attached to GSHP and experimental data on collector dynamic behaviour.
2. Novel PHE attached GSHP investigation of poultry house (for agricultural sustainable heating).

The flow coming from the sources must have a temperature difference between brine out and brine in of 2-5°C when the system is balanced, five minutes after the compressor starts. A high difference indicates a low brine flow, and a low difference indicates a high brine flow. Flow rate specifications are shown in Table 4-11.

Table 4-11: Flow rate specifications

System components	*Flow rate range (L/m)*
PHE	1.5-7.5
Thermal ground array	15-27.5
Horizontal ground thermal pipe	10-15
Vertical ground thermal pipe	3.5-8.5

Because the PHE used in the PVT array has a large static pressure drop and it is mounted at a height of around 5-8 m, additional pumping power is required. The water flow in the PHE is between 1.5 L/min (0.1 m^3/hr) and 7.5 L/min (0.43 m^3/hr). Using pressure drop data for the PHE and taking into account the mounting height of the

array, the effective head the pump will need to provide is 10 m at 1.5 L/min, and 26 m at 7.5 L/min (98 – 252 kPa). Two Wilo–Star STG pumps were installed in series to provide the required head for this application.

4.4.2.2. Buffer Tank- Fan coil

At the load-side, variable-speed water pumps are utilised to allow the heat pumps to reach the energy required to run HP efficiently. The flow in the fan coil must have a suitable temperature difference for the operation (heating operation: 5-10°C) between flow temperature and the return temperature. A high difference indicates a low heating medium flow, and a low difference indicates a high heating medium flow. The fan coil unit requires a water flow rate of 34 L/min (~2 m^3/hr). Grundfos UPS2 15-50/60 pumps were added in the system to achieve flow adjustment

4.5. Results and discussions

A solar assisted ground source heat pump system and its novel solar and soil heat exchangers are theoretically and experimentally investigated on daily basis under one operating mode condition in order to achieve the contribution of each novel component to the heating system.

4.5.1. Temperature and energy from PVT and ground array

4.5.1.1. PVT Array

The ambient temperature, irradiance, and flow and return temperatures were monitored during the test day, 18/12/2016, and presented in Figure 4-7. The relationship between thermal efficiency and the reduced mean temperature is more complicated in transient test consideration. During the test, the collector flow temperature is required to be maintained close to the ambient temperature to investigate the transient PVT efficiency. However, during the test the heat pump was

operated, and the flow temperature equalled the brine return from the heat pump, which fluctuated between -1 and 1˚C under winter weather conditions (Figure 4-8).

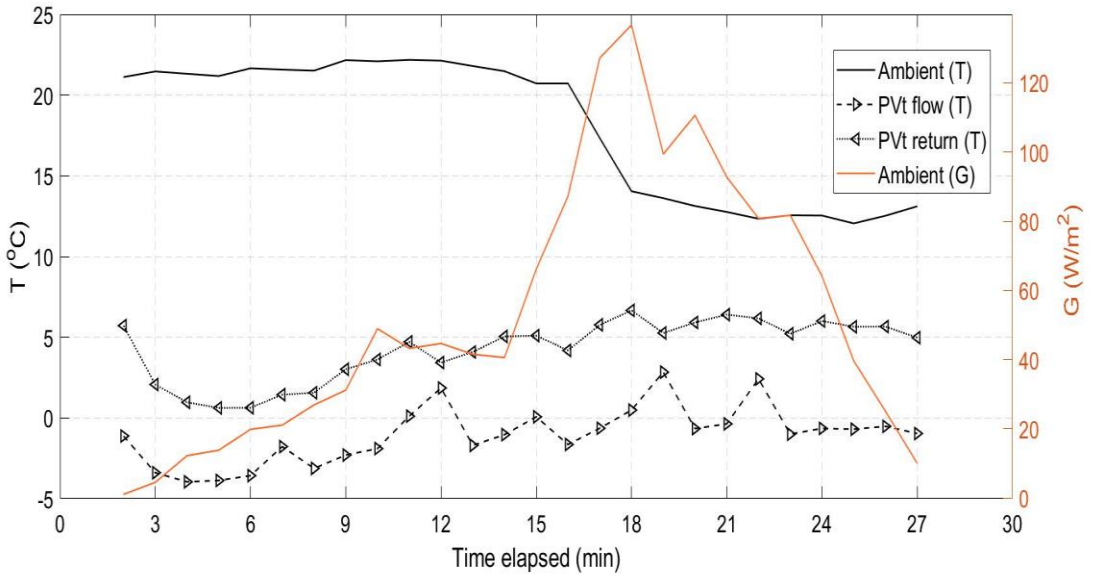


Figure 4-8: Time step change of incident irradiance/temperature on the PVT module (18/12/2016, 9.00-17.00)

Energy balance equations are used to determine PVT Thermal energy during the winter day (18/12/2016). The mean temperature of the inlet and outlet fluid in the PHE mat is determined for each time elapse measured 10-min time interval The results, presented in Figure 4-8, show that PHE achieved a maximum temperature difference of 9.6°C with 3 kW heating capacity (Figure 4-9). The thermal energy and electrical output were 10 kWh and 18 kWh respectively within 4.3 hours HP-on time between 9.00 and 17.00, 18/12/2016.

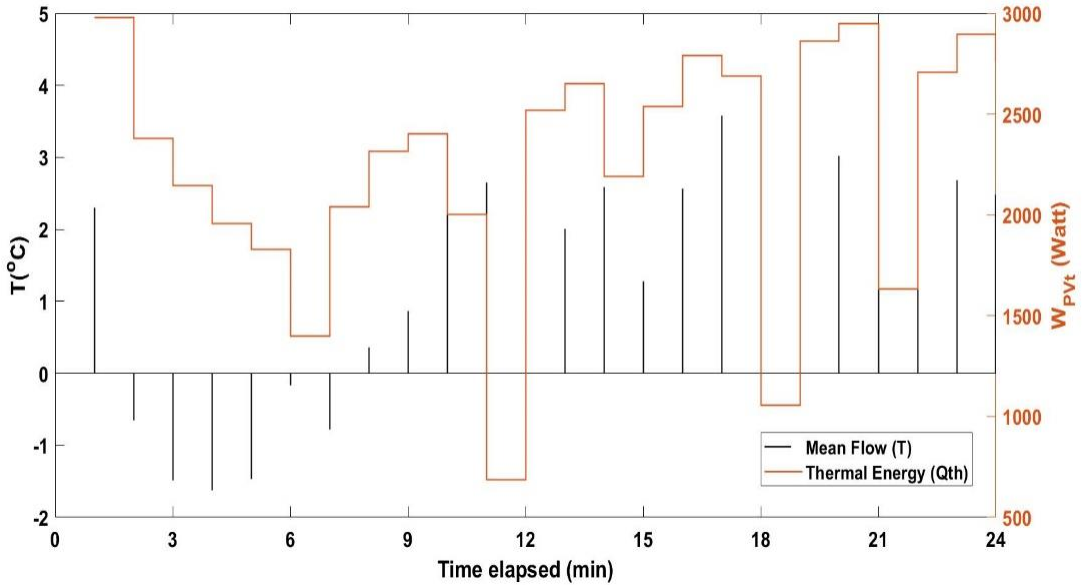


Figure 4-9: Collected Solar heat during day, 18/12/2016

4.5.1.2. Ground array

The ground array has two main components; Horizontal ground pipes and novel vertical ground heat exchangers. As stated, the flow coming from horizontal ground heat exchangers and VGHE are mixed at the main manifold. During the test, the temperatures of the fluid at the inlet of the horizontal and vertical ground heat exchangers are equalled the brine return from the heat pump, which fluctuated between -1 and 1^{o}C under winter weather conditions (Figure 4-10). The theoretical and experimental results show that the temperature of the fluid at outlet of the novel VGHE is 7.5 ^{o}C on average during peak heating season. The thermal difference at the inlet and outlet of the VGHEs highly depend on the temperature of the return fluid from the evaporator of the heat pump under the existing design condition.

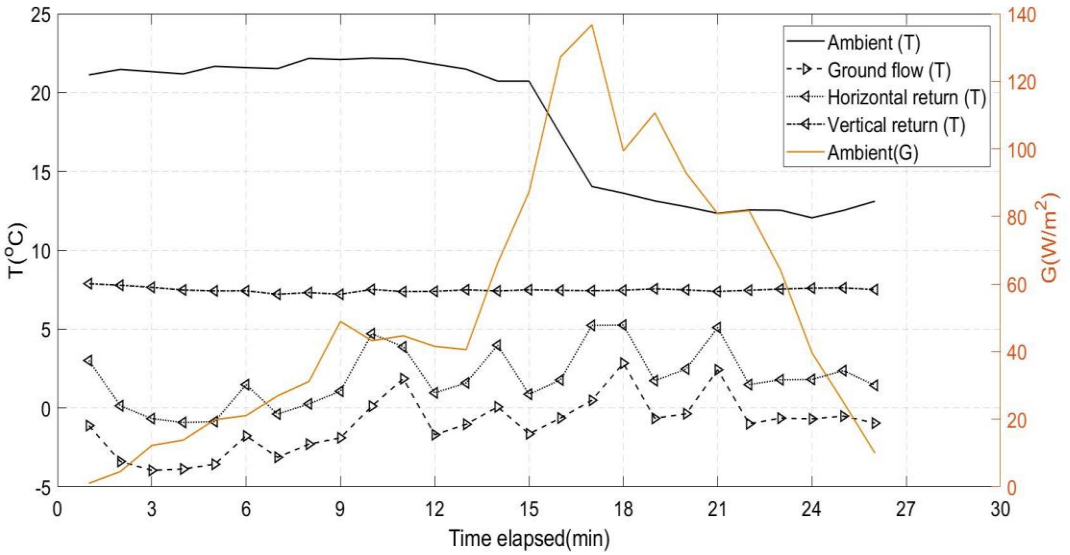


Figure 4-10 Temperature variation of the flow and return fluid at inlet and outlet of the ground array fluid , 18/12/2016

The results from testing on 18/12/2016 showed that horizontal ground array has 3.7kW heating capacity with 3.68˚C mean temperature difference. And, the VGHE array has a maximum amount of 6.7 kW heat transfer rate with 8.4˚C mean temperature (Figure 4-10 and Figure 4-11). Daily heat transfer rate of the VGHE per meter of thermal pipe is determined as 38.8 W/m on 18th December 2016 and the heat flux of the horizontal ground heat exchanger on average is 22.9 W/m^2.

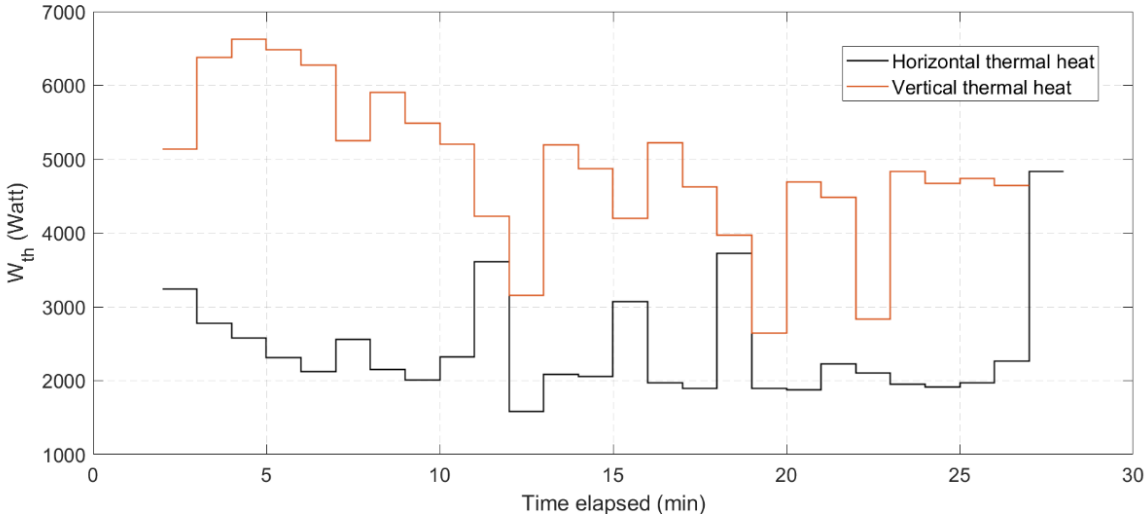


Figure 4-11: Collected ground heat during day

4.5.2. Temperature and energy from novel PVT-GSHP and COPs

Hourly assessment of the heat pump is important to observe the flow and return fluid temperature on the evaporator and condenser part of the heat pump during the different times of the day under dynamic conditions. Because of the temperature and radiation fluctuation, the COP fluctuated from 1.7 to 5.6 during the winter day. The return temperature from the PVT and Horizontal ground array causes the drop on the temperature of the flow at the HP (Figure 4-8 to 4-11).

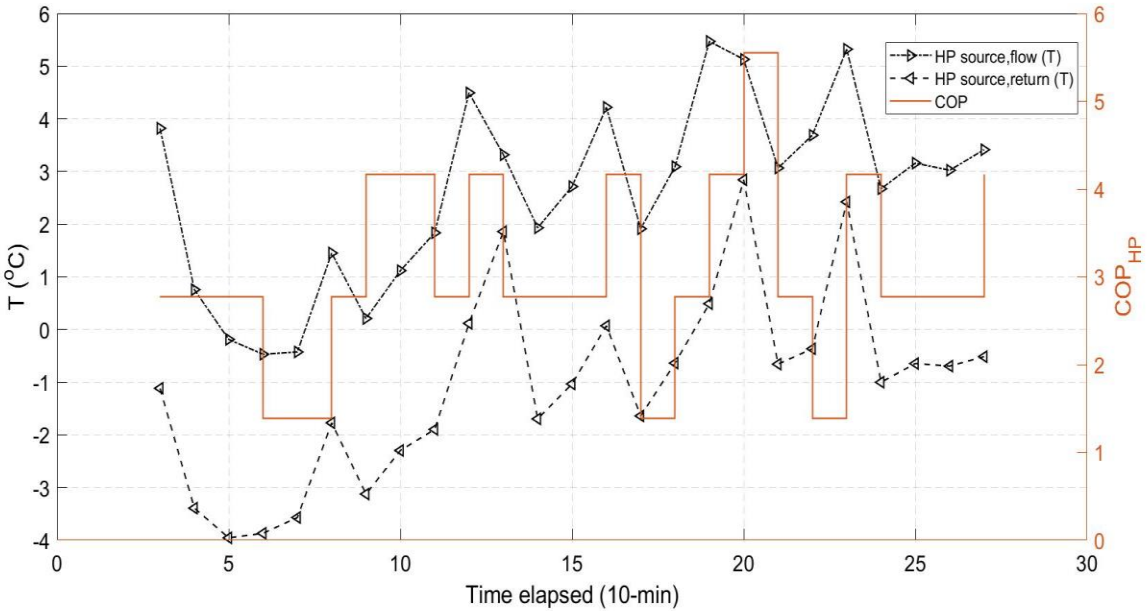


Figure 4-12: The temperatures of the fluid at inlet and outlet of the heat pump and COP variation, 18/12/2016

In order to evaluate heat pump performances, COP variations of the novel system is both systems' evaporation sides are given in Figure 4-12. Since the novel systems' evaporation sides during the daytime are exposed to almost the same condition, difference comes from the condensing temperatures under the daily basis simulations.

The main reason of the difference is the brine temperature variation during the day of the heat pump. COP reduces when the source inlet temperature (brine flow) comes with lower values into the evaporator side of the heat pump. However, considering the

fluctuation on the inside temperature of the poultry house during the 7-week production cycle, the fluctuation on the condensing temperature will affect the seasonal heating efficiency of the heating system. As it has been mentioned before, the system has a simple control mechanism with a mixed flow coming from solar PVT and ground array. More solar or soil heat can be utilized by the heat pump by adding more control mechanism like seasonal soil storage. Total daily energy consumption is 11.9 kWh from the heat pump compressor.

4.6. Summary

This chapter presents results from the simplified model and the application of the designed new PVT-GSHP heating system, retrofitted, installed, and tested in a poultry house in Kirton. The design consists of novel Polyethylene heat exchanger (PHE) underneath PV panels and novel copper vertical ground heat exchanger (VGHE) attached to the heat pump. The advantages of novel ground pipes include no need for deep digging, resulting in considerably lower installation costs than incurred using conventional vertical ground heat exchangers, like energy piles and boreholes. Moreover, these pipes require less land spaces than conventional horizontal thermal pipes and have much higher thermal efficiencies. PHEs attached underneath the PV panels also offer significant advantages over solar-based ground heat pumps, particularly in PV system applications. Although, there are some clear operational advantages to the use of these vertical ground thermal pipes over conventional heat pumps in building applications, the hybrid design was a combination of conventional horizontal pipes and novel vertical pipes in order to extract more heat from the ground and make the system more sustainable.

The efficiencies of the novel technologies attached to the heat pump to adapt the agricultural HVAC sector were investigated under the real conditions in the poultry farm on daily basis under winter conditions. The efficiencies of the novel PHE attached PVT modules and heat fluxes of the ground thermal array, in particular the novel low-cost copper VGHEs, were tested under real UK weather conditions. Novel solar and soil heat exchangers, heat pump, buffer vessel and fan coil unit have been evaluated under the weather conditions of the peak heating season in UK**.** The selected day is 18th December to evaluate the heat capacities of the novel heat exchangers, heat pump COPs and electricity generation rate of the system under the peak heating condition. The results from testing on 18/12/2016 showed that PHE has a maximum mean temperature difference of 9.6°C with 3 kW heating capacity. Horizontal ground pipe and vertical ground pipes have a maximum heating capacity of 3.7 kW with 3.68°C mean temperature and a maximum of 6.7 kW heat transfer rate with 8.4°C mean temperature difference, respectively.

It is important to highlight that the simplified model shows an acceptable match with the measured COP, although some inaccuracies exists because of the assumptions and steady simple calculations on the model. The COP fluctuate from 1.7 to 5.6 during the selected winter day under transient condition. The efficiency of the heat pump depends on the entering water temperature of the evaporative section of the heat pump, and the fan coil return temperature which flows to the condenser section. The supply heating from the gas boiler provides extra heat for the building's internal air management. This additional heat is important because of the thermal balance stored in the buffer tank between 9.00 and 17.00.

Moreover, all-day heat pump operation during the winter causes rapid cooling in the buffer tank, because of the limited heat from sources like soil and ambient air during the night. On the studied day under winter conditions, the COP of 2.54 with 4.3 running hours was observed. Some negligible effects were observed in the relationship between inflowing water temperature to the evaporator section of the heat pump and its COP, because of the method of using 10-minute measurement intervals. In addition to the system run with two operating modes, the effective soil thermal conductivity can be enhanced by adding conductive materials and solutions in order to increase the daily system efficiency.

CHAPTER 5 PERFORMANCE ASSESSMENT OF NOVEL PVT-GSHP: LONG TERM MONITORING

5.1. Introduction

The results of the field trial of the proposed heating system design applied to a poultry house are evaluated under a variety of environmental conditions to achieve annual/long-term efficiency of the heating system. As mentioned previously, poultry farm environmental management is much more challenging than residential buildings due to the great variety of required heat demand, from 22-32°C, depending on the physiological requirements of chickens by age, alongside the general complications of high stocking density and high-cost air conditioning and heating demands. As stated, there are 4 observed seasons on the poultry house management; the peak heating season (December-January, February-March), the heating season (April-May, October-November), the minimum heating season (June-July, August-September) under UK weather conditions.

As stated, the system in this study involves the PVT system cycle and ground array cycle in one operation mode with their parallel connection. Additional assessments have been done on designing methods of PVT-GSHP system based on heating dominated building. The solar and ground heat exchangers are novel components, this study presents experimental results on peak heating season and start and end of the heating seasons in one operating mode.

A full quantitative examination was undertaken for each novel component in relation to chicken peak heating season and start and end of the heating seasons, as presented in this chapter. Test investigations were performed on-site with the novel

designed system to investigate 8 hours of system-on between 9.00 and 17.00 from October to April. This chapter observes and discusses the runtime of the heat pump for peak heating season and start and end of the heating season, explaining the heat pump efficiency and runtime and assessing the impact of this on long-term performance.

Recently agreed standards for specifically testing PVT-HP are not available for dynamic conditions, nor are borehole standards available for novel vertical copper ground pipes. Therefore, dynamic on-site monitoring and performance assessment of the novel heating system was applied to observe on seasonal approach.

- The Experimental validation of novel hybrid PVT-GSHP heating system during peak heating season and start and end of the heating season.
- The potential energy saving of the novel design PVT-GSHP and copper vertical ground heat exchangers (VGHE) on agriculture under the variable of outside weather and the inside air temperature of chicken PCs during peak heating season and start and end of the heating season.
- Each of the following novel components' dynamic behaviour under seasonal dynamic conditions:
  - Solar polyethylene heat exchanger (PHE)
  - Copper vertical ground heat exchanger (VGHE)
  - Ground horizontal polyethylene heat exchanger

The field test was conducted under transient conditions with the settings on between the hours of 09.00 and 17.00 during the peak heating season from 2017 to 2019 (inclusive), to determine the overall energy output of the heating system over the

critical heating period. The minimum rate of ambient temperature was observed in January and the minimum rate of the solar irradiation was in December.

5.2. Methodology

The field test was conducted under transient conditions between the hours of 09.00 and 17.00 from 5th December to 24th January 2017 (inclusive) to determine the overall energy output of the heating system under autumn and winter weather conditions. This study focused on two PCs, one in autumn and one in winter weather conditions. 6700 chickens were placed in S1 both on 1st October and 5th December 2016 (i.e. 6700 chickens per occasion, net 13,400 chickens). The pullets (females) left at 4 weeks old and the cocks (males) left at 7 weeks.

There are many interactions affecting the system efficiency, such as inside, outside, and ground temperature, and the number of chickens. PHE harnesses the solar energy to provide a stable heat source for the heat pump evaporator and a new low-cost ground pipe array which provides the heat required when the thermal output from the PVT array is low. The brine fluid passes down and then back up each copper pipe in the ground array loop to extract relatively stable heat from the soil. Therefore, considering the existing heat extracted from available sources. Besides, existing gas boilers in the chicken house were used to back-up the HP for days requiring particularly high heating energy. The handicap of monthly simulation is to neglect the effects of extreme hourly or daily variations in the loads, which can cause unrecoverable errors in the system sizing process. Especially if a novel system is aimed to investigate, a simulation is recommended based on an hourly or smaller time step to provide the best possible design of the actual system (Spitler and Bernier

2016)**.**The system can be enhanced by using 3-valves and applying more operation conditions.

5.2.1. Chicken production cycles

Table 5-1 shows the conditions and achievements from PC 1 and PC 2. PC 1 assessment focused on the total energy production in autumn conditions. Since winter conditions are more challenging in the industry, the COP of the HP including variable pump inside the heat pump was calculated for PC 2, which was observed between 2.3 and 2.43, defined as heat pump system COP in the following annual section. This section was published in *International Low Carbon Technologies* (Gurler et al. 2018).

Table 5-1: PC 1 and PC 2 tests

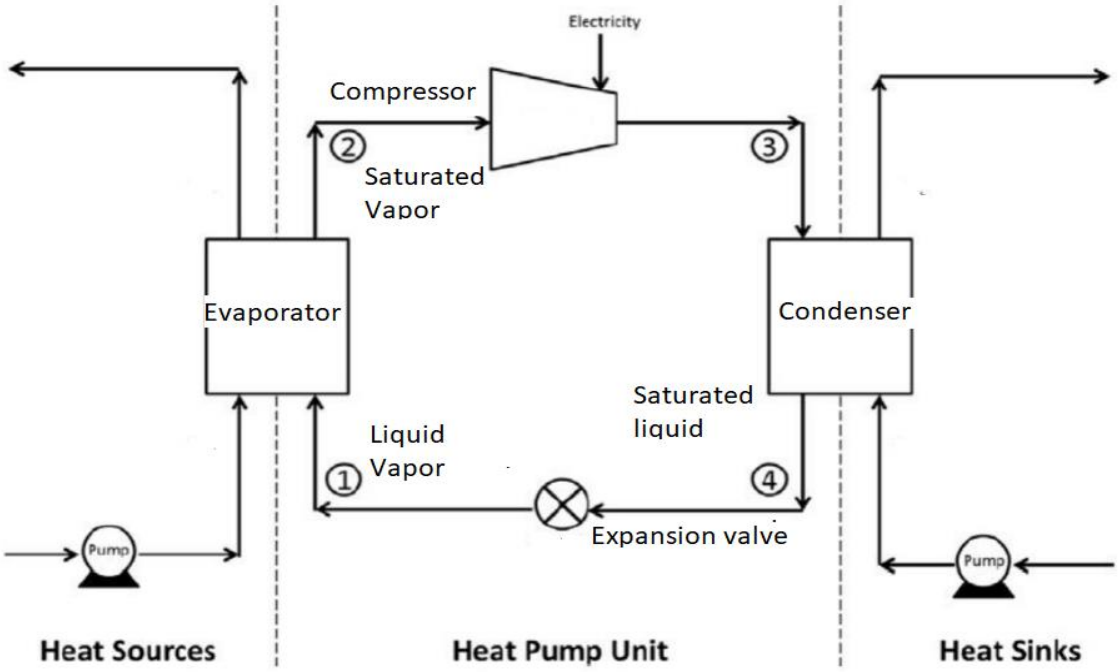
Test	*Assessment date*	*Production period*	*Chicken number*	*Chicken type*	*House*	*Heat provided (kWh)*	*Electricity generated (kWh)*
PC1	05/10/2016-24/11/2016	7-week	6700*	Cocks & pullets	S1	3733	1283**
PC2	05/12/2019-24/01/2017	7-week	6700*	Cocks & pullets	S1	2432	593***

* 3350 pullets (females) leave at 4 weeks old and 3350 cocks (males) leave at 7 weeks old

** Between 01/10/2016 and 30/11/2016

*** Between 01/12/2016 and 31/01/2017

5.2.2. PVT-GSHP system design



(a)

STAGES	Energy balance	equations
Stage 1 1→2	Q_{cold}=H2-H1	Q=m(h1-h2)
Stage 2 2→3	$-W_S$=H3-H2	-Ws=m(h3-h2)
Stage 3 3→4	Q_{hot}=H4-H3	Q=m(h4-h3)
Stage 4 4→ 1	H1-H4=0	H4=H1

(b)

Figure 5-1 (a) Heat pump stages (b) Heat pump working principle

The Figure 5-1 present the main stages of the heat pump working and energy balance equation of its components. To calculate the heat pump COP some assumptions and simplifications are made. Vapor compression cycle has four main components and stages to calculate the efficient. The first stage of the heat pump is to isobaric heat change process on the evaporator side of the heat pump. The second stage is

Isentropic compression in the compressor of the heat pump. And the third is isobaric heat change on the condensate of the heat pump. The final ,fourth stage is the adiabatic expansion in the expansion valve.

The system in this study involves the PVT system cycle and ground array cycle in one operation mode with their parallel connection. Design of the GSHP systems requires life cycle simulation because of the unbalanced thermal energy stored in the soil. The novel PVT-GSHP drawing is presented in Figure 5-2 (a more detailed version can be seen in Appendix A)

PHE and ground pipe arrays (novel vertical and commercial horizontal) and heat transfer rate of each component; the whole heating system's thermal and electrical energy output and its operating time; and the energy needs of the chicken farm. The indoor temperature is maintained with heat pump and gas boilers during the peak heating season. Various control strategies were performed in MATLAB to optimize the system on heating mode during whole year of operation considering hourly consumption profiles. The solar and ground heat exchangers are novel components, this study presents hourly validation of these components in MATLAB on daily basis in one operating mode. The next step of this study will be developing a control strategy for the novel hybrid system and optimizing each operating mode for each season., but this study focuses on the efficiency enhancement when the novel PHE and VGHE heat exchangers are added to the heating system and annual run with mixed in the heat exchanger coils on the evaporator side of the heat pump (Figure 5-2).

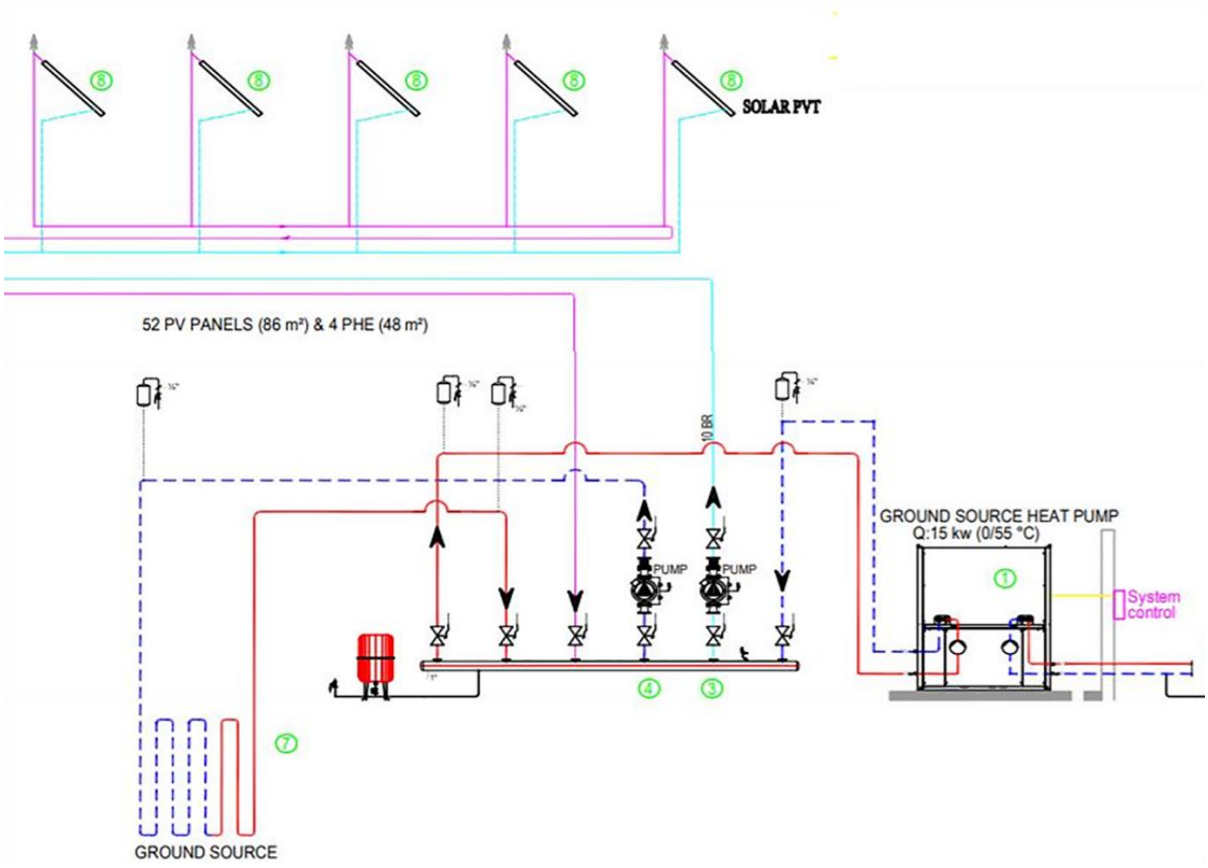


Figure 5-2 PVT-GSHP heating system drawing (APPENDIX A)

The results from the annual heating loads of the poultry farm in UK shows that the poultry houses in UK are heating dominated buildings. Therefore, the thermal responses on the chicken houses are strongly recommended to be determined based on the model of the ground array on hourly basis in the future of the study.

Additional assessments have been done on designing methods of PVT-GSHP system based on heating dominated building. The heating system is run for each heating season with one operation mode, which is the mixture of the outlet PHE flow coming from the roof and the flow coming from the ground array (Figure 5-3).

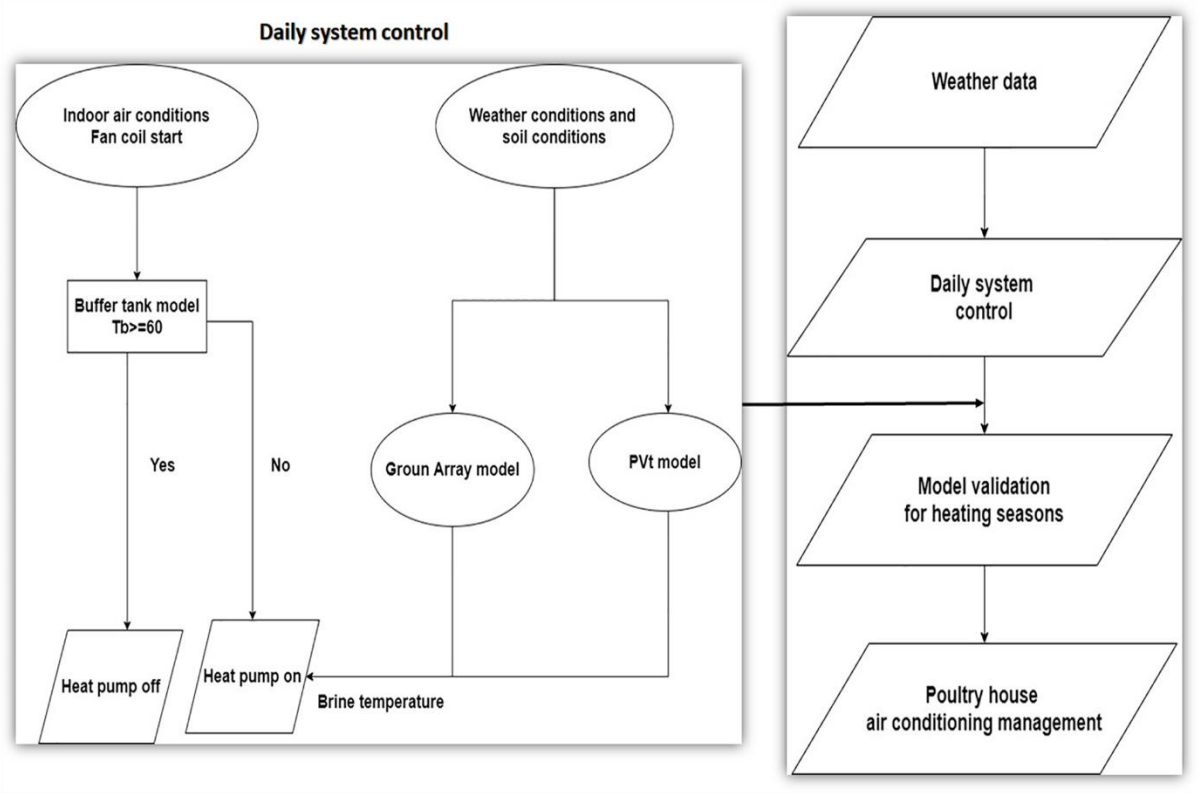


Figure 5-3 Basic schematic diagram of System control: One operation mode

The standard for experimental investigation of the collector dynamic-response requires specific conditions. But this can be costly and time-consuming to perform outdoors**.** Based on the test standards. However, because this system is a combined system with the heat pump, it is challenging to provide the required standard test condition for the PVT dynamic investigation such as constant flow temperature, which requires to be close to the ambient temperature, within different radiation level.

5.2.3. Experimental mathematical procedure

The temperature rise causes the performance drop on the PV panels, the difference between the array and reference temperatures is considered to calculate the electrical efficiency as shown in Equation 5-1 and Equation 5-2.

$$\eta_{el} = \frac{P_e}{G \cdot A} \quad \textit{5-1}$$

$$\eta_{TH} = \frac{Q_{TH}}{G \cdot A} \quad \textit{5-2}$$

Where, A_{eff} (m^2) is the effective temperature of the PV Cover plate temperature is estimated and correlated by the heat removal factor equation. Q_{loss} is the thermal energy losses and obtained using:

$$\mathrm{Q_{loss}} = \mathrm{U_{loss}A(T_{cell} - T_{amb})} \quad \textit{5-3}$$

Where, is the ambient temperature in Kirton, UK. Useful energy inside the PV Panel ($Q_{useful,pv}$):

$$Q_{useful,pv} = Q_{solar} - Q_{loss} - Q_{power} \quad \textbf{\textit{5-4}}$$

The rate of heat extracted by the PHE, is also calculated from the following equation:

$$Q_{useful,pv} = Q_{solar} - Q_{loss} - Q_{power} \quad \textbf{\textit{5-5}}$$

Where,(°C) and (°C) is the inlet and outlet temperature of the brine circulating through the PHE. (J/kg·°C) is the thermal capacity of brine in PHE, (kg/s) is the mass flow rate of the brine in PHE.

$$\dot{m}_b c_p\left(T_{f-o} - T_{f-i}\right) = \eta_o AG - c_1 A(T_m - T_a) - c_2 A(T_m - T_a)^2 - C\frac{dT_m}{dt} \quad 5\text{-}6$$

$$\dot{m}_b c_p\left(T_{f-o} - T_{f-i}\right) = \eta_{(o)} AG - UA\left(T_{f-o} - T_{f-i}\right) - C\frac{dT_m}{dt} \quad 5\text{-}7$$

Where, C is the effective heat capacity, a parameter used to characterize the transient performance of solar collectors. And is the measured flow out and flow in temperatures respectively.

The rate of heat extracted by the ground array, is calculated using the following equation.

$$Q_{useful,ground} = \dot{m}_{ground} \times C_{ground} \times \left(T_{ground,,out} - T_{ground,in}\right) \quad 5\text{-}8$$

Where (°C) and (°C) are the inlet and outlet temperature of the brine circulating through the ground array. (J/kg·°C) is the thermal capacity of brine in the ground array, and (kg/s) is the mass flow rate of the brine in the ground array.

5.3. Results and discussions

5.3.1. Performance evaluation of the PVT-GSHP system during peak heating seasons

The minimum rate of ambient temperature was observed in January and the minimum rate of the solar irradiation was in December. Each weather conditions are hourly validated under peak heating season and start and end of the heating season with 7-

week production cycles. For these seasons, since the fan coil return temperature will be assumed the same for all days during the PC of the chickens. Therefore, the condensing temperature will not affect the system COP during these heating seasons. However, for the annual and long-term approach. the condensing temperature will be affected by the heat load of the building during start and the end of the heating seasons. In this chapter, therefore, the main parameter affecting the system COP is the evaporating temperature on the evaporator side of the heat pump.

The temperature variation of the circulating fluid in the ground array over the critical heating period in 2017 is given in Figure 5-4. The blue and orange dashes on the figure are the inlet temperatures of the ground array and the outlet temperature from the horizontal GHE, respectively. On the other hand, the red continues line is the temperature of the return flow from the VGHE. Although the return temperature from the horizontal ground pipe varies from 0.97 °C to 17.6 °C depending upon the shallow temperature during the day based on the measurements of the 10- minute intervals during the time of the heat pump is on, the temperature of the return flow VGHE is stable and consistent and varied from 4.7 °C to 9 °C.

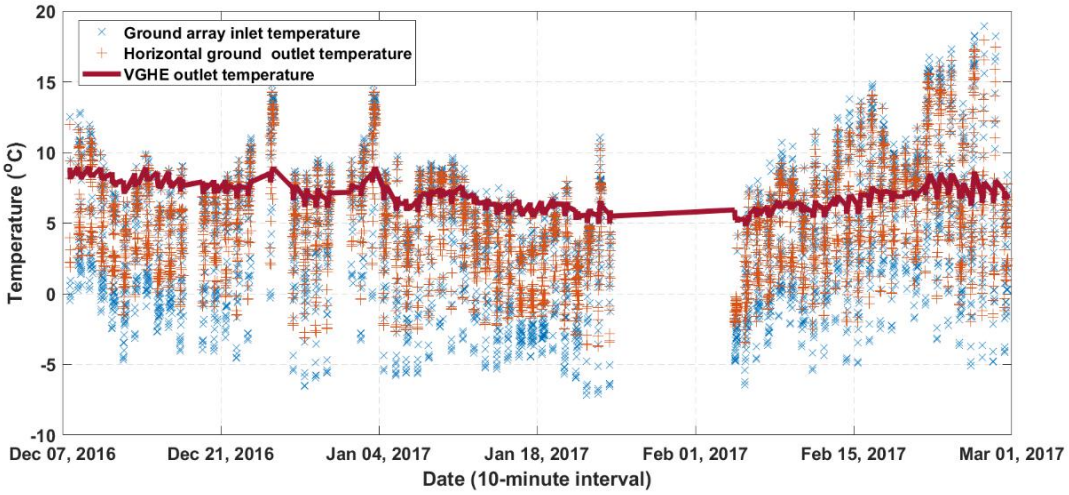


Figure 5-4 The temperature variation of the fluid at inlet and outlet of the Ground Array (Monitored data)

During the peak heating season, the seasonal average heat transfer rate of each hybrid tube of novel VGHEs equals 52 W according to the simplifications adopted, which yields 21 W/m per unit GHE length heat extraction at 2.5 m length and 5m soil depth in continues operation.

5.3.1.1. PVT collector

Figure 5-5 shows the PHE flow and return temperature in the innovative heating system on the PVT collector during critical heating season under UK weather condition. The minimum return temperature of PHE is -3.7 °C on January. within 10-minute interval. During the peak heating season, the average heat transfer rate of 46 m^2 PHE equals to 2.3 kW, with corresponding heat fluxes of 50 W/m^2 per unit PHE area in continues operation.

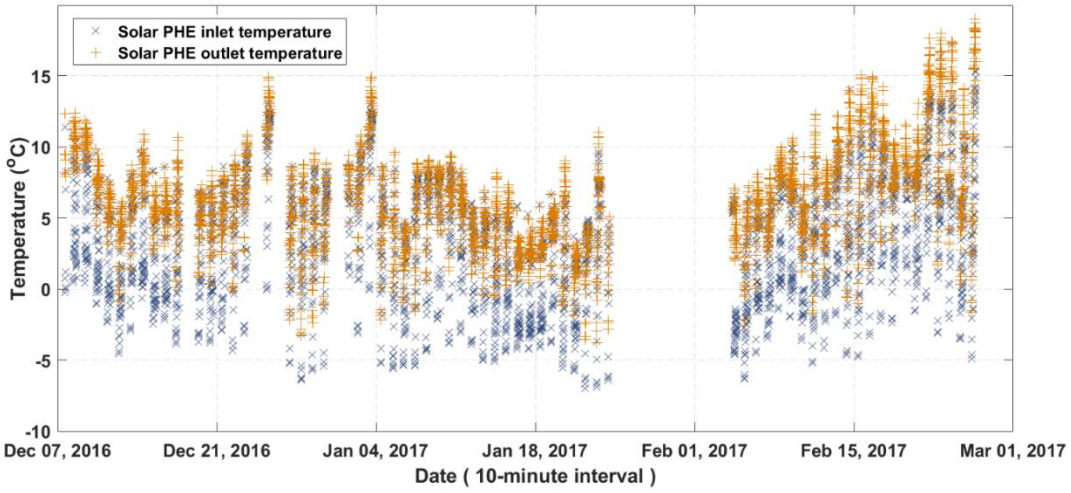


Figure 5-5: The temperature variation of the fluid at inlet and outlet of the Solar PVT Array (Monitored data)

Figure 5-6 indicates daily electricity output of the PV and used electrical energy by HP during the second production cycle. PV array achieved 593 kWh power supply over the winter period including cleaning time of poultry house, with a corresponding electrical saving 58%.

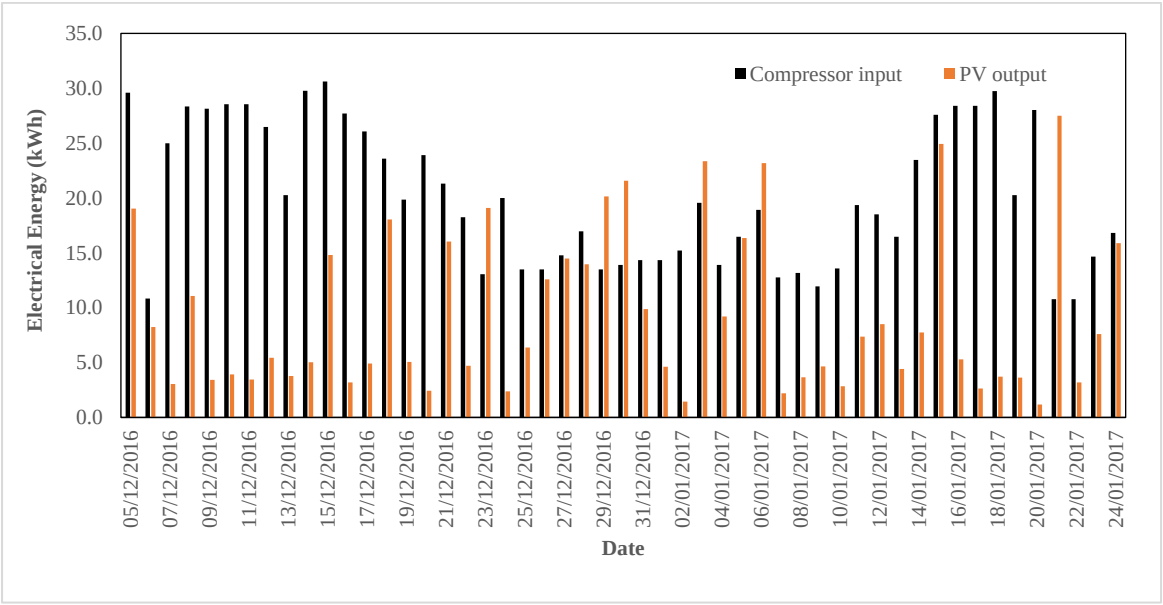


Figure 5-6: Electrical HP input and PV output energy

5.3.1.2. Heat pump-buffer tank-fan coil

Figure 5-7 presents the fan coil inlet flow and outlet return temperature at fan coil. Moreover, the results from the buffer tank bottom, middle and top temperature are presented in Figure 5-8. The results show that there is fluid of a constant inlet temperature 60^{o}C of temperature to the fan coil and fluid of outlet temperature between 45 and 49^{o}C from the fan coil. The dashes between 30 and 45 ^{o}C in Figure 5-7 and Figure 5-8 present the temperature of the buffer tank and fan coil when the system is off during night-time. Chickens were not inside from 25th January to 4th February during winter peak season**.**

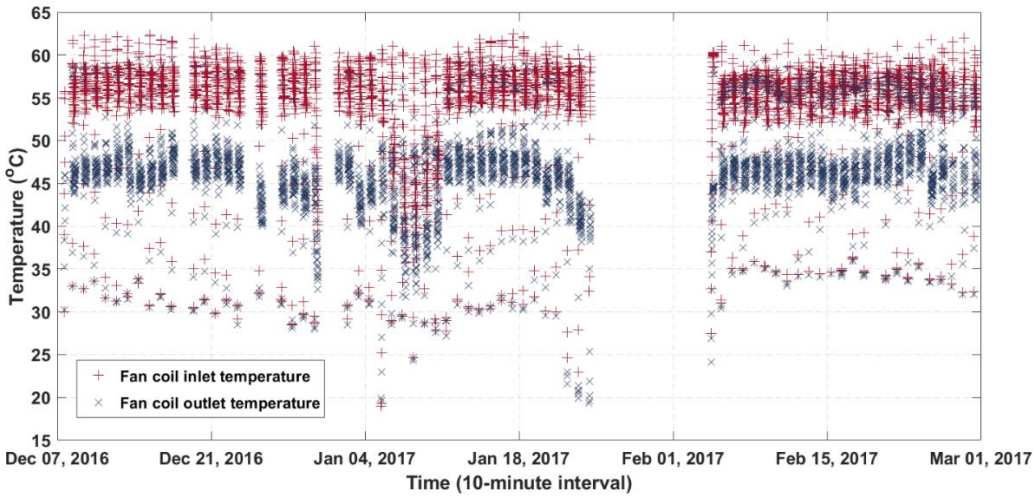


Figure 5-7 The temperature variation of the fluid at inlet and outlet of the Fan coil unit (Monitored data)

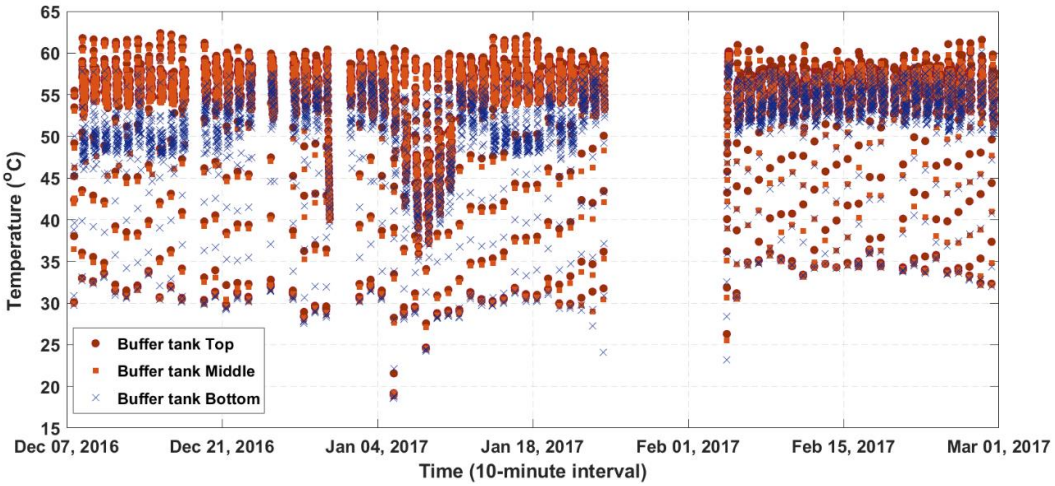


Figure 5-8 The temperature variation of the fluid at inlet and outlet of the top, middle and bottom of the Buffer Vessel (Monitored data)

The heat output of the HP is presented in Figure 5-9 during the cycle in winter weather conditions. Although many interactions were affecting the heat output and COP of the heat pump (COP), the main parameter was the ambient weather condition in and out of the poultry house as expected. The mean weekly COP ranged between 2.3 and 2.43 during the PC1, as shown in Figure 5-9. A Maximum daily energy 74 kWh was reached on 15th January, while the minimum mean weekly COP fluctuated between 2.3 and 2.32 on the days when the mean outdoor air temperature dropped to 5°C during the daytime.

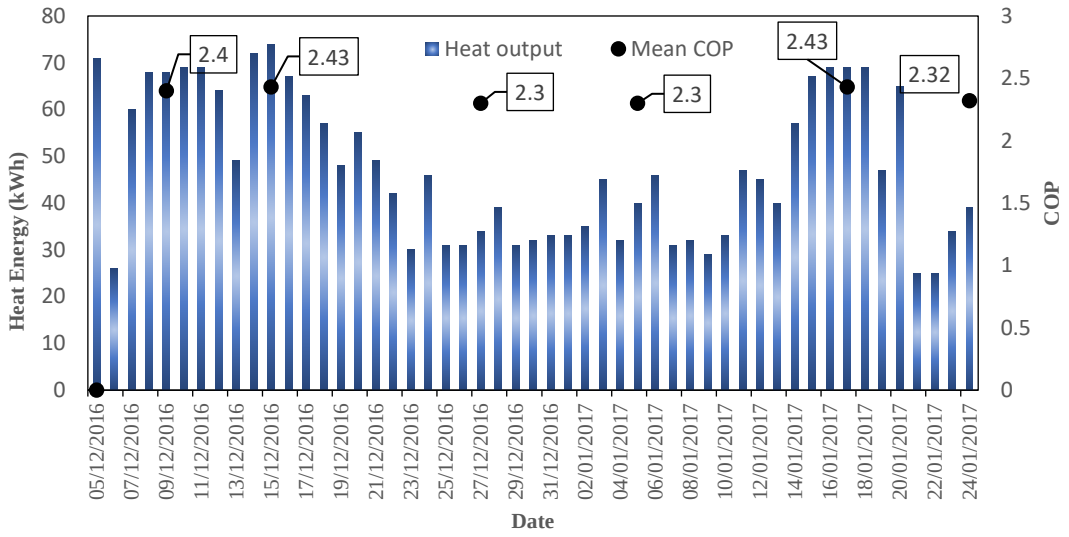


Figure 5-9: Heat output of the heat pump and COP_{sys} mean

5.3.2. Performance evaluation of the PVT-GSHP system during the heating season (middle of the heating)

5.3.2.1. PVT and Ground array

The results of the experimental study revealed that 6165 kWh heat and 1876 kWh electricity energy were provided under autumn/winter weather conditions. Figure 5-10 shows daily generated electricity and provided heat output for the autumn PC. The maximum daily heat output of 200 kWh energy was reached on 6th October while the minimum was 13 kWh on 20th November in the autumn weather condition. A PV array power supply of 984 kWh was achieved over the period in the PC. There followed a two-week cleaning period, as mentioned previously, with no chicken or heat requirement in S1. However, PV systems continued to generate electricity during this time. Therefore, it is worth noting that 1283 kWh total electrical energy was produced in the autumn cycle, including the cleaning period from 1st October to 30th November, in Kirton. Because the poultry farm is in the countryside, wifi and therefore automatic control of the heat pump is challenging. For future consideration, a special system design is strongly recommended for the agriculture industry, like relatively high

scheduled hours, and seasonal storage from PVT to ground by the attachment of a three-way valve between PHE and the ground array

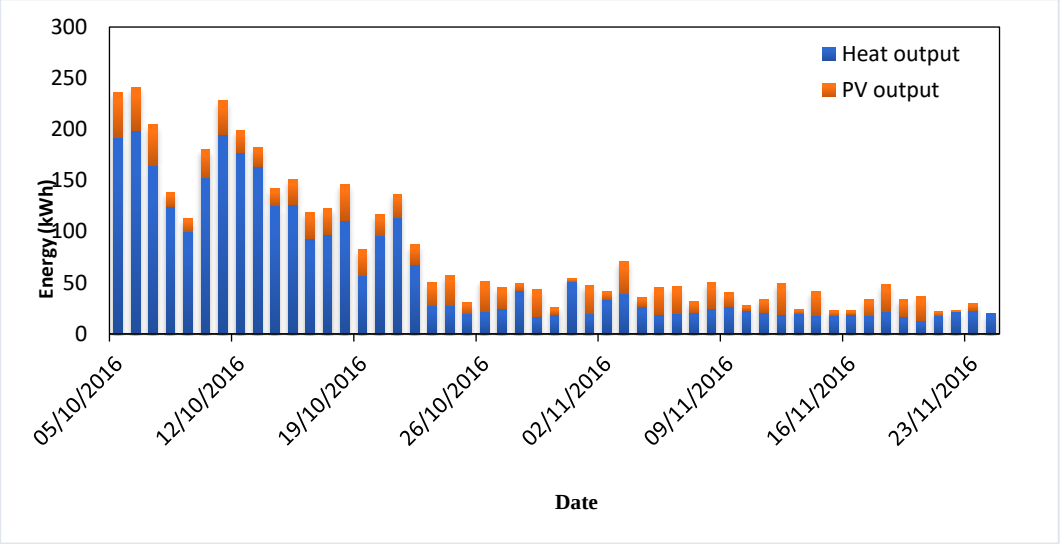


Figure 5-10: Electricity and heat output

Figure 5-10 also displays the daily electricity output of the PV and electrical energy used by the HP during the second PC. The PV array achieved 593 kWh power supply over the winter period, including the cleaning time of S1 in December, with a corresponding electrical saving of 58%.

5.3.2.2. Heat pump-buffer tank-fan coil

The analysis included the daily PV electrical output and heat pump performance. The results of the experimental study reveal that 3733 kWh and 2432 kWh heat energy were supplied in the 7-week growth cycle of studied chicken conditions. The COP of the HP was calculated for the winter period and observed to be between 2.3 and 2.43. The PV system was able to generate 1876 kWh electrical energy with the potential to meet HP energy needs in the most critical energy period.

Considering heating back-up requirement, the new heating system in the chicken house required two 65-70 kW heater units, similar to the existing gas boiler system.

However, PVT and ground array size were limited due to roof parameters, including its load-carrying capacity and available surface area. Therefore, 15 kW HP was selected, considering the existing heat extracted from available sources. Besides, existing gas boilers in the chicken house were used to back-up the HP for days requiring particularly high heating energy.

5.4. Further experimental observation of the PVT-GSHP

As an initial investigation, the heat pump was scheduled for full-time heat pump and fan coil power-on settings. The system was designed as a PHE and ground array mixed brine circuit for all-time run (Day and night). However, a sudden temperature drop was observed in the temperature of the return brine during night-time. The temperature at the buffer tank considerably decreased, and the stored heat was unbalanced after a while of night-time, when the heat pump was running all the time. Therefore, it was scheduled to switch on for 8 hours between 9.00 and17.00 hours.

The heat pump outputs of autumn PCs are presented in Figure 5-11, recorded in 2016 and 2017. Case 1 presents the experimental heat pump results from the scheduled time (on-power) from 9.00 to 17.00 during October and November in 2017, which equals to 3.2 daily running hours in October. Case 2 presents results from full time power-on during October and November 2016, which equals to 5.25 daily running hours on October. The heat pump outputs were 3733 kWh and 1786 kWh in 2016 and 2017, respectively. Although the heat output from 2016 covers more heating load of the building, the heat balance of the system decreased the system efficiency considerably. It is important to take into account the efficiency degradation of the heat pumps in case of partialisation of the loads described in the European EN14825 standards(DIN EN 14825 2019). These results highlight that in addition to the outdoor

air temperature fluctuations, the high temperature requirement of the chicken buildings and their inside air temperature fluctuations during PC causes more system heat balance challenges. Although the novel components have good results based on the daily experiments, the system requires a smart flow management to achieve more efficient long-term heat pump performance.

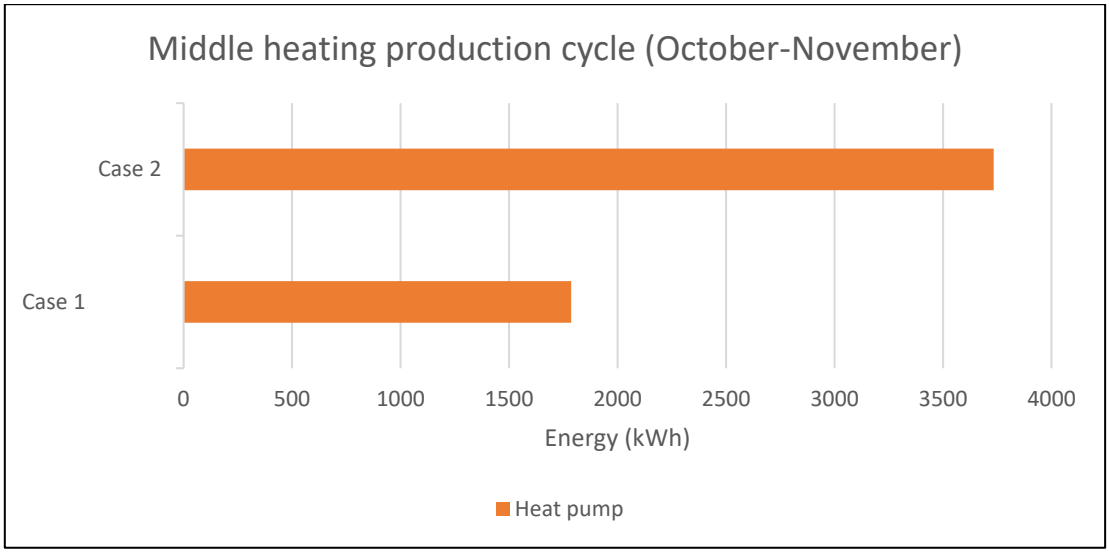


Figure 5-11: Heat pump output during middle of heating season

The solar and ground heat exchangers are novel components, and this study presents hourly validation of these components in MATLAB on daily basis in one operating mode. The next step of this study will be developing a control strategy for the novel hybrid system and optimizing each operating mode for each season to increase the running hours., but this study focuses on the efficiency enhancement when the novel PHE and VGHE heat exchangers are added to the heating system and annual run with mixed in the heat exchanger coils on the evaporator side of the heat pump.

5.5. Summary

The system aims to use novel solar and ground heat exchangers to increase the energy efficiency of the heating system. The system can be improved with various

system's operation modes, but this chapter focuses on the efficiency enhancement when the novel PHE and VGHE heat exchangers are added to the heating system. The heat pump run with mixed flow in the heat exchanger coils on the evaporator side of the heat pump.

This section discussed the results of the innovative heating system based on solar PV assisted HP system installed in a UK poultry farm, analysing the test results of the system field trailed for two PCs in the heating season. In conclusion, there are three main findings from this research. The first one is that the system achieved 6165 kWh of the heating energy requirement of the selected house (S1) during the system operation for winter conditions. Since the system is designed to meet only part of the load, the design could cut down the fair amount of the gas usage in the winter cycle and thus reduce the carbon footprint of the poultry farm, in addition to conventional energy bills. The second is that innovative PHE and thermal pipes were able to provide sufficient heat at the evaporator section of HP energy at the rated COP. Finally, the generated electricity has the potential to meet HP's energy needs in the most energy critical period. The future study would be developing a control strategy for the novel hybrid system and optimizing each operating mode for each season and PVT -ground storage.

The observation from the seasonal site investigation indicates that the heat demand of the building varied, depending upon the time of day, season, and PC. The results of the seasonal experimental study revealed that 6165 kWh heat and 1876 kWh electricity energy was provided under autumn/winter weather conditions. This study focused on two PCs, one in autumn and one in winter weather conditions. 6700 chickens were placed in S1 both on 1st October and 5th December 2016. The pullets

(females) left at 4 weeks old and the cocks (males) left at 7 weeks. There were many interactions affecting the system efficiency, such as inside, outside, and ground temperature, and the number of chickens. The minimum daily COP of the whole system was 2.05 on the monitored days throughout the peak heating season. Daily average heat pump minimum capacity was 10.61 kW. When considering the total electrical and thermal energy monitored for whole week during the peak heating season, the mean weekly COP ranged between 2.3 and 2.43 through the poultry production cycle (PC1).

The analysis from October 2016 for the autumn assessment, showed that the running time of the heat pump was relatively high, with the time scheduled as full-day switched on. The heat pump can theoretically provider heat demand of both PC 4 and 5 during the time from May to August. However, because of the settings and working hours on the heat pump, the field measurements shows lower heat capacity than the future design with two or more operating mode with dual source. The whole system efficiency can be increased by utilising the storage capacity and the seasonal smart storage method.

The errors in the real time measurement were determined considering errors in the measurement devices. Despite the inaccuracies caused by assumptions in the simplified model, it was found to be adequate for design purposes. Total daily heat energy received was 55 kWh during winter PC1. Total heat transfer of novel VGHE per meter of borehole is determined as 38.8 W/m on poultry farm under the weather and ground conditions in Kirton, UK.

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CHAPTER 6 TECHNO-ECONOMIC AND ENVIRONMENTAL ASSESSMENT OF NOVEL PVT-GSHP

6.1. Introduction

A comprehensive literature review was carried out and it was found that recent studies have worked on various types of heat pumps, including air, water, vertical or horizontal based GSHPs, and PV roof or land arrays. However, in addition to being relatively expensive to manufacture and install, borehole ground thermal pipes as used in borehole heat exchangers have certain drawbacks, especially when heat pumps run inefficiently during the start and end of the heating season (e.g. if they are oversized, a common mistake in the industry) (Technology Fact Sheet 2002).

The results of the field trial of the proposed heating system design applied to a poultry house are evaluated under a variety of environmental conditions to achieve annual/long-term efficiency of the heating system. As mentioned previously, poultry farm environmental management is much more challenging than residential buildings due to the great variety of required heat demand, from 22-32˚C, depending on the physiological requirements of chickens by age, alongside the general complications of high stocking density and high-cost air conditioning and heating demands. As mentioned, there are 4 observed seasons on the poultry house management; the peak heating season (December-January, February-March), heating season (April-May, October-November), the minimum heating season (June-July, August-September) under UK weather conditions.

A full quantitative examination was undertaken for each novel component in relation to chicken PCs and season, as presented in this chapter. Test investigations were

performed on-site with the novel designed system to investigate 8 hours of power-on between 9.00 and 17.00 from October to April, and variable condition between 8-12 hours in summer, depending on the sunset time from May to September. This chapter observes and discusses the runtime of the heat pump for the whole year, explaining the heat pump efficiency and runtime and assessing the impact of this on long-term performance.

Moreover, this section assesses the economic and environmental feasibility of the novel thermal regulation systems. Technical economic and environmental analyses of system performance are presented in the following chapter, in sections 6.2 and 6.3, respectively. Although there is no specific mechanical cooling in poultry farms in the UK, the farmers reported that the chickens are noticeably uncomfortable during the summer, particularly under the weather conditions during the end of the heating season PC in June and July. The evaporative cooling system is a common technology and has a valuable impact on chicken welfare, thus its feasibility for the design was evaluated in section 7.2 Moreover, the novel heat exchangers (VGHE) and PV panels (PHE) are assessed to determine their economic and environmental feasibility.

To show effect of the using novel solar and ground heat exchangers and their control methodology, performances of the conventional solar based ground heat pump units and novel PVT-GSHP are analysed and compared.

6.2. Methodology

6.2.1. System design and measurements

The field test was conducted under transient conditions with the settings on between the hours of 09.00 and 17.00 during the middle of the heating season and from 7.00 to 19.00 during the end of the heating season from 2017 to 2019 (inclusive), to

determine the overall energy output of the heating system over a long-term period. The minimum rate of ambient temperature was observed in January and the minimum rate of the solar irradiation was in December. The experiments aimed to characterise PVT collectors of various designs in terms of their dynamic behaviour, to provide reliable data for agricultural HVAC system designers over a wide range of time-varying operating conditions. Novel PVT-GSHP Annual performance assessment. Based on the observation, following an exhaustive analysis, the most representative periods for the heat pump operation seasons based on the outside and inside weather conditions were extracted for the poultry houses from the total monitoring period: the peak heating season (December-January, February-March), the heating season (April-May, October-November), the minimum heating season (June-July, August-September) under UK weather conditions. On-site experimental investigation of the PVT-GSHP under the environmental conditions of different PCs in S1 addressed the main annual chicken cycles: Production Cycles; PC1 (December- January); PC2 (February-March); PC3 (April- May); PC4 (June- July); PC5 (August- September); PC6 (October-November).

The seasonal performance factor (SPF) is a measure of the operating performance of an heat pump heating system over a year. There are four types of SPF's defined by Nordman *et al*. (Nordman 2012) The current system design focuses on part load of design of novel hybrid heating system and evaluates its possible use in agriculture industry. Therefore, this study uses the basic classification of the SPF , SPF1, which is similar to Seasonal COP (SCOP) and involves only the heat pump unit itself.

Seasonal coefficient of performance (SCOP) is determined by usable energy and consumed energy during the measured period.

$$SPF_1 = \frac{Q_{HP,season}}{W_{HP,sseason}} \quad 6\text{-}1$$

6.2.2. Economic and environmental investigation of PVT-GSHP

The economic and environmental assessment aims to establish the future feasibility of the proposed thermal regulation systems for agricultural buildings under real conditions in comparison to the existing gas boiler system in a poultry farm. Economic and environmental reliability and future feasibility of the novel solar PHE and copper VGHE must consider projected changes in global energy resources, conversion techniques, and associated costs

6.2.2.1. Economic Assessment

Net Present Value (NPV) and simple payback period (SPBP) are used to assess the economic performance of the novel PVT-GSHP heating system. The main parameters are PV export ratio and renewable heat incentive (RHI) for renewable energy generation, in addition to net present cost, equivalent uniform annual payback, and simple payback period, which are the most common indicators of project feasibility and reliability.

- Net present value (NPV)

Net present value (NPV) is a good indicator of project value, widely used to assess feasibility as an initial step in proposed projects. The total cost of a project over a specified period is the net present cost, which considers the time value of money. The determination of net present cost (NPC) is presented in Equation 6-2: Unlike NPV method, the results NPC analysis will be positive. The lowest value of NPC will be considered for the optimum system selection.

$$NPC = \sum_{t=0}^{N} \frac{\mathrm{AA_{TC}}}{(1+i_r)^n} + I_{cc} \qquad 6\text{-}2$$

Where AA_{TC} is the adjusted annual total costs (£), i_r is the discount rate, n is the number of the years, and I_{cc} is the initial capital cost (£).

The adjusted annual total cost (AA_{TC}) is calculated using Equation 6-3:

$$AA_{TC} = A_{TC}\left(1+i_f\right)^n \qquad 6\text{-}3$$

Where A_{TC} is the non-adjusted annual total costs (£), i_f is the inflation rate and n is the number of years.

The guideline interest rate is 7% in most engineering analysis. And, economic lifetime used in the analysis is 20 years because heat pumps are typically expected to last 20 years (Sommerfeldt and Madani 2019). Moreover, in this study, two more parameters are considered in addition to the common feasibility calculations, in order to consider the system's sustainability and environmentally friendly characteristics: export-tariffs, and RHI. Export tariffs include state support provided by the UK government for renewable electricity generation. The generation tariff is 0.0439 £/kWh, and the export tariff is 0.0491 £/kWh for PV modules(Waldau 2016). RHI is financial support from the UK government to encourage renewable heat generation, which is 0.0895 £/kWh for the heat pump(Department of Energy&Climate Change 2016).

The analysis undertaken can be given as follows:

$$SPBP = \frac{I_{cc}}{Annual\ savings + Goverment\ \text{revenue}} \qquad 6\text{-}4$$

Where I_{cc} is the initial capital cost of the system (£). Annual savings are calculated by subtracting the annual total cost (A_{TC}) of the gas boiler system from the cost using the proposed heat pump heating unit, derived from long-term testing data under real site conditions. The PVT-GSHP is compared with the base case system, which is a commercially available system for poultry farms.

6.2.2.2. Environmental assessment

The main advantage of the heat pump is considerable gas emission saving compared with gas boiler, although the electricity use increases by using heat pump. In this nov5 5el hybrid PVT-GSHP has PV panels o eliminate electrical usage and covers all HP electricity consumption. Environmental assessment is applied by the multiplication of the annual natural gas and electrical demand by their respective emission factors.

Electrical emission factor can be varied, the assumptions used in the environmental assessment are the same as those presented in the economic assessment in previous section. The emission factors of natural gas and electricity are taken from the latest UK government announcement (Department for Business Energy & Strategy Industrial 2018). The emission factors used are presented below;

Gas emission factor: 0.21. kg.CO_2 per kWh thermal energy

Electricity emission factor: 0.35 kg.CO_2 per kWh electricity energy

6.3. Results and discussion

6.3.1. Temperature variations and energy from PVT -GSHP

The minimum and maximum average daily inlet flow temperatures of solar and ground array were 12.7°C and -2.8°C during January and July, respectively. The uncertainty on the annual energy consumption obtained with adjustment factor has been decreased from around 7 % down to 3%.

6.3.1.1. PVT Array

The experiments aimed to characterise not only novel solar PHE collectors but also PVT collectors in general of various designs in terms of their dynamic behaviour, to provide reliable data for agricultural HVAC system designers over a wide range of time-varying operating conditions. The electricity production of the PVT panels is also considered, in accordance with the electricity consumption of the heat pump. Figure 6-1 shows the daily output from the PV with the maximum value of 98 kWh in June from December 2016 to November 2017. The annual electrical output from PV panels are 11.99 MWh throughout the year.

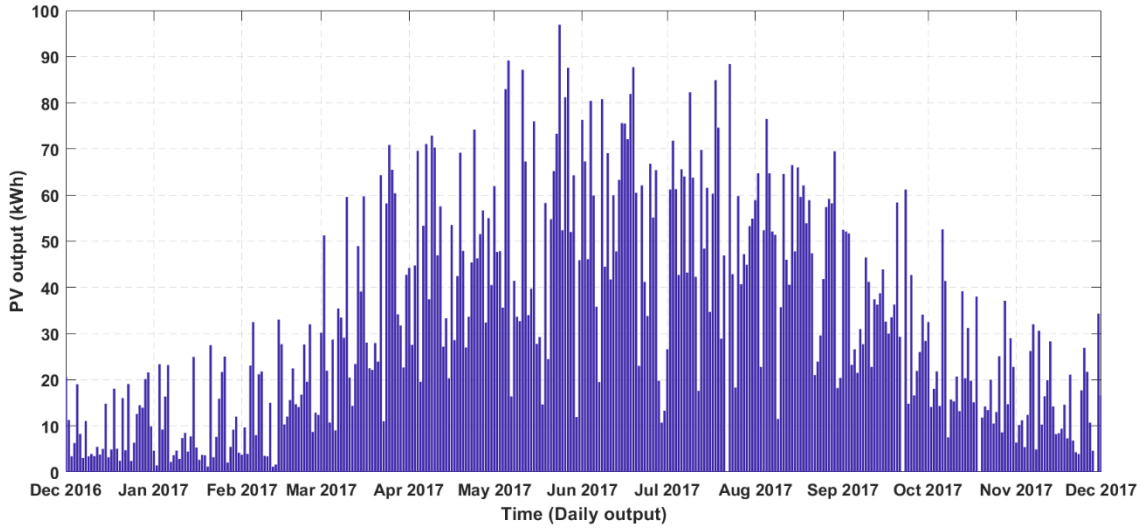


Figure 6-1: Solar PV output (Monitored data)

The annual temperature variation of the circulating fluid in the PVT array is given in Figure 6-2. The blue and orange fluctuated lines on the figure are the temperatures of the fluid at inlet and outlet of the PVT array, respectively. The maximum temperature was 35°C on July while heat pump turned off. This heat can be utilized if the novel heating system run with second or third operating mode with seasonal thermal energy storage approach.

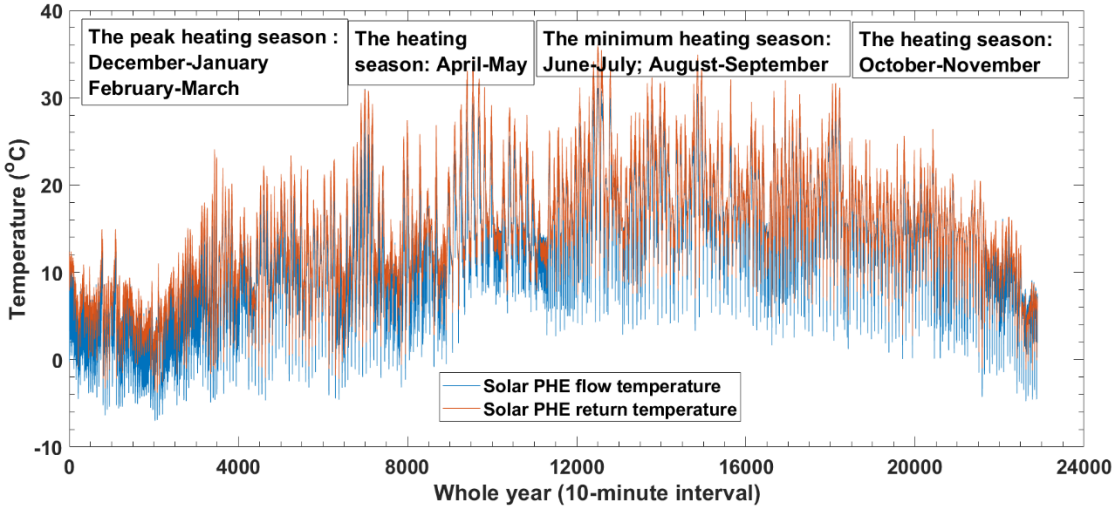


Figure 6-2: The temperature variation of the fluid at inlet and outlet of the Solar PVT Array (Whole year-Monitored data)

Annual average temperatures of the fluid at inlet and outlet of the PHE shows that the maximum PHE heat transfer rate is in April, with the value of 3.42 kW. Annual average values of the heat transfer rate of the Solar PHE are presented in Table 6-1. The maximum and minimum monthly averaged temperatures of solar PHE are 17.1˚C and -1.8˚C during August and January, respectively. The heat transfer amount extracted by 46 m^2 PHE fluctuates between 2.4 kW and 3.4 kW, with corresponding heat fluxes of 52.2 W/m^2 and 73.9 W/m^2 The heat transfer amount of PHE during winter PC was 2.5 kW, which is insufficient because of the temperature condition of the mean temperature of the fluid at inlet during December and January.

Table 6-1: Monthly averaged results from Solar

	Solar PHE	*Ambient mean temp.*	*Flow mean temp.*	*Solar radiation*	*Wind velocity*
	(kW)	(Tam) ˚C	(Tm) ˚C	(G) W/m^2	(Vw) m/s
Jan	2.51	5.1	-1.8	3.39	5.1
Feb	3.3	4.9	1.47	8.04	4.9
Mar	2.94	6.7	3.09	8.98	6.7
Apr	3.42	8.9	6.4	12.97	8.9
May	3.33	12.3	5.83	12.47	12.3
Jun	3.37	15.1	10.12	16.71	15.1
Jul	3.11	16.8	12.73	18.58	16.8
Aug	3.23	17.1	9.53	15.92	17.1
Sep	2.87	14.4	7.49	13.27	14.4
Oct	2.26	11.2	2.95	7.47	11.2
Nov	2.21	7.3	3.27	7.65	7.3
Dec	2.8	4.7	0.29	6.02	4.7

6.3.1.2. Ground array

The annual temperature variation of the circulating fluid in the ground array is given in Figure 6-3. The blue and orange fluctuated lines on the figure are the inlet temperatures of the ground array and the outlet temperature from the horizontal GHE,

respectively. On the other hand, the light orange continues line is the temperature of the return flow from the VGHE. Although the return temperature from the horizontal ground pipe varies from -3.7 °C to 34.1 °C depending upon the shallow temperature during the day based on the measurements of the 10-minute intervals during the time of the heat pump is on or off, the temperature of the return flow VGHE is consistent and varied from 4.7 °C to 18.7 °C.

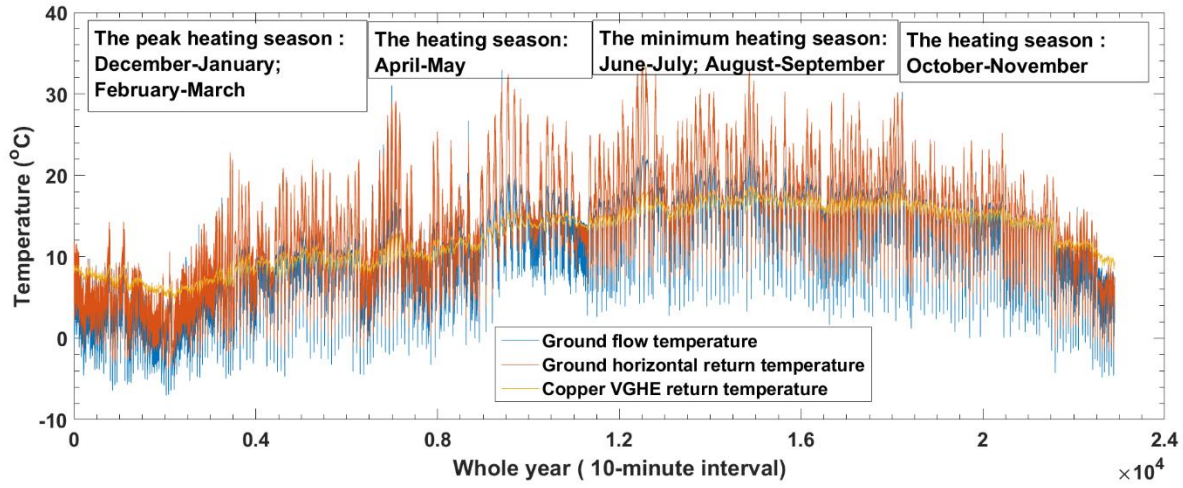


Figure 6-3: The temperature variation of the fluid at inlet and outlet of the Ground Array (Whole year-Monitored data)

Annual average temperature of the fluid at inlet and outlet of the ground array showed that the average maximum horizontal heat transfer rate was 3.03 kW on average, and maximum mean value of the heat transfer rate of the VGHE was 5.07 kW. The heat transfer value absorbed by the VGHE yielded 29.2 W/m on average at 2.5 m soil depth with 3.65 kW heat transfer capacity. Monthly averaged results are presented in Table 6-2.

Table 6-2: Monthly averaged results from Ground Array

	Flow temp. (°C).	*Horizontal (kW)*	*Vertical (kW)*	*Horizontal ground return temp. (°C)*	*Vertical ground return temp. (°C)*
Jan	-1.80	2.15	4.29	0.97	6.24
Feb	1.47	2.18	4.52	3.75	6.59
Mar	3.09	2.21	4.48	6.09	8.84
Apr	6.40	1.93	3.00	8.78	9.45
May	5.83	1.83	1.78	8.88	11.07
Jun	10.12	2.37	2.95	13.51	14.63
Jul	12.73	2.65	2.76	16.06	16.44
Aug	9.53	2.62	2.11	13.15	16.29
Sep	7.49	2.84	3.87	11.17	15.15
Oct	2.95	2.86	4.24	5.85	10.90
Nov	3.27	3.03	5.37	6.01	10.91
Dec	0.29	2.22	4.47	3.07	7.80

6.3.1.3. Heat pump-buffer tank-fan coil

Whole system is assessed under the design condition of whole year. The total thermal output from the heat pump was 15.020 kWh with 1030 hours of operating time throughout the year. The reason of the low annual heat output from the heat pump was the day-time heat pump settings which means the heat pump did not run at night during peak heating season and the heating (middle of the heating) season. Moreover, because the heat pump run on dual source but one operation mode. In other word, the brine going to the evaporator of the HP was the mixture of the fluid at outlet of the Solar PHE coming from the roof and the fluid at the outlet of the ground array coming from the soil. There was also no cooling mode during summer. Therefore, the hours of the operating time of the heat pump was required to be decreased by the fan coil settings to let the ground to recharge during summer season.

The COP of a heat pump at any moment in time depends largely on the temperatures of the heat sink and the sources. To ensure the COP is optimised, this temperature

difference must be kept as small as possible. The cold source temperature is affected by external air and soil temperature. However, the temperature of the hot source is set by the temperature of the return flow from the fan coil system. In the proposed system integrated to the poultry house, a buffer vessel was attached between the fan coil and heat pump to make the system more stable and decrease the impact of the load intermittency on daily purpose. If integrated supplementary heating is used, the return temperature will be increased, and COP will reduce. Therefore, there are external back-up heaters for the heat pump during winter conditions. Potentially more significantly, heat pumps have a high-pressure cut-out to prevent excessive refrigerant pressure. This may cause the heat pump to switch off if return temperatures exceed some maximum limit.

One of the main parameters affecting HP efficiency is the inflowing water temperature (i.e. brine flow temperature) and the temperature of the fluid at the outlet of the fan coil unit. Figure 6-4 presents the fluid at inlet and outlet of the evaporator side of the heat pump, which is the mixture of the fluid at inlet of the Solar PHE and ground array (horizontal ground heat exchangers and VGHEs). The highest and lowest mean temperature of the brine(glycol-water) were 17˚C and 1.7˚C in July and January, respectively, while the maximum temperature ate the fan coil outlet was 48.8˚C in June. It is important to highlight that there is a buffer vessel between the fan coil and heat pump, to provide more stable heat to the fan coil, and decrease the return temperature’s direct impact on the heat pump outputs, like soil storage methodologies during the non-thermal regulation term.

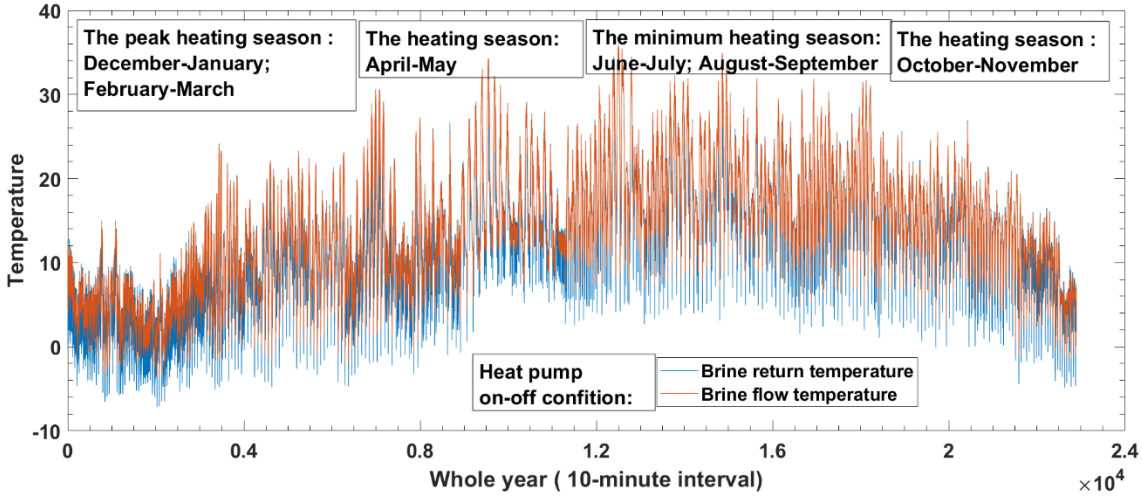


Figure 6-4 The temperature variation of the fluid at inlet and outlet of the evaporator side of the heat pump (Whole year)

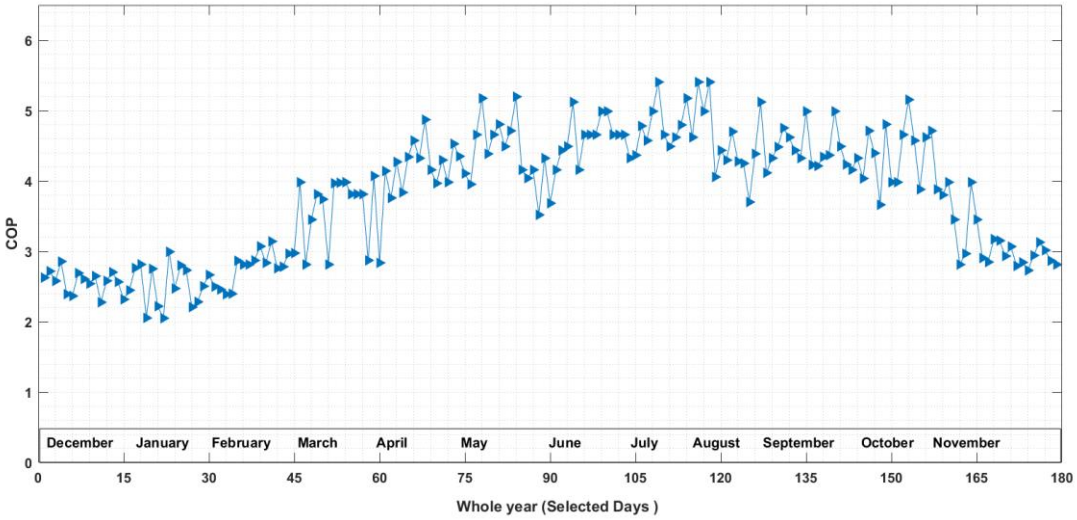


Figure 6-5 : *Average daily COPs over the year (sampled monitored data)*

The highest value of the average monthly COP of the heat pump was -4.58 in PC 5 during June and July, while the minimum value of the monthly COP of the whole system was 2.45 in PC 1 during December and January. (Figure 6-5). The GSHP

system was run with the maximum daily COP of 5.17 in June and the minimum of 2.05 in January, corresponding with 18.6 kW and 10.61 kW respectively.

The main parameters affecting HP efficiency is the inflowing water temperature (i.e. brine flow temperature) and return flow temperature from the fan coil, which means the temperature difference between the condenser and the evaporator of the heat pump. The system performance data under real conditions showed daily average heat pump capacity between 10.61 kW and 18.6 kW, and actual running hours on the site with a PV array with long-term high performance**.** The average daily COP of the heat pump ranged between 2.05 and 5.17, with the maximum value of 5.17 in July, while the minimum is 2.05 in the January. The average daily COP of the heat pump was determined by using the total electricity used by heat pump and heat energy produced during the monitored day. When considering the total electrical and thermal energy monitored for each week during the peak heating season, the mean weekly COP ranged between 2.3 and 2.43 through the PC1. It is important to highlight that there is a buffer vessel between the fan coil and heat pump to decrease the return temperature's direct impact on the heat pump outputs. The Seasonal COP is determined by dividing annual heat output of the PVT-GSHP to the electricity consumption of the system. The seasonal COP, which is SPF1, is 3.73 through a year. The novel PVT-GSHP design and components are very promising and relatively cheap compared to existing heat pump and solar systems. Chicken Production cycles

The highest average monthly thermal output of the HP is 1258 kWh in August, whereas the lowest value is approximately 813 kWh in September (Table 6-3).

Table 6-3: Monthly energy from whole system

	Average COPs	*Running hours*	*HP outpu (kWh)t*	*PV generation (kWh)-2016-17*	*PV generation (kWh)-2017-18*
Jan	2.50	91.31	1181.04	290.4	540
Feb	2.73	108.56	1420.30	428.27	694
Mar	3.47	73.60	1017.30	1102.5	1082
Apr	4.13	95.45	1485.01	1400.2	1984
May	4.43	96.60	1541.72	1633.4	1994
Jun	4.55	46.00	755.52	1637	1648
Jul	4.44	55.20	883.92	1602.9	1361
Aug	4.40	80.50	1276.91	1503.3	2497
Sep	4.37	92.00	1448.47	1018.9	767
Oct	4.09	98.90	1521.48	650.9	314
Nov	3.06	96.60	1364.26	428.3	305
Dec	2.58	86.48	1155.37	322	

During these three years, the heat pump efficiency and its working hours were affected by many parameters, such as heat required during the chicken cycle, ambient temperature, and entering water temperature to the HP (as explained previously with regard to the operating conditions of the HP).

Table 6-4: Annual chicken production cycle, 2016-2019

Test	*Assessment period*	*Production period*	*Chicken number*	*Chicken type*	*House*	*Electricity generated (kWh)*
Production year	2018-2019	7-week	6700*	Cocks & pullets	S1	-
Production year	2017-2018	7 -week	6700*	Cocks & pullets	S1	12499
Production year	2016-2017	7 -week	6700*	Cocks & pullets	S1	11993**

* 3350 pullets (females) leave at 4 weeks old and 3350 cocks (males) leave at 7 weeks old

** Between 01/10/2016 and 30/11/2016

*** Between 01/12/2016 and 31/01/2017

The COP and maximum output of a heat pump depend on the conditions in which it is operating, including external air temperature, return flow temperature (from the heating system), and relative humidity. Energy balance equations were applied based on temperature (˚C), flow rate (l/m), heat meter (Wh), and electricity meter (Wh) measurements for the calculation of the coefficient of the performance of the heat pump (COP), the seasonal coefficient of performance (SCOP), and the heat taken from solar and ground by using novel solar and soil heat exchangers. The instant electrical power consumed by the heat pump compressor (W_e) was determined from interpolated HP data.

Table 6-5 presents all COPs, PV generation for each cycle, and the annual achievement of each PC of the poultry farm. Because there is a break time for the cleaning process between PCs, the heat pump was turned off for around 15 days at the end of each cycle, which has an impact on the total running hours of each PC.

Table 6-5: Annual heat production under real condition (-2017)

Test	Period	Running hour	Heat (kWh)	PV generation (kWh-)		COP
				2017	2018	HP
PC1	Dec-Jan	178	2257.9	567	586	2.45
PC2	Feb-Mar	182	2659.5	1453	1233	3.05
PC3	Apr-May	192	2984.5	2927	3066	3.98
PC4	Jun-Jul	101	1669.8	3137	3642	4.58
PC5	Aug-Sep	173	2644.4	3887	3859	4.26
PC6	Oct-Nov	196	2801.5	1018	1081	3.1

* S1: 6700 cocks & pullets (3350 pullets (females) leave at 4 weeks old and 3350 cocks (males) leave at 7 weeks old)

** House temp. … (e.g. Dec-Jan cycle has 15 days' cleaning)

* * Production period 7 weeks

Moreover, it is important to investigate the dynamic responses and average energy achievement of the innovative components for each PC. The maximum heat transfer rate of the novel Solar PHE was 3.4 kW and a minimum value was 2.2 kW. The maximum heat flux of the PHE, with 46 m^2 total area was 73 W/m^2, and the minimum value was 47 W/m^2. The annual energy extracted by PHE and utilised in the HP was 3062.85 kWh. This is the renewable heat energy used indirectly for chicken PCs. From the ground heat exchangers, the heat transfer rate of the patented VGHE was 5.07 kW with 50 units of 2.5 m vertical pipes. PHE and VGHE are the novel and cheap products when compared to the existing solar and ground heat exchangers (assessed with regard to economic and environmental system performance- Table 6-6).

Table 6-6: Monthly averaged heat from Solar and Ground

	Cycle	*PHE (kW)*	*Horizontal (kW)*	*Vertical (kW)*
Jan	PC 1	2.51	2.15	4.29
Feb	PC 2	3.3	2.18	4.52
Mar	PC 2	2.94	2.21	4.48
Apr	PC 3	3.42	1.93	3.00
May	PC 3	3.33	1.83	1.78
Jun	PC 4	3.37	2.37	2.95
Jul	PC 4	3.11	2.65	2.76
Aug	PC 5	3.23	2.62	2.11
Sep	PC 5	2.87	2.84	3.87
Oct	PC 6	2.26	2.86	4.24
Nov	PC 6	2.21	3.03	5.37
Dec	PC 1	2.8	2.22	4.47

6.3.2. Economic and environmental investigation of PVT-GSHP

The system performance data under real conditions presented in the previous section are used for the economic and environmental assessments, operating between 10.61 kW and 18.6 kW on average heat pump capacity, with a PV array with long-term high performance.

The external temperature ranged from 2-25°C, and the external relative humidity from 40-95%. The total thermal output from the heat pump was approximately 15 MWh per annum. The daily COP ranges between 2.05 and 5.17, with the maximum value of 5.17 in PC 4**,** while the minimum is 2.05 in January and the maximum is in July. Figure 6-6 presents the heat energy generation; the heat output of the heat pump and part load provided by the heat pump.

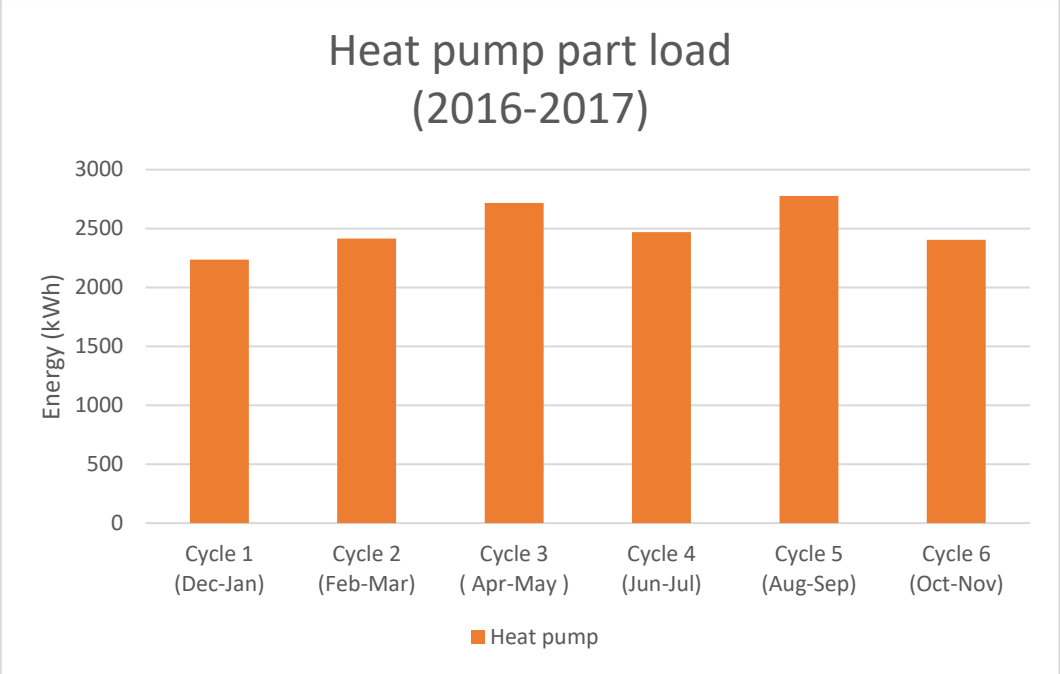


Figure 6-6: Heat pump output during annual PCs

In the field measurements, the heat pump provided part load of the house's energy demand as designed. The system can be enhanced with more operating modes in settings and component circuits. The main issue observed in the design for the crucial winter period was the temperature of the fluid at the outlet of the PHE and the horizontal ground pipe being lower than the novel VGHEs, which makes the system less efficient than its potential, causing a rapid temperature drop underground. However, scheduled hours can be increased relative to the seasonal soil store temperature. Employing a 3-way valve attachment between PHE and the ground array is recommended for future assessment. Gas boiler efficiency is considered as 0.85.

6.3.2.1. Economic assessment

Life cycle cost assessments involves the sum of initial costs of PVT-GSHP system, annual energy cost and maintenance costs of GSHP systems are not periodic or regularly occurring.

Simple payback period is the most common indicator to know the project feasibility and reliability. The heating system has an installed cost of £39,812. The system used for heating demand. (Table 6-7).

Table 6-7: Heating system parts cost

Item	*Size*	*Cost (£)*
PV	13.5 kW PVT array	20,100.
PHE	x7 12 m^2 mats	1,715.
Heat pump	15 kW	8,294.
Buffer vessel	UKV 300 L	723
Fan coil and accessories	14.5 kW heating	670
Accessories	Controller, valves	310
Ground array (Making good)	100 m”	8,000
	Total	*39,812*

I_{cc} is the initial capital cost of the system (£). Annual savings are calculated by subtracting the annual total cost (A_{TC}) of the gas boiler system from the long-term tests.

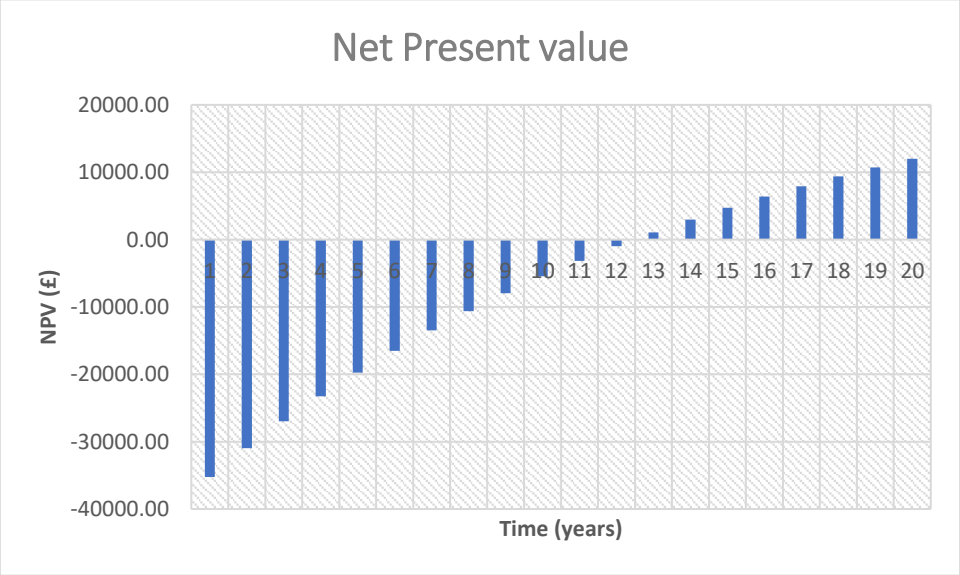


Figure 6-7: Net present values in UK

In the first year the annual saving was £780.86 electricity savings. The current design condition of the heat pump had £ 2,830 annual savings. However, the projected pay-back period is determined under two operating modes conditions of hours of the heat pump in order to see the real value of the system. The simple payback period of the hybrid heating system is 8.14 years, with £4,890 total annual savings with government revenues under the condition of efficient running hours potential with two operating modes. The saving parameters are presented in Table 6-7. The gas cost and electricity cost are 0.029 £.kWh and 0.1097 kWh (high rate), respectively (Gandolfi 2016). The main parameters and savings are presented in Table 6-8 with a 7% interest rate (see Figure 6-7).

Table 6-8: PVT-GSHP heating system saving cost (The current design condition)

	Cost (£/kWh)	HP saving (MWh)	PV saving (MWh)	Annual total (£)
Gas	0.029	14.95	-	510.9
Electricity (low rate-night)	0.0683	-	8.74	596.9
Electricity (high rate-day)	0.1097	-	8.74	958.8

6.3.2.2. Environment assessment

The saving of CO_2 emissions of the PVT-GSHP heating system are determined as 3.06 tons/year savings for electricity and 3.70 tons/year savings for gas. The current condition of the heat pump had 6.58 tons annual savings. However, two operating conditions of hours of the heat pump has been considered to for the projected total savings in order to see the real value of the environmentally friendly system. The projected total saving of CO_2 emissions is 8.47 tons/year under the condition of efficient running hours potential with two operating modes of the heat pump.

6.4. Summary

This chapter evaluates the economic and environmental feasibility of the novel PVT-GSHP system designed for a poultry house in Kirton. Long-term site investigation including testing PVT-GSHP and exploring its running hours and COPs from 2016 to 2019. The novel components of the design are the novel PHE attached to the PV panels on the roof, and copper VGHE attached to the heat pumps. Each component of the system and the novel hybrid design was experimentally investigated with long-term monitoring under transient conditions. The performance assessment of the novel system design and its components presented in this chapter is used to determine its economic and environmental applicability in the poultry house over the long-term.

As stated in previous section, the hybrid design can provide much higher heating energy savings when the system operates with different controlling methodology like two or three operating modes to maintain the PVT-GSHP system with only ground or solar array, and therefore more efficient operating hours, by using the seasonal smart storage method. This will be assessed in depth in the next step of the study for this novel hybrid system for agriculture HVAC industry.

The observation from the long-term site investigation indicates that UK poultry farms mainly require heating for about 10 months, particularly when growing chickens. Buildings occupied by young chickens require heat in every season and PC. The external temperature ranged between 2-25°C, and the external relative humidity ranged from 40-95%. The heat demand of the building varied, depending upon the time of day, season, and PC.

The efficiencies of each novel component attached to the heat pump were investigated under the long-term real conditions in the poultry farm. The maximum heat transfer

rate of the novel PHE was 3.42 kW on average on April. The total thermal output from the heat pump was approximately 15 MWh per annum. The COP ranged between 2.05 and 5.17, with a maximum value of 5.17 in July, while the minimum is 2.05 in January. The projected simple payback period is 8.14 years, with £4,890 net annual savings and government supports based on efficient running hours.

The experimental results show that the horizontal polyethylene heat exchanger (PHE) has a maximum 3.03 kW of heat transfer rate. The heat transfer rate of novel copper ground heat exchanger (VGHE) is 5.07 kW at 2.5 m length. The average heat transfer rate of the novel VGHE is 3.65 kW through a year, with a corresponding heat transfer amount of 29.2 W/m. The average heat transfer rate of novel 46 m2 solar polyethylene heat exchanger (PHE) is 2.9 kW, with a corresponding heat flux of 64 W/m^2 through a year.

The analysis from October 2016, described for the autumn assessment, showed that the running time of the heat pump was relatively high, with the time scheduled as full-day on-power. The heat pump can theoretically cover both PC 4 and 5 heat demand during the time from May to August when the system operates more operating conditions.

The whole system efficiency can be increased by utilising the storage capacity and the seasonal smart storage method. Because the poultry farm is in the countryside, wifi and therefore automatic control of the heat pump is challenging. For future consideration, a special system design is strongly recommended for the agriculture industry, like relatively high scheduled hours, and seasonal storage from PVT to ground by the attachment of a three-way valve between PHE and the ground array.

This study presents the results of a field trial of an innovative and cost-effective HVAC system design applied to a poultry house from the monitored between 2016 and 2018. The observation from the long-term site investigation indicates that UK poultry farms mainly require heating for about 10 months, particularly when growing chickens. Buildings occupied by young chickens require heat in every season and production cycle (PC). The heat demand of the building varied, depending upon the time of day, season and PC. The system provides a highly efficient solution to improve thermal regulation and air quality in poultry houses. It is fully adaptable and customisable by its nature to also address other markets.

The study concludes that:

- 15 kW heat pump was selected, considering the existing heat extracted from available sources to provide part load to the poultry house. The total thermal output from the heat pump was 15.02 MWh per annum with 1030 hours of operating time when it run with one operation mode throughout the year.
- The average daily COP of the whole system ranged between 2.05 and 5.17 on the monitored days throughout the year. Daily average heat pump capacity between 10.61 kW and 18.6 kW.
- The annual electrical output from PV panels are 11.99 MWh throughout the whole year. Solar PV can cover all the heat pump's annual electrical energy requirement and generated 8.74 MWh of extra electricity exported to the grid.
- When considering the total electrical and thermal energy monitored for whole week during the peak heating season, the mean weekly COP ranged between 2.3 and 2.43 through the poultry production cycle (PC1).

CHAPTER 7

PERFORMANCE ASSESSMENT OF AIR-COOLING SYSTEMS FOR LOW CARBON POULTRY HOUSES

7.1. Introduction

Heat stress is one of the most significant environmental concerns in poultry production in summertime all over the world. In addition to causing suffering and death among chickens, it decreases production and adversely affects chicken welfare. In UK weather conditions, poultry farms are managed with natural and mechanical ventilation during summer. Even though the site did not use a cooling system, the farmer reported that the chickens were uncomfortably hot during summer. Evaporative cooling is an effective method to protect broilers from heat stress and its catastrophic results during hot weather conditions, with great potential to save energy. On-site transient test investigation observed direct evaporative and dew point characteristics of evaporative cooling to assess:

1. The potential energy saving of the combined PEC unit with 2-fan control design under real outside weather conditions and inside air temperature variations of chicken PCs under transient condition.
2. The effects of dew point cooler using forced draft fan while changing environmental conditions for steady state tests.

From the review of the evaporative cooler units applied on the poultry farms, it is observed that in addition to being relatively expensive to manufacture and install, current available evaporative coolers have certain drawbacks, especially when water is drawn completely through the pads into the house interior, causing damage to

adjacent equipment and increasing microbial growth with associated footpad, joint, and respiratory infections. Therefore, a novel dew point cooling system was experimentally investigated under steady state conditions to apply to poultry houses. This offers multiple benefits for increasing efficiency and utilising renewable sources in agricultural buildings. Moreover, the results of a direct evaporative cooler designed for the University of Nottingham are presented in this chapter.

7.2. Methodology

7.2.1. System design and experimental setup

7.2.1.1. Direct evaporative cooling system design and setup

This section evaluates the feasibility of an evaporative cooling unit designed for both ventilation and air conditioning purposes for poultry farms under UK weather conditions. A schematic of the system is illustrated in Figure 7-1.

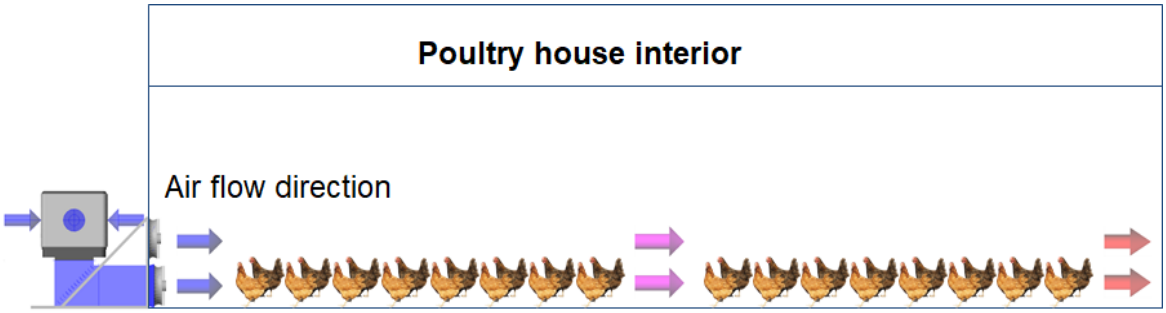


Figure 7-1: Direct evaporative cooler design: 2-fan attached (Ecocooling)

In summer the farmer installs an Alfan fan with 1270 mm diameter, as shown in Figure 7-2, to make the facility relatively cold using a tunnel ventilation scheme. This causes a rise on the volumetric airflow through the house and chickens to minimise heat stress during the summer. However, the farmer reported that this method is still insufficient for the welfare of the chickens in the facility. Therefore, the PEC unit was designed for application in the poultry farm, as experimentally investigated in this chapter. It can be

used for both ventilation and cooling purposes with its multiple function design. This system is capable of meeting cooling demand during summer in UK climate conditions. It runs as ventilation only, or both cooling and ventilation mode, based on the welfare requirement of the chickens in terms of temperature, humidity, and air flow.



(a) (b)



(c)

Figure 7-2: (a) Current 50" supply Alfan PEC cooler, (b) Front view, (c) Back view

The experimental setup is designed, constructed and validated to carry out a wide range of operating and transient real conditions such as air velocity, inside-outside air temperature, and humidity rate fluctuations. One Ecocooling wet box consisting of

cooling pads integrated with two 630 mm Ziehl Abegg Fans with associated airflow rate and cooling/ventilation settings was used (Figure 7-2c).

7.2.1.2. Dew point evaporative cooling system rig description and lab set-up

The system aims to cool air through an evaporative process under psychrometric conditions at atmospheric pressure. An environmental chamber is used for providing steady-state conditions during testing. Inlet air is ducted from the environmental chamber and passed through the PEC unit. Water is pumped from a reservoir to the top of the unit.

The dew point indirect evaporative cooling device is suitable for cooling a supply air stream with no moisture addition. The unit is 900 mm in length, 570 mm wide, and 500 mm tall. The dew point indirect evaporative cooling unit, demonstrated in Figure 7-3 consists of a PEC, pump, fan, internal water reservoir, and ballcock valve.

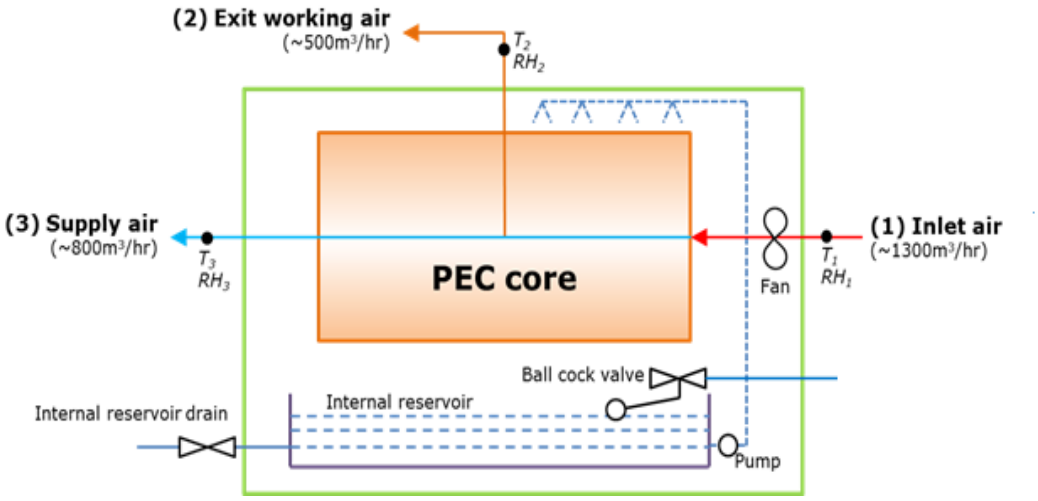


Figure 7-3: PEC schematic and airflows

The PEC psychrometric process is shown in Figure 7-3. There are three airflow rates in this system: environmentally conditioned inlet air, evaluated outlet air, and working air. Inlet air is supplied to the unit from the environmental chamber. The air is split into a working and supply air after entering the PEC. The water, pumped from the internal

tank, evaporates into the working air stream by creating a cooling effect. After that, this cold and humid working air cools the supply air through the exchanger wall with no moisture addition. Working air and outlet air both supply to the environment practically during testing.

The recent Maisotsenko cycle (or M-cycle) design is the most effective of recently developed methods (Wani, Ghodke, and Shrivastava 2012). Dew point indirect evaporative cooler (DPIEC) is a type of heat and mass exchanger run with the M-cycle using two working air channels, including holes to recycle the working humidified air and mix it with dry air. The wet side of the working air interacts with the water and uses latent heat. Unlike DEC and IEC, the holes in the channels can be stepped at points 1 to 3, as shown in Figure 7-4. The wet air could be pre-cooled, depending on the channel design, such as in terms of the hole sizes, numbers, and angles, and the product air can be cooled from 1 to 2 points with much higher efficiency.

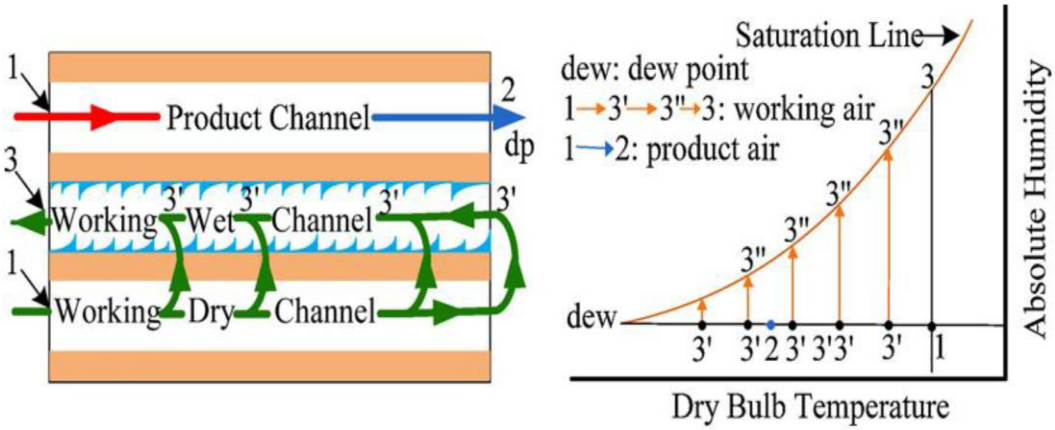


Figure 7-4: Working principle of M-Cycle (Wani, Ghodke, and Shrivastava 2012)

As mentioned previously, the pertinent equipment comprises the dew point cooler (including PEC core, fan, and pump), computer, data logger, temperature/relative humidity probes, and an environmental chamber. The rig was instrumented with Vaisalia HMP110 sensors (0-1VDC) connected to a DT85 Data Taker (12VDC)

connected with a computer, as presented in Figure 5-6a. The PEC uses binding of cloth and plastic materials, shown in Figure 5-6b. The electrical consumption of fan and pump is measured using a Brennenstuhl PM^{2}30 electricity monitor.



(a) (b)

Figure 7-5 (a) System instruments, (b)System materials

Figure 7-6 shows a picture of labelled air and water flows and switches. The rig has a portable frame. The pump is IP68 rated (dust-tight and protected against complete, continuous submersion in water). Fan and pump terminals are enclosed in a watertight box, with RCD (residual current device) protection. The rig is PAT (Portable appliance testing) tested. All pipes and fittings were checked for leakage before operation. The ballcock valve prevents the reservoir over-filling.

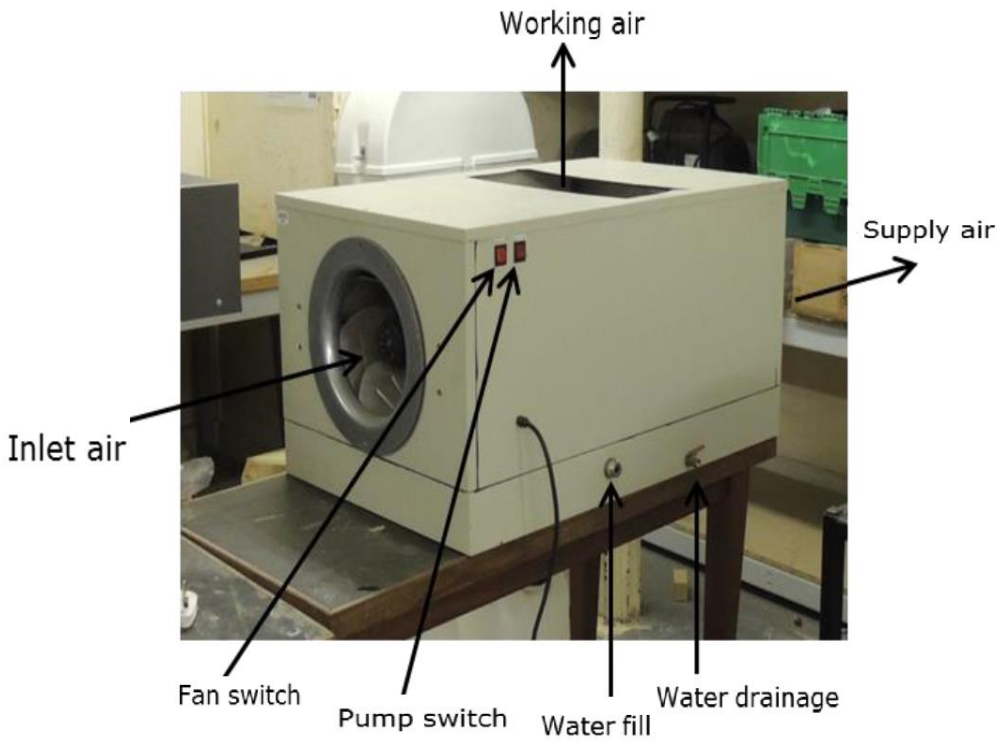


Figure 7-6: Photograph of PEC with labelled airflows

In the laboratory testing of the PEC, an environmental chamber was employed to provide consistent inlet air and achieve steady-state conditions to evaluate various environmental conditions. The environmental chamber can create air conditions from 0-40°C and 10-80% relative humidity. The PEC unit was placed outside and ducted to the environmental chamber, thus causing some small fluctuations of inlet temperature and system humidity. In operating the system, the relative humidity and temperature of the environmental chamber are first set, then the PEC unit is run at that condition. Getting the stable and homogeneous air condition can take up to 30 minutes, depending on latent and target humidity and temperature conditions. Once the required air conditions are achieved, recording begins.

7.2.2. Mathematical analysis

7.2.2.1. Mathematical analysis of direct evaporative cooler

Investigating the cooling capacity and effectiveness of the system requires focusing on the temperature and humidity rate of the outlet and working air. Working principles of the direct evaporative cooler are shown in Figure 7-7. The initial condition of the air is states as A. The dry-bulb temperature of the air decreases, while the humidity in the air increases. Wet bulb temperature remains constant until the is reached B (Pistochini and Modera 2011).

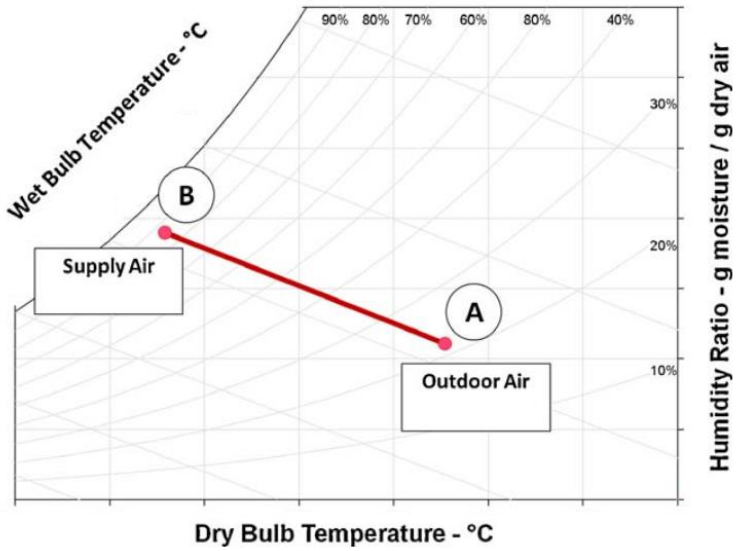


Figure 7-7: Direct evaporative cooler working principle (Pistochini and Modera 2011)

The samples of this study are the temperature and humidity rates of the inlet, outlet, and working air at transient condition. The purpose of the test is to assess the performance of the PEC system under varying conditions (i.e. temperature and relative humidity and air velocity) using fan speed and on-off controller. There are three principal effects that have to be considered in PEC core: wet-bulb effectiveness (ε_{wb}), humidification effectiveness (ε_{hum}) and energy efficiency ratio (EER), represented in

the following. The outlet air temperature and the humidity rate are important factors in the studies (X. Cui et al. 2014).

$$\varepsilon_{wb} = \frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{wb,in}}_{v} \quad 7\text{-}1$$

$$\varepsilon_{hum} = \frac{w_{out} - w_{in}}{w_{sat} - w_{in}} \quad 7\text{-}2$$

Where $T_{db,in}$ and $T_{db,out}$ are the dry bulb temperature of the inlet, outlet air is, $T_{wb,in}$ is the wet bulb temperature of the outlet fresh air, and w_{in} and w_{out} are the humidity ratio of the inlet and the outlet fresh air, respectively.

The cooling (wet bulb) effectiveness shows the relationship between the output product air temperature and the wet-bulb temperature in various inside conditions. As can be seen in Equation 7-1 and Equation 7-2, the outlet air temperature and the humidity rate are important factors (X. Cui et al. 2014).

7.2.2.2. Mathematical analysis of dew points evaporative cooler

On the other hand, the dew points evaporative cooling system, based on the other proposed design, was also designed for application in poultry houses to maintain a stable humidity rate, particularly in hot weather zones. In addition to the type of the evaporative cooler, the difference between these two systems is the testion condition of the system. Unlike the transient DEC test, the dew point cooler test explained in this section is to assess the performance of the PEC system under varying steady state conditions (i.e. temperature, relative humidity, and air velocity) using a fan speed controller and an environmental chamber. The dew point cooler performance parameters are slightly different than the parameters of the direct evaporative cooler because the differences in their system operating mechanism. There are two principal

effects to be considered in PEC core: wet-bulb effectiveness (ε_{wb}) dew point effectiveness (ε_{dp}), and EER (X. Cui et al. 2014):

In general, effectiveness is the ratio of a temperature differential between the inlet and outlet air, and a temperature differential between the inlet and working air. Equation 7-3 and Equation 7-4 are used to determine wet-bulb and dew point effectiveness, respectively. The cooling (wet bulb) effectiveness shows the relationship between the output product air temperature and the wet-bulb temperature in various inside conditions. As can be seen in Equation 7-3 and Equation 7-4**,** the outlet air temperature and the humidity rate are important factors in the studies (Riangvilaikul and Kumar 2010; X. Zhao, Liu, and Riffat 2008):.

$$\varepsilon_{\mathrm{wo}} = \frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{wb,work_v}} \quad 7\text{-}3$$

$$\varepsilon_{dp} = \frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{dp,work}} \quad 7\text{-}4$$

Where $t_{db,in}$ and $t_{db,out}$ are the dry bulb temperature of the inlet and outlet air in Kelvin, $T_{wb,work}$ and $t_{db,work}$ are the wet-bulb temperature and dew point temperature of the working air in Kelvin.

Q_{sen} is the sensible cooling of intake air in Watts (defined in Equation 7-5), and Cp is the specific heat of air at constant pressure in kJ/(kg K). Air density is given in kg/m^3 to represent the airflow rate of supply air in m^3/h:

$$Q_{sen} = \frac{\mathrm{C_{air}} \times \rho_{\mathrm{air}} \times V_{air} \times (T_{db,in} - T_{db,out})}{3.6} \quad 7\text{-}5$$

EER is the proportion of sensible cooling capacity in Watts to the total power consumption by the product air fan and pump in the unit, given by: Equation 7-6. Where W is the total power consumption of the PEC cooler through fans and pumps;

$$EER = \frac{Q_{sen}}{W} = \frac{C_{air} \times \rho_{\text{air}} \times V_{air} \times (T_{db,in} - T_{db,out})}{3.6 \times W} \quad 7\text{-}6$$

7.3. Results and discussion

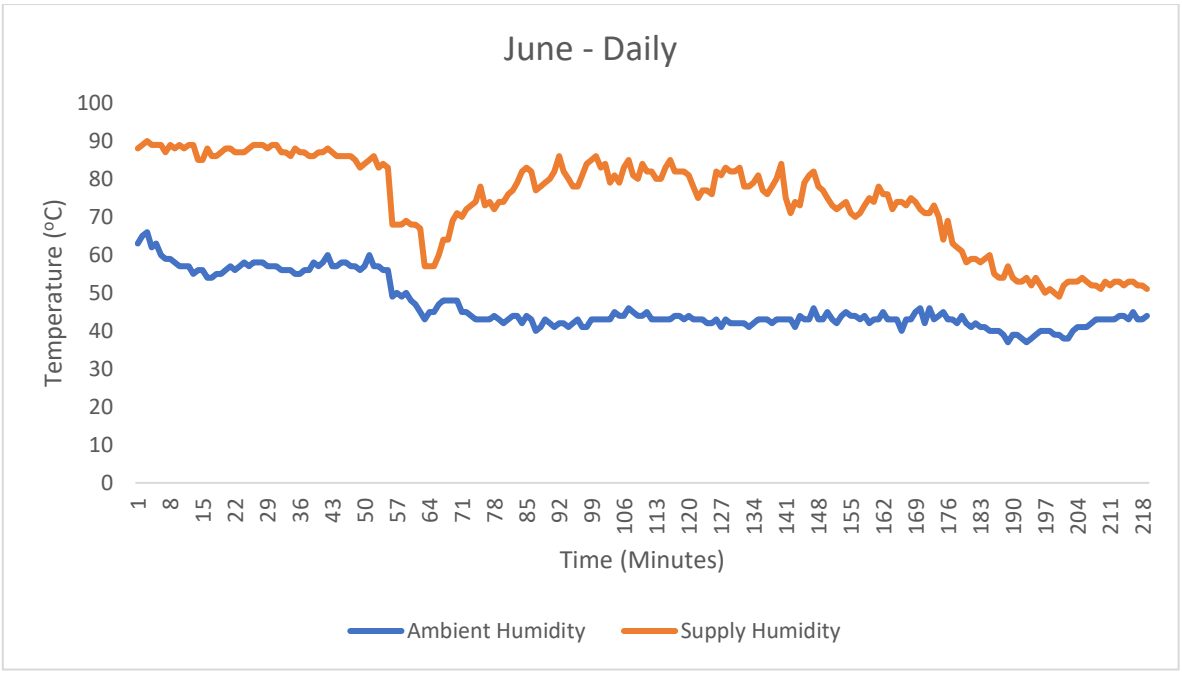
7.3.1.1. Direct evaporative cooling

The testing results of the design under real site condition are presented in this section to understand the impact of assessed parameters (i.e. inlet temperature, relative humidity, and airspeed) on system performances. The cooling requirement of the chicken house was observed from 17th July to 24th July in 2017, and from 26th June to 17th July in 2018 which are the dates of the end of the PC 4 (June-July). The comfort temperature required is the maximum value of 24˚C for this stage of chicken growth.

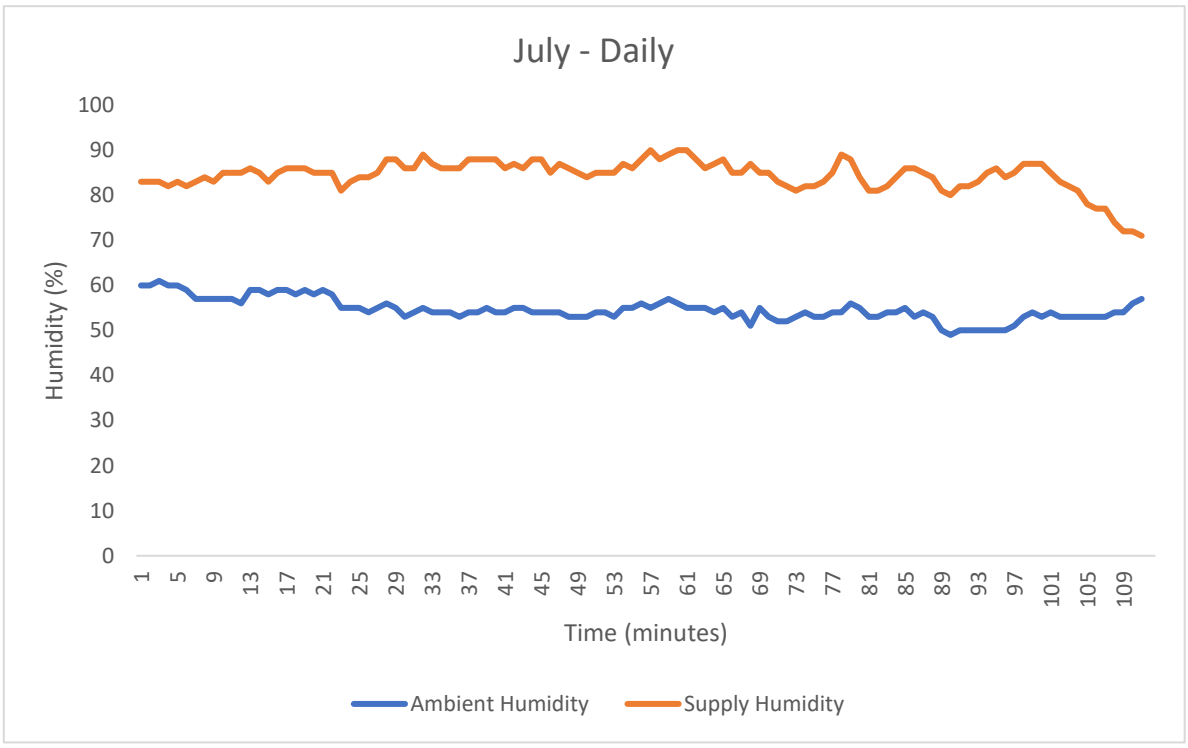
These fans' output matches with the supply fan output, with a maximum 18,000 m^3/h airflow rate capacity. The other important consideration is to match this flow rate within the required evaporative pads (Table 7-1).

Table 7-1: Maximum output with defined temperature increase and airflow (Eco-Cooling Manual)

Airflow	Temp. increase				
	5˚C	7.5˚C	10˚C	12.5˚C	15˚C
18,000 m^3/hr	31	46	61	76	91
13,500 m^3/hr	23	34	46	57	68
9,000 m^3/hr	16	23	31	38	46
4,500 m^3/hr	8	12	16	19	23



(a)



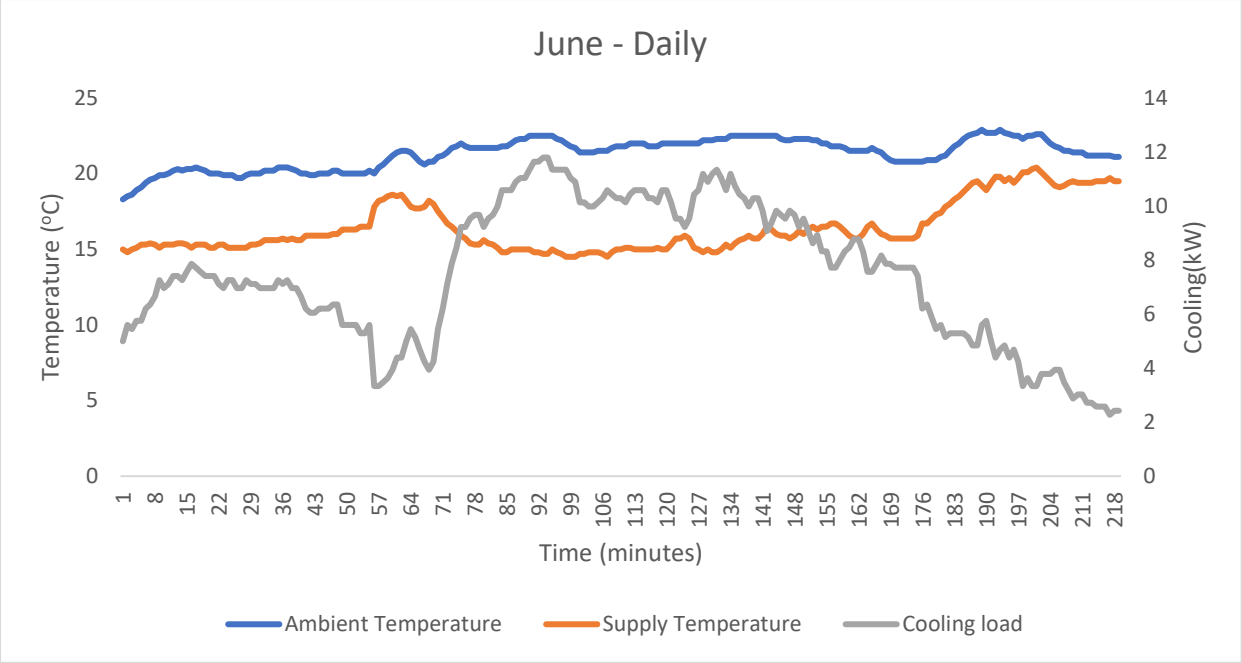
(b)

Figure 7-8: Ambient and supply air during test period (a) June, (b) July PC

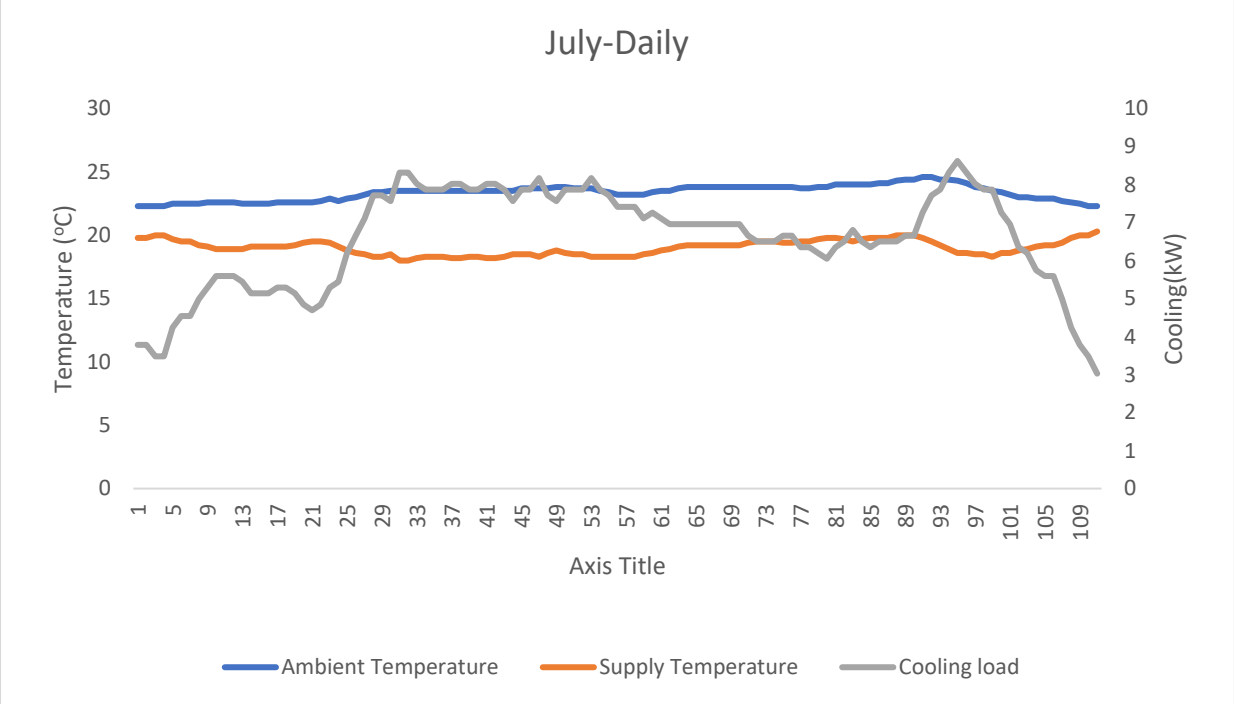
As can be seen in the Table7-1, total cooling output values fluctuate between 8 kW and 91 kW from a minimum flow of 4500 m^3/hr to a maximum flow of 18,000 m^3/hr, with wet box and free air fans working together for cooling mode at up to 30% fan speed capacity. This capacity increases up to 88% when the unit runs only for ventilating purposes, when the wet-box is off. CelDEK 5090 Evaporative Cooling Pad is the main material of the direct evaporative cooler in the cooling process

The ambient environment has a significant impact on the performance of evaporative cooling systems, especially the relative humidity, which is one of the main parameters for an adequate performance of the system. Ambient and supply air RH during a part of the test period are shown in Figure 7-8, fluctuating between 40% and 65%.

Figure 7-8 shows the ambient and supply temperature profile of the chicken house during the summertime, in June and July. The cooling mode starts when the inside temperature of the house is higher than 24˚C, from the 5^{th} week of the chicken growth cycle. The temperature difference is around 5˚C, with a 22% fan speed condition in the test data taken form the PEC cooler. The cooling mode running hours range from 2.5-4.18 hours, depending on the daily cooling requirement of the chicken facility. Daily running hours were observed for 3 hours on average. Cooling loads also varied from 6.5 kW to 7 kW for the two last weeks of chickens' growth cycle (Figure 7-9).



(a)



(b)

Figure 7-9: Ambient and supply air and cooling load during (a) June PC, (b) July PC

7.3.1.2. Dew point evaporative cooling system

Throughout the testing process, consideration is made to investigate the cooling capacity and effectiveness of the system. This requires focusing on the temperature and humidity rate of the outlet and working air. The samples of this study are the

temperature and humidity rates of the inlet, outlet, and working air in the steady-state condition. System performance parameters assessed in the testing include inlet temperature, relative humidity, and airspeed. However, it is important to mention that some parameters fluctuate unexpectedly, which is attributable to these results being derived from initial testing.

The efficiency has a linear relationship with air temperature. The results of testing for three cases are presented below;

- **Case 1: Constant inlet temperature and air velocity**

The methodology of the test is to keep constant of the temperature and the air velocity during test time. The temperature and Air velocity are 28 ^{o}C and 0.8 m/s, respectively. The conditions are also good to use the technology in Europe countries. Experiments were conducted at five different air humidity conditions; 22.39**%**, 27.00**%**, 33.85**%**, 42.69**%** and 48.3**%**. There was a considerable relation between evaporative cooling pad and cooling efficiency, as expected. The results from the experiment can be seen in Table 7-2. The results presented in Table 7-2 show that the RH of the working fluid increases with the rise of the RH in inlet air

Table 7-2: Testing results at steady state condition during humidity variation 22-48%

Flow air RH (%)	*Flow air temp. (˚C)*	*Released air temp. (˚C)*	*Released air_RH (%)*	*Working air temp. (˚C)*	*Working air RH (%)*	*Water temp. (˚C)*
22.39	28.06	19.96	51.69	21.75	60.76	19.94
27.00	28.57	19.86	55.16	21.68	63.09	19.77
33.85	28.90	20.17	62.59	22.02	67.22	20.13
42.69	28.58	21.12	70.35	22.88	72.52	21.13
48.38	28.13	22.29	73.86	24.00	75.48	22.41

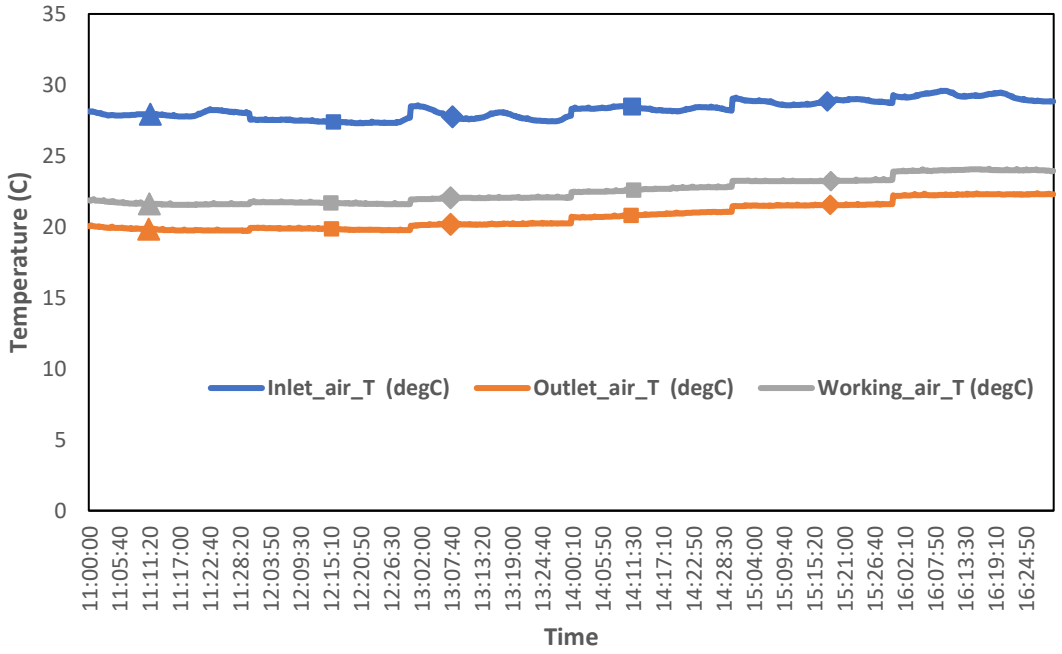


Figure 7-10: Inlet-outlet and working temperature sampling

Figure 7-10 presents the temperature of the flow (inlet) air to the evaporative cooler, outlet air released from the cooler and the working fluid in the system in the condition of 22.39% RH of inlet air condition within 1-minute interval. The effectiveness of the system under the variation of the RH condition are presented in Figure 7-11. The effectiveness's are slightly affected on the rise of the RH because of the efficient dew point cooler design with a working air.

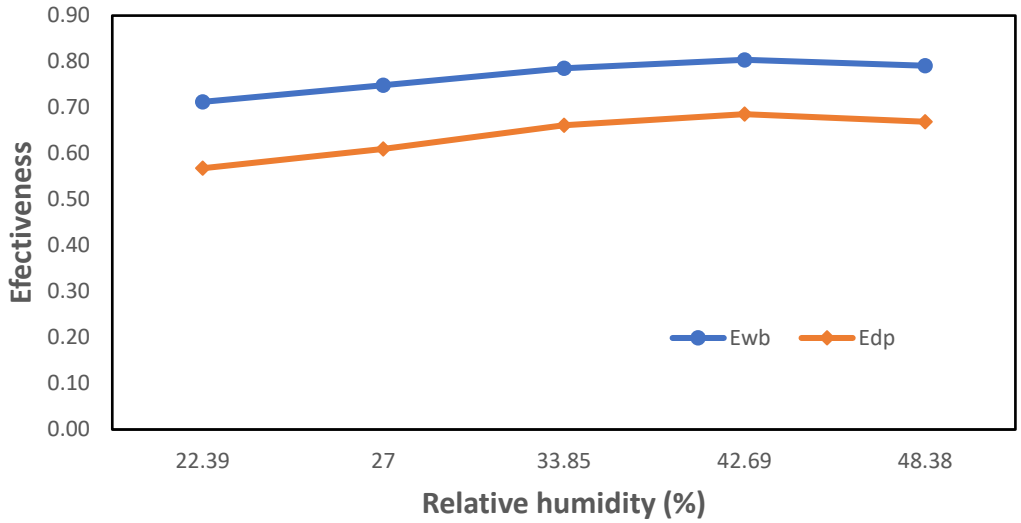


Figure 7-11 Initial The values of effectiveness at steady state conditions

❖ Case 2: Constant relative humidity (RH) and air velocity

The methodology of the test is to keep constant of the RH and the air velocity during test time. The RH and Air velocity are 27% and 0.8 m/s, respectively. Experiments were conducted at five different air humidity conditions; 23.17^{o}C ,26.03 ^{o}C, 27.98 ^{o}C, 29.95 ^{o}C and 32.76 ^{o}C. The results from the experiment can be seen in Table 7-3.

Table 7-3: Testing results at steady state condition during temperature variation 23-32˚C

Flow air temp (^{o}C)	*Flow air temp. (˚C)*	*Released air temp. (˚C)*	*Released air_RH (%))*	*Working air temp. (˚C)*	*Working air RH (%)*	*Water temp. (˚C)*
23.17	25.86	19.84	54.00	20.96	61.15	19.38
26.03	25.85	20.50	55.00	22.05	60.19	19.87
27.98	26.09	21.08	57.00	22.82	60.93	20.47
29.95	28.28	21.15	60.00	22.84	64.06	20.79
32.76	28.02	21.79	63.00	23.89	65.60	21.67

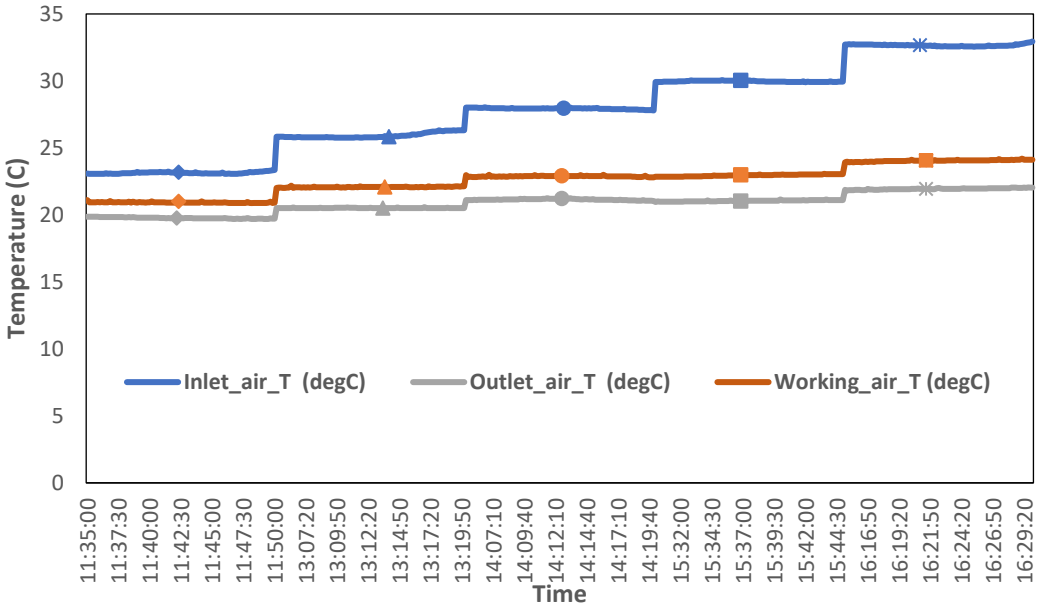


Figure 7-12: Inlet-Outlet and working temperature sampling

Figure 7-12 presents the temperature of the flow (inlet) air to the evaporative cooler, outlet air released from the cooler and the working fluid in the system in the variation of inlet temperature of inlet air condition within 1-minute interval under steady-state condition. The temperature between inlet and outlet air increases with when the temperature of the flow air rises.

The effectiveness of the system under the variation of the inlet temperature condition are presented in Figure 7-13. The effectiveness's are slightly impacted on the rise of the RH because of the efficient dew point cooler design with a working air. EER is .90 under 32.7^{o}C of temperature of the inlet air through to the cooler.

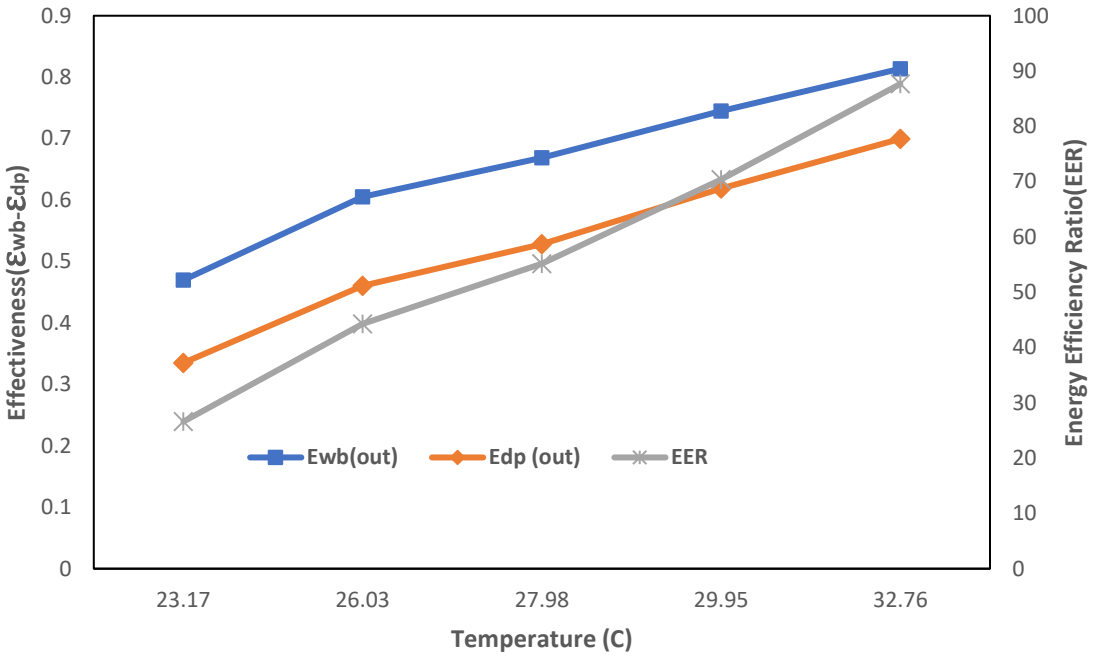


Figure 7-13: Initial effectiveness and EER values at steady state conditions

❖ Case 3: Constant relative humidity (28%) and temperature (28˚C)

Table 7-4: Testing results at steady state condition during aiir velocity variation 0.8-2.7 m/s

Air speed (m/s)	*Flow air RH (%)*	*Flow air temp. (°C)*	*Released air temp. (°C)*	*Released air RH (%)*	*Working air temp. (°C)*	*Working air RH (%)*	*Water temp. (°C)*
0.8	28.25	26.49	20.27	60.83	22.30	63.74	20.35
0.9	27.75	27.08	21.20	54.14	23.04	61.23	21.10
1.3	28.31	25.97	21.24	55.41	23.42	57.13	20.67
1.8	28.59	26.29	22.46	49.15	24.77	50.84	21.17
2.4	28.58	29.45	23.72	45.80	25.99	47.15	21.70
2.7	28.53	30.76	24.35	45.27	26.54	46.57	22.12

Experiments were conducted at six different air velocities; 0.8 ms^{-1}, 0.9 ms^{-1}, 1.3 ms^{-1}, 1.8 ms^{-1}, 2.4 ms^{-1} and 2.7 ms^{-1} under the constant 28% RH and 28 ˚C**.** of inlet air temperature. Figure 7-14 presents the temperature of the flow (inlet) air to the evaporative cooler, outlet air released from the cooler and the working fluid in the

system in the variation of inlet temperature of inlet air condition within 1-minute interval under steady-state condition. The temperature between inlet and outlet air increases with when the temperature of the flow air drops (Figure 7-14).

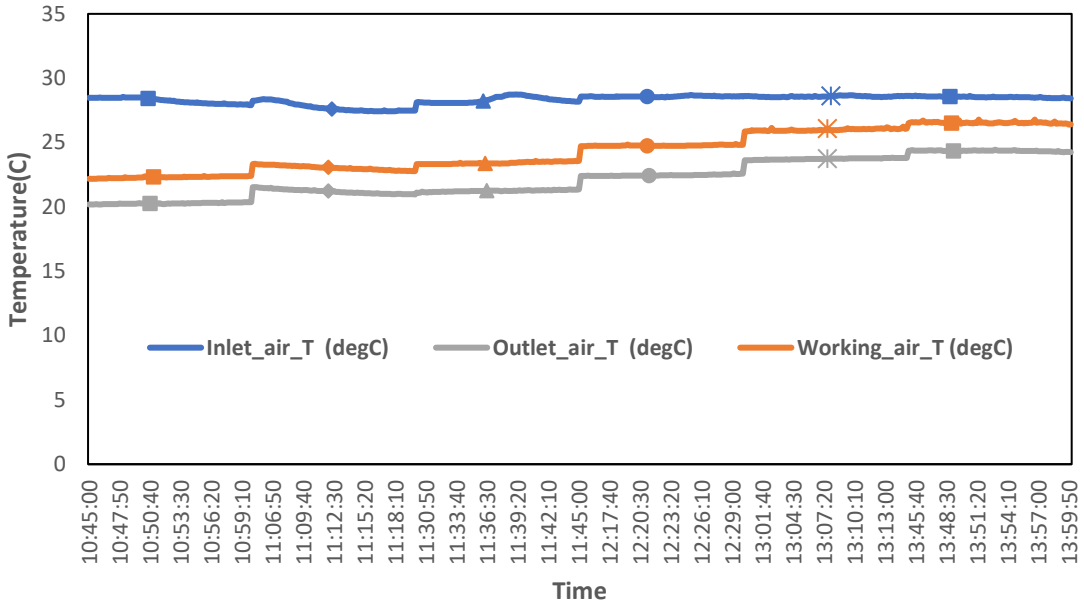


Figure 7-14: Inlet-outlet and working temperature sampling

The effectiveness of the system under the variation of the inlet temperature condition are presented in Figure 7-15. The rise of the air causes the drop on effectiveness.

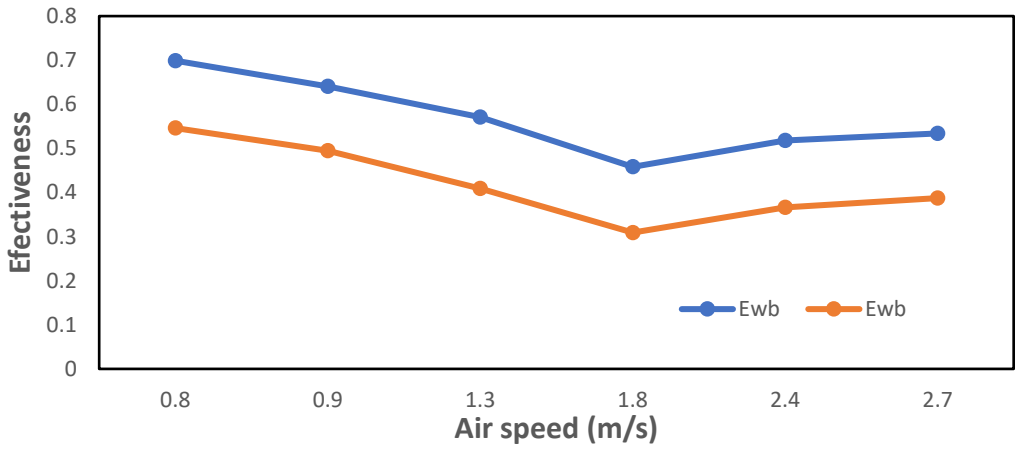


Figure 7-15 Initial The values of effectiveness at steady state conditions

7.4. Summary

The testing aimed to evaluate the potential improvement for animal health and welfare in poultry houses using direct PEC. The results presented for the operation of a direct evaporative cooler installed in a commercial poultry farm application indicate that the evaporative cooling system can cover 300 kWh of the cooling requirement of chickens, with the potential to provide optimised welfare conditions in poultry houses.

The cooling design combines an evaporative cooler and ventilation management. Moreover, a novel dew point evaporative cooler was investigated experimentally to maintain the humidity in the poultry house with less energy consumption strategy. Experimental investigations in lab and on-site contexts are explained in detail in the following chapter. Experimental work mainly focused on the effects of the novel dew point evaporative cooling pads by using forced draft fan while changing the environmental conditions. The experimental work involved the measurement of environmental parameters such as temperature, relative humidity, air velocity, and water temperature and energy consumption at different times under various test conditions.

The cooling pads achieved maximum cooling efficiency of 90%. For higher air temperature and lower air velocity it was found that the maximum reduction in temperature between cooling pad inlet and outlet was 11°C with the highest air temperature, and an acceptable increase in humidity. The results obtained for environmental factors indicated that there was a significant difference relation on environmental factors and cooling efficiency.

Finally, the present study concluded that this dew point cooler unit with cooling pads is not suitable for UK weather conditions, since the unit will have low efficiency in this context, based on measured observations at the studied site. However, the outcomes of this study can provide an effective and low-cost evaporative cooling system solution for other climate contexts, with humid or arid conditions and high temperatures, such as parts of Turkey, a major agricultural producer.

CHAPTER 8 CONCLUSIONS AND FUTURE RESEARCH

8.1. Conclusions

Conventional HVAC systems in poultry houses have unacceptably high fossil fuel consumption to meet the energy demand of thermal regulation and provide deficient animal welfare, given high density stocks, which adversely affects livestock health and thus economic productivity and sustainability. A good understanding of the nature and the structure of the energy consumption of poultry houses is essential to develop solutions for future energy audits and climate change policies in agriculture. The agricultural industry is responsible for 30% of total global energy consumption, and it is set to require ever more energy to meet global food demand with an expanding global population committed to consuming animal protein from poultry.

Commensurate with its remarkable energy consumption levels, the agricultural sector is a major contributor to GHG emissions. Agricultural sustainability depends on increasing use of sustainable energy resources, and the adaptation of HVAC systems for agricultural buildings will play a key role in reducing GHG emissions from the sector, particularly given that poultry production does not have carbon offsetting options (e.g. unlike arable farming, which uses a lot of energy but which consumes a lot of carbon in plant growth, poultry farming is essentially a massive producer of carbon).

In this respect, poultry houses are ripe for review to assess main failures to provide improved welfare conditions in poultry houses, and their impact on the chickens. Industrial players commonly report heat stress, high ammonia ratio, high stocking density, high energy consumption, and related high carbon emission in poultry houses

as major problems. Reviewing the latest technologies adapted to poultry houses indicated that evaporative cooling technology is quite common in the industry and is considered as a promising cooling system for poultry houses in the coming years.

This study reviewed recent evaporative cooling technologies and proposed, tested, and monitored two types of evaporative cooling units. It is important to achieve possible energy saving and CO_2 emissions reduction by using environmentally friendly thermal regulation technology and improving the welfare of chickens and the quality of production in poultry houses. Under UK weather conditions the main challenge is the high heating energy demand of poultry houses. Although heat pumps are not a common application in the industry, some heat pump applications met with positive feedback from theoretical and experimental investigations. This study analysed recent developments in heat pumps to identify the most promising avenues for future research.

Within the scope of this thesis, energy efficient HVAC systems have been introduced, and their performance assessed under real site conditions. The novel renewable applications are a PVT-GSHP system and a direct evaporative cooling unit design with dew point evaporative cooling. Relevant comparisons were made to highlight their reliability and feasibility. The following salient points can be concluded from the findings of this study in terms of implications for the agricultural and HVAC industries.

8.1.1. Current HVAC systems' air distribution is inadequate

Current HVAC systems utilised in poultry houses do not provide sufficient quality of animal welfare conditions for chickens, and they consume massive amounts of energy, with high GHG emissions. The results from the theoretical investigations of the poultry house in UK show that the air exchange in the building does not match the required

fresh air intake, although the highest heat losses are incurred from incoming ventilation.

The failure of conventional HVAC systems in poultry houses causes many issues pertaining to chickens' temperatures. The distribution of the air among the chickens is another concern in poultry management, since they require well distributed and warm air in winter, and chilled air in summer, to let them make sensible heat losses during summer and prevent heat stress. The results from CFD analysis using Design Builder show that the commonly used system, positive-pressurised ventilation, is not suitable because the air distribution flow tends to crash on the floor and then distribute throughout the room, rather than a gentler and more even heat and air distribution. It is concluded that the ventilation management can be enhanced to provide improved welfare, but this is not a certain solution. Future study should focus on air cleaning technologies, which can provide safer and more ambient air for chickens with reduced need for new air input, which would increase the thermal efficiency of poultry houses (i.e. cleaning existing air at a steady temperature rather than replacing it from outside).

8.1.2. Thermal regulation energy requirements

This study highlighted that poultry house internal air quality (IAQ) management is a challenging process with many parameters to be considered, including temperature set points, ambient humidity, and airflow fluctuations. Building heating, cooling, and airflow simulations were presented and the main parameters affecting poultry management were discussed in this study. A poultry farm in Kirton was assessed and theoretically investigated. Building simulations show that the field chicken building requires 100-120 kW heating energy on the cold days with high rate ventilation. With the development of the chicken PC, the theoretical results showed the maximum

heating demand is determined 34.4 MWh/PC while the minimum is 11,1 MWh/PC in the studied poultry farm. These results were compared with the energy consumption of the chicken house taken from real consumption data with the assumption of the 85% efficiency of the existing gas boiler. From the minimum ventilation rate determination and airflow simulations, it was observed that high rates of the ventilation are the main losses in winter conditions, in particular at the fourth week of the growth cycle and end of the chicken PC, when they are adults.

8.1.3. Evaporative cooling

Evaporative cooling units can operate through direct and indirect systems, and can be enhanced with dew point evaporative cooling design in poultry houses. In a direct evaporative cooler unit, heat is directly transferred between hot airflow and cold fluid. On the other hand, in an indirect evaporative cooling unit there is a splitting surface between hot and cold flows. This can be increased with the holes on the surface between the flows using fluids' dew point properties. Compared to direct evaporative cooling and dew point evaporative cooling, dew point cooling systems can provide improved welfare conditions for chickens, in particular high humidity conditions.

The development of more efficient evaporative cooling units is required to provide improved welfare conditions The experimental on-site investigations of the direct evaporative cooling unit demonstrated the efficiency of using ventilating only or cooling and ventilating mode, but this is not a long-term solution for high humidity conditions. Recent studies have made intensive efforts to develop evaporative cooling systems to provide efficient results under humid air conditions. Dew point cooling method is a very promising method to improve chickens' welfare under hot and humid air conditions. The reliability of the system and accuracy of measurements was verified by performing

a lab test under steady state conditions. The results showed 0.6 effectiveness at a temperature of 29˚C, with relative humidity of 27% and a fan speed of 0.8 m/s.

8.1.4. Renewable energy technology implications of PVT-GSHP

8.1.4.1. Heat pump

Associated with conventional, inefficient attempts to mitigate the uncomfortable and unhealthy conditions of poultry houses, the structures have high energy consumption. Renewable energy sources are not common in the industry because of their intermittency and high set-up costs. The novel PVT-GSHP system designed in this study provides sufficient performance within the current design. The heat pump size is highly dependent upon the energy consumption profile, therefore the optimum level of the insulation and proper management of ventilation are important in the building. The 15 kW heat pump sizing provides part of the load in winter, considering the parameters affecting the heat pump efficiency (i.e. return brine temperature, heat sink, and source temperature), and full load during other seasons. The PEC unit on-site investigation revealed that direct evaporative cooling methods palpably increase the welfare condition of the chickens.

8.1.4.2. Application locations and installation

PVT-GSHP system is very promising for locations where thermal regulation loads in the ground are balanced. However, poultry production entails a much greater heating than cooling load in UK weather conditions, and vice versa under hot air conditions (particularly with high humidity). It is essential to improve designs to increase the actual running hours of the heat pump for optimum sustainability of this solution. Therefore, for further investigations hybrid designs are recommended to develop seasonal storage, as a control approach for further investigation, with a smart control.

Traditional vertical GSHP systems use relatively large boreholes that require much larger lengths of pipe, usually made from high density polyethylene (HDPE), to be installed into the ground. The processes become increasingly expensive as the length of the bore increases. Thus, the initial costs for a GSHP system are very high in a traditional system. This system has less area than horizontal ground heat exchangers, with much more functional efficiency and cheaper economic costs than traditional vertical boreholes.

Horizontal ground heat exchangers have a lower cost in comparison to vertical ones; however, they require more land area, and they are relatively less efficient. For the sake of reducing the required land area for horizontal GHEs, flat panel GHEs have recently been introduced. In this study, a novel copper vertical ground heat exchangers (VGHE) was presented with serial connections. This study therefore also focused on the development of the low-cost PHE heat exchangers for the PV panels and investigated cheap and easy to apply VGHEs, instead of conventional boreholes. The experiments aimed to characterise ground thermal arrays and PVT collector integrated designs in terms of their dynamic behaviour, to provide reliable data for agriculture HVAC system designers over a wide range of time-varying conditions.

The novel PVT-GSHP design and components are low-cost when compared to existing heat pump and solar systems. Given the UK climate profile and the novelty of the heat pump components, this study mainly focused on the developed PVT-GSHP for poultry farms with a high heating requirement, land restrictions/limitations for boreholes, and noise restrictions on air heat exchangers, which limit the latent heat pump market. Moreover, conventional ground borehole heat exchangers have high costs due to their relatively complicated design and excavation process.

8.1.5. Academic contributions of validated experimental results

The novel PVT-GSHP design and components are cheaper compared to existing heat pump and solar systems, and the validated results indicate that the PEC cooling unit and novel and cheap PVT-GSHP integrated designs are reliable systems for agricultural applications in terms of their dynamic behaviour.

A full quantitative validation was undertaken made and each innovative component was experimentally investigated under the transient conditions for the long term (>1 year). The dynamic behaviours of the whole integrated system and each novel component were investigated, leading to the following key findings:

- The annual heat transfer rate of the VGHE at 2.5 m is 3.65 kW on average. The heat transfer amount per meter is 29.2 W/m on average.
- The heat transfer rate of the 46 m^2 novel solar PHE achieved 3.42 kW, with a corresponding heat flux of 73.9 W/m^2.
- The total thermal output from the heat pump was approximately 15 MWh per annum, with a corresponding Seasonal COP of 3.73.
- The daily COP of the heat pump ranged from 2.05 to 5.17 during the PCs of the chickens.
- The projected simple payback period is 8.14 years, with £4,890 total annual savings with government revenues under two operating modes of the hybrid system. The dual-source design will be assessed in-depth as a continuation of this work in the future.

Ventilation constitutes a major source of thermal regulation inefficiency in poultry houses, which in the UK context mainly comprises heat losses due to the influx of colder air. Air ventilation has intrinsically high energy consumption, and also increases

the cost of thermal regulation. Current ventilation systems are inadequate for optimum animal welfare. Furthermore, there is an increasing environmental concern about localised dust pollutants emitted from poultry houses during the ventilation process. Direct (local) environmental impacts and carbon emissions can be decreased with lower reliance on basic ventilation, such as by using air filtration systems within houses to reduce the need for new air input.

The novel PVT-GSHP system is a very promising solution for agriculture HVAC industry. As future work of this study, the system will be assessed to run with three operating modes to have more efficient hours of operation: (1) Only ground source (2) only solar source with seasonal storage from solar to the ground and (3) the current operation mode by the attachment of a three-way valve on the PVT-GSHP system.

8.2. Future research directions

HVAC is currently the key tool in the maintenance of animal housing temperature and internal air quality. In this study, the results from the in-situ performance assessments and theoretical and experimental investigations of the poultry house by applying heat pump unit and PEC cooler were mainly based on broiler houses. Further works are expected to extend the research on the different types of agricultural buildings, such as layer poultry farming, pig farms, and greenhouse buildings. In this respect, more precise monitoring and theoretical analysis are recommended for future study. More generally, in order to decrease the high energy consumption of existing HVAC systems in the poultry houses, there are still two main issues to be addressed, as discussed below.

The reduction of fuel consumption entails a commensurate reduction in GHG emissions after retrofitting, and the overall CO_2 rate decreases in the field. Therefore,

the birds' welfare and productivity is expected to increase, which also has positive long-term implications for the economic and environmental output dimensions. In order to extend the use of GSHPs and their novel hybrid designs in poultry houses, efficient ground charging is needed. In the UK poultry farms mainly require heating, which makes seasonal soil storage more challenging, and ground source systems become less efficient year by year. It is recommended that studies attempt to improve smart soil storage approaches under different ambient conditions. Novel applications like nanomaterials can be used to enhance soil conductivity and charge capacity, with the use of novel ground vertical pipes in addition to smart soil storage approaches.

More theoretical investigations under different climate condition and CFD simulations are recommended to improve the accuracy of the results. Moreover, in addition to the experimental work carried out in this study, more experimental investigations on the novel ground pipes presented in this thesis with the soil injected novel applications are recommended to improve thermal conductivity and provide a more stable charging process.

The main heat losses were attributable to the high rate of air ventilation, to provide sufficient fresh air per kg of chicken, as defined in the ASHRAE standards. The assessment of the existing building shows that the welfare conditions of chickens are still in need of improvement. Put simply, current ventilation aiming to provide amenable air quality for chickens entails massively inefficient energy consumption for ventilation itself and for thermal regulation, with disproportionately limited achievement of actual welfare for chickens. Furthermore, there is increasing concern about localised health impacts of pollutants emitted from poultry houses. Chickens produce ammonia and odour emissions, which are harmful to themselves and to eco-systems surrounding

farms. In areas where there is a high density of poultry farms, owners are seeing their planning applications delayed and sometimes rejected due to difficulties over the controls of ammonia emissions.

More air circulation can be provided through internal fans (inside the houses) with higher internal air velocities, but the impact of this would be limited, considering the air distribution limitations and the high density of the chickens in the building. Effective IAQ control management without adjustment of ventilation should be improved in future to overcome the high cost of ventilation management while decreasing mortality rates, reducing waste and loss, improving metabolic efficiency and animal welfare, and maximising feed-conversion ratios. A novel design with fibre filters and coating technologies could provide improved IAQ and lighting for poultry, over and above all current standards. This would present a substantive contribution to increase the efficiency of poultry house operation, reducing carbon consumption and improving the conditions of chickens and thus product quality, and offering a more environmentally friendly, sustainable, and economically efficient solution for poultry farming to meet the global food security needs of the coming years.

The novel PVT-GSHP system is a very promising solution for agriculture HVAC industry. As future work of this study, the system will be assessed to run with three operating modes to have more efficient hours of operation: (1) Only ground source (2) only solar source with seasonal storage from solar to the ground and (3) the current operation mode by the attachment of a three-way valve on the PVT-GSHP system. Moreover, soil thermal conductivity will be assessed to increase the system efficiency by adding materials and solutions into the soil in order to enhance the system efficiency.

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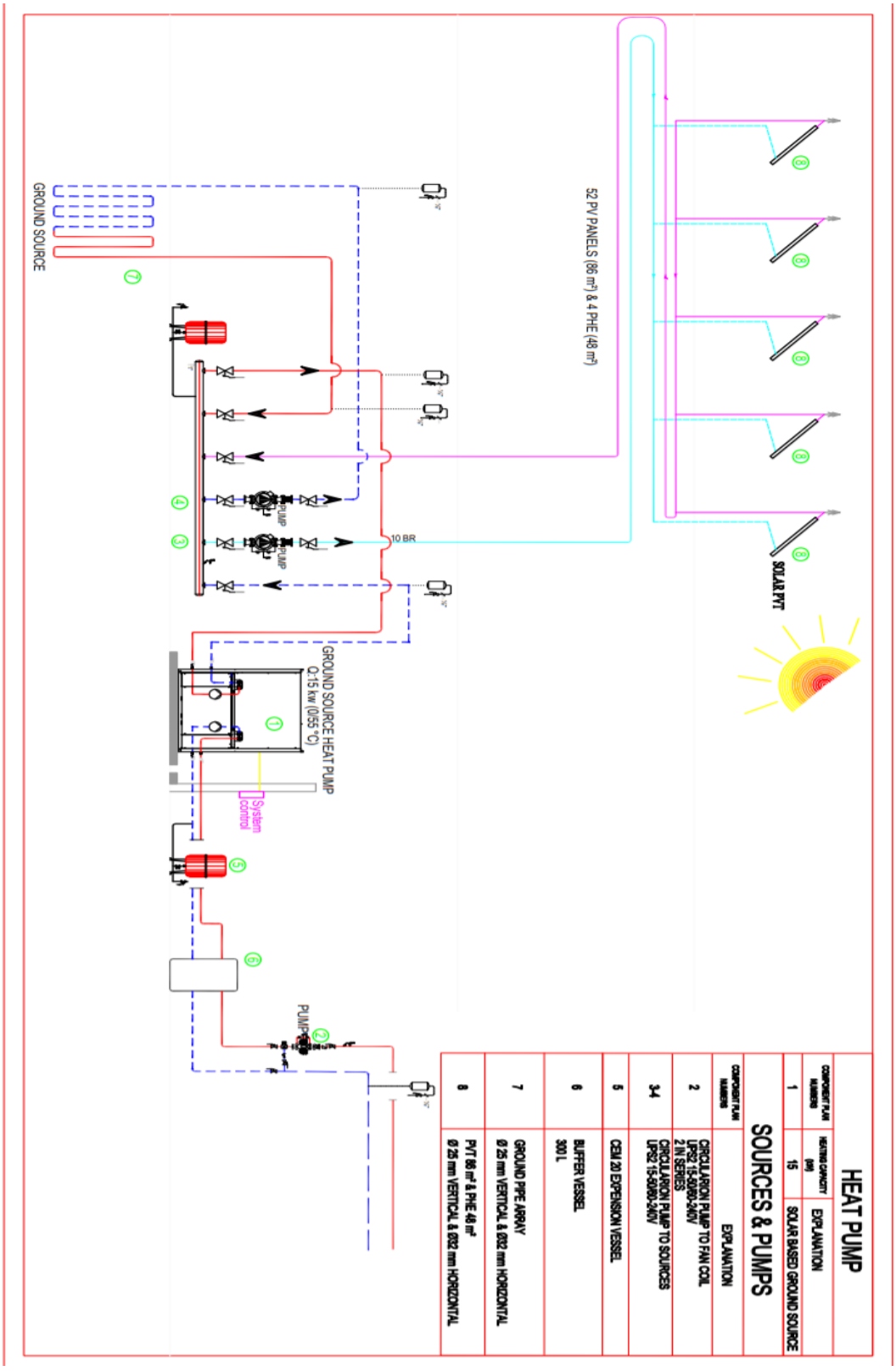
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APPENDIX A


HEAT PUMP
COMPONENT PLAN NUMBERS
HEATING CAPACITY (kW)
EXPLANATION
1
15
SOLAR BASED GROUND SOURCE
SOURCES & PUMPS
COMPONENT PLAN NUMBERS
EXPLANATION
2
CIRCULARION PUMP TO FAN COIL UPS2 15-50/80-240V 2 IN SERIES
3-4
CIRCULARION PUMP TO SOURCES UPS2 15-50/80-240V
5
CEM 20 EXPENSION VESSEL
6
BUFFER VESSEL 300 L
7
GROUND PIPE ARRAY Ø 25 mm VERTICAL & Ø32 mm HORIZONTAL
8
PVT 86 m² & PHE 48 m² Ø 25 mm VERTICAL & Ø32 mm HORIZONTAL
GROUND SOURCE
52 PV PANELS (86 m²) & 4 PHE (48 m²)
SOLAR PVT
10 BR
GROUND SOURCE HEAT PUMP Q:15 kw (0/55 °C)
System Control
PUMP